TECHNICAL SPECIFICATION

ISO/TS 6336-21

Second edition
2022-05

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

Reference number
ISO/TS 6336-21:2022(E)

Contents

Page

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).

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. Details of any patent rights identified during the development of the document will be in the Introduction and/or on the ISO list of patent declarations received (see www.iso.org/patents).

Any trade name used in this document is information given for the convenience of users and does not constitute an endorsement.

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
$g_{an1,2}$	Recess path of contact of pinion, wheel	mm
$g_{fn1,2}$	Approach path of contact of pinion, wheel	mm
g^*	Sliding factor	—
K_A	Application factor	—
K_v	Dynamic factor	—
$K_{B\alpha}$	= $K_{H\alpha}$ transverse load factor (scuffing)	—
$K_{B\beta}$	= $K_{H\beta}$ face load factor (scuffing)	—
$K_{B\gamma}$	Helical load factor (scuffing)	—
$K_{H\alpha}$	Transverse load factor	—
$K_{H\beta}$	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
$v_{g\gamma l}$	Maximum sliding velocity at tip of pinion	m/s
v_{gs}	Sliding velocity at pitch point	m/s
$v_{g1,2}$	Sliding velocity	m/s
$v_{g\alpha 1}$	Sliding velocity	m/s
$v_{g\beta 1}$	Sliding velocity	m/s
$v_{\Sigma C}$	Sums of tangential speeds at pitch point	m/s
$v_{\Sigma s}$	Tangential speed	m/s
$v_{\Sigma h}$	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	°
Γ	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
ϑ_{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
$\vartheta_{flaintT}$	Mean flash temperature of the test gear	K
ϑ_{oil}	Oil sump or spray temperature	°C
$\vartheta_{M\text{-}C}$	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.

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

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).

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

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).

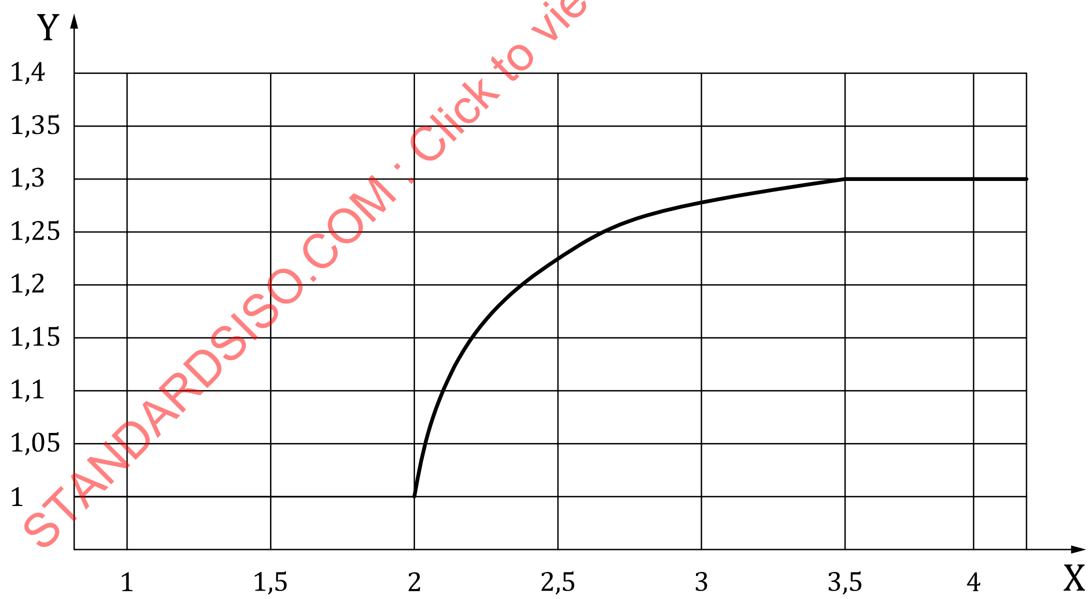
$$v_{\Sigma C} = 2 \cdot v \cdot \tan\alpha_t' \cdot \cos\alpha_t \tag{2}$$

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

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

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, $K_{B\gamma}$

$K_{B\gamma} = 1$ for $\varepsilon_\gamma \leq 2$

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

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

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).

5.2 Run-in factor, X_E

These calculation methods presume that the gears are well run-in. In practice, scuffing failure occurs very often during the first few hours in service, e.g. in a full load test run, the acceptance run of vessels or when a new set of gears is built into a production machinery when the gears are run under full load conditions before a proper run-in. Investigations[4] show a 1/4 to 1/3 load carrying capacity of a newly manufactured gear flank compared to a properly run-in flank. This should be taken into account by a run-in factor, X_E, as shown in Formula (9):

$$X_E = 1 + (1 - \varphi_E) \cdot \frac{30 \cdot Ra}{\rho_{redC}} \tag{9}$$

where

φ_E is 1, full run-in (for carburized and ground gears full run-in can be assumed if $Ra_{run\text{-}in} \approx 0{,}6\ Ra_{new}$);

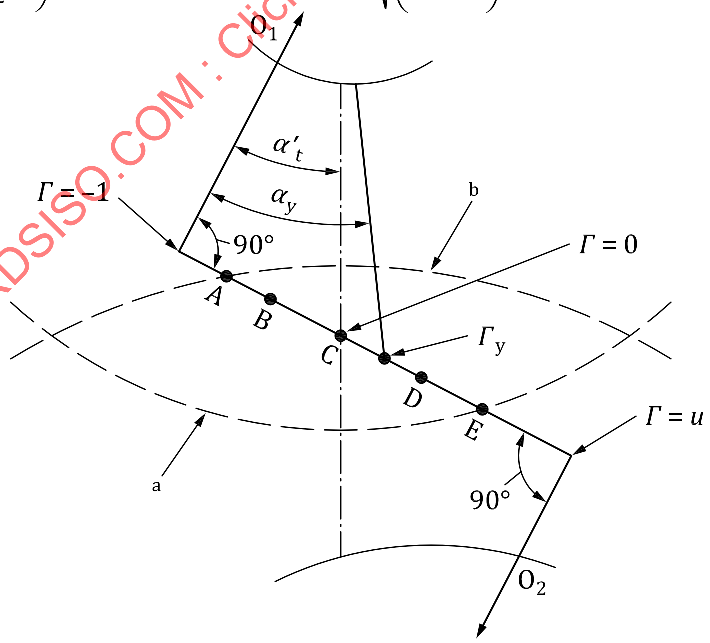
φ_E is 0, newly manufactured.

5.3 Thermal flash factor, X_M

The thermal flash factor, X_M, accounts for the influence of the properties of the pinion and gear materials on the flash temperature.

Calculation of the thermal flash factor for an arbitrary point (index y) on the line of action (see Figure 2) is shown in Formula (10):

$$X_M = \left(\frac{2}{\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}} \right)^{0,25} \cdot \frac{\sqrt{(1+\Gamma)} + \sqrt{\left(1 - \frac{\Gamma}{u}\right)}}{B_{M1} \cdot \sqrt{(1+\Gamma)} + B_{M2} \cdot \sqrt{\left(1 - \frac{\Gamma}{u}\right)}} \tag{10}$$



a Tip circle 1.

b Tip circle 2.

Figure 2 — Parameter Γ on the line of action

$$\Gamma = \frac{\tan \alpha_y}{\tan \alpha_t'} - 1 \tag{11}$$

If the materials of pinion and wheel are the same, Formula (10) can be simplified as shown in Formula (12):

$$X_M = \frac{E^{0,25}}{\left(1-\nu^2\right)^{0,25} \cdot B_M} \tag{12}$$

In the above formula, the thermal contact coefficient, B_M, is shown in Formula (13):

$$B_M = \sqrt{\left(\lambda_M \cdot c_v\right)} \tag{13}$$

For case hardened steels with the following typical characteristic values:

λ_M = 50 N/(s · K), c_v = 3,8 N/(mm^2 · K), E = 206 000 N/mm^2 and ν = 0,3

follows

X_M = 50,0 K · N$^{-0,75}$ · s^{0,5} · m$^{-0,5}$ · mm

For the characteristic values of other materials, see Reference [10].

5.4 Pressure angle factor, $X_{\alpha\beta}$

The pressure angle factor, $X_{\alpha\beta}$, is used to account for the conversion of load and tangential speed from reference circle to pitch circle.

Method A: Factor $X_{\alpha\beta\text{-A}}$ is shown in Formula (14):

$$X_{\alpha\beta-A} = 1,22 \cdot \frac{\left(\sin \alpha_t'\right)^{0,25} \cdot \left(\cos \alpha_n\right)^{0,25} \cdot \left(\cos \beta\right)^{0,25}}{\left(\cos \alpha_t'\right)^{0,5} \cdot \left(\cos \alpha_t\right)^{0,5}} \tag{14}$$

Table 3 shows the values for the pressure angle factor, $X_{\alpha\beta}$, for a standard rack with pressure angle, α_n = 20°, the typical range of standard working pressure angles, α_t', and helix angles, β.

Table 3 — Method B: Factor $X_{\alpha\beta\text{-B}}$

α_t'	β = 0°	10°	20°	30°
19°	0,963	0,960	0,951	0,938
20°	0,978	0,975	0,966	0,952
21°	0,992	0,989	0,981	0,966
22°	1,007	1,004	0,995	0,981
23°	1,021	1,018	1,009	0,995
24°	1,035	1,032	1,023	1,008
25°	1,049	1,046	1,037	1,012

As an approximation, for gears with normal pressure angle, α_n = 20°, the pressure angle factor can be approximated as follows:

$X_{\alpha\beta\text{-B}}$ = 1

6 Calculation

6.1 Cylindrical gears

6.1.1 General

Clause 6 contains formulae which enable the assessment of the "probability of scuffing" (warm scuffing) of oil-lubricated, involute spur and helical gears. Various examples of calculations using the following formulae are given in Annex A.

It is assumed that the total tangential load is equally distributed between the two helices of double helical gears. When, due to application of forces such as external axial forces, this is not the case, the influences of these shall be taken into account separately. The two helices shall be treated as parallel single helical gears. Influences affecting scuffing probability, for which quantitative assessments can be made, are included.

The formulae are valid for gears with external or internal teeth conjugated to a basic rack which shall follow the characteristics given in ISO 53. For internal gears, negative values shall be introduced for the determination of the geometry factor, X_{BE}, as presented in 6.1.11. They may also be considered as valid for similar gears of other basic rack forms, of which the transverse contact ratio is $\varepsilon_\alpha \leq 2{,}5$.

6.1.2 Scuffing safety factor, S_{intS}

As uncertainties and inaccuracies in the assumptions cannot be excluded, it is necessary to introduce a safety factor, S_{intS}. The scuffing safety factor is temperature-related and is not a factor by which gear torque may be multiplied to arrive at the same values for the integral temperature number, ϑ_{int}, and the scuffing integral temperature number, ϑ_{intS}.

$$S_{intS} = \frac{\vartheta_{intS}}{\vartheta_{int}} \geq S_{Smin} \tag{15}$$

Recommendation for choosing S_{Smin}:

$S_{Smin} < 1$	High scuffing risk
$1 \leq S_{Smin} \leq 2$	Critical range with moderate scuffing risk, influenced by the operating conditions of the actual gear. Influencing factors are, e.g. the tooth flank roughness, run-in effects, the accurate knowledge of the load factors, the load capacity of lubricating oil.
$S_{Smin} > 2$	Low scuffing risk

Given the relationship between the actual load and the integral temperature number, the corresponding load safety factor, S_{Sl}, can be approximated by Formula (16):

$$S_{Sl} = \frac{w_{Btmax}}{w_{Bteff}} \approx \frac{\vartheta_{intS} - \vartheta_{oil}}{\vartheta_{int} - \vartheta_{oil}} \tag{16}$$

6.1.3 Permissible integral temperature, ϑ_{intP}

$$\vartheta_{intP} = \frac{\vartheta_{intS}}{S_{Smin}} \tag{17}$$

The minimum required scuffing safety factor, S_{Smin}, shall be separately determined for each application.

6.1.4 Integral temperature, ϑ_{int}

$$\vartheta_{\mathrm{int}} = \vartheta_{\mathrm{M}} + C_2 \cdot \vartheta_{\mathrm{flaint}} \leq \vartheta_{\mathrm{intP}} \tag{18}$$

where C_2 is the weighting factor derived from experiments. For spur and helical gears, C_2 = 1,5.

$$\vartheta_{\mathrm{flaint}} = \vartheta_{\mathrm{flaE}} \cdot X_{\varepsilon} \tag{19}$$

6.1.5 Flash temperature at pinion tooth tip, $\vartheta_{\mathrm{flaE}}$

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.

$$\vartheta_{\mathrm{flaE}} = \mu_{\mathrm{mC}} \cdot X_{\mathrm{M}} \cdot X_{\mathrm{BE}} \cdot X_{\alpha\beta} \cdot \frac{\left(K_{\mathrm{B}\gamma} \cdot w_{\mathrm{Bt}}\right)^{0,75} \cdot v^{0,5}}{|a|^{0,25}} \cdot \frac{X_{\mathrm{E}}}{X_{\mathrm{Q}} \cdot X_{\mathrm{Ca}}} \tag{20}$$

6.1.6 Bulk temperature, ϑ_{M}

6.1.6.1 General

The bulk temperature is the temperature of the tooth surfaces immediately before they come into contact.

The bulk temperature is established by the thermal balance of the gear unit. There are several sources of heat in a gear unit of which the most important are tooth and bearing friction. Other sources of heat, such as seals and oil flow, contribute to some extent. At pitch line velocities in excess of 80 m/s, heat from the churning of oil in the mesh and windage losses can become significant and should be taken into consideration (see Method A). The heat is transferred to the environment via the housing walls by conduction, convection and radiation and for spray lubrication conditions through the oil into an external heat exchanger.

Values obtained using the different calculation methods described below shall be distinguished by the subscripts A, B and C.

6.1.6.2 Method A, $\vartheta_{\text{M-A}}$

The bulk temperature as a mean value or as temperature distribution over the facewidth can be measured experimentally or be determined by a theoretical analysis based on known power loss and heat transfer data, i.e. by using thermal network methods.

6.1.6.3 Method B, $\vartheta_{\text{M-B}}$

This method is not used for the integral temperature method (the flash temperature method is given in ISO/TS 6336-20).

6.1.6.4 Method C, $\vartheta_{\text{M-C}}$

An approximate value for the bulk temperature consists of the sum of the oil temperature and a part of a mean value derived from the flash temperature over the path of contact according to method C, as shown in Formula (21):

$$\vartheta_{\mathrm{M-C}} = \vartheta_{\mathrm{oil}} + C_1 \cdot X_{\mathrm{mp}} \cdot \vartheta_{\mathrm{flaint}} \cdot X_{\mathrm{S}} \tag{21}$$

where

X_S is 1,2 for spray lubrication;

X_S is 1,0 for dip lubrication;

X_S is 0,2 for gears submerged in oil;

C_1 is the constant accounting for heat transfer conditions, from test results C_1 = 0,7.

$$X_{mp} = \frac{1+n_p}{2} \tag{22}$$

where n_p is the number of meshing gears.

6.1.7 Mean coefficient of friction, μ_{mC}

See 5.1.

6.1.8 Run-in factor, X_E

See 5.2.

6.1.9 Thermal flash factor, X_M

See 5.3.

6.1.10 Pressure angle factor, $X_{\alpha\beta}$

See 5.4.

6.1.11 Geometry factor at tip of pinion, X_{BE}

The geometry factor, X_{BE}, takes account for Hertzian stress and sliding velocity at the pinion tooth tip. X_{BE} is a function of the gear ratio, u, and the radius of curvature, ρ_E, at the pinion tooth tip, E.

For internal gears, the following parameters shall be introduced as negative values:

— number of teeth, z_2;

— gear ratio, u;

— centre distance, a;

— all diameters.

$$X_{BE} = 0{,}51 \cdot \sqrt{\frac{|z_2|}{z_2} \cdot (u+1)} \cdot \frac{\sqrt{\rho_{E1}} - \sqrt{\frac{\rho_{E2}}{u}}}{(\rho_{E1} \cdot |\rho_{E2}|)^{0{,}25}} \tag{23}$$

$$\rho_{E1} = 0{,}5 \cdot \sqrt{d_{a1}^2 - d_{b1}^2} \tag{24}$$

$$\rho_{E2} = a \cdot \sin\alpha_t' - \rho_{E1} \tag{25}$$

6.1.12 Approach factor, X_Q

The approach factor, X_Q, takes into account impact loads at the ingoing mesh (at the tooth tip of driven gear) in areas of high sliding. It is represented by a function of the quotient of the approach contact ratio, ε_f, over the recess contact ratio, ε_a (see Figure 3).

$$X_Q = 1{,}00 \qquad \text{for } \frac{\varepsilon_f}{\varepsilon_a} \le 1{,}5 \tag{26}$$

$$X_Q = 1{,}40 - \frac{4}{15} \cdot \frac{\varepsilon_f}{\varepsilon_a} \qquad \text{for } 1{,}5 < \frac{\varepsilon_f}{\varepsilon_a} < 3 \tag{27}$$

$$X_Q = 0{,}60 \qquad \text{for } 3 \le \frac{\varepsilon_f}{\varepsilon_a} \tag{28}$$

When the pinion drives the wheel:

$$\varepsilon_f = \varepsilon_2 \text{ and } \varepsilon_a = \varepsilon_1 \tag{29}$$

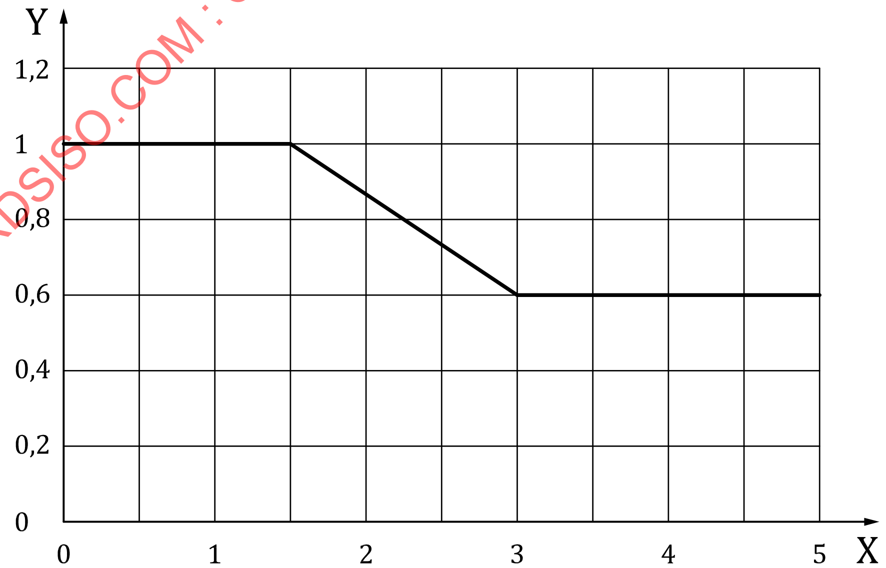
When the pinion is driven by the wheel:

$$\varepsilon_f = \varepsilon_1 \text{ and } \varepsilon_a = \varepsilon_2 \tag{30}$$

$$\varepsilon_1 = \frac{z_1}{2 \cdot \pi} \cdot \left[\sqrt{\left(\frac{d_{a1}}{d_{b1}}\right)^2 - 1} - \tan \alpha_t' \right] \tag{31}$$

$$\varepsilon_2 = \frac{|z_2|}{2 \cdot \pi} \cdot \left[\sqrt{\left(\frac{d_{a2}}{d_{b2}}\right)^2 - 1} - \tan \alpha_t' \right] \tag{32}$$

When tooth tips are chamfered or rounded, the tip diameter, d_a, shall be substituted by the effective tip diameter, d_{Na}, at which the recess is starting.



Key

X approach contact ratio, ε_f/recess contact ratio, ε_a

Y approach factor, X_Q

Figure 3 — Approach factor, X_Q

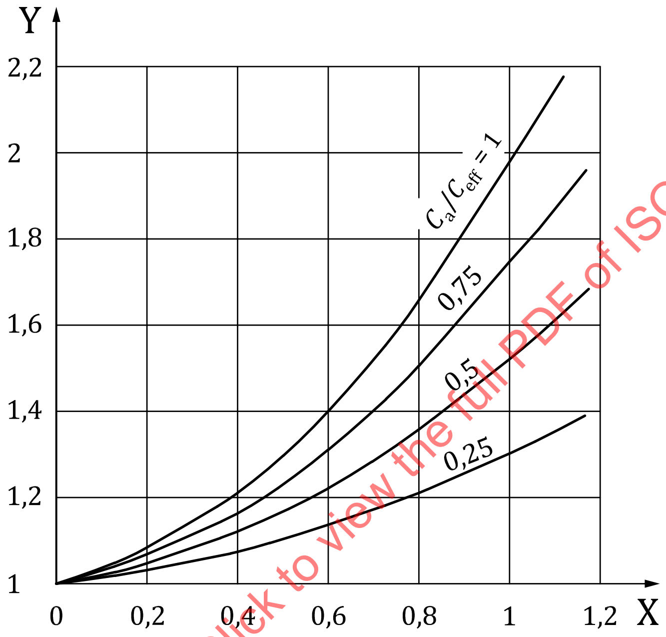
6.1.13 Tip relief factor, X_{Ca}

Elastic deformations of loaded teeth can cause high impact loads at tooth tips in areas of relatively high sliding. The tip relief factor, X_{Ca}, takes account of the influences of profile modifications on such loads. X_{Ca} is a relative tip relief factor which depends on the actual amount of tip relief, C_a, related to the effective tip relief due to elastic deformation, C_{eff} (see Figure 4).

The curves in Figure 4 can be approximated by Formula (33).

$$X_{Ca} = 1 + \left[0{,}06 + 0{,}18 \cdot \left(\frac{C_a}{C_{eff}}\right)\right] \cdot \varepsilon_{max} + \left[0{,}02 + 0{,}69 \cdot \left(\frac{C_a}{C_{eff}}\right)\right] \cdot \varepsilon_{max}^2 \tag{33}$$

where ε_{max} is the maximum value, ε_1 or ε_2.



Key

X maximum value, ε_{max}, of ε_1 or ε_2

Y tip relief factor, X_{ca}

Figure 4 — Tip relief factor, X_{Ca}, due to experimental data[11][12]

The nominal amount of tip relief, C_a, to be introduced into Formula (33) depends on the actual values of tip relief, C_a and, C_{a2}, the effective tip relief, C_{eff}, the ratio of addendum contact ratios and the direction of power flow.

When the pinion drives the wheel and $\varepsilon_1 > 1{,}5 \cdot \varepsilon_2$ or the pinion is driven by the wheel and $\varepsilon_1 > (2/3) \cdot \varepsilon_2$,

$$C_a = C_{a1} \quad \text{for} \quad C_{a1} \le C_{eff} \tag{34}$$

$$C_a = C_{eff} \quad \text{for} \quad C_{a1} > C_{eff} \tag{35}$$

When the pinion drives the wheel and $\varepsilon_1 \le 1{,}5 \cdot \varepsilon_2$ or the pinion is driven by the wheel and $\varepsilon_1 < (2/3) \cdot \varepsilon_2$,

$$C_a = C_{a2} \quad \text{for} \quad C_{a2} \le C_{eff} \tag{36}$$

$$C_a = C_{eff} \quad \text{for} \quad C_{a2} > C_{eff} \tag{37}$$

where C_{eff} is the effective tip relief, that amount of tip relief which compensates for the elastic deformation of the teeth in single pair contact.

$$C_{eff} = \frac{K_A \cdot F_t}{b \cdot c'} \quad \text{for spur gears} \tag{38}$$

$$C_{eff} = \frac{K_A \cdot F_t}{b \cdot c_\gamma} \quad \text{for helical gears} \tag{39}$$

where b is the facewidth.

If the facewidth of the pinion is different from that of the wheel, the smaller is determinant.

Tip relief, as described above, applies to gears of ISO tolerance class 6 or better, in accordance with ISO 1328-1. For less accurate gears, X_{Ca} is to be set equal to 1 (see also ISO 6336-1).

6.1.14 Contact ratio factor, X_ε

The contact ratio factor, X_ε, converts the flash temperature value at the pinion tooth tip, when load sharing is neglected, to a mean value of the flash temperature over the path of contact. The contact ratio factor can be expressed in terms of addendum contact ratios, ε_1 and ε_2, and their sum ε_α. The formulae for X_ε are based on an assumed linearity of the flash temperature over the path of contact. Possible errors due to this approach are unlikely to exceed 5 % and are always on the safe side.

For $\varepsilon_\alpha < 1, \varepsilon_1 < 1, \varepsilon_2 < 1$:

$$X_\varepsilon = \frac{1}{2 \cdot \varepsilon_\alpha \cdot \varepsilon_1} \cdot \left(\varepsilon_1^2 + \varepsilon_2^2\right) \tag{40}$$

For $1 \le \varepsilon_\alpha < 2, \varepsilon_1 < 1, \varepsilon_2 < 1$ (see Figure 5):

$$X_\varepsilon = \frac{1}{2 \cdot \varepsilon_\alpha \cdot \varepsilon_1} \cdot \left[0{,}70 \cdot \left(\varepsilon_1^2 + \varepsilon_2^2\right) - 0{,}22 \cdot \varepsilon_\alpha + 0{,}52 - 0{,}60 \cdot \varepsilon_1 \cdot \varepsilon_2\right] \tag{41}$$

For $1 \le \varepsilon_\alpha < 2, \varepsilon_1 \ge 1, \varepsilon_2 < 1$:

$$X_\varepsilon = \frac{1}{2 \cdot \varepsilon_\alpha \cdot \varepsilon_1} \cdot \left(0{,}18 \cdot \varepsilon_1^2 + 0{,}70 \cdot \varepsilon_2^2 + 0{,}82 \cdot \varepsilon_1 - 0{,}52 \cdot \varepsilon_2 - 0{,}30 \cdot \varepsilon_1 \cdot \varepsilon_2\right) \tag{42}$$

For $1 \le \varepsilon_\alpha < 2, \varepsilon_1 < 1, \varepsilon_2 \ge 1$:

$$X_\varepsilon = \frac{1}{2 \cdot \varepsilon_\alpha \cdot \varepsilon_1} \cdot \left(0{,}70 \cdot \varepsilon_1^2 + 0{,}18 \cdot \varepsilon_2^2 - 0{,}52 \cdot \varepsilon_1 + 0{,}82 \cdot \varepsilon_2 - 0{,}30 \cdot \varepsilon_1 \cdot \varepsilon_2\right) \tag{43}$$

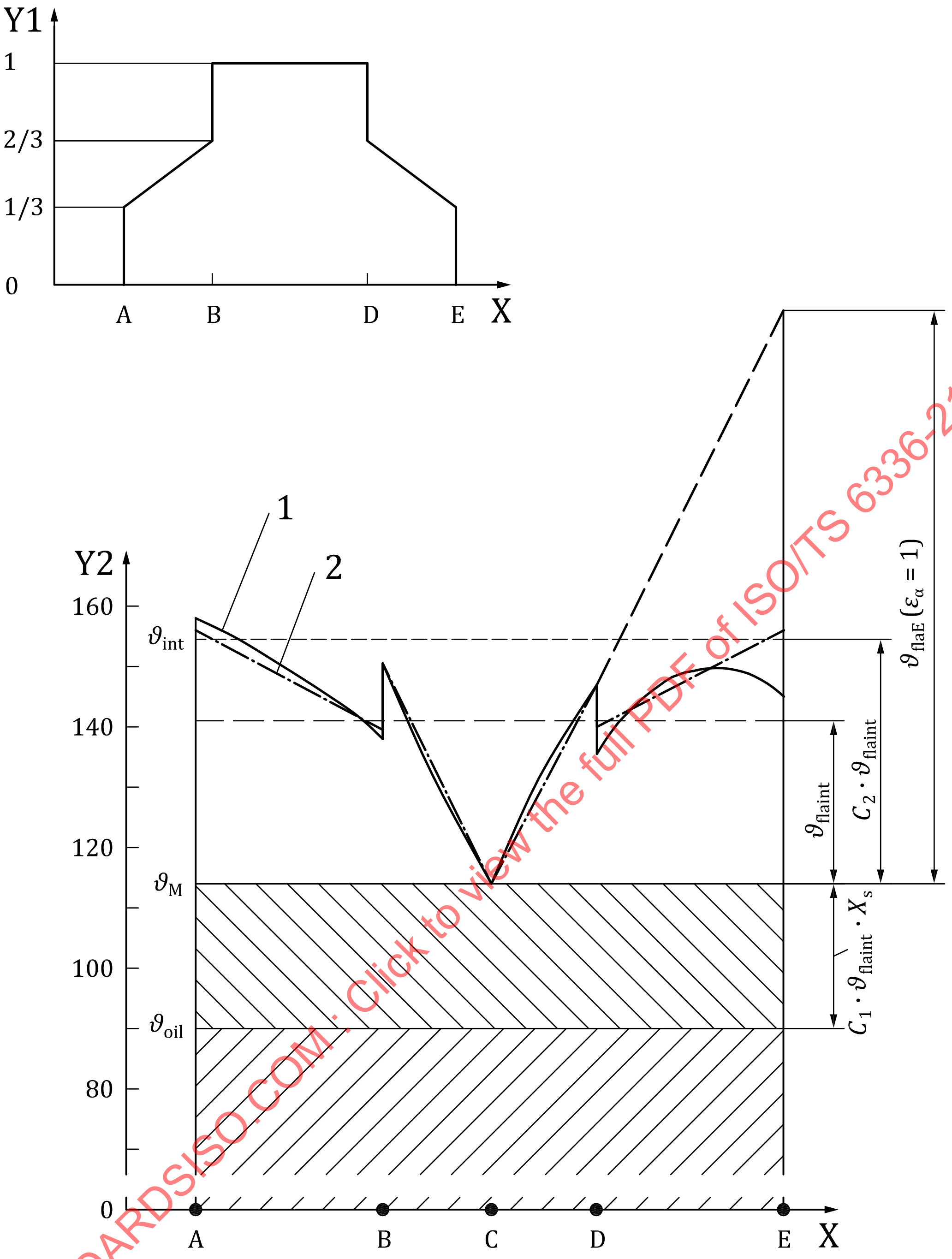
For $2 \le \varepsilon_\alpha < 3, \varepsilon_1 \ge \varepsilon_2$ (see Figure 6):

$$X_\varepsilon = \frac{1}{2 \cdot \varepsilon_\alpha \cdot \varepsilon_1} \cdot \left(0{,}44 \cdot \varepsilon_1^2 + 0{,}59 \cdot \varepsilon_2^2 + 0{,}30 \cdot \varepsilon_1 - 0{,}30 \cdot \varepsilon_2 - 0{,}15 \cdot \varepsilon_1 \cdot \varepsilon_2\right) \tag{44}$$

For $2 \le \varepsilon_\alpha < 3, \varepsilon_1 < \varepsilon_2$ (see Figure 6):

$$X_\varepsilon = \frac{1}{2 \cdot \varepsilon_\alpha \cdot \varepsilon_1} \cdot \left(0{,}59 \cdot \varepsilon_1^2 + 0{,}44 \cdot \varepsilon_2^2 - 0{,}30 \cdot \varepsilon_1 + 0{,}30 \cdot \varepsilon_2 - 0{,}15 \cdot \varepsilon_1 \cdot \varepsilon_2\right) \tag{45}$$

$$\varepsilon_\alpha = \varepsilon_1 + \varepsilon_2 \tag{46}$$



Key

X path of contact

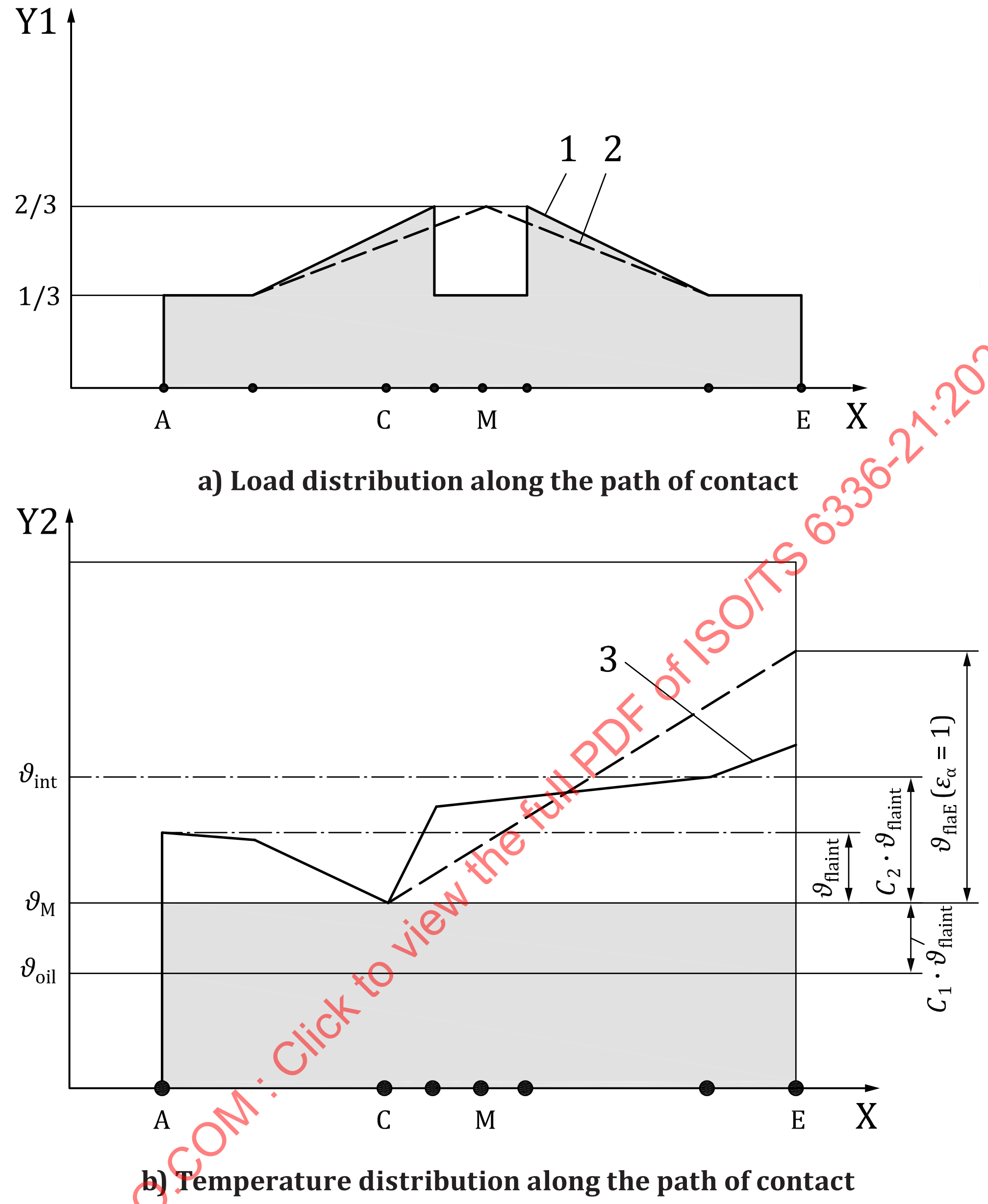
Y1 load

Y2 temperature, °C

1 distribution of contact temperature

2 approximated distribution

Figure 5 — Load and temperature distribution for $1{,}0 \leq \varepsilon_\alpha < 2{,}0$



Key

X path of contact
Y1 load
Y2 temperature
1 real load distribution
2 approximated load distribution
3 distribution of contact temperature

Figure 6 — Load and temperature distribution for $2{,}0 \leq \varepsilon_\alpha < 3{,}0$

6.2 Scuffing integral temperature

6.2.1 General

The scuffing integral temperature is the limiting value of the temperature at which scuffing occurs. It can be calculated on the basis of test results.

This method is valid for all types of oils (pure mineral oils, EP-oils, synthetic oils) for which the scuffing load capacity has been determined in a test gear (suitable tests are, for example, the FZG-test A/8,3/90, the FZG L-42 test, the Ryder gear oil test or the IAE gear oil test), or by an actual case of damage.

The scuffing temperature shall be corrected when material and heat treatment of the test gear are not identical with that of the actual gear, as the limiting temperature is a function of the material-oil system.

6.2.2 Scuffing integral temperature, ϑ_{intS}

6.2.2.1 General

According to the integral temperature postulate, gears are likely to scuff when the mean flank temperature exceeds a value termed the scuffing integral temperature number. This number is assumed to be characteristic for the lubricant and gear material combination of a gear pair and shall be determined by testing a similar lubricant and gear material combination.

A scuffing integral temperature number can be derived from the results of any gear oil scuffing test by entering the test data into the formulae in 6.1 and 6.2. Thus, scuffing integral temperature numbers for any oil, straight mineral, EP or synthetic, can be evaluated.

6.2.2.2 Calculation of the scuffing integral temperature

The approximate scuffing integral temperature number of heat or surface-treated gear steels in combination with a mineral oil, can be derived from that of a combination of gear steels with other heat or surface treatments and the same lubricant.

$$\vartheta_{\text{intS}} = \vartheta_{\text{MT}} + X_{\text{WrelT}} \cdot C_2 \cdot \vartheta_{\text{flaintT}} \tag{47}$$

where C_2 = 1,5; derived from experiments.

6.2.2.3 Determination of ϑ_{MT}, $\vartheta_{\text{flaintT}}$ from test results

Figure 7 shows the diagram for mineral oils in case that the scuffing load capacity is determined in an FZG-Test A/8,3/90 in accordance with DIN 51354, in a Ryder[6] or an FZG-Ryder test[7] and in an FZG L-42 test[8].

For computer calculations the diagrams in Figures 7 to 9 can be approximated by the following formulae:

a) For the FZG test A/8,3/90:

$$\vartheta_{\text{MT}} = 80 + 0{,}23 \cdot T_{1\text{T}} \cdot X_{\text{L}} \tag{48}$$

$$\vartheta_{\text{flaintT}} = 0{,}2 \cdot T_{1\text{T}} \cdot \left(\frac{100}{\nu_{40}} \right)^{0{,}02} \cdot X_{\text{L}} \tag{49}$$

$$T_{1\text{T}} = 3{,}726 \cdot (\text{FZG load stage})^2 \tag{50}$$

b) For the Ryder and the FZG-Ryder test R/46,5/74:

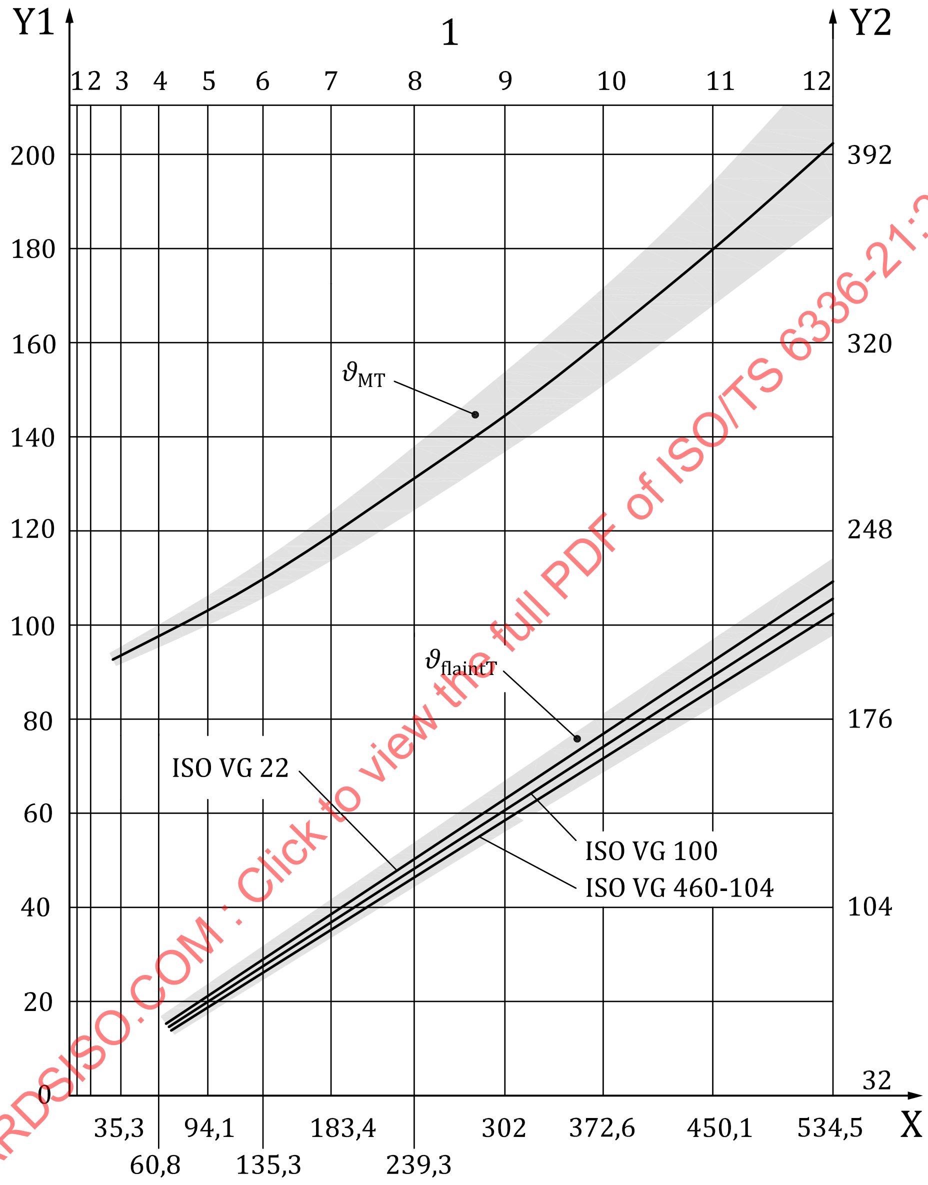
$$\vartheta_{\text{MT}} = 90 + 0{,}012\,5 \cdot \left(\frac{F_{\text{bt}}}{b} \right)_{\text{T}} \cdot X_{\text{L}} \tag{51}$$

with F_{bt}/b in lb/in.

c) For the FZG L-42 test 141/19,5/110:

$$\vartheta_{MT} = 110 + 0{,}02 \cdot T_{1T} \cdot X_L \tag{52}$$

$$\vartheta_{flaintT} = 0{,}48 \cdot T_{1T} \cdot \left(\frac{100}{\nu_{40}}\right)^{0{,}02} \cdot X_L \tag{53}$$



Key

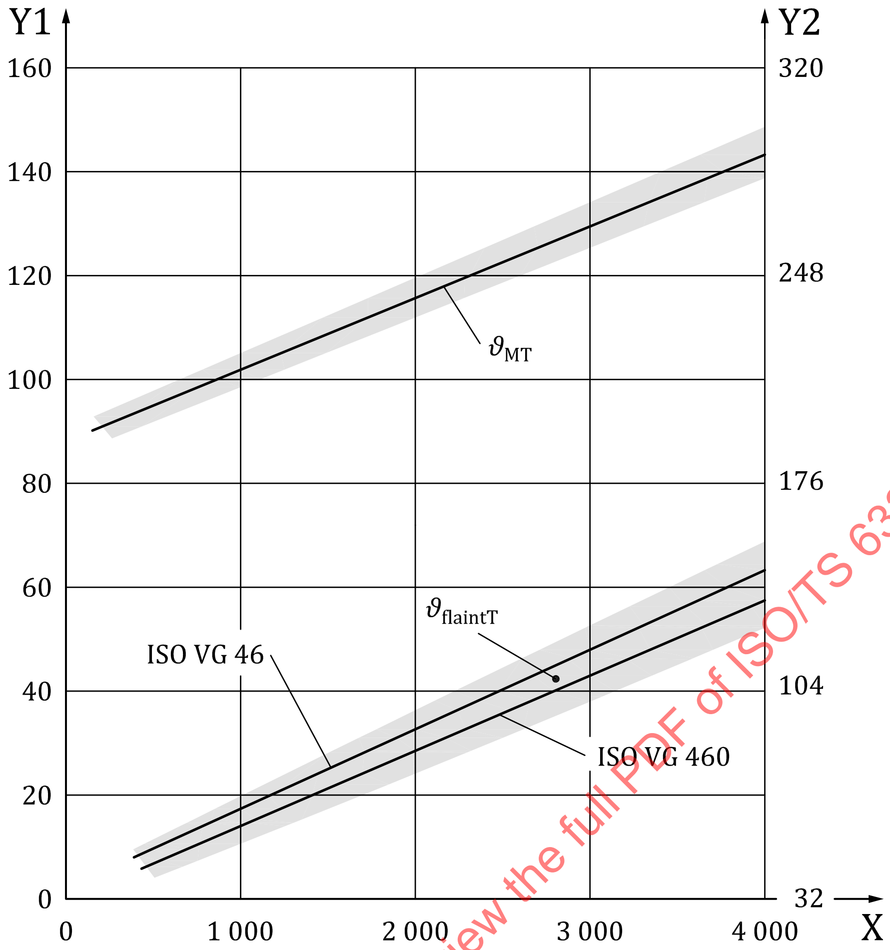
X pinion test torque T_{1T}, N·m

Y1 temperature, in °C

Y2 temperature, in °F

1 load stages FZG-test

Figure 7 — Scuffing temperature, ϑ_{intS}, for the FZG test A/8,3/90



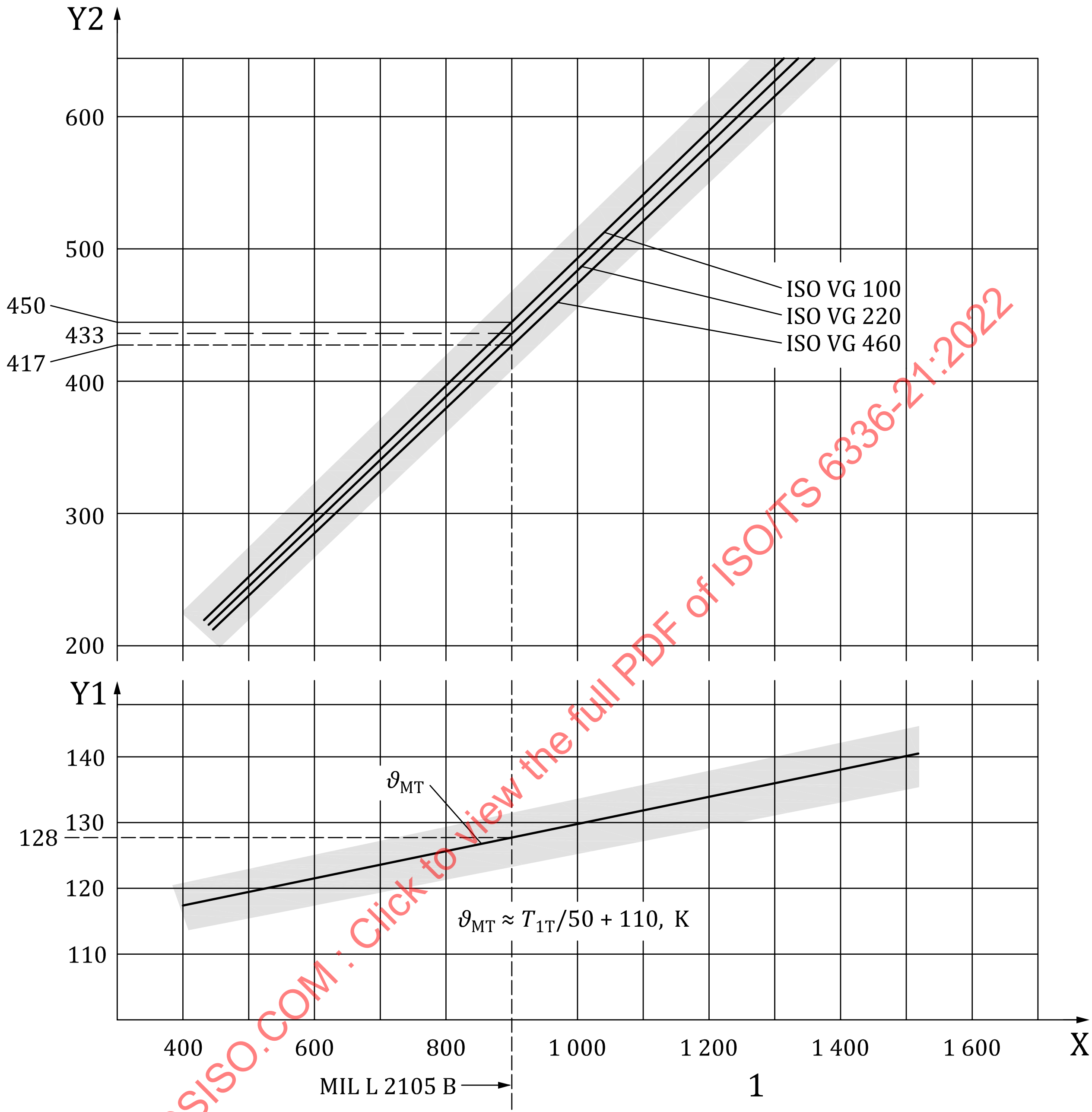
Key

X normal load per facewidth, $(F_{bt}/b)_T$, in ppi

Y1 temperature, in °C

Y2 temperature, in °F

Figure 8 — Scuffing temperature, ϑ_{intS}, for the Ryder and the FZG-Ryder gear test R/46,5/74



Key

X scuffing torque of test pinion T_{1T}, in N·m

Y1 test bulk temperature, ϑ_{MT}, in K

Y2 mean flash temperature of the test gear, $\vartheta_{flaintT}$, in K

1 test torque of pinion

NOTE $\vartheta_{flaintT} \approx 0{,}75 T_{1T}^{\ 0{,}95} \eta_{MT}^{\ -0{,}05}$, K

Approximation

$\eta_{MT} \approx \eta_{40} \left(40 / \vartheta_{MT}\right)^{2{,}85}$

Figure 9 — Scuffing temperature, ϑ_{intS}, for the FZG L-42 test 141/19,5/110

6.2.3 Relative welding factor, X_{WrelT}

The relative welding factor, X_{WrelT}, is an empirical factor for the influence of the heat or surface treatment on the scuffing integral temperature, as shown in Formula (54):

$$X_{\mathrm{WrelT}} = \frac{X_{\mathrm{W}}}{X_{\mathrm{WT}}} \tag{54}$$

where

X_{WT} is 1 for the FZG gear test, the Ryder gear test and the FZG L-42 test;

X_{W} is the welding factor of the actual gear material as given in Table 4.

Table 4 — Welding factor, X_{W}

Gear material	X_{W}
Through-hardened steel	1,00
Phosphated steel	1,25
Copper-plated steel	1,50
Bath and gas nitrided steel	1,50
Case carburized steel:	
— average austenite content less than 10 %	1,15
— average austenite content 10 % to 20 %	1,00
— average austenite content greater than 20 % to 30 %	0,85
Austenitic steel (stainless steel)	0,45

Annex A
(informative)

Examples

Verifying the accuracy of the integral temperature method, the scuffing resistance of the following gear sets is calculated by using the methods according to this document. The examples contain cylindrical gear drives, with centre distances between a = 22,07 mm and a = 2 419,63 mm. The module range includes modules from m = 1,25 mm up to m = 20 mm. Some of the selected gear units are damaged by a scuffing failure, or near to the scuffing limit (borderline scuffing). In other gear drives, no scuffing failure is observed. The data of the gear units and the results of the scuffing calculation are given in Tables A.1 to A.8.

Table A.1 — Helical gear: Turbine Gear (No. 3 from the Michaelis dissertation)

Description		Symbol	Unit	Value
Number of teeth	pinion	z_1	—	73
	gear	z_2	—	325
Operating centre distance		a	mm	1 419,00
Normal module		m_n	mm	7,000
Normal pressure angle		α_n	°	20,00
Helix angle at standard PD		β	°	11,00
Profile shift factor	pinion	x_1	—	0,010 0
Net facewidth		b	mm	280,00
Outside diameter	pinion	d_{a1}	mm	534,40
	gear	d_{a2}	mm	2 331,00
Tip relief	pinion	C_{a1}	µm	0
	gear	C_{a2}	µm	0
Index of driving gear		—	—	2
Transmitted power		P	kW	10 295
Pinion speed		n_1	min^{-1}	4 450
Flank surface roughness		Ra	µm	2,00
Tooth root surface roughness		Rz	µm	-
Oil temperature		ϑ_{oil}	°C	40
Lubricant kinematic viscosity at 40 °C		ν_{40}	mm^2/s	32
Scuffing torque in FZG standard test A/8,3/90 according to DIN 51354		T_{1T}	Nm	239
Lubrication factor		X_S	—	1,2
Relative material factor		X_{WrelT}	—	1,00
Run-in factor		X_E	—	1,0
Application factor		K_A	—	1,20
Dynamic factor		K_v	—	1,15
Face load factor		$K_{B\beta}$	—	1,20
Transverse load factor		$K_{B\alpha}$	—	1,10
Coefficient of friction		μ_{mC}	—	0,023
Bulk temperature		ϑ_M	°C	45,6
Integral temperature		ϑ_{int}	°C	55,5
Scuffing safety factor		S_{intS}	—	3,8
Observed failures		No scuffing		

Table A.2 — Helical gear: Steel Mill Gear (No. 5 from the Michaelis dissertation)

Description		Symbol	Unit	Value
Number of teeth	pinion	z_1	—	28
	gear	z_2	—	28
Operating centre distance		a	mm	580,00
Normal module		m_n	mm	20,000
Normal pressure angle		α_n	°	20,00
Helix angle at standard PD		β	°	10,00
Profile shift factor	pinion	x_1	—	0,303 5
Net facewidth		b	mm	330,00
Outside diameter	pinion	d_{a1}	mm	619,20
	gear	d_{a2}	mm	619,20
Tip relief	pinion	C_{a1}	µm	0
	gear	C_{a2}	µm	0
Index of driving gear		—	—	1
Transmitted power		P	kW	2 200
Pinion speed		n_1	min^{-1}	150
Flank surface roughness		Ra	µm	1,50
Tooth root surface roughness		Rz	µm	-
Oil temperature		ϑ_{oil}	°C	32
Lubricant kinematic viscosity at 40 °C		ν_{40}	mm^2/s	220
Scuffing torque in FZG standard test A/8,3/90 according to DIN 51354		T_{1T}	Nm	239
Lubrication factor		X_S	—	1,2
Relative material factor		X_{WrelT}	—	1,00
Run-in factor		X_E	—	1,0
Application factor		K_A	—	1,20
Dynamic factor		K_v	—	1,00
Face load factor		$K_{B\beta}$	—	1,20
Transverse load factor		$K_{B\alpha}$	—	1,00
Coefficient of friction		μ_{mC}	—	0,048
Bulk temperature		ϑ_M	°C	59,6
Integral temperature		ϑ_{int}	°C	109,0
Scuffing safety factor		S_{intS}	—	1,9
Observed failures		No scuffing		

Table A.3 — Helical gear: Machine Tool Gear (No. 11 from the Michaelis dissertation)

Description		Symbol	Unit	Value
Number of teeth	pinion	z_1	—	5
	gear	z_2	—	28
Operating centre distance		a	mm	22,07
Normal module		m_n	mm	1,250
Normal pressure angle		α_n	°	20,00
Helix angle at standard PD		β	°	20,00
Profile shift factor	pinion	x_1	—	0,350 0
Net facewidth		b	mm	10,00
Outside diameter	pinion	d_{a1}	mm	9,98
	gear	d_{a2}	mm	38,45
Tip relief	pinion	C_{a1}	µm	0
	gear	C_{a2}	µm	0
Index of driving gear		—	—	1
Transmitted power		P	kW	3,3
Pinion speed		n_1	min^{-1}	15 000
Flank surface roughness		Ra	µm	1,00
Tooth root surface roughness		Rz	µm	—
Oil temperature		ϑ_{oil}	°C	50
Lubricant kinematic viscosity at 40 °C		ν_{40}	mm^2/s	220
Scuffing torque in FZG standard test A/8,3/90 according to DIN 51354		T_{1T}	Nm	450
Lubrication factor		X_S	—	1,0
Relative material factor		X_{WrelT}	—	1,00
Run-in factor		X_E	—	1,0
Application factor		K_A	—	1,00
Dynamic factor		K_v	—	1,00
Face load factor		$K_{B\beta}$	—	1,00
Transverse load factor		$K_{B\alpha}$	—	1,00
Coefficient of friction		μ_{mC}	—	0,144
Bulk temperature		ϑ_M	°C	84,8
Integral temperature		ϑ_{int}	°C	159,4
Scuffing safety factor		S_{intS}	—	2,0
Observed failures		No scuffing		

Table A.4 — Helical gear: Marine Gear (No. 13 from the Michaelis dissertation)

Description		Symbol	Unit	Value
Number of teeth	pinion	z_1	—	21
	gear	z_2	—	87
Operating centre distance		a	mm	900,00
Normal module		m_n	mm	16,00
Normal pressure angle		α_n	°	20,00
Helix angle at standard PD		β	°	10,00
Profile shift factor	pinion	x_1	—	0,790 0
Net facewidth		b	mm	370,00
Outside diameter	pinion	d_{a1}	mm	394,50
	gear	d_{a2}	mm	1 465,50
Tip relief	pinion	C_{a1}	µm	0
	gear	C_{a2}	µm	0
Index of driving gear		—	—	1
Transmitted power		P	kW	4 412
Pinion speed		n_1	min^{-1}	520
Flank surface roughness		Ra	µm	2,00
Tooth root surface roughness		Rz	µm	—
Oil temperature		ϑ_{oil}	°C	60
Lubricant kinematic viscosity at 40 °C		ν_{40}	mm^2/s	150
Scuffing torque in FZG standard test A/8,3/90 according to DIN 51354		T_{1T}	Nm	450
Lubrication factor		X_S	—	1,2
Relative material factor		X_{WrelT}	—	1,00
Run-in factor		X_E	—	1,0
Application factor		K_A	—	1,30
Dynamic factor		K_v	—	1,05
Face load factor		$K_{B\beta}$	—	1,40
Transverse load factor		$K_{B\alpha}$	—	1,00
Coefficient of friction		μ_{mC}	—	0,058
Bulk temperature		ϑ_M	°C	105,1
Integral temperature		ϑ_{int}	°C	185,7
Scuffing safety factor		S_{intS}	—	1,7
Observed failures		Borderline scuffing		

Table A.5 — Helical gear: Steel Mill Gear (No. 16 from the Michaelis dissertation)

Description		Symbol	Unit	Value
Number of teeth	pinion	z_1	—	24
	gear	z_2	—	79
Operating centre distance		a	mm	700,00
Normal module		m_n	mm	12,00
Normal pressure angle		α_n	°	20,00
Helix angle at standard PD		β	°	27,00
Profile shift factor	pinion	x_1	—	0,550 0
Net facewidth		b	mm	175,00
Outside diameter	pinion	d_{a1}	mm	360,00
	gear	d_{a2}	mm	1 087,50
Tip relief	pinion	C_{a1}	µm	0
	gear	C_{a2}	µm	0
Index of driving gear		—	—	1
Transmitted power		P	kW	200
Pinion speed		n_1	min^{-1}	240
Flank surface roughness		Ra	µm	2,00
Tooth root surface roughness		Rz	µm	—
Oil temperature		ϑ_{oil}	°C	40
Lubricant kinematic viscosity at 40 °C		ν_{40}	mm^2/s	150
Scuffing torque in FZG standard test A/8,3/90 according to DIN 51354		T_{1T}	Nm	135
Lubrication factor		X_S	—	1,2
Relative material factor		X_{WrelT}	—	1,00
Run-in factor		X_E	—	1,0
Application factor		K_A	—	1,50
Dynamic factor		K_v	—	1,05
Face load factor		$K_{B\beta}$	—	1,40
Transverse load factor		$K_{B\alpha}$	—	1,00
Coefficient of friction		μ_{mC}	—	0,051
Bulk temperature		ϑ_M	°C	48,9
Integral temperature		ϑ_{int}	°C	64,9
Scuffing safety factor		S_{intS}	—	2,3
Observed failures		Borderline scuffing		

Table A.6 — Helical gear: Turbine Gear (No. 19 from the Michaelis dissertation)

Description		Symbol	Unit	Value
Number of teeth	pinion	z_1	—	43
	gear	z_2	—	44
Operating centre distance		a	mm	161,40
Normal module		m_n	mm	3,628
Normal pressure angle		α_n	°	20,00
Helix angle at standard PD		β	°	12,00
Profile shift factor	pinion	x_1	—	0,000 0
Net facewidth		b	mm	51,00
Outside diameter	pinion	d_{a1}	mm	166,75
	gear	d_{a2}	mm	170,50
Tip relief	pinion	C_{a1}	µm	40
	gear	C_{a2}	µm	40
Index of driving gear		—	—	1
Transmitted power		P	kW	195
Pinion speed		n_1	min^{-1}	3 900
Flank surface roughness		Ra	µm	0,75
Tooth root surface roughness		Rz	µm	—
Oil temperature		ϑ_{oil}	°C	70
Lubricant kinematic viscosity at 40 °C		ν_{40}	mm^2/s	68
Scuffing torque in FZG standard test A/8,3/90 according to DIN 51354		T_{1T}	Nm	140
Lubrication factor		X_S	—	1,2
Relative material factor		X_{WrelT}	—	1,00
Run-in factor		X_E	—	1,0
Application factor		K_A	—	1,20
Dynamic factor		K_v	—	1,00
Face load factor		$K_{B\beta}$	—	1,20
Transverse load factor		$K_{B\alpha}$	—	1,00
Coefficient of friction		μ_{mC}	—	0,360
Bulk temperature		ϑ_M	°C	77,7
Integral temperature		ϑ_{int}	°C	91,5
Scuffing safety factor		S_{intS}	—	1,7
Observed failures		Borderline scuffing		