TECHNICAL SPECIFICATION

ISO/TS 6336-21

Second edition
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Calculation of load capacity of spur and helical gears —

Part 21: Calculation of scuffing load capacity — Integral temperature method

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

Partie 21: Calcul de la capacité de charge au grippage — 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 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 6336-21:2017), 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;

— subclause 5.1 has been rearranged.

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 6336-1 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.			

This document describes the surface damage "warm scuffing" for cylindrical (spur and helical) 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", generally associated with low temperature and low speed, under approximately 4 m/s, and through-hardened, heavily loaded gears, the formulae 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 can 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 increases 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/TS 6336-20), based on contact temperatures which vary along the path of contact;

— the integral temperature method (presented in this document), 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 influencing 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 can be necessary. The development of such methods is the objective of ongoing research.

Calculation of load capacity of spur and helical gears —

Part 21: Calculation of scuffing load capacity — Integral temperature method

1 Scope

This document specifies the integral temperature method for calculating the scuffing load capacity of cylindrical gears.

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

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

3 Terms and definitions

3.1 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 1122-2 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 this document are given in Table 2.

Table 2 — Symbols and units

Symbol	Description	Unit
a	Centre distance	mm
B_M	Thermal contact coefficient	$N/(mm \cdot s^{1/2} \cdot K)$
b	Facewidth, smaller value of pinion or wheel	mm
C_1, C_2, C_{2H}	Weighting factors	—
C_a	Nominal tip relief	µm
C_{eff}	Effective tip relief	µm
c_v	Specific heat capacity per unit volume	$N/(mm^2 \cdot K)$

Table 2 *(continued)*

Symbol	Description	Unit
c'	Single stiffness	N/(mm·µm)
c_γ	Mesh stiffness	N/(mm·µm)
d	Reference circle diameter	mm
d_{Na}	Effective tip diameter	mm
d_{a}	Tip diameter	mm
d_{b}	Base diameter	mm
E	Module of elasticity (Young's modulus)	N/mm^2
F_{n}	Normal tooth load	N
F_{t}	Nominal tangential load at reference circle	N
$g_{\mathrm{an1,2}}$	Recess path of contact of pinion, wheel	mm
$g_{\mathrm{fn1,2}}$	Approach path of contact of pinion, wheel	mm
g^*	Sliding factor	—
K_{A}	Application factor	—
K_{v}	Dynamic factor	—
$K_{\mathrm{B}\alpha}$	= $K_{\mathrm{H}\alpha}$ transverse load factor (scuffing)	—
$K_{\mathrm{B}\beta}$	= $K_{\mathrm{H}\beta}$ face load factor (scuffing)	—
$K_{\mathrm{B}\gamma}$	Helical load factor (scuffing)	—
$K_{\mathrm{H}\alpha}$	Transverse load factor	—
$K_{\mathrm{H}\beta}$	Face load factor	—
m	Module	mm
m_{sn}	Normal module of virtual crossed axes helical gear	mm
n_{p}	Number of meshing gears	—
p_{en}	Normal base pitch	mm
Ra	Arithmetic mean roughness	µm
S_{intS}	Scuffing safety factor	—
S_{Smin}	Minimum required scuffing safety factor	—
T_1	Torque of the pinion	Nm
T_{1T}	Scuffing torque of test pinion	Nm
u	Gear ratio	—
v	Reference line velocity	m/s
$v_{\mathrm{g}\gamma\mathrm{l}}$	Maximum sliding velocity at tip of pinion	m/s
v_{gs}	Sliding velocity at pitch point	m/s
$v_{\mathrm{g1,2}}$	Sliding velocity	m/s
$v_{\mathrm{g}\alpha 1}$	Sliding velocity	m/s
$v_{\mathrm{g}\beta 1}$	Sliding velocity	m/s
$v_{\Sigma\mathrm{C}}$	Sums of tangential speeds at pitch point	m/s
$v_{\Sigma\mathrm{s}}$	Tangential speed	m/s
$v_{\Sigma\mathrm{h}}$	Tangential speed	m/s
w_{Bt}	Specific tooth load, scuffing	N/mm
X_{BE}	Geometry factor at pinion tooth tip	—
X_{E}	Run-in factor	—
X_{Ca}	Tip relief factor	—
X_{L}	Lubricant factor	—
X_{M}	Thermal flash factor	—

Table 2 *(continued)*

Symbol	Description	Unit
X_Q	Approach factor	—
X_R	Roughness factor	—
X_S	Lubrication factor	—
X_W	Welding factor of executed gear	—
X_WT	Welding factor of test gear	—
X_WrelT	Relative welding factor	—
X_mp	Contact factor	—
$X_{\alpha\beta}$	Pressure angle factor	—
X_ε	Contact ratio factor	—
z	Number of teeth	—
α	Pressure angle	°
α_n	Normal pressure angle	°
α_sn	Normal pressure angle of crossed axes helical gear	°
α_st	Transverse pressure angle of crossed axes helical gear	°
α_t	Transverse pressure angle	°
α_t'	Transverse working pressure angle	°
α_y	Arbitrary angle	°
β	Helix angle	°
β_b	Helix angle at base circle	°
β_s	Helix angle of virtual crossed axes helical gear	°
$\varGamma$	Parameter on the line of action	—
γ	Auxiliary angle	°
ε_a	Recess contact ratio	—
ε_f	Approach contact ratio	—
ε_n	Contact ratio in normal section of virtual crossed axes helical gear	—
ε_1	Addendum contact ratio of the pinion	—
ε_2	Addendum contact ratio of the wheel	—
ε_α	Contact ratio	—
η	Hertzian auxiliary coefficient	—
η_oil	Dynamic viscosity at oil temperature	mPa · s
ϑ	Hertzian auxiliary angle	°
ϑ_flaE	Flash temperature at pinion tooth tip when load sharing is neglected	K
$\vartheta_\mathrm{flaint}$	Mean flash temperature	K
ϑ_int	Integral temperature	K
ϑ_intP	Permissible integral temperature	K
ϑ_intS	Scuffing integral temperature (allowable integral temperature)	K
$\vartheta_\mathrm{flaintT}$	Mean flash temperature of the test gear	K
ϑ_oil	Oil sump or spray temperature	°C
$\vartheta_\mathrm{M\text{-}C}$	Bulk temperature	°C
ϑ_MT	Test bulk temperature	°C
λ_M	Heat conductivity	N/(s · K)
μ_mC	Mean coefficient of friction	—
ν	Poisson's ratio	—
ν_{40}	Kinematic viscosity of the oil at 40 °C	mm²/s; cSt

Table 2 *(continued)*

Symbol	Description	Unit
ξ	Hertzian auxiliary coefficient	—
$\rho_{E1,2}$	Radius of curvature at tip of the pinion, wheel	mm
ρ_{Cn}	Relative radius of curvature at pitch point in normal section	mm
$\rho_{n1,2}$	Radius of curvature at pitch point in normal section	mm
ρ_{redC}	Relative radius of curvature at pitch point	mm
Σ	Crossing angle of virtual crossed axes helical gear	°
φ	Axle angle of virtual crossed axes helical gear	°
φ_E	Run-in grade	—
Subscript		
$_1$	Pinion	
$_2$	Wheel	
$_a$	Tip diameter of the gear	
$_b$	Base circle of the gear	
$_n$	Normal section	
$_s$	Virtual crossed axes helical gear	
$_t$	Tangential direction	
$_T$	Test gear	

4 Field of application

4.1 General

The calculation methods are based on results of the rig testing of gears run at pitch line velocities less than 80 m/s. The formulae 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, as speeds exceed the range with experimental background.

4.2 Scuffing damage

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 can 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 extreme pressure (EP) properties. It is important, however, to be aware that some disadvantages attend the use of EP oils, e.g. corrosion of copper, embrittlement of elastomers, lack of world-wide availability. These disadvantages shall be taken into consideration if optimum lubricant choice shall be made, which means as few additives as possible, as much as necessary.

NOTE EP-additives are also known as anti-scuff-additives.

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, shall 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.

4.3 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 6.1.6 and the mean value of the flash temperature is approximated by substituting mean values of the coefficient of friction, the dynamic loading, 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 is derived from measurements[4] and is approximated by Formula (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 Formula (1).

$$\mu_{mC} = 0{,}045 \cdot \left(\frac{w_{Bt} \cdot K_{B\gamma}}{v_{\Sigma C} \cdot \rho_{redC}} \right)^{0{,}2} \cdot \eta_{oil}^{-0{,}05} \cdot X_R \cdot X_L \tag{1}$$

where

w_{Bt} is the specific tooth load for scuffing;

$K_{B\gamma}$ is the helical load factor for scuffing;

$v_{\Sigma C}$ is the sum of tangential speeds at pitch point;

ρ_{redC} is the relative radius of curvature at pitch point;

η_{oil} is the dynamic viscosity at oil temperature;

X_R is the roughness factor;

X_L is the lubricant factor.

NOTE Formula (1) is derived from testing of gears with centre distance $a \approx 100$ mm. An alternative formula for calculating μ_{mC} based on tests within a range of a = 91,5 mm to 200 mm is given in Formula (8).

The coefficient of friction of the integral temperature method takes account of the size of the gear in a different way as the coefficient of friction of the flash temperature method. Formula (1) for calculating the coefficient of friction should not be applied outside the field of the part where it is presented, e.g. coefficient of friction for thermal rating.

The formula for the calculation of μ_{mC} is derived from experiments in the following range of operating conditions. Extrapolation can lead to deviations between the calculated and the real coefficient of friction.

$1\ \text{m/s} \le v \le 50\ \text{m/s}$

At reference line velocities, v, lower than 1 m/s, higher coefficients of friction are expected. At reference line velocities, v, higher than 50 m/s, the limiting value of $v_{\Sigma C}$ at v = 50 m/s shall be used in Formula (1).

$w_{Bt} \ge 150\ \text{N/mm}$

For lower values of the specific normal tooth load, w_{Bt}, the limiting value, w_{Bt} = 150 N/mm, shall be used in Formula (1).

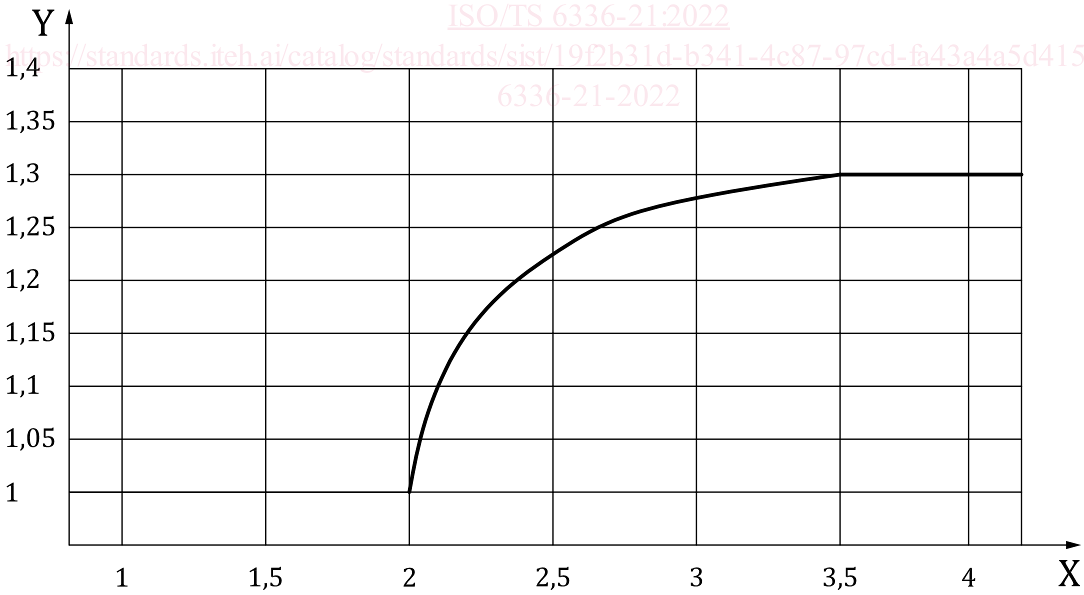
$$v_{\Sigma C} = 2 \cdot v \cdot \tan \alpha_t' \cdot \cos \alpha_t \tag{2}$$

$$\rho_{redC} = \frac{u}{(1+u)^2} \cdot a \cdot \frac{\sin \alpha_t'}{\cos \beta_b} \tag{3}$$

$$w_{Bt} = K_A \cdot K_v \cdot K_{B\beta} \cdot K_{B\alpha} \cdot \frac{F_t}{b} \tag{4}$$

The following definitions for the parameters $K_{B\gamma}$, X_R and X_L in Formula (1) apply.

$K_{B\gamma}$ is the helical load factor. Scuffing takes account of increasing friction for increasing total contact ratio (see Figure 1).



Key

X total contact ratio, ε_γ

Y helical load factor, $K_{B\gamma}$

Figure 1 — Helical load factor, $K_{B\gamma}$

$$K_{B\gamma} = 1 \qquad \text{for } \varepsilon_\gamma \le 2$$

$$K_{B\gamma} = 1 + 0{,}2 \cdot \sqrt{(\varepsilon_\gamma - 2) \cdot (5 - \varepsilon_\gamma)} \qquad \text{for } 2 < \varepsilon_\gamma < 3{,}5 \tag{5}$$

$$K_{B\gamma} = 1{,}3 \qquad \text{for } \varepsilon_\gamma \geq 3{,}5$$

X_R is the roughness factor

where

$$X_R = 2{,}2 \cdot (Ra / \rho_{redC})^{0{,}25} \tag{6}$$

$$Ra = 0{,}5 \cdot (Ra_1 + Ra_2) \tag{7}$$

Ra_1 and Ra_2 are the tooth flank roughness values of pinion and wheel measured on the new flanks as manufactured (e.g. reference test gear roughness values, *Ra*, are ≈ 0,35 µm).

X_L is the lubricant factor

where

X_L is 1,0 for mineral oils;

X_L is 0,8 for polyalfaolefins;

X_L is 0,7 for non-water-soluble polyglycols;

X_L is 0,6 for water-soluble polyglycols;

X_L is 1,5 for traction fluids;

X_L is 1,3 for phosphate esters.

Formula (8) represents results of tests within a range of a = 91,5 mm to 200 mm. The application of this formula makes it necessary to adjust Figures 9, 10 and 11 for the scuffing temperature, ϑ_{intS}, accordingly.

$$\mu_{mC} = 0{,}048 \cdot \left(\frac{F_{bt} / b}{v_{\Sigma C} \cdot \rho_{redC}} \right)^{0{,}2} \cdot \eta_{oil}^{-0{,}05} \cdot Ra^{0{,}25} \cdot X_L \tag{8}$$

where

X_L is $0{,}75 \cdot \left(\frac{6}{v_{\Sigma C}} \right)^{0{,}2}$ for polyglycols;

X_L is 1,0 for mineral oils;

X_L is 0,8 for polyalfaolefins;

X_L is 1,5 for traction fluids;

X_L is 1,3 for phosphate esters;

Ra see Formula (7).

