TECHNICAL REPORT

ISO/TR 13989-1

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2000-03-15

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

Part 1: Flash temperature method

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

Partie 1: Méthode de la température-éclair

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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 Technical Reports of type 3 do not necessarily have to be reviewed until the data they provide are considered to be no longer valid or useful.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 13989-1, 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 13989 are for information only.

Introduction

Since 1990 the flash temperature method, presented in this part of ISO/TR 13989, was enriched with research for short exposure times, consideration of transition diagrams, new approximations for the coefficient of friction, and completely renewed load sharing factors. In 1991 Prof. Blok contributed an extension of the flash temperature formula which made it directly applicable to hypoid gears.

The integral temperature, presented in ISO/TR 13989-2, averages the flash temperature and supplements empirical influence factors to the hidden load sharing factor. The resulting value approximates the maximum contact temperature, thus yielding about the same assessment of scuffing risk as the flash temperature method of this part of ISO/TR 13989. The integral temperature method is less sensitive for those cases where there are local temperature peaks, usually in gearsets that have low contact ratio or contact near the base circle or other sensitive geometries.

The 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. 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. According to Blok [12][13][14][15][16][17], high contact temperatures of lubricant and tooth surfaces at the instantaneous contact position may effect a break-down of the lubricant film at the contact interface.

The interfacial contact temperature is conceived as the sum of two components:

— the interfacial bulk temperature of the moving interface, which, if varying, does so only comparatively slowly. For evaluating this component, it may be suitably averaged from the two overall bulk temperatures of the two rubbing teeth. The latter two bulk temperatures follow from the thermal network theory [18].

— the rapidly fluctuating flash temperature of the moving faces in contact. Special attention has to be paid to the coefficient of friction. A common practice is the use of a coefficient of friction valid for regular working conditions, although it may be stated that at incipient scuffing the coefficient of friction has significantly higher values.

The complex relationship between mechanical, hydrodynamical, thermodynamical and chemical phenomena was the objective of extensive research and experiments, which may induce various empirical influence factors. A direct suppletion of empirical influence factors may enforce the related functional factors in the main formula to be fixated to average values. However, correct treatment of functional factors (e.g. coefficient of friction, load sharing factor, thermal contact coefficient) keeps the main formula intact, in confirmation with the experiments and practice.

Next to the maximum contact temperature, the progress of the contact temperature along the path of contact provides necessary information to the gear design.

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

Part 1:
Flash temperature method

1 Scope

This part of ISO/TR 13989 specifies methods and formulae for evaluating the risk of scuffing, based on Blok's contact temperature concept.

The fundamental concept according to Blok is applicable to all machine elements with moving contact zones. The flash temperature formulae are valid for a band-shaped or approximately band-shaped Hertzian contact zone and working conditions characterized by sufficiently high Péclet numbers.

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

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*.

ISO 10825:1995, *Gears — Wear and damage to gear teeth — Terminology*.

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 and ISO 10825 apply.

3.2 Symbols and units

The symbols used in this part of ISO/TR 13989 are given in Table 1. The units of length metre, millimetre and micrometre are chosen in accordance with common practice. To achieve a "coherent" system, the units for B_M, c_γ, X_M are adapted to the mixed application of metre and millimetre or millimetre and micrometre.

1) To be published.

Table 1 — Symbols and units

Symbol	Description	Unit	Reference
a	centre distance	mm	Eq. (A.5)
b	facewidth, smaller value for pinion or wheel [a]	mm	Eq. (11)
b_{eff}	effective facewidth	mm	Eq. (12)
b_H	semi-width of Hertzian contact band	mm	Eq. (3)
B_M	thermal contact coefficient	$N/(mm^{½} \cdot m^{½} \cdot s^{½} \cdot K)$	Eq (A.13)
B_{M1}	thermal contact coefficient of pinion	$N/(mm^{½} \cdot m^{½} \cdot s^{½} \cdot K)$	Eq. (3)
B_{M2}	thermal contact coefficient of wheel	$N/(mm^{½} \cdot m^{½} \cdot s^{½} \cdot K)$	Eq. (3)
C_{a1}	tip relief of pinion	µm	Eq. (48)
C_{a2}	tip relief of wheel	µm	Eq. (46)
C_{eff}	optimal tip relief	µm	Eq. (46)
C_{eq1}	equivalent tip relief of pinion	µm	Eq. (B.2)
C_{eq2}	equivalent tip relief of wheel	µm	Eq. (B.3)
C_{f1}	root relief of pinion	µm	Eq. (B.3)
C_{f2}	root relief of wheel	µm	Eq. (B.2)
c_{M1}	specific heat per unit mass of pinion	J/(kg·K)	Eq. (9)
c_{M2}	specific heat per unit mass of wheel	J/(kg·K)	Eq. (10)
c_γ	mesh stiffness	N/(mm·µm)	Eq. (B.1)
d_1	reference diameter of pinion	mm	Eq. (34)
d_2	reference diameter of wheel	mm	Eq. (35)
d_{a1}	tip diameter of pinion	mm	Eq. (34)
d_{a2}	tip diameter of wheel	mm	Eq. (35)
E_1	modulus of elasticity of pinion	N/mm^2	Eq. (A.10)
E_2	modulus of elasticity of wheel	N/mm^2	Eq. (A.10)
E_r	reduced modulus of elasticity	N/mm^2	Eq. (A.9)
F_{ex}	external axial force	N	Eq. (18)
F_n	normal load in wear test	N	Fig. 1
F_t	nominal tangential force	N	Eq. (11)
H_1	auxiliary dimension	mm	Eq. (B.3)
H_2	auxiliary dimension	mm	Eq. (B.2)
h_{am1}	tip height in mean cone of pinion	mm	Eq. (43)
h_{am2}	tip height in mean cone of wheel	mm	Eq. (44)

Table 1 — Symbols and units *(continued)*

Symbol	Description	Unit	Reference
K_A	application factor	—	Eq. (11)
$K_{B\alpha}$	transverse load factor (scuffing)	—	Eq. (11)
$K_{B\beta}$	face load factor (scuffing)	—	Eq. (11)
$K_{H\alpha}$	transverse load factor (contact stress)	—	Eq. (15)
$K_{H\beta}$	face load factor (contact stress)	—	Eq. (14)
K_{mp}	multiple path factor	—	Eq. (11)
K_v	dynamic factor	—	Eq. (11)
m_n	normal module	mm	Eq. (B.2)
n_1	revolutions per minute of pinion	r/min	Eq. (5)
n_p	number of mesh contacts	—	Eq. (16)
Pe_1	Péclet number of pinion material	—	Eq. (9)
Pe_2	Péclet number of wheel material	—	Eq. (10)
Q	quality grade	—	Eq. (57)
Ra_1	tooth flank surface roughness of pinion	µm	Eq. (28)
Ra_2	tooth flank surface roughness of wheel	µm	Eq. (28)
R_m	cone distance of mean cone	mm	Eq. (A.16)
r_{m1}	reference radius in mean cone of pinion	mm	Eq. (43)
r_{m2}	reference radius in mean cone of wheel	mm	Eq. (44)
S_B	safety factor for scuffing	—	Eq. (100)
S_{FZG}	load stage (in FZG test)	—	Eq. (99)
t_1	contact exposure time of pinion	µs	Eq. (95)
t_2	contact exposure time of wheel	µs	Eq. (96)
t_c	contact exposure time at bend of curve	µs	Eq. (97)
t_{max}	longest contact exposure time	µs	Eq. (95)
u	gear ratio	—	Eq. (A.6)
u_v	virtual ratio	—	Eq. (B.6)
v_g	sliding velocity	m/s	Fig. 1
v_{g1}	tangential velocity of pinion	m/s	Eq. (3)
v_{g2}	tangential velocity of wheel	m/s	Eq. (3)
$v_{g\Sigma C}$	sum of tangential velocities in pitch point	m/s	Eq. (25)
v_t	pitch line velocity	m/s	Eq. (26)
w_{Bn}	normal unit load	N/mm	Eq. (3)
w_{Bt}	transverse unit load	N/mm	Eq. (5)

Table 1 — Symbols and units *(continued)*

Symbol	Description	Unit	Reference
X_{but}	buttressing factor	—	Eq. (54)
X_{butA}	buttressing value	—	Eq. (51)
X_{butE}	buttressing value	—	Eq. (51)
X_{G}	geometry factor	—	Eq. (A.5)
X_{J}	approach factor	—	Eq. (3)
X_{L}	lubricant factor	—	Eq. (25)
X_{M}	thermo-elastic factor	$\mathrm{K \cdot N^{-3/4} \cdot s^{-1/2} \cdot m^{-1/2} \cdot mm}$	Eq. (5)
X_{mp}	multiple mating pinion factor	—	Eq. (22)
X_{R}	roughness factor	—	Eq. (25)
X_{S}	lubrication system factor	—	Eq. (22)
X_{W}	structural factor	—	Eq. (94)
$X_{\alpha\beta}$	angle factor	—	Eq. (A.6)
X_{Γ}	load sharing factor	—	Eq. (3)
X_{Θ}	gradient of the scuffing temperature	—	Eq. (97)
z_1	number of teeth of pinion	—	Eq. (30)
z_2	number of teeth of wheel	—	Eq. (30)
α_{a1}	transverse tip pressure angle of pinion	°	Eq. (31)
α_{a2}	transverse tip pressure angle of wheel	°	Eq. (30)
α_{t}	transverse pressure angle	°	Eq. (34)
α_{wn}	normal working pressure angle	°	Eq. (A.2)
α_{wt}	transverse working pressure angle	°	Eq. (7)
α_{y1}	pinion pressure angle at arbitrary point	°	Eq. (29)
β	helix angle	°	Eq. (18)
β_{b}	base helix angle	°	Eq. (49)
β_{bm}	base helix angle in midcone	°	Eq. (50)
β_{w}	working helix angle	°	Eq. (A.2)
Γ_{A}	parameter on the line of action at point A	—	Eq. (24)
Γ_{AA}	parameter on the line of action at point AA	—	Eq. (68)
Γ_{AB}	parameter on the line of action at point AB	—	Eq. (66)
Γ_{AU}	parameter on the line of action at point AU	—	Eq. (49)
Γ_{B}	parameter on the line of action at point B	—	Eq. (31)
Γ_{BB}	parameter on the line of action at point BB	—	Eq. (70)
Γ_{D}	parameter on the line of action at point D	—	Eq. (32)
Γ_{DD}	parameter on the line of action at point DD	—	Eq. (72)
Γ_{DE}	parameter on the line of action at point DE	—	Eq. (67)

Table 1 — Symbols and units *(continued)*

Symbol	Description	Unit	Reference
Γ_E	parameter on the line of action at point E	—	Eq. (24)
Γ_{EE}	parameter on the line of action at point EE	—	Eq. (74)
Γ_{EU}	parameter on the line of action at point EU	—	Eq. (49)
Γ_M	parameter on the line of action at point M	—	Eq. (86)
Γ_y	parameter on the line of action at arbitrary point	—	Eq. (7)
γ_1	angle of direction of tangential velocity of pinion	—	Eq. (3)
γ_2	angle of direction of tangential velocity of wheel	—	Eq. (3)
δ_1	pitch cone angle of pinion	°	Eq. (37)
δ_2	pitch cone angle of wheel	°	Eq. (39)
ε_α	transverse contact ratio	—	Eq. (76)
ε_β	overlap ratio	—	Eq. (52)
η_{oil}	absolute (dynamic) viscosity at oil temperature	mPa·s	Eq. (27)
Θ_B	contact temperature	°C	Eq. (1)
Θ_{Bmax}	maximum contact temperature	°C	Eq. (2)
Θ_{fl}	flash temperature	K	Eq. (1)
Θ_{flm}	average flash temperature	K	Eq. (22)
Θ_{flmax}	maximum flash temperature	K	Eq. (2)
Θ_{flmaxT}	maximum flash temperature at test	K	Eq. (94)
Θ_M	bulk temperature	°C	Eq. (22)
Θ_{Mi}	interfacial bulk temperature	°C	Eq. (1)
Θ_{M1}	bulk temperature of pinion teeth	°C	Eq. (20)
Θ_{M2}	bulk temperature of wheel teeth	°C	Eq. (20)
Θ_{MT}	bulk temperature at test	°C	Eq. (94)
Θ_{oil}	oil temperature before reaching the mesh	°C	Eq. (22)
Θ_S	scuffing temperature	°C	Eq. (94)
Θ_{Sc}	scuffing temperature at long contact time	°C	Eq. (97)
λ_{M1}	heat conductivity of pinion	N/(s·K)	Eq. (9)
λ_{M2}	heat conductivity of wheel	N/(s·K)	Eq. (10)
μ	coefficient of friction in pin-and-ring test	—	Fig. 1
μ_m	mean coefficient of friction	—	Eq. (3)
ν_1	Poisson's ratio of pinion material	—	Eq. (A.10)
ν_2	Poisson's ratio of wheel material	—	Eq. (A.10)

Table 1 — Symbols and units *(concluded)*

Symbol	Description	Unit	Reference
ρ_{M1}	density of pinion material	kg/m³	Eq. (9)
ρ_{M2}	density of wheel material	kg/m³	Eq. (10)
ρ_{relC}	relative radius of curvature at pitch point	mm	Eq. (25)
ρ_{y1}	radius of curvature at arbitrary point of pinion	mm	Eq. (5)
ρ_{y2}	radius of curvature at arbitrary point of wheel	mm	Eq. (5)
ρ_{yrel}	relative radius of curvature at arbitrary point	mm	Eq. (5)
Σ	shaft angle	°	Eq. (A.15)
Φ	quill shaft twist	°	Eq. (17)
a The term wheel is used for the mating gear of a pinion.			

4 Scuffing and wear

4.1 Occurrence of scuffing and wear

When gear teeth are completely separated by a full fluid film of lubricant, there is no contact between the asperities of the tooth surfaces, and usually there is no scuffing or wear. Here, the coefficient of friction is rather low. In exceptional cases a damage similar to scuffing may be caused by a sudden thermal instability [19] in a thick oil film, which phenomenon is not treated here.

For thinner elastohydrodynamic films, incidental asperity contact takes place. As the mean film thickness decreases, the number of contacts increases accordingly. Abrasive wear, adhesive wear or scuffing becomes possible. Abrasive wear may occur due to the rolling action of the gear teeth or the presence of abrasive particles in the lubricant. Adhesive wear occurs by localized welding and subsequent detachment and transfer of particles from one or both of the meshing teeth. Abrasive or adhesive wear may not be harmful if it is mild and if it subsides with time, as in a normal run-in process.

In contrast to mild wear, scuffing is a severe form of adhesive wear that can result in progressive damage to the gear teeth. In contrast to pitting and fatigue breakage which show a distinct incubation period, a short transient overloading can result in scuffing failure.

Excessive aeration or the presence in the lubricant of contaminants such as metal particles in suspension, or water, also increases the risk of scuffing damage. After scuffing, high speed gears tend to suffer high levels of dynamic loading due to vibration which usually cause further damage by scuffing, pitting or tooth breakage.

In most cases, the resistance of gears to scuffing can be improved by using a lubricant with enhanced anti-scuff[2)] additives. It is important, however, to be aware that some disadvantages attend the use of anti-scuff additives: corrosion of copper, embrittlement of elastomers, lack of world-wide availability, etc.

The methods described are not suitable for "cold scuffing" which is in general associated with low speed, under approx. 4 m/s, through-hardened heavily loaded gears of rather poor quality.

4.2 Transition diagram

The lubrication condition of sliding concentrated steel contacts, which operate in a liquid lubricant, can be described [20][21][22][23] in terms of transition diagrams. A transition diagram according to Figure 1 is considered to be applicable to contacts functioning at constant oil bath temperature.

2) The less correct designation Extreme Pressure, EP, is replaced by anti-scuff.

At combinations of normal force F_n and relative sliding velocity v_g which fall below the line A1-S, in region I, see Figure 1, the lubrication condition is characterized by a coefficient of friction of about 0,1 and a specific wear rate of 10^{-2} mm^3/(N·m) to 10^{-6} mm^3/(N·m) (i.e. volume wear per unit of normal force, per unit of sliding distance).

If, with v_g not above a value according to point S, the load is increased into region II, a transition into a second condition of lubrication occurs. This mild wear lubrication condition is characterized by a coefficient of friction of about 0,3 to 0,4 and a specific wear rate of 1 mm^3/(N·m) to 5 mm^3/(N·m).

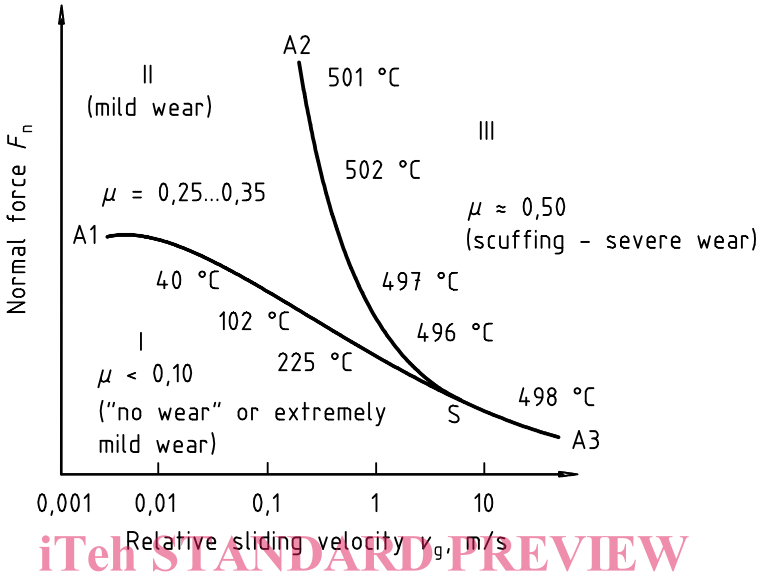

Figure 1 — Transition diagram for contraform contacts with example of calculated contact temperatures

If load is increased still further, a transition into a third condition of lubrication, region III, occurs at intersection of the line A2-S. This region is characterized by a coefficient of friction equal to 0,4 to 0,5. The wear rate, however, is considerably higher, i.e. 100 mm^3/(N·m) to 1 000 mm^3/(N·m), than in regions I and II and the worn surfaces show evidence of severe wear in the form of scuffing. If load increases at relative sliding velocities beyond point S, a direct transition from region I to region III takes place.

There is strong evidence that the position of the line A1-S-A3 depends upon lubricant viscosity [24] as well as upon Hertzian contact pressure [20][21]. At combinations of F_n and v_g that fall below this line, it is believed that the surfaces are kept apart by a thin lubricant film which is, however, penetrated by roughness asperities. In this context, the term "partial elastohydrodynamic lubrication" has been used [21].

In region III liquid film effects are completely absent. This region is identical to the region of "incipient scuffing" [25]. There is evidence that the transition which occurs at intersecting the line A2-S is associated with reaching a critical value of the contact temperature. This is the fundamental concept according to Blok.

The transition diagram shown is applicable to newly assembled, i.e. unoxidized steel contacts, as occur in gears, cams and followers, etc. It has been found that the diagram is applicable to four-ball as well as to pin-and-ring test results.

Along curve A1-S-A3 temperature ranges from an oil bath, respectively overall bulk, respectively interfacial bulk temperature, of 28 °C at v_g = 0,001 m/s to a contact temperature of 498 °C at v_g = 10 m/s. This temperature behaviour strongly suggests that the collapse of (partial) elasto-hydrodynamical lubrication does not occur at a constant contact or interfacial bulk temperature, for instance being associated with melting of chemisorbed material. Instead, the pronounced decrease of load carrying capacity with increasing sliding velocity is supposed to be due to decreasing viscosity [24][26][27][28][29].

Contrary to the above, calculated contact temperatures along curve A2-S-A3 tend to attain a constant value, e.g. in the case of AISI 52100 steel specimens approximately 500 °C; see Figure 1. This suggests that the II-III transition is associated with a transformation in the steel, causing the wear mechanism of surfaces to change from mildly adhesive to severely adhesive, perhaps involving a mechanism of thermo-elastic instability [30][31].