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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.

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 2.

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is entirely informative in nature and does not have to be reviewed until the data it provides are considered to be no longer valid or useful.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-geared 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
$b_{2\text{H}}$	effective wheel facewidth	mm	Fig. 4	
$b_{2\text{H,std}}$	Standard worm wheel facewidth	mm		52
$b_{2\text{R}}$	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
$d_{\text{a }1}$	worm tip diameter	mm		13
$d_{\text{a }2}$	worm wheel tip diameter	mm		34
$d_{\text{b }1}$	base diameter of involute helicoid (for I profile)	mm		21
$d_{\text{e }2}$	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
$d_{\text{f }1}$	worm root diameter	mm		14
$d_{\text{f }2}$	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
$e_{\text{mx }1}$	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
$h_{\mathrm{min\ m}}$	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	$\mathrm{W/(m^2 \cdot K)}$		
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
$m_{\mathrm{x\,1}}$	axial module	mm		2/G.1
Δm	material loss	mg		
Δm_{lim}	material loss limit	mg		
n_1	rotational speed of the worm shaft	$\mathrm{min^{-1}}$		
n_2	rotational speed of the wheel	$\mathrm{min^{-1}}$		
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
$p_{x\,1}$	axial pitch	mm	Fig. 2	1
$p_{z\,1}$	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
$\vec{v}_\mathrm{g}$	sliding velocity at mean reference diameter	m/s		51
v_Σ	sum velocity	m/s		53
$v_{\Sigma\mathrm{n}}$	sum velocity in normal direction	m/s		53
x_2	worm wheel profile shift coefficient	-		28
$x_{2\mathrm{max}}$	maximum worm wheel profile shift coefficient for tooling selection	-		H.3
$x_{2\mathrm{min}}$	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	$\mathrm{N/mm}^2$		
E_2	modulus of elasticity of the worm wheel	$\mathrm{N/mm}^2$		
E_red	equivalent modulus of elasticity	$\mathrm{N/mm}^2$		62
E_steel	modulus of elasticity for steel	$\mathrm{N/mm}^2$		62
$F_{\mathrm{xm}1}$	axial force to the worm shaft	N		46/49
$F_{\mathrm{xm}2}$	axial force to the worm wheel	N		45/48
$F_{\mathrm{rm}1}$	radial force to the worm shaft	N		47
$F_{\mathrm{rm}2}$	radial force to the worm wheel	N		53
$F_{\mathrm{tm}1}$	circumferencial or tangential force to the worm shaft	N		45/48
$F_{\mathrm{tm}2}$	circumferencial or tangential force to the worm wheel	N		46/49
$\mathrm{d}F/\mathrm{d}b$	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
$K_{\mathrm{H}\alpha}$	transverse load distribution factor	-		
$K_{\mathrm{H}\beta}$	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_ν	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
$P_\mathrm{Vz1\text{-}2}$	meshing power loss in reducer	W		104
$P_\mathrm{Vz2\text{-}1}$	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
$S_\mathrm{F\,min}$	minimum tooth breakage safety factor	-		149
S_H	pitting safety factor	-		133
S_T	temperature safety factor	-		157/167
$S_\mathrm{T\,min}$	minimum temperature safety factor	-		158/168
S_W	wear safety factor	-		107
$S_\mathrm{W\,min}$	minimum wear safety factor	-		108
S_δ	deflection safety factor	-		143
$S_{\delta\,\mathrm{lim}}$	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	$W/(m^2K)$		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
$\delta_{\mathrm{W\ lim}}$	limiting value of flank loss	mm		132
$\delta_{\mathrm{W\ lim\ n}}$	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
$\eta_{\mathrm{z1\text{-}2}}$	gear efficiency in reducer	-		88
$\eta_{\mathrm{z2\text{-}1}}$	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		
$\Delta\theta_{\mathrm{oil}}$	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
$\theta_{\mathrm{S\ lim}}$	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		
$\Delta_{s\ \mathrm{lim}}$	allowable tooth thickness loss	mm		129
$\sigma_{\mathrm{H\ lim\ T}}$	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
$\tau_{\mathrm{F\ lim\ T}}$	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.

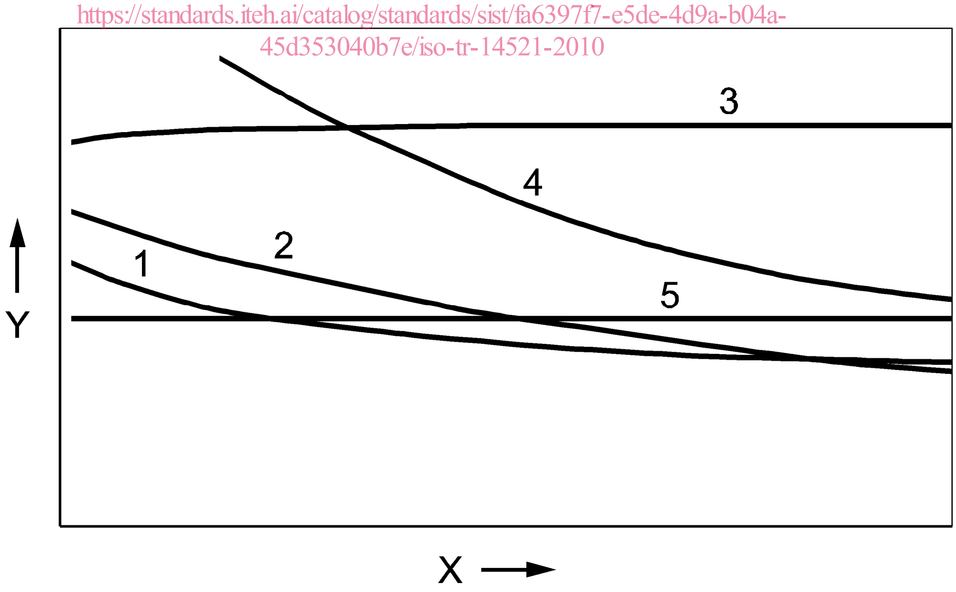
The different rating criteria are calculated independently and not in combination (see Figure 1). For a given worm gear pair, the zone of contact could change with loading. At a steady load, fatigue pits can develop which may subsequently be reduced by wear. This can be followed by further pitting, additional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