TECHNICAL SPECIFICATION

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Calculation of load capacity of spur and helical gears —

Part 20:
Calculation of scuffing load capacity — Flash temperature method

Calcul de la capacité de charge des engrenages cylindriques à dentures droite et hélicoïdale —

Partie 20: Calcul de la capacité de charge au grippage — 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 procedures used to develop this document and those intended for its further maintenance are described in the ISO/IEC Directives, Part 1. In particular, the different approval criteria needed for the different types of ISO documents should be noted. This document was drafted in accordance with the editorial rules of the ISO/IEC Directives, Part 2 (see www.iso.org/directives).

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

This document was prepared by Technical Committee ISO/TC 60, *Gears*, Subcommittee SC 2, *Gear capacity calculation*.

This second edition cancels and replaces the first edition (ISO/TS 6336-20:2017), which has been technically revised.

The main changes are as follows:

— bevel gear related content has been removed after the publication of ISO/TS 10300-20:2021 which precisely covers bevel gears;

— unit of the thermo-elastic factor, X_M, has been corrected in 5.2 and A.4;

— Formula (30) to calculate the parameter on the line of action at point D, Γ_D, has been revised;

— Formula (A.10) to calculate the reduced modulus of elasticity, E_r, has been corrected;

— Formulae (A.11) and (A.12) to calculate the thermal contact coefficients, B_{M1} and B_{M2}, have been corrected;

— Bibliography has been updated.

A list of all parts in the ISO 6336 series can be found on the ISO website.

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

Introduction

The ISO 6336 series consists of International Standards, Technical Specifications (TS) and Technical Reports (TR) under the general title Calculation of load capacity of spur and helical gears (see Table 1).

— International Standards contain calculation methods that are based on widely accepted practices and have been validated.

— TS contain calculation methods that are still subject to further development.

— TR contain data that is informative, such as example calculations.

The procedures specified in ISO 6336-1 to ISO 6336-19 cover fatigue analyses for gear rating. The procedures described in ISO 6336-20 to ISO 6336-29 are predominantly related to the tribological behaviour of the lubricated flank surface contact. ISO 6336-30 to ISO 6336-39 include example calculations. The ISO 6336 series allows the addition of new parts under appropriate numbers to reflect knowledge gained in the future.

Requesting standardized calculations according to ISO 6336 without referring to specific parts requires the use of only those parts that are currently designated as International Standards (see Table 1 for listing). When requesting further calculations, the relevant part or parts of ISO 6336 need to be specified. The use of a technical specification as acceptance criteria for a specific design needs to be agreed in advance between the manufacturer and the purchaser.

Table 1 — Overview of ISO 6336

Calculation of load capacity of spur and helical gears	International Standard	Technical Specification	Technical Report
Part 1: *Basic principles, introduction and general influence factors*	X		
Part 2: *Calculation of surface durability (pitting)*	X		
Part 3: *Calculation of tooth bending strength*	X		
Part 4: *Calculation of tooth flank fracture load capacity*		X	
Part 5: *Strength and quality of materials*	X		
Part 6: *Calculation of service life under variable load*	X		
Part 20: *Calculation of scuffing load capacity — Flash temperature method*		X	
Part 21: *Calculation of scuffing load capacity — Integral temperature method*		X	
Part 22: Calculation of micropitting load capacity (replaces ISO/TR 15144-1)		X	
Part 30: Calculation examples for the application of *ISO 6336 parts 1,2,3,5*			X
Part 31: Calculation examples of micropitting load capacity (replaces ISO/TR 15144-2)			X
At the time of publication of this document, some of the parts listed here were under development. Consult the ISO website.			

Since 1990, the flash temperature method has been enriched with research for short exposure times, consideration of transition diagrams, new approximations for the coefficient of friction, and completely renewed load sharing factors.

The integral temperature, presented in ISO/TS 6336-21, 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 document. 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[8][9][10][11][12][13], high contact temperatures of lubricant and tooth surfaces at the instantaneous contact position can effect a breakdown 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. The bulk temperature, θ_{M}, is the equilibrium temperature of the surface of the gear teeth before they enter the contact zone. For evaluating this component, it can 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[17].

— The rapidly fluctuating flash temperature of the moving faces in contact. The flash temperature is the calculated increase in gear tooth surface temperature at a given point along the path of contact resulting from the combined effects of gear tooth geometry, load, friction, velocity and material properties during operation. The coefficient of friction can significantly influence the result and it is recommended to closely pay attention to its calculation. A common practice is the use of a coefficient of friction valid for regular working conditions, although it can be stated that at incipient scuffing, the coefficient of friction has significantly higher values.

The complex relationship between mechanical, hydrodynamical, thermodynamical and chemical phenomena has been the object of extensive research and experiment. Experimental investigations can induce empirical influence factors. A direct substitution of empirical influence factors can 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 load capacity of spur and helical gears —

Part 20: Calculation of scuffing load capacity — Flash temperature method

1 Scope

This document specifies methods and formulae for evaluating the risk of scuffing, based on Blok's contact temperature concept.

The fundamental concept 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 documents are referred to in the text in such a way that some or all of their content constitutes requirements of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

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

ISO 1328-1, *Cylindrical gears — ISO system of flank tolerance classification — Part 1: Definitions and allowable values of deviations relevant to flanks of gear teeth*

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

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

3 Terms and definitions

3.1 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 1122-1 and ISO 10825 apply.

ISO and IEC maintain terminology databases for use in standardization at the following addresses:

— ISO Online browsing platform: available at https://www.iso.org/obp

— IEC Electropedia: available at https://www.electropedia.org/

3.2 Symbols and units

The symbols used in the formulae are shown in Table 2. The units of length, metre, millimetre and micrometre, have been chosen in accordance with common practice. To achieve a “coherent” system, the units for B_M, c_γ and X_M have been adapted to the mixed application of metre and millimetre or millimetre and micrometre.

NOTE The term *wheel* is used for the mating gear of a pinion.

Table 2 — Symbols and units

Symbol	Description	Unit
A	Tolerance class in accordance with ISO 1328-1	—
a	Centre distance	mm
B_{M}	Thermal contact coefficient	$\mathrm{N/(mm^{1/2}{\cdot}m^{1/2}{\cdot}s^{1/2}{\cdot}K)}$
B_{M1}	Thermal contact coefficient of pinion	$\mathrm{N/(mm^{1/2}{\cdot}m^{1/2}{\cdot}s^{1/2}{\cdot}K)}$
B_{M2}	Thermal contact coefficient of wheel	$\mathrm{N/(mm^{1/2}{\cdot}m^{1/2}{\cdot}s^{1/2}{\cdot}K)}$
b	Facewidth, smaller value for pinion or wheel	mm
b_{H}	Semi-width of Hertzian contact band	mm
C_{a1}	Tip relief of pinion	µm
C_{a2}	Tip relief of wheel	µm
C_{eff}	Optimal tip relief	µm
C_{eq1}	Equivalent tip relief of pinion	µm
C_{eq2}	Equivalent tip relief of wheel	µm
C_{f1}	Root relief of pinion	µm
C_{f2}	Root relief of wheel	µm
c_{M1}	Specific heat per unit mass of pinion	J/(kg·K)
c_{M2}	Specific heat per unit mass of wheel	J/(kg·K)
c_{γ}	Mesh stiffness	N/(mm·µm)
d_{a1}	Tip diameter of pinion	mm
d_{a2}	Tip diameter of wheel	mm
d_1	Reference diameter of pinion	mm
d_2	Reference diameter of wheel	mm
E_1	Modulus of elasticity of pinion	$\mathrm{N/mm^2}$
E_2	Modulus of elasticity of wheel	$\mathrm{N/mm^2}$
E_{r}	Reduced modulus of elasticity	$\mathrm{N/mm^2}$
F_{ex}	External axial force	N
F_{n}	Normal load in wear test	N
F_{t}	Nominal tangential force	N
H_1	Auxiliary dimension	mm
H_2	Auxiliary dimension	mm
K_{A}	Application factor	—
$K_{\mathrm{B}\alpha}$	Transverse load factor (scuffing)	—
$K_{\mathrm{B}\beta}$	Face load factor (scuffing)	—
$K_{\mathrm{H}\alpha}$	Transverse load factor (contact stress)	—
$K_{\mathrm{H}\beta}$	Face load factor (contact stress)	—
K_{mp}	Multiple path factor	—
K_{v}	Dynamic factor	—
m_{n}	Normal module	mm
n_{p}	Number of mesh contacts	—
n_1	Revolutions per minute of pinion	$\mathrm{min^{-1}}$
$Pé_1$	Péclet number of pinion material	—
$Pé_2$	Péclet number of wheel material	—
Ra_1	Tooth flank surface roughness of pinion	µm
Ra_2	Tooth flank surface roughness of wheel	µm
S_{B}	Safety factor for scuffing	—

Table 2 *(continued)*

Symbol	Description	Unit
S_{FZG}	Load stage (in FZG test)	—
t_c	Contact exposure time at bend of curve	µs
t_{max}	Longest contact exposure time	µs
t_1	Contact exposure time of pinion	µs
t_2	Contact exposure time of wheel	µs
u	Gear ratio	—
u_v	Virtual ratio	—
v_g	Sliding velocity	m/s
v_{g1}	Tangential velocity of pinion	m/s
v_{g2}	Tangential velocity of wheel	m/s
$v_{g\Sigma C}$	Sum of tangential velocities in pitch point	m/s
v_t	Pitch line velocity	m/s
w_{Bn}	Normal unit load	N/mm
w_{Bt}	Transverse unit load	N/mm
$X_{but,\Gamma}$	Buttressing factor	—
$X_{but,A}$	Buttressing value	—
$X_{but,E}$	Buttressing value	—
X_G	Geometry factor	—
X_J	Approach factor	—
X_L	Lubricant factor	—
X_M	Thermo-elastic factor	$K \cdot N^{-3/4} \cdot s^{1/2} \cdot m^{-1/2} \cdot mm$
X_{mp}	Multiple mating pinion factor	—
X_R	Roughness factor	—
X_S	Lubrication system factor	—
X_W	Structural factor	—
$X_{\alpha\beta}$	Angle factor	—
X_Γ	Load sharing factor	—
X_Θ	Gradient of the scuffing temperature	—
z_1	Number of teeth of pinion	—
z_2	Number of teeth of wheel	—
α_{a1}	Transverse tip pressure angle of pinion	°
α_{a2}	Transverse tip pressure angle of wheel	°
α_t	Transverse pressure angle	°
α_{wn}	Normal working pressure angle	°
α_{wt}	Transverse working pressure angle	°
α_{y1}	Pinion pressure angle at arbitrary point	°
β	Helix angle	°
β_b	Base helix angle	°
β_w	Working helix angle	°
Γ_A	Parameter on the line of action at point A	—
Γ_{AA}	Parameter on the line of action at point AA	—
Γ_{AB}	Parameter on the line of action at point AB	—
Γ_{AU}	Parameter on the line of action at point AU	—
Γ_B	Parameter on the line of action at point B	—

Table 2 *(continued)*

Symbol	Description	Unit
Γ_{BB}	Parameter on the line of action at point BB	—
Γ_{D}	Parameter on the line of action at point D	—
Γ_{DD}	Parameter on the line of action at point DD	—
Γ_{DE}	Parameter on the line of action at point DE	—
Γ_{E}	Parameter on the line of action at point E	—
Γ_{EE}	Parameter on the line of action at point EE	—
Γ_{EU}	Parameter on the line of action at point EU	—
Γ_{M}	Parameter on the line of action at point M	—
Γ_{y}	Parameter on the line of action at arbitrary point	—
γ_1	Angle of direction of tangential velocity of pinion	—
γ_2	Angle of direction of tangential velocity of wheel	—
ε_α	Transverse contact ratio	—
ε_β	Overlap ratio	—
ε_γ	Total contact ratio	—
η_{oil}	Absolute (dynamic) viscosity at oil temperature	mPa·s
Θ_B	Contact temperature	°C
Θ_{Bmax}	Maximum contact temperature	°C
Θ_{fl}	Flash temperature	K
Θ_{flm}	Average flash temperature	K
Θ_{flmax}	Maximum flash temperature	K
Θ_{flmaxT}	Maximum flash temperature at test	K
Θ_M	Bulk temperature	°C
Θ_{Mi}	Interfacial bulk temperature	°C
Θ_{MT}	Bulk temperature at test	°C
Θ_{M1}	Bulk temperature of pinion teeth	°C
Θ_{M2}	Bulk temperature of wheel teeth	°C
Θ_{oil}	Oil temperature before reaching the mesh	°C
Θ_S	Scuffing temperature	°C
Θ_{Sc}	Scuffing temperature at long contact time	°C
λ_{M1}	Heat conductivity of pinion	N/(s·K)
λ_{M2}	Heat conductivity of wheel	N/(s·K)
μ	Coefficient of friction in pin-and-ring test	—
μ_m	Mean coefficient of friction	—
ν_1	Poisson's ratio of pinion material	—
ν_2	Poisson's ratio of wheel material	—
ρ_{M1}	Density of pinion material	kg/m^3
ρ_{M2}	Density of wheel material	kg/m^3
ρ_{relC}	Transverse relative radius of curvature at pitch point	mm
ρ_{y1}	Radius of curvature at arbitrary point of pinion	mm
ρ_{y2}	Radius of curvature at arbitrary point of wheel	mm
ρ_{rely}	Relative radius of curvature at arbitrary point y	mm
Φ	Quill shaft twist	°

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 can be caused by a sudden thermal instability[15] in a thick oil film. This phenomenon is not treated here.

For thinner elastohydrodynamic films, incidental asperity contact takes place. Accordingly, as the mean film thickness decreases, the number of contacts increases. Abrasive wear, adhesive wear or scuffing becomes possible. Abrasive wear can 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 additives.

NOTE The less correct designation Extreme Pressure (EP) is replaced by anti-scuff.

It is important to be aware that the use of anti-scuff additives can equally lead to some disadvantages, e.g. corrosion of copper, embrittlement of elastomers, lack of world-wide availability.

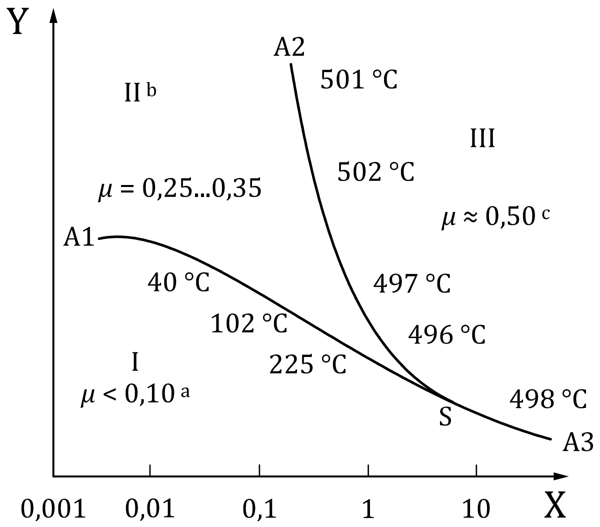
The methods described are not suitable for "cold scuffing" which is in general associated with low speed, under approximately 4 m/s, through hardened heavily loaded gears of rather poor quality.

4.2 Transition diagram

The lubrication condition of sliding concentrated steel contacts, which operate in a liquid lubricant, can be described[16][17][18][19] 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.

At combinations of normal force, F_n, and 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).



Key

X sliding velocity, v_g, in m/s

Y normal force, F_n

a "No wear" or extremely mild wear.

b Mild wear.

c Scuffing — severe wear.

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

If load is further increased, 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 mm³/(N·m) to 1 000 mm³/(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 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[20] as well as upon Hertzian contact pressure[16][17]. At combinations of F_n and v_g that fall below this line, 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" is used[17].

In region III, liquid film effects are completely absent. This region is identical to the region of "incipient scuffing"[21]. 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 References [8],[9],[10],[11],[12],[13],[14] and [15].

The transition diagram shown is applicable to newly assembled, i.e. unoxidized steel contacts, as occur in gears, cams and followers. 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 to A3, the temperature ranges from an oil bath, overall bulk and 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[20][22][23][24][25].

Contrary to the above, calculated contact temperatures along curve A2-S to A3 tend to attain a constant value, e.g. in the case of AISI 52100[18][20], steel specimens are 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[26][27].

Therefore, the results indicate scuffing is associated with a critical magnitude of the contact temperature. For steel lubricated with mineral oils, the critical magnitude does not depend on load, velocity and geometry, and equals near 500 °C.

4.3 Friction at incipient scuffing

As shown in the transition diagram in Figure 1, in the case of scuffing, the coefficient of friction increases from about 0,25 to about 0,5. The corresponding contact temperature proves to be about 500 °C. This contact temperature is the sum of a measured interfacial bulk temperature of 28 °C and a calculated flash temperature of 470 °C. During the flash temperature calculation, the coefficient of friction just before transition, $\mu = 0{,}35$ is used. If this method has to be applied not only for pin-and-ring tests but also (during the design stage) for gear transmissions, the choice of the value of the critical magnitude of the contact temperature shall be agreed on one hand and the value of the coefficient of friction to be used in the calculations on the other.

A gear load capacity can be predicted

- on the safe side, with the coefficient of friction of $\mu = 0{,}50$,
- accurately, with the coefficient of friction between $\mu = 0{,}25$ and $\mu = 0{,}35$, dependent on the lubricant, and
- according to previous practice, with a low coefficient of friction of regular working conditions, provided that the limiting contact temperature is correspondingly low.

In terms of previous practice, for non-additive and low-additive mineral oils, each combination of oil and rolling materials has a critical scuffing temperature which, in general, is constant regardless of the operating conditions, load, velocity and geometry.

For high-additive and certain kinds of synthetic lubricants, the critical scuffing temperature can well vary from one set of operating conditions to another. So, this critical temperature shall then be determined for each such set separately from tests which closely simulate the operating condition of the gearset.

5 Basic formulae

5.1 Contact temperature

As mentioned in the introduction, the contact temperature is the sum of the interfacial bulk temperature, Θ_{Mi}, (see 5.4) and the flash temperature, Θ_{fl}, (see 5.2), as shown in Formula (1):

$$\Theta_{\mathrm{B}} = \Theta_{\mathrm{Mi}} + \Theta_{\mathrm{fl}} \tag{1}$$