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Izraļ un nosilnosti glede na toplotno razjedanje zobnih bokov valjastih, stoģļ astih in hipoidnih zobnikov - 2. del: Metoda povpreļ ne temperature

Calculation of scuffing load capacity of cylindrical, bevel and hypoid gears -- Part 2: Integral temperature method

Calcul de la capacité de charge au grippage des engrenages cylindriques, coniques et hypoïdes -- Partie 2: Méthode de la température intégrale

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TECHNICAL REPORT

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Calculation of scuffing load capacity of cylindrical, bevel and hypoid gears —

Part 2:
Integral temperature method

Calcul de la capacité de charge au grippage des engrenages cylindriques, coniques et hypoïdes —

Partie 2: Méthode de la température intégrale

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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 main task of technical committees is to prepare International Standards, but in exceptional circumstances a technical committee may propose the publication of a Technical Report of one of the following types:

— type 1, when the required support cannot be obtained for the publication of an International Standard, despite repeated efforts;

— type 2, when the subject is still under technical development or where for any other reason there is the future but not immediate possibility of an agreement on an International Standard;

— type 3, 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).

Technical Reports of types 1 and 2 are subject to review within three years of publication, to decide whether they can be transformed into International Standards. Technical Reports of type 3 do not necessarily have to be reviewed until the data they provide are considered to be no longer valid or useful.

Technical Reports are drafted in accordance with the rules given in the ISO/IEC Directives, Part 3.

Attention is drawn to the possibility that some of the elements of this part of ISO/TR 13989 may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights.

ISO/TR 13989-2, which is a Technical Report of type 2, was prepared by Technical Committee ISO/TC 60, *Gears*, Subcommittee SC 2, *Gear capacity calculation*.

This document is being issued in the Technical Report (type 2) series of publications (according to subclause G.3.2.2 of Part 1 of the ISO/IEC Directives, 1995) as a “prospective standard for provisional application” in the field of scuffing load capacity of gears because there is an urgent need for guidance on how standards in this field should be used to meet an identified need. In 1975, two methods to evaluate the risk of scuffing were documented to be studied by ISO/TC 60. It was agreed that after a period of experience one method shall be selected. Since the subject is still under technical development and there is a future possibility of an agreement on an International Standard, the publication of a type 2 Technical Report was proposed.

This document is not to be regarded as an “International Standard”. It is proposed for provisional application so that information and experience of its use in practice may be gathered. Comments on the content of this document should be sent to the ISO Central Secretariat.

A review of this Technical Report (type 2) will be carried out not later than three years after its publication with the options of: extension for another three years; conversion into an International Standard; or withdrawal.

ISO/TR 13989 consists of the following parts, under the general title *Calculation of scuffing load capacity of cylindrical, bevel and hypoid gears*:

— *Part 1: Flash temperature method*

— *Part 2: Integral temperature method*

Annexes A and B of this part of ISO/TR 13989 are for information only.

Introduction

This part of ISO/TR 13989 describes the surface damage "warm scuffing" for cylindrical (spur and helical), bevel and hypoid gears for generally used gear materials and different heat treatments. "Warm scuffing" is characterized by typical scuffing and scoring marks, which can lead to increasing power loss, dynamic load, noise and wear. For "cold scuffing", in general associated with low temperature and low speed, under approximately 4 m/s, and through-hardened, heavily loaded gears, the equations are not suitable.

There is a particularly severe form of gear tooth surface damage in which seizure or welding together of areas of tooth surfaces occurs, due to absence or breakdown of a lubricant film between the contacting tooth flanks of mating gears, caused by high temperature and high pressure. This form of damage is termed "scuffing" and most relevant when surface velocities are high. Scuffing may also occur for relatively low sliding velocities when tooth surface pressures are high enough, either generally or, because of uneven surface geometry and loading, in discrete areas.

Risk of scuffing damage varies with the properties of gear materials, the lubricant used, the surface roughness of tooth flanks, the sliding velocities and the load. Excessive aeration or the presence of contaminants in the lubricant such as metal particles in suspension, also increase the risk of scuffing damage. Consequences of the scuffing of high speed gears include a tendency to high levels of dynamic loading due to increase of vibration, which usually leads to further damage by scuffing, pitting or tooth breakage.

High surface temperatures due to high surface pressures and sliding velocities can initiate the breakdown of lubricant films. On the basis of this hypothesis two approaches to relate temperature to lubricant film breakdown are presented:

— the flash temperature method (presented in ISO/TR 13989-1), based on contact temperatures which vary along the path of contact;

— the integral temperature method (presented in this part of ISO/TR 13989), based on the weighted average of the contact temperatures along the path of contact.

The integral temperature method is based on the assumption that scuffing is likely to occur when the mean value of the contact temperature (integral temperature) is equal to or exceeds a corresponding critical value. The risk of scuffing of an actual gear unit can be predicted by comparing the integral temperature with the critical value, derived from a gear test for scuffing resistance of lubricants. The calculation method takes account of all significant influence parameters, i.e. the lubricant (mineral oil with and without EP-additives, synthetic oils), the surface roughness, the sliding velocities, the load, etc.

In order to ensure that all types of scuffing and comparable forms of surface damage due to the complex relationships between hydrodynamical, thermodynamical and chemical phenomena are dealt with, further methods of assessment may be necessary. The development of such methods is the objective of ongoing research.

Calculation of scuffing load capacity of cylindrical, bevel and hypoid gears —

Part 2: Integral temperature method

1 Scope

This part of ISO/TR 13989 specifies the integral temperature method for calculating the scuffing load capacity of cylindrical, bevel and hypoid gears.

2 Normative references

The following normative documents contain provisions which, through reference in this text, constitute provisions of this part of ISO/TR 13989. For dated references, subsequent amendments to, or revisions of, any of these publications do not apply. However, parties to agreements based on this part of ISO/TR 13989 are encouraged to investigate the possibility of applying the most recent editions of the normative documents indicated below. For undated references, the latest edition of the normative document referred to applies. Members of ISO and IEC maintain registers of currently valid International Standards.

ISO 53:1998, *Cylindrical gears for general and heavy engineering — Standard basic rack tooth profile*.

ISO 1122-1:1998, *Vocabulary of gear terms — Part 1: Definitions related to geometry*.

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

ISO 6336-1:1996, *Calculation of load capacity of spur and helical gears — Part 1: Basic principles, introduction and general influence factors*.

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

3 Terms, definitions, symbols and units

3.1 Terms and definitions

For the purposes of this part of ISO/TR 13989, the terms and definitions given in ISO 1122-1 apply.

3.2 Symbols and units

The symbols used in this part of ISO/TR 13989 are given in Table 1.

1) To be published.

Table 1 — Symbols and units

Symbol	Description	Unit	Reference
a	centre distance	mm	—
a_v	virtual centre distance of virtual cylindrical gear	mm	ISO 10300-1
b	face width, smaller value of pinion or wheel	mm	—
b_{eB}	effective facewidth for scuffing	mm	Eq. (46)
c_v	specific heat capacity per unit volume	$N/(mm^2 \cdot K)$	—
c'	single stiffness	N/(mm·µm)	ISO 6336-1
c_γ	mesh stiffness	N/(mm·µm)	ISO 6336-1
d	reference circle diameter	mm	—
d_{Na}	effective tip diameter	mm	—
d_a	tip diameter	mm	Eq. (69)
d_b	base diameter	mm	Eq. (70)
d_m	diameter at mid-facewidth	mm	—
d_s	reference circle of virtual crossed axes helical gear	mm	Eq. (68)
d_v	reference diameter of virtual cylindrical gear	mm	ISO 10300-1
d_{va}	tip diameter of virtual cylindrical gear	mm	ISO 10300-1
d_{vb}	base diameter of virtual cylindrical gear	mm	ISO 10300-1
$g_{an1,2}$	recess path of contact of pinion, wheel	mm	Eqs. (90), (91)
$g_{fn1,2}$	approach path of contact of pinion, wheel	mm	Eqs. (90), (91)
g^*	sliding factor	—	Eq. (62)
h_{am}	addendum at mid-facewidth of hypoid gear	mm	—
m	module	mm	—
m_{mn}	normal module of hypoid gear at mid-facewidth	mm	—
m_{sn}	normal module of virtual crossed axes helical gear	mm	Eq. (73)
n_p	number of meshing gears	—	—
p_{en}	normal base pitch	mm	Eq. (74)
u	gear ratio	—	—
u_v	gear ratio of virtual cylindrical gear	—	ISO 10300-1
v	reference line velocity	m/s	—
$v_{t1,2}$	tangential velocity of pinion, wheel of hypoid gear	m/s	Eqs. (77), (78)
$v_{g\gamma1}$	maximum sliding velocity at tip of pinion	m/s	Eq. (83)
v_{gs}	sliding velocity at pitch point	m/s	Eq. (82)
$v_{g1,2}$	sliding velocity	m/s	Eqs. (84), (85)
$v_{g\alpha1}$	sliding velocity	m/s	Eq. (87)

Table 1 *(continued)*

Symbol	Description	Unit	Reference
$v_{g\beta 1}$	sliding velocity	m/s	Eq. (88)
v_{mt}	tangential speed at reference cone at mid-facewidth of bevel gear	m/s	—
$v_{\Sigma C}$	sums of tangential speeds at pitch point	m/s	Eqs. (2), (47), (81)
$v_{\Sigma s}$	tangential speed	m/s	Eq. (79)
$v_{\Sigma h}$	tangential speed	m/s	Eq. (80)
w_{Bt}	specific tooth load, scuffing	N/mm	Eq. (4)
z	number of teeth	—	—
z_v	number of teeth of virtual cylindrical gear	—	ISO 10300-1
B_M	thermal contact coefficient	$N/(mm \cdot s^{1/2} \cdot K)$	Eq. (12)
C_1, C_2, C_{2H}	weighting factors	—	—
C_a	nominal tip relief	µm	—
C_{eff}	effective tip relief	µm	Eqs. (37), (38), (49)
E	module of elasticity (Young's modulus)	N/mm^2	—
F_{mt}	nominal tangential load at reference cone at mid-facewidth	N	—
F_n	normal tooth load	N	Eq. (51)
F_t	nominal tangential load at reference circle	N	—
K_A	application factor	—	ISO 6336-1, ISO 10300-1
K_v	dynamic factor	—	ISO 6336-1, ISO 10300-1
$K_{B\alpha}$	$= K_{H\alpha}$ transverse load factor (scuffing)	—	6.2.4, ISO 6336-1, ISO 10300-1
$K_{B\beta}$	$= K_{H\beta}$ face load factor (scuffing)	—	ISO 6336-1 ISO 10300-1, 6.2.4, Eqs. (52), (53)
$K_{B\gamma}$	helical load factor (scuffing)	—	Eq. (5), 6.2.4, 6.3.5
$K_{B\beta be}$	bearing factor	—	6.3.3
$K_{H\alpha}$	transverse load factor	—	ISO 6336-1, ISO 10300-1
$K_{H\beta}$	face load factor	—	ISO 6336-1, ISO 10300-1
$K_{H\beta be}$	bearing factor	—	ISO 10300-1
L	contact parameter	—	Eq. (55)
Ra	arithmetic mean roughness	µm	Eq. (6)
S_{intS}	scuffing safety factor	—	Eq. (14)
S_{Smin}	minimum required scuffing safety factor	—	—

Table 1 *(continued)*

Symbol	Description	Unit	Reference
T_1	torque of the pinion	Nm	—
T_{1T}	scuffing torque of test pinion	Nm	Eq. (96)
X_{BE}	geometry factor at pinion tooth tip	—	Eq. (22)
X_E	run-in factor	—	Eq. (8)
X_{Ca}	tip relief factor	—	Eq. (32)
X_G	geometry factor of hypoid gears	—	Eq. (54)
X_L	lubricant factor	—	5.1
X_M	thermal flash factor	—	Eq. (9)
X_Q	approach factor	—	Eqs. (25), (26), (27)
X_R	roughness factor	—	Eq. (7)
X_S	lubrication factor	—	6.1.5.3
X_W	welding factor of executed gear	—	Table 3
X_{WT}	welding factor of test gear	—	6.4.2
X_{WrelT}	relative welding factor	—	Eq. (102)
X_{mp}	contact factor	—	Eq. (21)
$X_{\alpha\beta}$	pressure angle factor	—	Eqs. (13), (48)
X_ε	contact ratio factor	—	Eqs. (39) to (44)
α	pressure angle	°	—
α_{mn}	normal pressure angle at mid-facewidth of hypoid gear	°	—
α_n	normal pressure angle	°	—
α_{sn}	normal pressure angle of crossed axes helical gear	°	Eq. (64)
α_{st}	transverse pressure angle of crossed axes helical gear	°	Eq. (66)
α_t	transverse pressure angle	°	—
α_t'	transverse working pressure angle	°	—
α_{vt}	transverse pressure angle of virtual cylindrical gear	°	ISO 10300-1
α_y	arbitrary angle	°	Figure 2
β	helix angle	°	—
β_b	helix angle at base circle	°	Eqs. (67), (71)
β_m	helix angle at reference cone at mid-facewidth of hypoid gear	°	—
β_s	helix angle of virtual crossed axes helical gear	°	Eq. (63)
γ	auxiliary angle	°	Eq. (86)
δ	reference cone angle	°	—

Table 1 *(continued)*

Symbol	Description	Unit	Reference
ε_a	recess contact ratio	—	Eqs. (28), (29)
ε_f	approach contact ratio	—	Eqs. (28), (29)
ε_n	contact ratio in normal section of virtual crossed axes helical gear	—	Eqs. (92), (93)
ε_1	addendum contact ratio of the pinion	—	Eq. (30)
ε_2	addendum contact ratio of the wheel	—	Eq. (31)
ε_α	contact ratio	—	Eq. (45)
$\varepsilon_{v\alpha}$	transverse contact ratio of virtual cylindrical gear	—	ISO 10300-1
ε_{v1}	tip contact ratio of virtual cylindrical pinion	—	ISO 10300-1
ε_{v2}	tip contact ratio of virtual cylindrical wheel	—	ISO 10300-1
ξ	Hertzian auxiliary coefficient	—	Figure 7, Eqs. (57), (59)
μ_{mC}	mean coefficient of friction	—	Eqs. (1), (1a)
η_{oil}	dynamic viscosity at oil temperature	mPa·s	—
λ_M	heat conductivity	N/(s·K)	—
ν	Poisson's ratio	—	—
ν_{40}	kinematic viscosity of the oil at 40 °C	mm^2/s; cSt	—
$\rho_{E1,2}$	radius of curvature at tip of the pinion, wheel	mm	Eqs. (23), (24)
ρ_{Cn}	relative radius of curvature at pitch point in normal section	mm	Eq. (76)
$\rho_{n1,2}$	radius of curvature at pitch point in normal section	mm	Eq. (75)
ρ_{redC}	relative radius of curvature at pitch point	mm	Eq. (3)
η	Hertzian auxiliary coefficient	—	Figure 7, Eqs. (58), (60)
ϑ	Hertzian auxiliary angle	°	Eqs. (56) to (60)
ϑ_{flaE}	flash temperature at pinion tooth tip when load sharing is neglected	K	Eq. (19)
ϑ_{flaint}	mean flash temperature	K	Eq. (18)
$\vartheta_{flainth}$	mean flash temperature of hypoid gear	K	Eq. (50)
ϑ_{int}	integral temperature	K	Eq. (17)
ϑ_{intP}	permissible integral temperature	K	Eq. (16)
ϑ_{intS}	scuffing integral temperature (allowable integral temperature)	K	Eq. (94)
$\vartheta_{flaintT}$	mean flash temperature of the test gear	K	Eqs. (96), (99), (101)
ϑ_{oil}	oil sump or spray temperature	°C	—
$\vartheta_{M\text{-}C}$	bulk temperature	°C	Eq. (20)

Table 1 *(concluded)*

Symbol	Description	Unit	Reference
ϑ_{MT}	test bulk temperature	°C	Eqs. (95), (98), (100)
φ	axle angle of virtual crossed axes helical gear	°	Eq. (72)
Σ	axle angle of virtual crossed axes helical gear	°	Eq. (65)
Φ_E	run-in grade	—	5.2
Γ	parameter on the line of action	—	Eq. (10)

Subscripts:

1 pinion

2 wheel

a tip diameter of the virtual gear

b base circle of the virtual gear

m mid-facewidth of bevel or hypoid gears

n normal section

s virtual crossed axes helical gear

t tangential direction

T test gear

4 Field of application

The calculation methods are based on results of the rig testing of gears run at pitch line velocities less than 80 m/s. The equations can be used for gears which run at higher speeds, but with increasing uncertainty as speed increases. The uncertainty concerns the estimation of bulk temperature, coefficient of friction, allowable temperatures, etc. as speeds exceed the range with experimental background.

4.1 Scuffing damage

When once initiated, scuffing damage can lead to gross degradation of tooth flank surfaces, with increase of: power loss, dynamic loading, noise and wear. It can also lead to tooth breakage if the severity of the operating conditions is not reduced. In the event of scuffing due to an instantaneous overload, followed immediately by a reduction of load, e.g. by load redistribution, the tooth flanks may self-heal by smoothing themselves to some extent. Even so, the residual damage will continue to be a cause of increased power loss, dynamic loading and noise.

In most cases, the resistance of gears to scuffing can be improved by using a lubricant with enhanced E.P. (extreme pressure) properties. It is important however, to be aware that some disadvantages attend the use of E.P. oils — corrosion of copper, embrittlement of elastomers, lack of world-wide availability, etc. These disadvantages are to be taken into consideration if optimum lubricant choice is to be made, which means: as few additives as possible, but as many as necessary.

Due to continuous variation of different parameters, the complexity of the chemical properties and the thermo-hydro-elastic processes in the instantaneous contact area, some scatter in the calculated assessments of probability of scuffing risk is to be expected.

In contrast to the relatively long time of development of fatigue damage, one single momentary overload can initiate scuffing damage of such severity that affected gears may no longer be used. This should be carefully considered when choosing an adequate safety factor for gears, especially for gears required to operate at high circumferential velocities.

4.2 Integral temperature criterion

This approach to the evaluation of the probability of scuffing is based on the assumption that scuffing is likely to occur when the mean value of the contact temperatures along the path of contact is equal to or exceeds a corresponding "critical value" . In the method presented herein, the sum of the bulk temperature and the weighted mean of the integrated values of flash temperatures along the path of contact is the "integral temperature". The bulk temperature is estimated as described under 6.1.5 and the mean value of the flash temperature is approximated by substituting mean values of the coefficient of friction, the dynamic loading, etc., along the path of contact. A weighting factor is introduced accounting for possible different influences of a real bulk temperature value and a mathematically integrated mean flash temperature value on the scuffing phenomenon.

The probability of scuffing is assessed by comparing the integral temperature with a corresponding critical value derived from the gear testing of lubricants for scuffing resistance (e.g. different FZG test procedures, the IAE and the Ryder gear tests), or from gears which have scuffed in service.

5 Influence factors

5.1 Mean coefficient of friction μ_{mC}

The **actual** coefficient of friction between the tooth flanks is an instantaneous and local value which depends on several properties of the oil, surface roughness, lay of the surface irregularities such as those left by machining, properties of the tooth flank materials, tangential velocities, forces at the surfaces and the dimensions. Assessment of the instantaneous coefficient of friction is difficult since there is no method currently available for its measurement.

The **mean** value for the coefficient of friction μ_{mC} along the path of contact was derived from measurements [1] and approximated by Equation (1). Although the local coefficient of friction is near to zero in the pitch point C, the mean value can be approximated with the parameters at the pitch point and the oil viscosity η_{oil} at oil temperature ϑ_{oil} when introduced into Equation (1).