

PD ISO/TR 14521:2010



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Gears — Calculation of load capacity of wormgears

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TECHNICAL REPORT

PD ISO/TR 14521:2010

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Gears — Calculation of load capacity of wormgears

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.

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

In exceptional circumstances, when a technical committee has collected data of a different kind from that which is normally published as an International Standard ("state of the art", for example), it may decide by a simple majority vote of its participating members to publish a Technical Report. A Technical Report is entirely informative in nature and does not have to be reviewed until the data it provides are considered to be no longer valid or useful.

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.

ISO/TR 14521 was prepared by Technical Committee ISO/TC 60, *Gears*, Subcommittee SC 1, *Nomenclature and wormgearing*.

Introduction

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

This Technical Report 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 Technical Report can apply to worm gearing with cylindrical helicoidal worms having the following thread forms: A, C, I, N, K.

Other than the requirements of 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 worm and worm wheel be supplied by the same manufacturer.

In this Technical Report, 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 7.3. The permissible torque on the worm wheel is the least of the calculated values.

Gears — Calculation of load capacity of wormgears

WARNING — Special attention is required when establishing the tooth geometry especially for C type gear profile.

1 Scope

This Technical Report specifies equations 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 Technical Report.

The load rating and design procedures are valid for sliding velocities over tooth surfaces of up to 25 m/s and contact ratios equal to or greater than 2,1. For wear, sliding velocities over tooth surfaces are not below 0,1 m/s.

The rules and recommendations for the dimensioning, lubricants or materials selected by this Technical Report 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 Technical Report is intended for use by experienced gear designers who are able to make informed judgements concerning factors. It is not intended for use by engineers who lack the necessary experience. See 5.4.

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

2 Normative references

The following referenced documents are indispensable for the application 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 701:1998, *International gear notations — Symbols for geometrical data*

ISO 1122-2:1999, *Vocabulary of gear terms — Part 2: Definitions related to worm gears geometry*

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

ISO/TR 10828:1997, *Worm gears — Geometry of worm profiles*

DIN 3974-1:1995, *Accuracy of worms and wormgears — Part 1: General bases*

DIN 3974-2:1995, *Accuracy of worms and wormgears — Part 2: Tolerances for individual errors*

3 Symbols and terminology

3.1 Symbols

NOTE Where applicable, the symbols are in accordance with ISO 701 and the definitions are in accordance ISO 1122-2.

Table 1 — Symbols for worm gears

Symbol	Description	Unit	Figure	Equation Number
a	centre distance	mm		38/39
a_0, a_1, a_2	oil sump temperature coefficients, calculated according to method C	-		160 to 166
$a_{\min}, a_{\max}$	minimum and maximum centre distance for tooling selection	mm		G.2/G.3
a_T	centre distance of standard reference gear	mm		
b_1	worm facewidth	mm		22
b_2	facewidth of the wheel as specified in DIN 3975	mm		36
b_{2H}	effective wheel facewidth	mm	Fig. 4	
$b_{2H, \text{std}}$	Standard worm wheel facewidth	mm		52
b_{2R}	wheel rim width	mm	Fig. 4	
b_H	half hertzian contact width	mm	Fig.19	
c_1, c_2	tip clearance	mm		
c_1^*, c_2^*	tip clearance coefficient in axial section	mm		
c_{oil}	specific heat capacity of the oil (for temperature calculation with spray lubrication)	Ws/(kg.K)		170
c_α	proximity value for the viscosity pressure exponent α	m ² /N		64/66
d_{a1}	worm tip diameter	mm		13
d_{a2}	worm wheel tip diameter	mm		34
d_{b1}	base diameter of involute helicoid (for I profile)	mm		21
d_{e2}	worm wheel outside diameter	mm		35
dF	force transmitted by a segment of the contact line	N	Fig. B.2	B.3
dl	length of contact line segment	mm		B.1
d_{f1}	worm root diameter	mm		14
d_{f2}	worm wheel root diameter	mm		33
d_{m1}	worm reference diameter	mm	Fig. 2/5	9
d_{m1T}	reference diameter of the worm, from standard reference gear	mm		
d_{m2}	worm wheel reference diameter	mm	Fig 3/5	24
d_{m2T}	reference diameter of the wheel, from standard reference gear	mm		
d_{w1}	worm pitch diameter	mm		40
d_{w2}	worm wheel pitch diameter	mm		41
e_{mx1}	worm reference tooth space width in axial section	mm	Fig. 2	16
e_{n1}	worm normal tooth space width in normal section	mm		18

Table 1 (continued)

Symbol	Description	Unit	Figure	Equation Number
e_{m2}	worm wheel tooth space width in mid-plane section	mm		27
f_h	Worm wheel face width factor for the parameter for the minimum mean lubricant film thickness	-		58
f_p	Worm wheel face width factor for the parameter for the mean hertzian stress	-		59
h_1	worm tooth depth	mm		10
h_2	worm wheel tooth depth	mm		31
h_{am1}	worm tooth reference addendum in axial section	mm	Fig. 5	11
h_{am2}	worm wheel tooth reference addendum in mid-plane section	mm	Fig. 5	29
h_{am1}^*	worm tooth reference addendum coefficient in axial section	-		11
h_{am2}^*	worm wheel tooth reference addendum coefficient in mid-plane section	-		29
h_{e2}	worm wheel tooth external addendum	mm		32
h_{fm1}	worm tooth reference dedendum in axial section	mm		12
h_{fm2}	worm wheel tooth reference dedendum in mid-plane section	mm		30
h_{fm1}^*	worm tooth reference dedendum coefficient in axial section	-		
h_{fm2}^*	worm wheel tooth reference dedendum coefficient in mid-plane section	-		30
h_{min}	minimum lubricant film thickness	μm		C.1
$h_{min\ m}$	minimum mean lubricant film thickness	μm		63
h^*	parameter for minimum mean lubricant film thickness	-		56/57
h_T^*	parameter for minimum mean lubricant film thickness of the standard reference gear	-		
j_x	axial backlash	mm		
k	lubricant constant	1/K		69/71
k^*	mean heat transition coefficient	W/(m ² ·K)		
l_1	spacing of the worm shaft bearings	mm		
l_{11}, l_{12}	bearing spacing of the worm shaft	mm	Fig. 11	
m_{max}	maximum axial module for tooling selection	mm	Fig. 11	G.4
m_{min}	minimum axial module for tooling selection	mm		G.5
m_{xhob}	axial module for tooling selection	mm		Annex G
m_n	normal module	mm		8
$m_{x\ 1}$	axial module	mm		2/G.1
Δm	material loss	mg		
Δm_{lim}	material loss limit	mg		
n_1	rotational speed of the worm shaft	min ⁻¹		
n_2	rotational speed of the wheel	min ⁻¹		
N_S	number of starts per hour			112

Table 1 (continued)

Symbol	Description	Unit	Figure	Equation Number
p_0	environmental pressure	N/mm ²		
p_{b1}	base cylinder pitch for I profile	mm		22
p_{Hm}	hertzian stress; mean value for the total contact area	N/mm ²		B.7
p_m^*	parameter for the mean hertzian stress	-		53/54
p_{mT}^*	parameter for the mean hertzian stress of the standard reference gear	-		
p_{n1}	normal pitch	mm		7
p_{t2}	transverse pitch	mm		25
p_{x1}	axial pitch	mm	Fig. 2	1
p_{z1}	lead of worm threads	mm		3
q_1	diameter factor	mm		4
q_{hob}	diameter factor for hob	mm		Annex G
r_{g2}	worm wheel throat radius	mm		37
s_2	reference tooth thickness of the wheel teeth in the spur section	mm		153
s_{f2}	mean tooth root thickness of the wheel teeth in the spur section	mm		153
s_{ft2}	mean tooth root thickness of the wheel teeth in the spur section	mm		153
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	Fig. 3	26
s_K	rim thickness	mm	Fig. 12	
s_{Wm}	wear path inside of the required life expectancy	mm		71/D.1
s_{mx1}	worm tooth thickness in axial section	mm	Fig. 2	15
s_{mx1}^*	worm tooth thickness in axial section coefficient	-		15
s_{n1}	normal worm tooth thickness in normal section	mm		17
s^*	parameter for the mean sliding path	-		59/60/D.8
s_T^*	parameter for the mean sliding path of the standard reference gear	-		
Δs	tooth thickness loss	mm		
u	gear ratio			42
u_T	gear ratio of the standard reference gear			
v_1	velocity of a flank point of the worm	m/s	Fig. B.1	62
v_2	velocity of a flank point of a worm wheel	m/s	Fig. B.1	62
v_{1n}	worm velocity component normal to the contact line	m/s	Fig. B.2	
v_{2n}	wheel velocity component normal to the contact line	m/s	Fig. B.2	
$\vec{V}_{gB}$	sliding velocity at the reference diameter in flank direction	m/s		91/92/93/E.6

Table 1 (continued)

Symbol	Description	Unit	Figure	Equation Number
$\bar{v}_g$	sliding velocity at mean reference diameter	m/s		51
v_Σ	sum velocity	m/s		53
$v_{\Sigma n}$	sum velocity in normal direction	m/s		53
x_2	worm wheel profile shift coefficient	-		28
$x_{2\max}$	maximum worm wheel profile shift coefficient for tooling selection	-		H.3
$x_{2\min}$	minimum worm wheel profile shift coefficient for tooling selection	-		H.3
z_1	number of threads in worm	-		
z_2	number of teeth in worm wheel	-		
A	coefficient for kinematic viscosity			76
A_{ges}	free surface of the gear housing	m ²		
A_{fl}	total flank surface of the worm wheel	mm ²		131
A_R	dominant cooled surface of the gear set	m ²		174
B	coefficient for kinematic viscosity	-		76
c	immersion factor	-		
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		62
E_{steel}	modulus of elasticity for steel	N/mm ²		62
F_{xm1}	axial force to the worm shaft	N		46/49
F_{xm2}	axial force to the worm wheel	N		45/48
F_{rm1}	radial force to the worm shaft	N		47
F_{rm2}	radial force to the worm wheel	N		53
F_{tm1}	circumferencial or tangential force to the worm shaft	N		45/48
F_{tm2}	circumferencial or tangential force to the worm wheel	N		46/49
dF/db	specific loading	N/mm		
J_{OT}	reference wear intensity	-	Fig. 10	111 to 121
J_W	wear intensity	-		110
K_n	rotational speed factor / wheel bulk temperature	-		177
$K_{H\alpha}$	transverse load distribution factor	-		
$K_{H\beta}$	longitudinal load distribution factor	-		
K_S	size factor / wheel bulk temperature	-		179
K_A	application factor	-		
K_V	dynamic factor	-		
K_W	lubricant film thickness parameter	-		122
K_v	viscosity factor / wheel bulk temperature	-		178
L_h	life time	h		
N_L	number of stress cycles of the worm wheel	-		73
P_1	input power to the worm shaft	W		

Table 1 (continued)

Symbol	Description	Unit	Figure	Equation Number
P_2	output power from the worm wheel shaft	W		
P_K	cooling capacity of the oil with spray lubrication	W		169
P_V	total power loss of the worm gear unit	W		80
P_{VO}	idle running power loss	W		80/81/H.1
P_{Vz1-2}	meshing power loss in reducer	W		104
P_{Vz2-1}	meshing power loss in increaser	W		106
P_{VD}	sealing power loss	W		86/87
P_{VLP}	bearing power loss through loading	W		82 to 85
Q_{oil}	spray quantity	m ³ /s		
Ra_1	arithmetic mean roughness	μm		
Ra_T	arithmetic mean roughness for reference gear	μm		80
Rz_1	mean roughness depth	μm		
S_F	tooth breakage safety factor	-		148
$S_{F \min}$	minimum tooth breakage safety factor	-		149
S_H	pitting safety factor	-		133
S_T	temperature safety factor	-		157/167
$S_{T \min}$	minimum temperature safety factor	-		158/168
S_W	wear safety factor	-		107
$S_{W \min}$	minimum wear safety factor	-		108
S_δ	deflection safety factor	-		143
$S_{\delta \lim}$	limit of deflection safety factor	-		144
T_1	input torque to the worm shaft	Nm		43
T_{1N}	nominal input torque to the worm shaft	Nm		43
T_2	output torque from the worm wheel	Nm		44/B.4/ B.5
T_{2N}	nominal output torque from the worm wheel	Nm		44
W_H	pressure factor	-		126/127
W_{ML}	material - lubricant factor	-		
W_{NS}	start factor	-		125
W_S	lubricant structure factor	-		123/124
Y_F	form factor / tooth breakage	-		151/152
Y_G	geometry factor / coefficient of friction	-		101/102
Y_K	rim thickness factor / tooth breakage	-		155
Y_{NL}	life factor / tooth breakage	-	Fig 13a/b	Table 11
Y_R	roughness factor / coefficient of friction	-		103/104
Y_S	size factor / coefficient of friction	-		99/100
Y_W	material factor / coefficient of friction	-		
Y_ε	contact factor / tooth breakage	-		151
Y_γ	lead factor / tooth breakage	-		154
Z_h	life factor / pitting	-		136

Table 1 (continued)

Symbol	Description	Unit	Figure	Equation Number
Z_{oil}	lubricant factor / pitting	-		142
Z_S	size factor / pitting	-		138/139
Z_u	gear ratio factor	-		141/142
Z_v	velocity factor / pitting	-		137
α	pressure viscosity factor	m ² /N		
α_{ot}	axial pressure angle for A profile	°		
α_L	heat transition coefficient for immersed wheel teeth	W/(m ² K)		175
α_n	normal pressure angle	°		19
β_{m1}	reference helix angle of worm	°		6
γ_{m1}	reference lead angle of worm	°		5
γ_{b1}	base lead angle of worm thread (for I profile)	°		19
δ_{lim}	limiting value of deflection	mm		147
δ_m	incurred deflection	mm		145/146
δ_{Wn}	flank loss from wheel through abrasive wear in the normal section	mm		109
$\delta_{W \text{ lim}}$	limiting value of flank loss	mm		132
$\delta_{W \text{ lim } n}$	limiting value of flank loss in normal section	mm		128 to 130
η_{ges}	total efficiency in reducer	-		77
η'_{ges}	total efficiency in increaser	-		78
η_{z1-2}	gear efficiency in reducer	-		88
η_{z2-1}	gear efficiency in increaser	-		89
η_{0M}	dynamic viscosity of lubricant at ambient pressure and wheel bulk temperature	Ns/m ²		67
θ	temperature	°C		
$\Delta\theta$	temperature difference between oil sump and worm wheel bulk temperature	°C		173
θ_{in}	oil entrance temperature	°C		
θ_{out}	oil exit temperature	°C		
θ_0	ambient temperature	°C		
θ_{oil}	spray temperature	°C		
$\Delta\theta_{oil}$	oil temperature difference between input and output cooling system	°C		171
θ_M	wheel bulk temperature	°C		172/176
θ_S	oil sump temperature	°C		159/161
$\theta_{S \text{ lim}}$	limiting value of oil sump temperature	°C		
μ_{0T}	base coefficient of friction	-		91 to 93
μ_{zm}	mean tooth coefficient of friction	-		90
ν_1	POISSON ratio of the worm	-		
ν_2	POISSON ratio for the worm wheel	-		
ν_θ	kinematic viscosity at oil temperature θ	mm ² /s		74
ν_{40}	kinematic viscosity at 40 °C	mm ² /s		74

Table 1 (continued)

Symbol	Description	Unit	Figure	Equation Number
ν_{100}	kinematic viscosity at 100 °C	mm ² /s		
ν_M	kinematic viscosity at wheel bulk temperature	mm ² /s		67
ρ	profile radius of the grinding disk for C type	mm		
ρ_{oil}	lubricant density	kg/dm ³		
ρ_{oil15}	lubricant density at 15 °C	kg/dm ³		68
ρ_{oilM}	lubricant density at wheel bulk temperature	kg/dm ³		67
ρ_{red}	equivalent radius of curvature	mm		B.2
ρ_z	friction angle for the tooth coefficient of friction	°		
ρ_{Rad}	material density of the wheel	mg/mm ³		
$\Delta_s \lim$	allowable tooth thickness loss	mm		129
$\sigma_{H \lim T}$	pitting strength	N/mm ²		
σ_H	contact stress	N/mm ²		135
σ_{Hm}	mean contact stress	N/mm ²		61
σ_{HG}	limiting value for the mean contact stress	N/mm ²		135
τ_F	shear stress at tooth root	N/mm ²		150
$\tau_{F \lim T}$	shear endurance strength	N/mm ²		
τ_{FG}	limiting value for shear stress at tooth root	N/mm ²		156
ω_2	angular velocity	s ⁻¹		

3.2 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.

The following conditions can limit the rated load capacity:

- **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 thread and worm shaft breakage:** shaft breakage can occur as a result of bending fatigue or overload,
- **worm shaft deflection:** excessive deformation under load modifying 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,
- **working temperature:** when excessively high working temperature leads to accelerated degradation of the worm gear lubricant,
- **type of limitations in worm gear rating:** Table 2 indicates the relationship between different forms of capacity limits in combination with speed and torque.

When the many influence factors such as material properties, meshing conditions, (e.g. contact pattern under load), lubrication and etc. are considered, it is apparent that values of Hertzian pressure along the lines of contact are extremely significant.

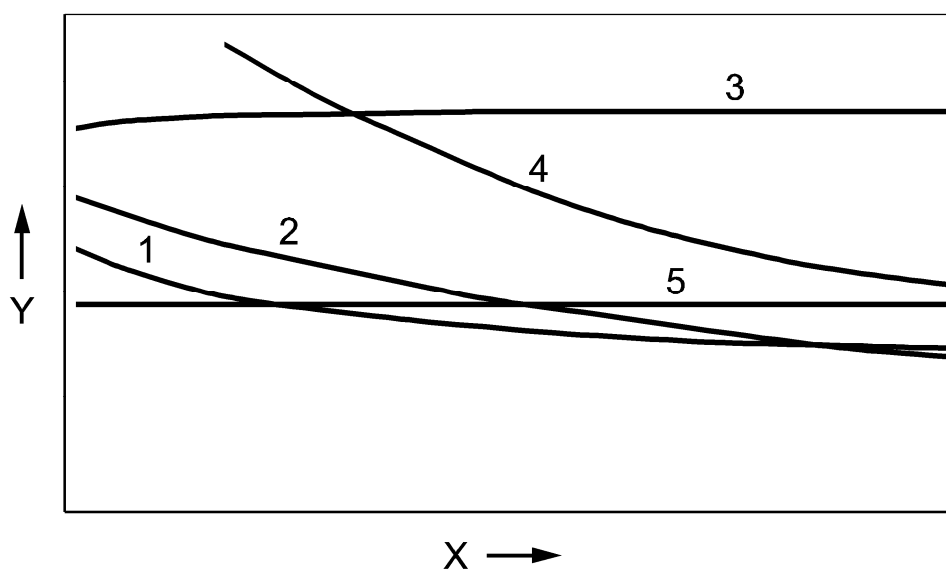
The different rating criteria are calculated independently and not in combination (see Figure 1). For a given worm gear pair, the zone of contact could change with loading. At a steady load, fatigue pits can develop which may subsequently be reduced by wear. This can be followed by further pitting, additional wear or a stable condition.

The most significant factors of gear tooth damage are shown in the first column of Table 2.

The load capacity of worm gearing is determined by calculations dealing with permissible stresses for pitting and wear, the deflection in worm, shafts, and the temperature. The permissible torque shall be determined from the least of the calculated values.

Table 2 — Most significant factors: failure mode according to influence factors

Influence factors	Failure modes					
	Wear	Pitting	Tooth-Breakage	Worm shaft Deflection	Scuffing	Low efficiency
Hertzian pressure	x	x	x	x	x	x
Worm speed	x	x			x	x
Oil film thickness	x	x			x	x
Oil	x	x			x	x
Contact Pattern	x	x	x		x	x
Worm surface roughness	x	x			x	x
Shearing value			x			

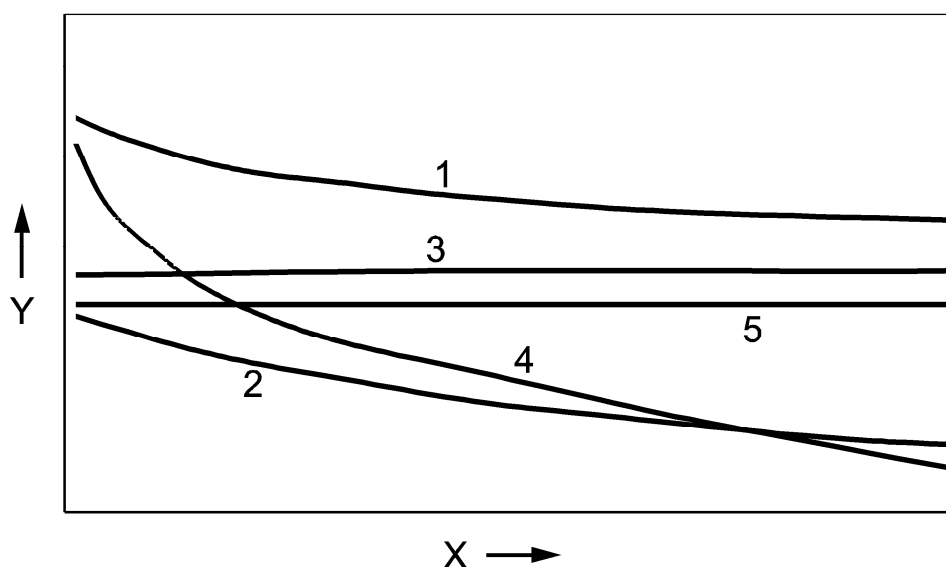


Key

- 1 wear
- 2 pitting
- 3 worm shaft deflection
- 4 temperature
- 5 tooth breakage

- X worm speed n_1
- Y output torque T_2

Figure 1a — Example for small center distance



Key

- 1 wear
- 2 pitting
- 3 worm shaft deflection
- 4 temperature
- 5 tooth breakage

- X worm speed n_1
- Y output torque T_2

Figure 1b — Example for large centre distance

Figure 1 — Limitations of worm gear torque

3.3 Basis of the method

The calculation methods are partly based on investigations of test gears (see, standard reference gear, 5.2), and partly on application experience. Investigations on test gears are mainly ascertained through varied test conditions and verified through practical experience. They are not however physically justified.

4 Formulae for calculation of dimensions

4.1 Parameters for a cylindrical worm

4.1.1 Axial pitch

$$p_{x1} = \pi \cdot m_{x1} \quad (1)$$

4.1.2 Axial module

$$m_{x1} = \frac{p_{x1}}{\pi} \quad (2)$$

4.1.3 Lead

$$p_{z1} = z_1 \cdot p_{x1} \quad (3)$$

4.1.4 Diametral factor

$$q_1 = \frac{d_{m1}}{m_{x1}} \quad (4)$$

4.1.5 Reference lead angle

$$\tan \gamma_{m1} = \frac{m_{x1} \cdot z_1}{d_{m1}} = \frac{z_1}{q_1} \quad (5)$$

4.1.6 Reference helix angle

$$\beta_{m1} = 90^\circ - \gamma_{m1} \quad (6)$$

4.1.7 Normal pitch on reference cylinder

$$p_{n1} = p_{x1} \cdot \cos \gamma_{m1} \quad (7)$$

4.1.8 Normal module

$$m_n = m_{x1} \cdot \cos \gamma_{m1} \quad (8)$$

4.1.9 Reference diameter

$$d_{m1} = q_1 \cdot m_{x1} \quad (9)$$

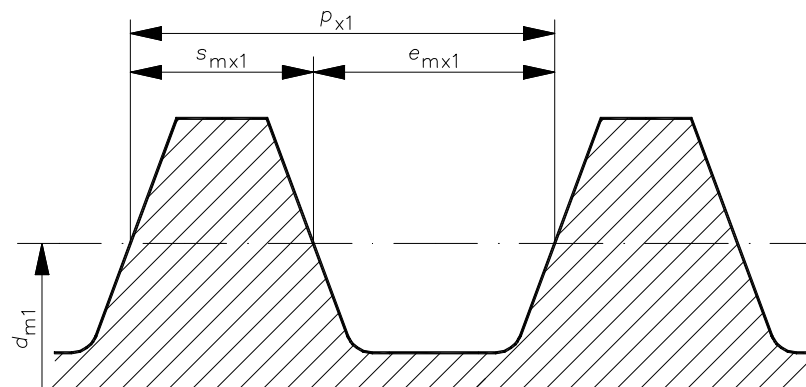


Figure 2 — Axial pitch, reference tooth thickness and reference tooth space for worm

4.1.10 Reference tooth depth

$$h_1 = h_{am1} + h_{fm1} = \frac{1}{2} \cdot (d_{a1} - d_{f1}) \quad (10)$$

4.1.11 Reference addendum

$$h_{am1} = h_{am1}^* \cdot m_{x1} = \frac{1}{2} \cdot (d_{a1} - d_{m1}) \quad (11)$$

where h_{am1}^* is the addendum coefficient = 1 (normally)

4.1.12 Reference dedendum

$$h_{fm1} = h_{fm1}^* \cdot m_{x1} = \frac{1}{2} \cdot (d_{m1} - d_{f1}) \quad (12)$$

where h_{fm1}^* = dedendum coefficient; generally $1,1 < h_{fm1}^* < 1,3$, the recommended value is 1,2

4.1.13 Tip diameter

$$d_{a1} = d_{m1} + 2 \cdot h_{am1} \quad (13)$$

4.1.14 Root diameter

$$d_{f1} = d_{m1} - 2 \cdot h_{fm1} \quad (14)$$

4.1.15 Tooth thickness coefficient s_{mx1}^*

A recommended value is $s_{mx1}^* = 0,5$

In general practice, this coefficient is very often less than 0,5 when there is a wish to increase the worm wheel tooth thickness to prevent wear of bronze and to increase strength of worm wheel.

See Figure 2

4.1.16 Reference tooth thickness in the axial section

$$s_{mx1} = s_{mx1}^* \cdot p_{x1} \quad (15)$$

4.1.17 Reference space width in the axial section

$$e_{mx1} = p_{x1} - s_{mx1} \quad (16)$$

4.1.18 Normal tooth thickness

$$s_{n1} = s_{mx1} \cdot \cos \gamma_{m1} \quad (17)$$

4.1.19 Normal space width

$$e_{n1} = e_{mx1} \cdot \cos \gamma_{m1} \quad (18)$$

4.1.20 Profile flank form

It is specified by a letter:

A is the straight sided axial thickness section

N is the straight sided normal space width section

I is the involute helicoid

K is the milled helicoid by double cone form

C is the milled helicoid by circular convex form

4.1.21 Normal pressure angle

$$\tan \alpha_n = \tan \alpha_{ot} \cdot \cos \gamma_{m1} \quad (19)$$

NOTE 1 For I, N, K, C profiles $\alpha_n = \alpha_{on}$ defined in ISO 10828

NOTE 2 For A profile α_n is defined by Equation (19)

4.1.22 Base lead angle for I profile

$$\cos \gamma_{b1} = \cos \gamma_{m1} \cdot \cos \alpha_n \quad (20)$$

4.1.23 Base diameter for I profile

$$d_{b1} = d_{m1} \cdot \frac{\tan \gamma_{m1}}{\tan \gamma_{b1}} = \frac{m_{x1} \cdot z_1}{\tan \gamma_{b1}} \quad (21)$$

4.1.24 Base cylinder pitch for I profile

$$p_{b1} = p_{x1} \cdot \cos \gamma_{b1} \quad (22)$$

4.1.25 Worm face width

$$b_1 \geq \sqrt{(d_{e2})^2 - (2 \cdot a - d_{a1})^2} \quad (23)$$

4.2 Parameters for a worm wheel

4.2.1 Reference diameter

$$d_{m2} = d_{w2} + 2 \cdot x_2 \cdot m_{x1} \text{ or } d_{m2} = 2 \cdot a - d_{m1} \quad (24)$$

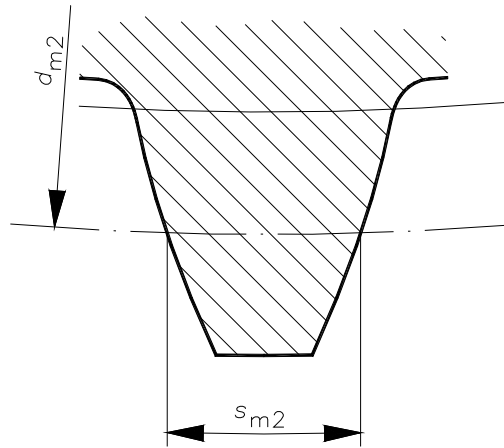


Figure 3 — Tooth thickness for worm wheel

4.2.2 Transverse pitch

$$p_{t2} = p_{x1} \quad (25)$$

4.2.3 Transverse tooth thickness at reference diameter

This value can be calculated only for a worm wheel without addendum modification as follows:

$$s_{m2} = e_{mx1} - j_x \quad (26)$$

where j_x = axial backlash

See Figure 3.

4.2.4 Space width at reference diameter

$$e_{m2} = p_{x1} - s_{m2} \quad (27)$$

4.2.5 Profile shift coefficient

$$x_2 = \frac{2 \cdot a - d_{m1} - m_{x1} \cdot z_2}{2 \cdot m_{x1}} \quad (28)$$

4.2.6 Addendum

$$h_{am2} = m_{x1} \cdot h_{am2}^* = \frac{1}{2} \cdot (d_{a2} - d_{m2}) \quad (29)$$

where h_{am2}^* is the addendum coefficient; $h_{am2}^* = 1$ (normally).

4.2.7 Dedendum

$$h_{fm2} = m_{x1} \cdot h_{fm2}^* = \frac{1}{2} \cdot (d_{m2} - d_{f2}) \quad (30)$$

where h_{fm2}^* is the dedendum coefficient; generally $1,1 < h_{fm2}^* < 1,3$, the recommended value is 1,2.

4.2.8 Tooth depth

$$h_2 = h_{am2} + h_{fm2} \quad (31)$$

4.2.9 Outside addendum

$$h_{e2} = \frac{1}{2} \cdot (d_{e2} - d_{a2}) \quad (32)$$

Generally: $0,4 \leq \frac{h_{e2}}{m_{x1}} \leq 1,5$ Normally: $h_{e2}/m_{x1} = 0,5$

4.2.10 Root diameter

$$d_{f2} = d_{m2} - 2h_{fm2} \quad (33)$$

4.2.11 Tip diameter

$$d_{a2} = d_{m2} + 2h_{am2} \quad (34)$$

4.2.12 Outside diameter

$$d_{e2} = d_{a2} + 2 \cdot h_{e2} \quad (35)$$

4.2.13 Worm wheel face width

$$b_{2H \max} = \sqrt{(2 \cdot a - d_{f2})^2 - (2 \cdot a - d_{e2})^2} \quad (36)$$

See Figure 4.

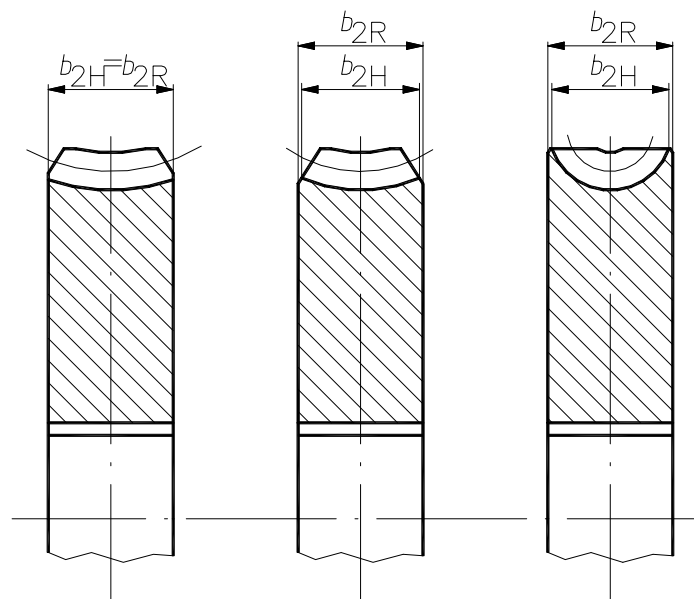


Figure 4 — Worm wheel face width

4.2.14 Throat radius

$$r_{g2} \geq a - \frac{d_{a2}}{2} \quad (37)$$

4.3 Meshing parameters

4.3.1 Centre distance

$$a = 0,5 \cdot (d_{m1} + d_{m2}) = 0,5 \cdot (d_{w1} + d_{w2}) \quad (38)$$

or

$$a = m_{x1} \cdot [0,5 \cdot (q_1 + z_2) + x_2] \quad (39)$$

See Figure 5.

4.3.2 Pitch diameter for worm wheel

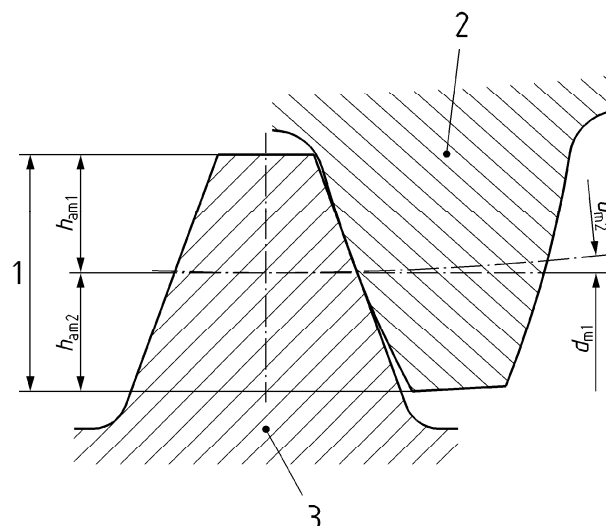
$$d_{w2} = z_2 \cdot m_{x1} \quad (40)$$

4.3.3 Pitch diameter for worm

$$d_{w1} = 2 \cdot a - d_{w2} \quad (41)$$

4.3.4 Worm gear ratio

$$u = \frac{z_2}{z_1} \quad (42)$$



Key

- 1 working depth
- 2 wheel
- 3 worm

Figure 5 — Reference diameters for worm gears ($h_{am1} = h_{am2}$)

4.3.5 Contact ratio

The calculation of the contact ratio is out of the scope of this standard. The definition of contact ratio is according to ISO 1122-2.

4.3.6 Relation to cutting tool parameters

See Annex G

5 General

The equations used for the calculation procedure in this Technical Report 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.

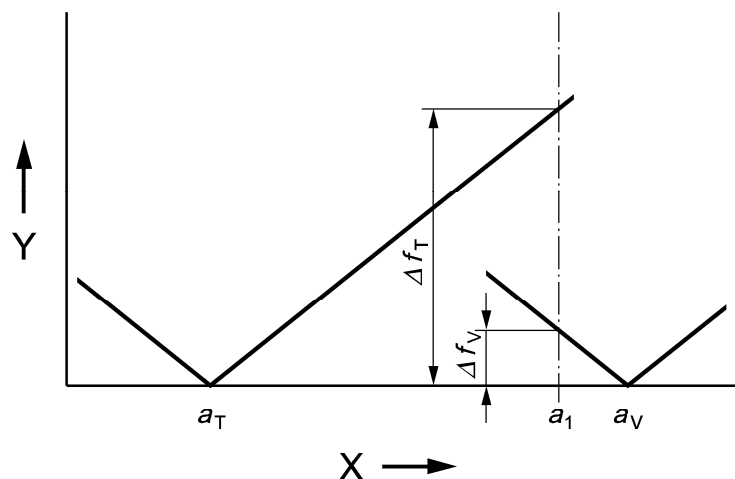
Relative parameters: The calculation offers the possibility to use investigation results in the corresponding calculation process directly. This enables the calculation procedure of the specific results to be adapted.

As the gear to be calculated concerning dimensions, materials, lubricants and operating conditions approach those for the standard reference gear or, if the corresponding test gear data is available, the deviation is decreased. Figure 6 shows an example of influence of centre distance.

The gear concerned 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 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 operating or test experience in which each operating condition (tooth form, material, lubricant, rotational speed, loading, etc.) are as similar as possible to those of the gear in question. In calculating the load capacity or in the calculation of various factors more methods are allowed (see 5.1).

The utilisation 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 must be increased accordingly (see Annex H).

For calculation example, see Annex J.



Key

- X centre distance
- Y relative deviation
- a_1 centre distance of the concerning gear
- 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 relative deviation between a quantity of the concerning gear and a reference gear
- Δf_T here used as relative deviation between the centre distance of the concerning gear and the standard reference gear
- Δf_V here used as relative deviation between the centre distance of the concerning gear and a gear operating or test experiences are available

Figure 6 — Deviation as a function of the centre (based on linear regression)

5.1 Applicability

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

- on knowledge and judgement acquired over years of experience in designing, manufacturing and operating worm gearing.
- on results of bench testing I form worm gears having centre distances from 65 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)

5.2 Validity

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

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 5000 r/min
- **gear ratio** from 5 to 100
- **sliding velocity** between tooth surfaces up to 25 m/s for wear not below 0,1 m/s
- **shaft angle** of 90°
- **accuracy grade:** the worm accuracy grade (according to DIN 3974) is assumed to be one accuracy grade better than wheel
- **worm materials:**
 - case hardened steels, case hardened (HRC = 58 ... 62);
 - through hardened steels, flame or induction hardened (HRC = 50...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 are as listed in Table 3.

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 6743-6:1990;
 - compounded oils (steam cylinder oils), no test results available, included as a mineral oil in this standard;
 - 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 ¹⁾³⁾	GZ-CuSn12Ni2 ¹⁾	GC-CuSn12Ni2 ¹⁾	GZ-CuAl10Ni ^{1) 2)}	GGG-40	GG-25
Wear	+	+	+	o	o	o
Pitting	+	+	+	o	-	-
Tooth Breakage	+	+	-	+	+	+
Temperature	+	+	+	+	-	-
+: covered study available o: known study -: empirical values ¹⁾ 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 minimum of 50 grains are needed to be observed on the area of active flanks. ²⁾ Forged aluminium bronze can be treated like GZ-CuAl10Ni ³⁾ 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.						

5.3 System considerations

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

5.4 Calculation methods A, B, C

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

- Factors which are concerned with meshing geometry or compatibility are calculated using given equations.
- 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.

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 it is recommended that the method used is agreed upon by manufacturer and purchaser.

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 must be known.

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 should be within the range of the given assumptions.

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 should be within the range of the given assumptions.

5.4.1 Notes on numerical equations

The equations specified in this Technical Report must be calculated in the specified units (see Table 1).

5.4.2 Base conditions, interaction

— Wear:

This procedure is in accordance with the investigation described in Bibliography [24] and is based on practical experience.

— Pitting damage:

The procedure follows the investigation described in Bibliography [31] 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. According to present day theory however, these need not be considered. For the above reasons 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 estimations of wear life under this condition cannot be considered at the moment.

— 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.

— 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.

5.4.3 Other notes

Proof of load capacities are provided by continual endurance operation. With implementation procedures, intermittent operation and alternating loading etc. the experience of the gear manufacturer must be considered.

5.5 Standard reference gear

With certain calculation procedures the absolute calculating equations are similar to the relative calculating equations. If the adopted sizes of the corresponding standard reference gear (subscript T) must be employed, the relative calculating equations can be transferred to the absolute calculating equations (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 (eq. (53))
Parameter for mean lubricant film thickness h_T^*	0,07 (eq. (55))
Parameter for mean sliding path s_T^*	30,8 (eq. (59))
Worm material	16MnCr5 case hardened
Wheel material	GZ-CuSn12Ni2
Lubricant density	1,048 kg/dm ³
Flank form	I
Normal pressure angle	20°
Worm surface roughness Ra_1	0,5 µm
Equivalent modulus of elasticity E_{red}	150 622 N/mm ²

6 Geometrical data to be known for calculation

6.1 Input variables

For the calculation the following variables have to be known:

Geometry data:

- Center distance, a
- Face width, b_{2H} , see Figure 7
- Wheel rim width, b_{2R} , see Figure 7
- Reference diameter, d_{m1} , d_{m2}
- Axial module of the worm, m_{x1}
- Number of teeth, z_1 , z_2
- Addendum modification factor, x_2
- Pressure angle α_n
- Profile flank form, (A, N, K, I, C)
- Outside diameter, d_{e2}
- Rim thickness, s_k , see Figure 7
- Worm tip diameter, d_{a1}
- Worm tooth thickness in axial section divided by axial pitch, s_{mx1}/p_{x1}

Loading:

- Nominal output torque, T_2
- Application factor, K_A
- Rotational speed of worm, n_1
- Life time, L_h
- Number of start per hour, N_S