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Enclosed gear drives for industrial applications

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Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

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

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

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

— type 3, when a technical committee has collected data of a different kind from that which is normally published as an International Standard ("state of the art", for example).

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 Technical Reports of type 3 do not necessarily have to be reviewed until the data they provide are considered to be no longer valid or useful.

ISO/TR 13593, which is a Technical Report of type 2, was prepared by Technical Committee ISO/TC 60, *Gears*.

This document is being issued in the Technical Report (type 2) series of publications (according to subclause G.3.2.2 of Part 1 of the ISO/IEC Directives, 1995) as a “prospective standard for provisional application” in the field of gearing because there is an urgent need for guidance on how standards in this field should be used to meet an identified need.

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

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

Annexes A to F of this Technical Report are for information only.

Enclosed gear drives for industrial applications

1 Scope

This Technical Report is applicable to enclosed speed reducers and increasers for industrial applications, where the designs include spur, helical, herringbone or double helical gears and their combination in single or multistage drives.

This Technical Report provides a method by which gear drive designs can be compared and selected. It is not intended to assure performance of assembled gear drive systems. It is intended for use by experienced gear designers capable of selecting reasonable values for the factors, based on performance knowledge of similar designs and the effects of such items as lubrication, deflection, manufacturing tolerances, metallurgy, residual stress and system dynamics. It is not intended for use by the engineering public at large.

Maintaining an acceptable temperature in the oil sump of an enclosed gear drive is critical to the life of the gear drive. Therefore, this Technical Report for enclosed gear drives considers not only the mechanical rating but also the thermal rating.

The rating methods and influences identified in this Technical Report are limited to enclosed drives of single and multiple stage designs where the pitch line velocities do not exceed 35 m/s and pinion speeds do not exceed 4 500 r/min. In this Technical Report, gear teeth rating is covered only as limited by tooth root bending and contact pressure.

This Technical Report does not cover the design and application of epicyclic drives. It is beyond the scope of this Technical Report to present a detailed analysis of efficiency.

Annexes A to F can be used to make a more detailed analysis of certain rating factors.

2 Normative references

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

ISO 76:1987, *Rolling bearings — Static load ratings*.

ISO 281:1990, *Rolling bearings — Dynamic load ratings and rating life*.

ISO 701, *International gear notation — Symbols for geometrical data*.

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

ISO 3448:1992, *Industrial liquid lubricants — ISO viscosity classification*.

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

ISO 6336-2:1996, *Calculation of load capacity of spur and helical gears — Part 2: Calculation of surface durability (pitting).*

ISO 6336-3:1996, *Calculation of load capacity of spur and helical gears — Part 3: Calculation of tooth bending strength*.

ISO 6336-5:1996, *Calculation of load capacity of spur and helical gears — Part 5: Strength and quality of materials*.

ISO 6743-6:1990, *Lubricants, industrial oils and related products (class L) — Classification — Part 6: Family C (Gears).*

ISO 8579-1, *Acceptance code for gears — Part 1: Determination of airborne sound power levels emitted by gear units*.

ISO 8579-2, *Acceptance code for gears — Part 2: Determination of mechanical vibrations of gear units during acceptance testing*.

ISO 8821:1989, *Mechanical vibration — Balancing — Shaft and fitment key convention*.

ISO 9085:—[1], *Calculation of load capacity of spur and helical gears — Application for industrial gears*.

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

ISO 12925-1:1996, *Lubricants, industrial oils and related products (class L) — Family C (gears) — Part 1: Specifications for lubricants for enclosed gear systems*.

3 Symbols, terms and definitions

NOTE The symbols, terms, and definitions contained in this document may vary from those used in other ISO standards. Users of this Technical Report should verify that they are using these symbols and terms in the manner indicated herein.

3.1 Symbols

For the purposes of this Technical Report, the symbols given in Table 1 apply.

[1] To be published.

Table 1 — Symbols used in equations

Symbol	Meaning	Units	Where first used	Subclause
A_C	surface area of gear drive	m^2	Eq 40	7.4.3
A_R	fit holding capacity	N	Eq 21	5.6.3
A_s	stress cross section of fastener	mm^2	Eq 27	5.7.2
a_1	life adjustment factor for reliability	—	Eq 3	5.4.3.3
B_A	altitude factor	—	Eq 41	7.5
B_D	operation time factor	—	Eq 41	7.5
B_{ref}	ambient temperature factor	—	Eq 41	7.5
B_T	non-standard oil sump temperature factor	—	Eq 41	7.5
B_V	ambient air velocity factor	—	Eq 41	7.5
b_k	width of key	mm	Eq 17	5.6.2
D_f	nominal diameter of threaded fastener	mm	Eq 28	5.7.2
d_{he}	outside diameter of hub	mm	Eq 24	5.6.3
d_{hi}	inside diameter of hub	mm	Eq 24	5.6.3
d_{max}	maximum nominal fastener diameter	mm	Table 3	5.7.2
d_{sh}	shaft diameter	mm	Eq 16	5.6.2
d_{she}	shaft outside diameter	mm	Eq 6	5.5.2
d_{shi}	shaft inside diameter	mm	Eq 6	5.5.2
E_H	modulus of elasticity for hub material	N/mm^2	Eq 23	5.6.3
E_S	modulus of elasticity for shaft material	N/mm^2	Eq 23	5.6.3
F_A	applied tensile load	N	Eq 31	5.7.4
F_M	fastener tensile preload	N	Eq 27	5.7.2
f_L	load peak frequency factor	—	Eq 20	5.6.3
h_k	height of key	mm	Eq 16	5.6.2
I	actual or minimum possible interference fit	mm	Eq 23	5.6.3
i	number of keys	—	Eq 16	5.6.2
K_A	application factor	—	9.5.1	9.5.1
K_J	joint stiffness factor	—	Eq 30	5.7.3
K_{sf}	selection factor	—	Eq 1	4.5.3
K_{tc}	torque coefficient	—	Eq 29	5.7.2
k	heat transfer coefficient	$kW/(m^2 \cdot K)$	Eq 40	7.4.3
L	length of hub	mm	Eq 22	5.6.3
L_{na}	adjusted rating life at $100 - n = R$ % reliability	h	Eq 3	5.4.3.3
L_{10a}	rating life at basic 90 % reliability	h	Eq 3	5.4.3.3
l_g	length of fastener grip	mm	5.7.3	5.7.3
l_{tr}	bearing length of the key	mm	Eq 16	5.6.2
M	bending moment	Nm	Eq 7	5.5.2
M_A	fastener tightening torque	Nm	Eq 29	5.7.2
P_A	input power to gear drive	kW	Eq 34	7.4.1
P_B	bearing power loss	kW	Eq 38	7.4.2
P_H	pressure at common shaft/hub interface	N/mm^2	Eq 22	5.6.3
P_L	load-dependent power losses	kW	Eq 33	7.4.1
P_M	gear mesh power loss	kW	Eq 38	7.4.2
P_{mc}	minimum rated component power	kW	Eq 1	4.5.1
P_N	non-load-dependent power losses	kW	Eq 33	7.4.1
P_n	nominal power of the driven machine or the driving machine	kW	Eq 1	4.5.3

Table 1 — Symbols used in equations

Symbol	Meaning	Units	Where first used	Subclause
P_P	oil pump power consumption	kW	Eq 39	7.4.2
P_Q	heat dissipation of gear drive	kW	Eq 32	7.4.1
P_S	oil seal power loss	kW	Eq 39	7.4.2
P_T	thermal power rating	kW	Eq 37	7.4.1
P_Thm	modified application thermal power rating	kW	Eq 41	7.5
P_V	total power loss	kW	Eq 32	7.4.1
P_WB	bearing windage and oil churning power loss	kW	Eq 39	7.4.2
P_WG	gear windage and oil churning power loss	kW	Eq 39	7.4.2
p_f	fastener thread pitch	mm	Eq 28	5.7.2
R	reliability level	percent	Eq 4	5.4.3.3
R_e	tensile strength of the key material	$\mathrm{N/mm^2}$	Eq 18	5.6.2
$S_\mathrm{F\ min}$	minimum safety factor for bending strength	—	9.5.1	9.5.1
$S_\mathrm{H\ min}$	minimum safety factor for pitting resistance	—	9.5.1	9.5.1
T	shaft torque	Nm	Eq 6	5.5.2
T_a	allowable torque based on the lesser of T_C and T_s	Nm	5.6.2	5.6.2
T_C	allowable torque based on the allowable compressive stress	Nm	Eq 16	5.6.2
T_max	maximum torque	Nm	Eq 20	5.6.3
T_mc	minimum rated component torque	Nm	Eq 2	4.5.3
T_n	nominal torque of the driven machine or the driving machine	Nm	Eq 2	4.5.3
T_R	torque carried by friction in the interface of shaft and hub	Nm	Eq 21	5.6.3
T_s	allowable torque based on the allowable key shear stress	Nm	Eq 17	5.6.2
t_k	shaft keyway depth	mm	Eq 16	5.6.2
Y_NT	life factor for bending strength	—	9.5.1	9.5.1
Z_NT	life factor for pitting resistance	—	9.5.1	9.5.1
β_τ	torsional notch factor	—	Eq 10	5.5.3
β_σ	bending notch factor	—	Eq 12	5.5.3
ΔT	temperature differential	K	Eq 40	7.4.3
φ	share of the load	—	Eq 16	5.6.2
η	overall drive efficiency	percent	Eq 36	7.4.1
μ	coefficient of friction	—	Eq 22	5.6.3
ρ_H	Poisson's ratio for hub material	—	Eq 23	5.6.3
ρ_S	Poisson's ratio for shaft material	—	Eq 23	5.6.3
σ_B	material tensile strength	$\mathrm{N/mm^2}$	Eq 10	5.5.3
σ_b	calculated bending shaft stress	$\mathrm{N/mm^2}$	Eq 7	5.5.2
σ_ba	allowable bending stress	$\mathrm{N/mm^2}$	Eq 12	5.5.3
σ_f	calculated tensile stress in fastener	$\mathrm{N/mm^2}$	Eq 31	5.7.4
σ_fa	allowable tensile stress of fastener	$\mathrm{N/mm^2}$	Eq 30	5.7.3
σ_M	preload tensile stress, recommended	$\mathrm{N/mm^2}$	Eq 26	5.7.2
$\sigma_\mathrm{p0,2}$	fastener 0,2 % offset yield strength	$\mathrm{N/mm^2}$	Eq 26	5.7.2
σ_s	calculated torsional shaft stress	$\mathrm{N/mm^2}$	Eq 6	5.5.2
σ_sa	allowable torsional stress	$\mathrm{N/mm^2}$	Eq 10	5.5.3
σ_SC	allowable compressive stress	$\mathrm{N/mm^2}$	Eq 16	5.6.2
τ_ps	allowable shear stress	$\mathrm{N/mm^2}$	Eq 17	5.6.2

3.2 Terms and definitions

For the purposes of this Technical Report, the following terms and definitions apply.

3.2.1
gear unit rating
overall mechanical power rating of all static and rotating elements within the enclosed drive, as determined by the minimum rated component power, P_{mc} (weakest part, whether determined by gear teeth, shafts, bolting, housing, etc.)

3.2.2
thermal rating
maximum power that can be continuously transmitted through an enclosed gear drive without exceeding a specified oil sump temperature

NOTE The thermal rating equals or exceeds the actual service transmitted power. Selection factors are not used when determining thermal requirements, see 7.1.

4 Application and design considerations

4.1 Application limitations

In this Technical Report, the gear unit rating, as defined, is the mechanical capacity (selection factor, $K_{sf} = 1{,}0$) of the gear drive components. In some applications it may be necessary to select a gear drive with an increased mechanical rating in order to accommodate adverse effects of environmental conditions, thermal capacity of the drive, external loading or any combination of these factors.

4.2 Rating factors

The allowable stress numbers in this Technical Report are maximum allowed values. Some latitude based upon experience is permissible in the selection of specific factors within this Technical Report. Less conservative values for other rating factors in this Technical Report shall not be used.

4.3 Metallurgy

The factors for gears affected by material conditions and quality are defined in ISO 6336-5.

4.4 System analysis

The system of connected rotating parts shall be compatible, free from critical speeds, torsional or other types of vibration within the specified operating speed range, no matter how induced. The enclosed gear drive designer or manufacturer is not responsible for this analysis, unless agreed to in the purchase contract.

4.5 Gear unit rating

4.5.1 Unit rating application

The gear unit rating is the overall mechanical power rating of all static and rotating elements within the enclosed drive. The minimum rated component power, P_{mc} (weakest part, whether determined by gear teeth, shafts, bolting, housing, etc.), of the enclosed drive determines the gear unit rating. The load histogram for determining the gear unit rating shall consist of 10 000 cycles at 200 % load plus 10 000 h at 100 % load. The gear unit rating shall also include the effects of the allowable overhung load at a specified distance from the end of the gearbox where the overhung load is applied.

NOTE It is the responsibility of the user to specify peak load conditions so that the drive can be selected such that the peak torque does not exceed that specified in 4.6.

Unity selection factor ($K_{sf} = 1{,}0$) is used in determining the gear unit rating. Refer to clause 9 for a discussion of the selection factor, K_{sf}.

4.5.2 Gear unit rating requirements

The gear unit rating implies that all items within the unit have been designed to meet or exceed the unit rating. Gear and pinion ratings shall be in accordance with the bending strength and pitting resistance ratings as specified in 5.2.

4.5.3 Application of gear unit rating

The required gear unit rating of an enclosed drive is a function of the application and assessment of variable factors that affect the overall rating. These factors include environmental conditions, severity of service and life. Refer to clause 9 for further explanation.

The application of the enclosed drive requires that its unit rating meet the requirements of the actual service conditions. This is accomplished by the proper selection of a selection factor, K_{sf}, based on field data or experience.

The values shown in annex A may be used as a guide. The gear unit rating required for the considered application is then obtained by satisfying:

$$P_{\mathrm{mc}} \geqslant P_{\mathrm{n}} K_{\mathrm{sf}} \tag{1}$$

where

P_{n} is the nominal power of the driven machine or the driving machine. See clause 9 and annex A.

Similarly, when rating by torques:

$$T_{\mathrm{mc}} \geqslant T_{\mathrm{n}} K_{\mathrm{sf}} \tag{2}$$

If the nominal power or the nominal torque of the driven machine is used for the gear unit rating and $P_{\mathrm{n\ driver}}$ is greater than $P_{\mathrm{n\ driven\ machine}}$, the maximum torque appearing in the whole system should be checked. During acceleration (or at other times) the maximum torque should not exceed 200 % of the nominal torque of the driven machine, see 4.6.

4.6 Momentary overloads

When the enclosed drive is subjected to momentary overloads, direct on-line motor starts, braking, stall conditions and low-cycle fatigue, the conditions should be evaluated to assure that the strength limitations of any component are not exceeded.

With respect to the gear bending strength for momentary overloads, the maximum allowable stress is determined by the allowable fatigue limitations of the material. Shaft, bearing and housing deflections have a significant effect on gear mesh alignment during momentary overloads. The enclosed drive shall be evaluated to assure that the reactions to momentary overloads do not result in excessive misalignment causing localized high stress concentrations and/or permanent deformation. In addition, the effects of external loads such as overhung, transverse and thrust loads shall be evaluated.

Gear drives rated to this Technical Report shall be able to accommodate peak loads whose magnitude does not exceed 200 % of P_{mc} applied for a number of stress cycles not exceeding 10 000. The minimum face load factor, $K_{\mathrm{H}\beta}$, determined for 100 % load applies to the analysis at 200 %.

4.7 Efficiency estimate

When an efficiency estimate of the enclosed drive is calculated, it should be determined based on the transmitted power and specified operating conditions. The estimate should include the effects of the components within the enclosed drive and shaft driven accessories agreed to by manufacturer and user. Unless specifically agreed to between the user and manufacturer, the prime mover, couplings, external driven loads, motor driven accessories, etc., are not included in the enclosed drive efficiency estimate. See clause 7 for calculations.

4.8 Reverse loading

The effect of torque reversals on an enclosed drive is taken into account by choosing an adequate selection factor for the considered application, e.g. travel drive. In a detailed rating analysis, the effect of reverse loading may be considered alternatively at component level.

5 Components

5.1 Rating considerations

The components of a gear drive shall be designed with due consideration for all loads likely to be encountered during operation. These include not only the torque loads imposed on the components through the gearing, but also external loads, i.e. overhung loads, external thrust loads, dynamic loads such as from cast overhung pinions, etc. These components shall also be designed to withstand any assembly forces which might exceed the operating loads. During the design process, the operating loads shall be considered to occur in the worst possible direction and in the worst possible loading combinations, including a 200 % momentary peak starting load.

Component rating shall be within the limits specified in this Technical Report. Where user requirements or specifications dictate different design criteria, such as higher bearing life, this shall be by contractual agreement.

Alternative component rating methods based on test data or field experience are allowed. The gear manufacturer shall indicate and document all modifications which are used.

Gear unit ratings may also include allowable overhung load values which are usually designated to act at a distance of one shaft diameter from the face of the housing or enclosure component. Stresses in related parts resulting from these overhung loads shall also be within limits set by this Technical Report.

For the purposes of this Technical Report, where component capacities are being determined, the calculations are specifically related to the gear unit rating as defined in 4.5.1.

NOTE A separate computation is required to relate the gear unit rating to application conditions.

5.2 Housing

The combined assembly of gears, shafts and bearings shall be enclosed by a housing of such design and construction as to provide the rigidity required for proper gear alignment. The housing shall maintain alignment under rated internal and external loading.

For housings with low speed centre distances greater than 460 mm, at least two reference surfaces should be machined parallel to the mounting surfaces for the purpose of levelling the gear drive.

5.3 Gears

5.3.1 Rating criteria

The fundamental formulas for enclosed gear drives shall be in accordance with ISO 9085.

The calculation method for each gear rating factor has the ability to be modified. The gear designer shall indicate all modifications to ISO 9085 that are used.

Pitting resistance is a function of the Hertzian contact (compressive) stresses between two curved surfaces or tooth surfaces and is proportional to the square root of the applied tooth load. Bending strength is measured in terms of the bending (tensile) stress in a cantilever plate and is directly proportional to this same load. The difference in nature of the stresses induced in the tooth surface areas and at the tooth root is reflected in a corresponding difference in allowable limits of contact and bending stress numbers for identical materials and load intensities.

The term "gear failure" is itself subjective and a source of considerable disagreement. One observer's "failure" may be another observer's "wearing-in". For a more complete discussion see ISO 10825.

5.3.1.1 Reverse loading

For gears which are reverse loaded on every cycle, see ISO 6336-5.

5.3.1.2 Localized yielding

This Technical Report does not extend to stress levels greater than those permissible at 10^3 cycles or less, since stresses in this range can exceed the elastic limit of the gear tooth in bending or in surface compressive stress. Depending on the material and the load imposed, a single stress cycle greater than the limit level at < 10^3 cycles could result in plastic yielding of the gear tooth.

5.4 Bearings

5.4.1 Bearing selection

Shafts may be mounted in bearings, of any size, type and capacity to properly carry the radial and thrust loads which would be induced under the most severe operating conditions.

5.4.2 Fluid film bearings

Fluid film bearings should be designed for bearing pressures not in excess of 6 N/mm^2 on projected area. Journal velocities should not exceed 8 m/s with lubricant supplied un-pressurized. Higher values may be used when the manufacturer has experience or test data.

5.4.3 Roller and ball bearing selection

5.4.3.1 Selection criteria

Roller and ball bearings shall be selected to have a minimum L_{10a} life of 5 000 h based on gear unit rating and gear drive selection factor equal to unity, according to the bearing manufacturers calculations. The L_{10a} life is the operating time that 90 % of apparently identical bearings will equal or exceed before a subsurface originated fatigue spall reaches a predetermined size.

When selecting bearings, the following parameters shall be considered:

— lubrication,

— temperature,

— load zone,

— alignment,

— bearing material.

5.4.3.2 Other considerations

The life calculation methods used by bearing manufacturers are based upon subsurface fatigue damage which leads to spalling. Other types of bearing damage which may occur include, but are not limited to, surface originated spalling due to bruises from lubricant contamination, failure of cages, plastic yielding, brinelling due to extreme momentary overload, and scoring or scuffing due to momentary lack of lubricant film.

5.4.3.3 Reliability

Bearing life at reliability levels other than 90 % is calculated by:

$$L_{na} = a_1 \, L_{10a} \tag{3}$$

where

L_{na} is the adjusted rating life at 100 – n = R percent reliability;

L_{10a} is rating life at basic 90 % reliability, factors a_2 and a_3 included;

a_1 is life adjustment factor for reliability, as in ISO 281.

for reliability $R \geqslant 90$ %,

$$a_1 = 4{,}48 \sqrt[1{,}5]{\ln\left(\frac{100}{R}\right)} \tag{4}$$

for reliability $R < 90$ %,

$$a_1 = 6{,}84 \sqrt[1{,}17]{\ln\left(\frac{100}{R}\right)} \tag{5}$$

Equations 4 and 5 for a_1 are based on the Weibull distribution, fitted to the data of leading bearing manufacturers.

5.5 Shafting

5.5.1 Design criteria

Shafts should be designed to adequately withstand the internal loads (generated by the gear meshes) and the external loads. Both the strength and the stiffness of the shafts are important. Adequate shaft strength will prevent fatigue or plastic deformation, while adequate stiffness will maintain gear and bearing alignment.

5.5.2 Shaft stress calculation

Nominal shaft stresses are calculated as follows. The applicability of equations 6 and 7 to the design of thin wall shafts where the ratio $d_{\mathrm{shi}}/d_{\mathrm{she}} > 0{,}9$ has not been established.

$$\sigma_{\mathrm{s}} = \frac{16\,000\; T\; d_{\mathrm{she}}}{\pi\left(d_{\mathrm{she}}^4 - d_{\mathrm{shi}}^4\right)} \tag{6}$$

$$\sigma_{\mathrm{s}} = \frac{32\,000\; M\; d_{\mathrm{she}}}{\pi\left(d_{\mathrm{she}}^4 - d_{\mathrm{shi}}^4\right)} \tag{7}$$

where

σ_{s} is calculated torsional shaft stress, in N/mm²;

T is shaft torque, in Nm;

d_{she} is shaft outside diameter, in mm;

d_{shi} is shaft inside diameter, in mm;

σ_{b} is calculated bending shaft stress, in N/mm²;

M is bending moment, in Nm.

For solid shafting, equations 6 and 7 simplify to:

$$\sigma_{\mathrm{s}} = \frac{16\,000\; T}{\pi\; d_{\mathrm{she}}^3} \tag{8}$$

$$\sigma_{\mathrm{b}} = \frac{32\,000\; M}{\pi\; d_{\mathrm{she}}^3} \tag{9}$$

5.5.3 Allowable stress

The calculated stresses due to bending and torsion shall not exceed the allowable stress values determined by equations 10 through 15. These equations are a simplified version of DIN 743 and are subject to the following limitations.

— Equations 10 through 15 apply for shaft diameters in the following range:

$25 \leqslant d_{she} \leqslant 150$ mm

For shaft diameters outside of this range the following conditions apply:

If $d_{she} \leqslant 25$ let $d_{she} = 25$ mm

If $150 \leqslant d_{she} \leqslant 500$ let $d_{she} = 150$ mm

— Equations 14 and 15 apply only for:

$d_{she}^{0,36} \times \sigma_B > 2\,600$ N/mm^2

The equations for the allowable stress values have been developed based on the following conditions:

— state of the art shaft design is utilized which should result in keeping the effective stress concentration factors below the maxima listed for each equation;

— repeated torsional stress (zero to maximum) and reversed bending stress;

— equations 11 and 13 apply only to shaft sections with little stress concentration effect;

— the effects of a variable load spectrum is considered by the use of an appropriate selection factor, K_{sf};

— momentary overloads shall be limited to 200 % of P_{mc} applied for a number of stress cycles not exceeding 10 000;

— the material requirements are as specified in 5.4.3.

For through hardened materials:

if $0,09 \times (\sigma_B)^{0,4} < \beta_\tau \leqslant 0,113 \times (\sigma_B)^{0,4}$

$$\sigma_{sa} = [2,22 - 0,35 \times \log(d_{she})] \times \sigma_B^{0,6} \quad (10)$$

if $\beta_\tau \leqslant 0,09 \times (\sigma_B)^{0,4}$

$$\sigma_{sa} = [2,61 - 0,35 \times \log(d_{she})] \times \sigma_B^{0,6} \quad (11)$$

if $0,10 \times (\sigma_B)^{0,4} < \beta_\sigma \leqslant 0,175 \times (\sigma_B)^{0,4}$

$$\sigma_{ba} = [1,88 - 0,30 \times \log(d_{she})] \times \sigma_B^{0,63} \quad (12)$$

if $\beta_\sigma \leqslant 0,10 \times (\sigma_B)^{0,4}$

$$\sigma_{ba} = [2,40 - 0,31 \times \log(d_{she})] \times \sigma_B^{0,66} \quad (13)$$

For carburized and case hardened materials:

If $\beta_\tau \leqslant 0,113 \times \sigma_B{}^{0,4}$

$$\sigma_{sa} = [1,43 - 0,36 \times \log(d_{she})] \times \sigma_B^{0,68} \quad (14)$$

if $\beta_\sigma \leqslant 0,175 \times \sigma_B{}^{0,4}$

$$\sigma_{ba} = [6{,}02 - 1{,}58 \times \log(d_{she})] \times \sigma_B^{0{,}57} \tag{15}$$

where

σ_B is the material tensile strength, in N/mm^2;

σ_{sa} is the allowable torsional stress, in N/mm^2;

σ_{ba} is the allowable bending stress, in N/mm^2;

β_τ is the torsional notch factor;

β_σ is the bending notch factor.

For applications beyond the limits, a more detailed analysis may be required.

5.5.4 Material requirements

For through hardened materials the basis for defining allowable stress is the minimum surface hardness at the critically stressed section. The minimum hardness at a depth from the surface of 1/4 the radius of the critical section shall be 75 % of the minimum hardness at the surface.

For case hardened materials the basis for defining allowable stress is the minimum core hardness at a distance of three times the effective case depth below the surface in the critically stressed section.

For both through hardened and case hardened materials, the hardness will be converted to tensile strength by the conversion table in ISO 6336-5:1996, annex C.

The material for shafts shall meet the requirements of Grade ML of ISO 6336-5:1996. Materials with hardness greater than 241 BHN (255 HV) shall undergo magnetic particle inspection. Indications longer than 1 mm are not permitted in the critically stressed areas.

Ground surfaces shall be free from grinding temper in the critically stressed areas.

The hardness at the specified radius may be determined by measuring the hardness at the same radius on a representative test bar coupon of the same alloy which has been heat treated with the product shaft(s). The coupon shall have the same diameter as the shaft when it is heat treated. See ISO 6336-5:1996, 6.3.

Selection of the appropriate alloy grade shall be based on expected quench rate at the critical section, critical section size, and Jominy hardenability. See ISO 6336-5:1996, annex B for more information.

Statistical or other verifiable process control methods may be substituted for the detailed quality requirements when justified by the manufacturer's experience. See ISO 6336-5:1996, clauses 0, 4, 5.1, and 6.1 for more information.

5.5.5 Deflection

Shaft deflections shall be analyzed regardless of stress levels to ensure satisfactory tooth and bearing contact.

5.6 Keys

5.6.1 Application limits

This calculation method is applicable for keyed connections within the following limits (see Figure 1):

$b_k / d_{sh} \leqslant 0{,}36$

$(h_k - t_k) / t_k \leqslant 0{,}81$

$(h_k - t_k) / b_k \leqslant 0{,}45$

number of keys, $i \leqslant 2$

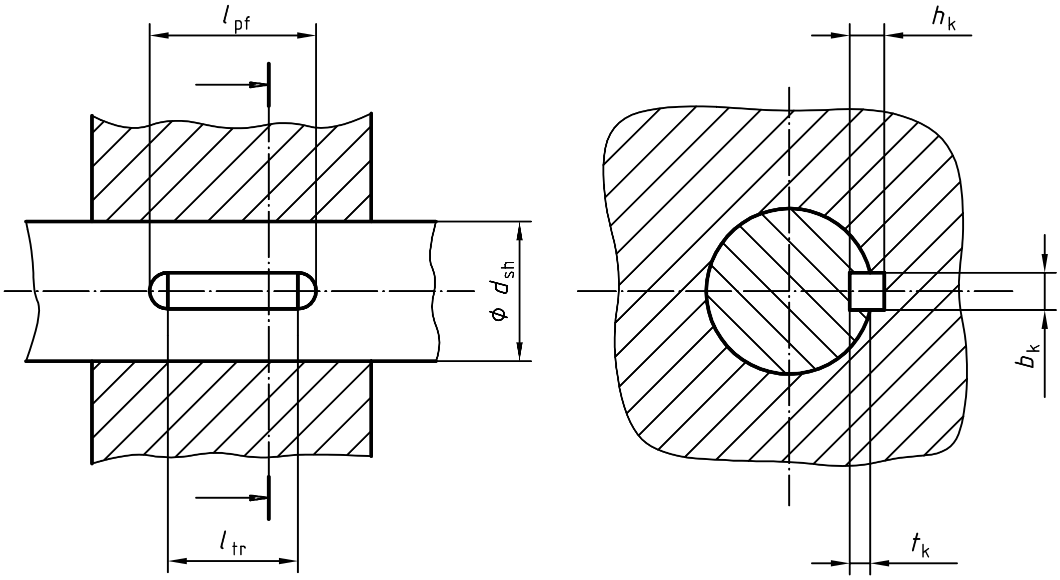


Figure 1 — Fitted key

In addition, the conditions that

a) $l_{tr} \leqslant 1{,}3 d_{sh}$ (a length longer than this does not make significant additional contribution to the strength of the fit), and

b) the direction of the torque does not change,

must be fulfilled.

If a) and b) are not fulfilled, than a more precise method should be used, such as DIN 6892:1995, Method B.

5.6.2 Allowable torque

The allowable torque, T_a, is dependent upon the lesser of the torques as calculated by equation 16 or equation 17.

$$T_C = \sigma_{SC} \frac{d_{sh}}{2\,000} \left(h_k - t_k\right) l_{tr}\ i\ \varphi \geqslant K_A T_n \tag{16}$$

$$T_s = \tau_{ps} \frac{d_{sh}}{2\,000} \left(b_k\ l_{tr}\ i\ \varphi\right) \geqslant K_A T_n \tag{17}$$

where

$$\sigma_{SC} = 0{,}9 R_{e\ min} \tag{18}$$

$$\tau_{ps} = 0{,}379 R_e \tag{19}$$

T_C is the allowable torque based on the allowable compressive stress, in Nm;

T_s is the allowable torque based on the allowable key shear stress, in Nm;

σ_{SC} is the allowable compressive stress, in N/mm^2;

τ_{ps} is the allowable shear stress in the key, in N/mm^2;

d_{sh} is shaft diameter, in mm;

T_n is the nominal torque of the driven machine, in Nm;

T_a is the allowable torque based on the lesser of T_C and T_s, in Nm;

R_e is the tensile strength of the key material, in N/mm^2;

b_k is the width of the key, in mm;

h_k is height of key, in mm;

t_k is shaft keyway depth, in mm;

l_{tr} is bearing length of the key, in mm;

i is the number of keys;

φ is the share of the load.

For one key (i = 1) the value of φ = 1,0 and for two keys (i = 2) the value of φ = 0,75.

5.6.3 Maximum torque

Momentary peak torques whose magnitude exceeds the allowable, T_a, by either equation 16 or 17, may be permitted for a limited number of cycles. The maximum torque value, T_{max}, is determined by:

$$T_{max} = (f_L T_a) + 0{,}8 T_R \qquad (20)$$

where

T_{max} is maximum torque, in Nm;

f_L is load peak frequency factor (see Table 2);

T_R is torque transmitted due to interference fit, in Nm.

Table 2 — Load peak frequency factor, f_L

Number of torque peaks	Load peak frequency factor, f_L	
	Ductile material	Brittle material
$\leqslant 10^3$	1,50	1,30
$> 10^3 \leqslant 10^4$	1,40	1,15
$> 10^4 \leqslant 10^5$	1,25	1,00
$> 10^5 \leqslant 10^6$	1,15	1,00
$> 10^6$	1,00	1,00

If an interference fit is used, T_R is calculated based on the minimum interference fit allowed by the tolerance range, unless the actual values are known.

$$T_R = A_R \frac{d_{she}}{2\,000} \qquad (21)$$

where

A_R is fit holding capacity, in N.

$$A_R = \pi \, P_H \, d_{she} \, L \, \mu \qquad (22)$$

where

P_H is pressure at common shaft/hub interface, in N/mm^2;

L is length of hub, in mm;

μ is coefficient of friction.

$$P_H = \frac{I}{d_{she}\left[\left(\frac{X - \rho_S}{E_S}\right) + \left(\frac{Y + \rho_H}{E_H}\right)\right]} \tag{23}$$

where

I is actual or minimum possible interference fit, in mm;

ρ_S is Poisson's ratio for shaft material;

ρ_H is Poisson's ratio for hub material;

E_S is modulus of elasticity for shaft material, in N/mm^2;

E_H is modulus of elasticity for hub material, in N/mm^2.

$$X = \frac{d_{she}^2 + d_{shi}^2}{d_{she}^2 - d_{shi}^2} \tag{24}$$

$$Y = \frac{d_{he}^2 + d_{hi}^2}{d_{he}^2 - d_{hi}^2} \tag{25}$$

where

d_{he} is outside diameter of hub, in mm;

d_{hi} is inside diameter of hub, in mm.

5.7 Threaded fasteners

5.7.1 Design considerations

The purpose of threaded fasteners is to clamp two or more joint members together. The fasteners shall be of sufficient tensile strength and quantity to withstand the maximum internal and external design loads and prevent movement between the joint members by the clamping force due to fastener tension. Fasteners may also be subjected to shear loading. This condition requires additional analysis and is beyond the scope of this Technical Report. The following simplified method of calculating fastener stresses is based upon VDI 2230.

5.7.2 Fastener preload

Preload is an initial load applied to the fastener to maintain a clamping force. The recommended preload tensile stress, σ_M, for fasteners used in enclosed gear drives is 70 % of the fastener 0,2 % offset yield strength, $\sigma_{p0,2}$ (see Table 3).

$$\sigma_M = 0{,}7\sigma_{p0,2} \tag{26}$$

Table 3 — Fastener preload tensile stress

ISO property class[a]	Maximum nominal fastener diameter d_{max} mm	0,2 % Offset yield strength[a] $\sigma_{p0,2}$ N/mm²	Preload tensile stress σ_M N/mm²
8,8	39	640	448
9,8	16	720	504
10,9	39	940	658
12,9	39	1 100	770

[a] ISO property class according to ISO 898-1.

The value of 70 % is used to provide an adequate safety factor against over stressing due to variations in the torque friction coefficient, accuracy of the assembly to produce the tightening torque and allow fastener reuse.

Tensile preload is considered to act at the tensile area of the fastener and can be calculated from:

$$F_M = A_S\ \sigma_M \tag{27}$$

$$A_S = 0{,}785(D_f - 0{,}938\ 2p_f)^2 \tag{28}$$

where

A_S is stress cross-section of fastener, in mm²;

D_f is nominal diameter of fastener, in mm;

p_f is fastener thread pitch, in mm.

Fastener preload is typically applied by torquing the fastener, or by other methods such as hydraulic stretching and heating. The following equation may be used to estimate the tightening torque for inducing fastener preload.

$$M_A = K_{tc}F_M D_f \tag{29}$$

where

M_A is tightening torque, in Nm;

K_{tc} is torque coefficient. Taking a typical overall friction coefficient of 0,12 into account, $K_{tc} = 0{,}16 \times 10^{-3}$;

F_M is tensile preload, in N.

5.7.3 Fastener allowable stress

The allowable tensile stress, σ_{fa}, is:

$$\sigma_{fa} = 0{,}35\sigma_M\ (K_J) \tag{30}$$

where

σ_M is preload tensile stress;

K_J is joint stiffness factor, see Table 4.

Table 4 — Joint stiffness factor

Joint stiffness factor	Joint material	
	Steel	Cast iron
K_J	1,14	1,28

The allowable tensile stress is based on the following conditions (for applications beyond these limits a more detailed analysis may be required):

— metal-to-metal joint;

— tensile preload equal to 0,7 ($\sigma_{p0,2}$), see Table 3;

— based on 40 % of fastener stress at joint opening, providing a safety factor of 1,25 on 200 %;

— $l_g \geqslant 4D_f$ (see Figure 2).

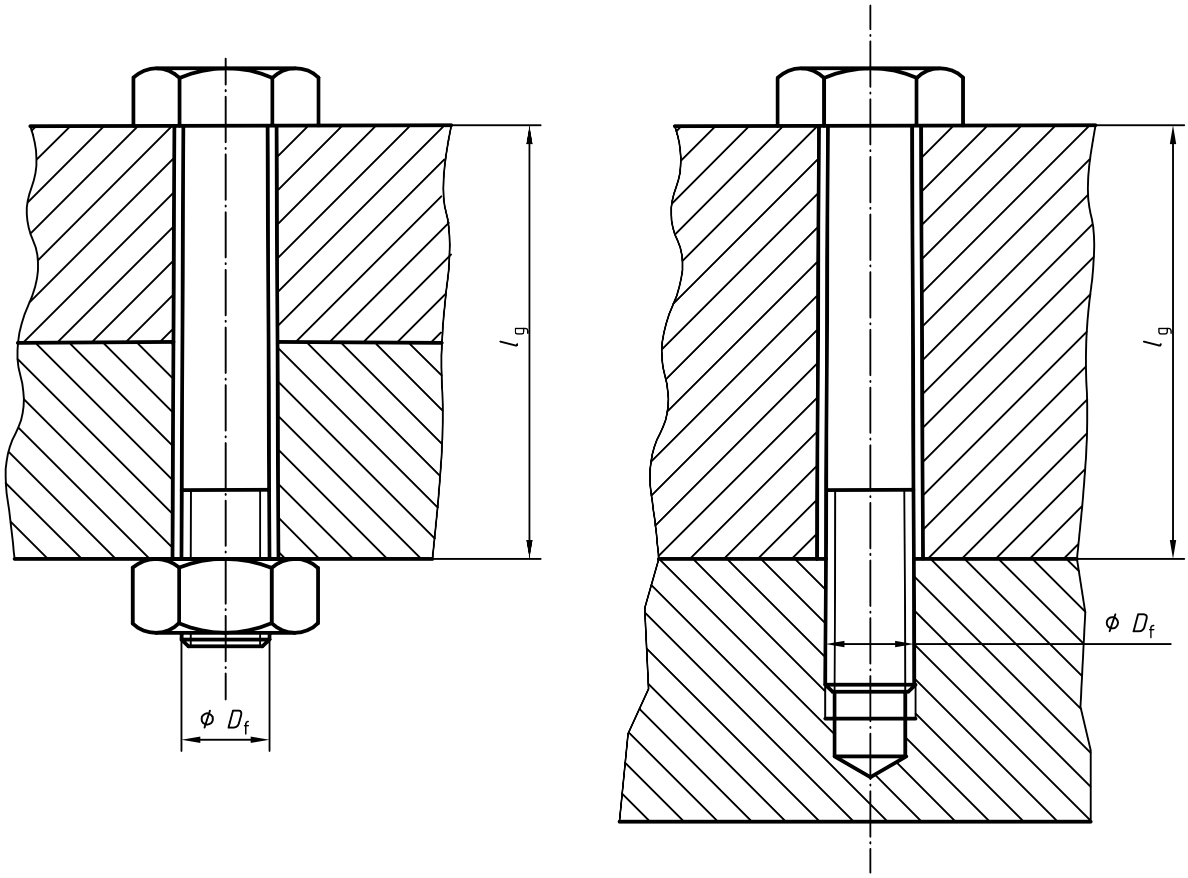


Figure 2 — Fastener grip requirement

5.7.4 Fastener tensile stress

The applied tensile load shall be based on forces developed by the mechanical rating of the gear drive. These forces, considered to act in the worst possible direction, shall include all internally and externally applied loads, i.e. overhung loads, thrust loads, etc., but shall not include tensile preload. The applied tensile load is considered to act at the tensile area of the fastener. Fastener tensile stress can be calculated from the following equation:

$$\sigma_f = \frac{F_A}{A_s} \leqslant \sigma_{fa} \tag{31}$$

where

σ_f is calculated tensile stress, in N/mm^2;

F_A is applied tensile load, in N.

5.7.5 Locking devices for fastener

Fasteners on housings and covers do not require locking devices for most industrial applications. Fasteners mounted on shafts should be locked for safety reasons. Typical locking methods include:

— lockwashers (various types);

— inserts in the engaged threaded area;

— self-locking type;

— locking compounds;

— locking tabs;

— lock wiring.

5.8 Other components

Information for other components which should be considered can be found in annex B.

6 Lubrication and lubricants

6.1 Lubrication

These lubrication recommendations apply only to enclosed gear drives which are designed and rated in accordance with current ISO standards. Additional information pertaining to enclosed gear lubrication is given in ISO 6743-6. These recommendations are not intended to replace any specific lubrication recommendations made by the gear drive manufacturer.

6.1.1 Ambient temperature

The ambient temperature range, –40 °C to + 50 °C, is defined as the air temperature in the immediate vicinity of the gear drive. Gear drives exposed to the direct rays of the sun or other radiant heat sources will run hotter and shall therefore be given special consideration.

6.1.2 Other ambient considerations

Gear drives operating outside of the temperature range of 6.1.1, or those operating in extremely humid, chemical, or dust laden atmospheres should be referred to the gear drive manufacturer.

6.1.3 Oil sump temperatures

The maximum allowable oil temperature, measured in the sump, is 95 °C for all types of mineral lubricants. See 6.5.

The use of higher temperatures for synthetic oils is permitted, if agreed upon between gear manufacturer, oil supplier and user.

6.1.4 Food and drug

The lubricants recommended in this Technical Report are not recommended for food and drug industry applications where incidental contact with the product being manufactured may occur.

The user shall assume the responsibility for selecting the proper lubricant for all food and drug industry applications.

6.1.5 Mounting position

All gear drives are considered to operate in the manufacturer's specified mounting position.

6.2 Lubricant kinematic viscosity

Lubricant kinematic viscosity classifications are specified by ISO 3448. The viscosity range related to the viscosity grade is the number minus 10 % for the minimum and plus 10 % for the maximum. For example, the viscosity grade VG 100 corresponds to a viscosity range from 90 mm^2/s (cSt) to 110 mm^2/s (cSt) at 40 °C.

6.3 Lubrication recommendations

Tables 5 and 6 show the grade of lubricant to be used. For multi-stage gear drives, the lowest pitch line velocity shall be used for lubricant grade selection.

6.4 Cold temperature starting

6.4.1 Low temperature conditions

Gear drive lubrication, either by splash or pump, shall be given special attention if the drive shall be started or operated at temperatures below which the oil can be effectively splashed or pumped. An acceptable low temperature gear oil, in addition to meeting ISO specifications, shall have a pour point at least 6 °C below the expected ambient temperature and a viscosity which is low enough to allow the oil to flow freely at the start-up temperature but high enough to carry the load at operating temperature. Gear drives operating in cold areas shall be provided with oil that circulates freely and does not cause high starting torques. Preheating the oil may be necessary under these low ambient temperature conditions. The gear manufacturer shall always be informed when drives are to operate under these conditions.

6.4.2 Sump heaters

If a suitable, low temperature gear oil is not available, the gear drive shall be provided with a sump heater to bring the oil up to a temperature at which it will circulate freely for starting. The heater should be so selected as to avoid excessive localized heating which could result in rapid degradation of the lubricant.

6.5 High temperature operation

When gear drives operate at or near the permissible maximum temperature, the probability of gear and bearing distress will increase, more frequent oil changes shall be required, and special consideration shall be given to seal selection.

EXAMPLE The life of a nitrile rubber seal is typically reduced from about 9 400 h at 68 °C to 3 000 h at 93 °C.

6.6 Lubricant types

6.6.1 Rust and oxidation inhibited gear oils

These lubricants are commonly referred to as rust and oxidation (R&O) gear oils. They are petroleum base lubricants which have been formulated to include special additives which resist rust and oxidation.

Table 5 — Minimum ISO lubricant grade recommendations or spur and helical gear drives

Pitch line velocity	ISO Lubricant grades for ambient temperature range		
m/s	–40 °C to –5 °C	–10 °C to 20 °C	10 °C to 50 °C
up to 10	See Table 6	150	320
10 to 20		68	150
20 to 35		32	68

6.6.2 Extreme pressure lubricants

Extreme pressure (EP) gear lubricants are petroleum based lubricants containing special chemical additives. EP gear lubricants recommended for enclosed gear drives are those containing sulphur, phosphorous, or similar type additives. EP gear lubricants should be used only when specified by the gear drive manufacturer (see Table 5).

WARNING — The lead naphthenate type is no longer recommended because of limited availability and poor stability in comparison to the more modern types of lubricants.

Do not use extreme pressure lubricant or lubricants with formulations including sulphur, chlorine, lead and phosphorous derivatives, as well as graphite and molybdenum disulfide, in gear drives equipped with an internal backstop, unless approved by the gear manufacturer or the backstop manufacturer.

6.6.3 Synthetic gear lubricants

Diesters, polyglycol and synthetic hydrocarbons have been used in enclosed gear drives for special operating conditions. Synthetic lubricants can be advantageous over mineral oils in that they are generally more stable, have a longer life, and operate over a wider temperature range. However, synthetics are not always appropriate. Each type has different characteristics, and many of them have distinct disadvantages. Such things as compatibility with other lubrication system components, behaviour in the presence of moisture, lubricating qualities, overall economics and compatibility with internal coatings should be carefully analyzed for each type of synthetic lubricant under consideration. In the absence of field experience in similar applications, the use of a synthetic lubricant should be carefully coordinated between the user, the gear manufacturer, and the lubricant supplier.

CAUTION — Special authorization is required from the manufacturer prior to using a synthetic lubricant in a drive equipped with an internal backstop.

6.6.4 Synthetic lubricant selection

The recommendations for synthetic lubricants in Table 6 are based on gear drive manufacturers' experience with synthetic hydrocarbons of the polyalphaolefin type. While other types of synthetic lubricants may be used, lack of experience prevents their recommendation. The viscosity recommendations in Table 6 may be used as a guide in selection of these other types of lubricants along with the considerations of 6.6.3.

Table 6 — Synthetic hydrocarbon (SHC) lubricant recommendations for spur and helical gears

Property	SHC recommended for ambient temperature range			
	–40 °C to –10 °C	–30 °C to 10 °C	–20 °C to 30 °C	–10 °C to 50 °C
ISO grade	32	68	150	220
Viscosity index, min.	130	135	135	145

6.7 Lubricant maintenance

6.7.1 Initial lubricant maintenance

The lubricant in a new gear drive should be drained after 500 h of effective operation. The gear case should be thoroughly cleaned with a commercial grade of flushing oil that is compatible with the seals and operating lubricant. The original lubricant can be used for refilling if it has been filtered through a filter of 30 µm or less; otherwise, new lubricant shall be used. Lubricants should not be filtered through Fuller's earth or other types of filters which remove lubricant additives.

6.7.2 Change intervals

Under normal conditions the lubricant should be changed as follows:

— petroleum at 5 000 h or one year, which ever occurs first;

— synthetic at 7 500 h or one year, which ever occurs first.

Extending the recommended change period may be permissible based on: the type of lubricant, system downtime, operating load and temperature, or environmental impact of used oil. This can be done through proper implementation of a comprehensive lubricant testing program. As a minimum, the program should include testing for:

- changes in appearance and colour;
- lubricant viscosity (oxidation);
- water concentration;
- contaminant concentration;
- sediment and sludge;
- additive concentration and condition.

In the absence of more specific limits, the following guidelines may be used to indicate when to change oil:

- water content greater than 0,05 % (500 parts per million);
- iron content exceeds 0,015 % (150 parts per million);
- silicon (dust/dirt) exceeds 0,002 5 % (25 parts per million);
- viscosity changes more than 15 %.

These tests should be performed on the initial charge of the gear drive to establish a baseline for comparison. Subsequent test intervals should be established based on the drive manufacturer's and lubricant supplier's recommendations.

6.7.3 Change intervals for unusual conditions

A rapid rise and fall in temperature can produce condensation, resulting in the formation of sludge. Dust, dirt, chemical particles or chemical fumes also react with the lubricant. Sustained sump temperatures in excess of 95 °C can result in accelerated degradation of the lubricant. Under these conditions the lubricant should be changed every one to three months depending on severity.

6.7.4 Cleaning and flushing

The lubricant should be drained while the gear drive is at operating temperature. The drive should be cleaned with a flushing oil. The use of a solvent should be avoided unless the gear drive contained deposits of oxidized or contaminated lubricant which cannot be removed with a flushing oil. When persistent deposits necessitate the use of a solvent, a flushing oil should then be used to remove all traces of solvent from the system.

6.7.5 Used lubricants

Used lubricant and flushing oil should be completely removed from the system to avoid contaminating the new charge.

6.7.6 Inspection

The interior surfaces should be inspected where possible, and all traces of foreign material removed. The new charge of lubricant should be added and circulated to coat all internal parts.

7 Thermal rating

7.1 Thermal rating application

Thermal rating is the maximum power that can be continuously transmitted through an enclosed gear drive without exceeding a specified oil sump temperature. The thermal rating shall equal or exceed the actual service transmitted power. Selection factors are not used when determining thermal requirements. The magnitude of the thermal rating depends upon the specifics of the enclosed drive, operating conditions, the maximum allowable sump temperature, as well as the type of cooling employed.

The primary thermal rating criterion is the maximum allowable oil sump temperature. Unacceptably high oil sump temperatures influence gear drive operation by increasing the oxidation rate of the oil and decreasing its viscosity. Reduced viscosity translates into reduced oil film thickness on the gear teeth and bearing contacting surfaces thereby reducing the pitting life of these elements. To achieve the required life and performance of a gear drive the operating oil sump temperatures shall be evaluated and limited.

Thermal ratings of enclosed gear drives rated by this Technical Report are limited to a maximum allowable oil sump temperature of 95 °C. However, based on the gear manufacturer's experience or application requirements selection can be made for oil sump temperatures above or below 95 °C, see 7.5.

Additional criteria that shall be applied in establishing the thermal rating for a specific gear drive with a given type of cooling are related to the operating conditions of the drive. The basic thermal rating, P_T, is established by test (Method A) or by calculation (Method B) under the following conditions:

— oil sump temperature at 95 °C;

— ambient air temperature of 25 °C;

— ambient air velocity of $\leqslant$ 1,4 m/s in a large indoor space;

— air density at sea level;

— operation continuous (100 % at thermal rating, P_T).

Modifying factors for deviation from these criteria are given in 7.5.

7.1.1 Thermal service considerations

7.1.1.1 Intermittent service

For intermittent service, the input power may exceed the manufacturer's thermal power ratings provided the oil sump temperature does not exceed 95 °C.

7.1.1.2 Adverse conditions

The ability of a gear drive to operate within its thermal rating can be reduced when adverse conditions exist. Some examples of adverse environmental conditions are:

— an enclosed space;

— a build-up of material that may cover the gear drive and reduce heat dissipation;

— a high ambient temperature, such as boiler or turbine rooms, or in conjunction with hot processing equipment;

— high altitudes;

— the presence of solar energy or radiant heat.

7.1.1.3 Favourable conditions

The thermal rating may be enhanced when operating conditions include increased air movement or a low ambient temperature.

7.1.2 Auxiliary cooling

Auxiliary cooling should be used when thermal rating is insufficient for operating conditions. The oil can be cooled by a number of means, such as:

— shaft fan cooling: the fan shall maintain the fan cooled thermal rating;

— heat exchanger: the heat exchanger used shall be capable of absorbing generated heat that cannot be dissipated by the gear drive by convection and radiation.

7.2 Methods for determining the thermal rating

The thermal rating can be determined by one of two methods: Method A, test; or Method B, calculation.

a) **Method A, test:** test of full scale gear drives at operating conditions is the preferred method for establishing the thermal rating of the gear drive. See 7.3.

b) **Method B, heat balance calculation:** the thermal rating of a gear drive can be calculated using the heat balance equation which equates heat generated, P_V, with heat dissipated, P_Q. The calculation of thermal rating is given in 7.4. The method for calculating heat generation is discussed in 7.4.2, and for heat dissipation in 7.4.3.

The thermal rating should be evaluated for both directions of rotation. The thermal rating of the gear drive is the minimum value obtained for either direction or the value for the application direction, if known.

7.3 Method A — Test

A test of a specific gear drive at its design operating conditions is the most reliable means to establish the thermal rating. Thermal testing involves measuring the steady-state oil sump temperature of the gear drive operating at its rated speed at no-load and at least one or two increments of load. Preferably one test should be at 95 °C sump temperature.

Some guidelines for acceptable thermal testing are as follows:

— the ambient air temperature and velocity shall be stabilized and measured for the duration of the test;

— the time required for the gear drive to reach a steady-state sump temperature depends upon the drive size and the type of cooling;

— steady-state conditions can be approximated when the change in oil sump temperature is 1 °C or less per hour.

The oil temperature in the sump at various locations can vary as much as 15 °C. Outer surface temperatures can vary substantially from the sump temperature. The opposite direction of rotation can create a different sump temperature.

During thermal testing the housing outer surface temperature can be surveyed if detailed analysis of the heat transfer coefficient and effective housing surface area is desired. Also, with shaft fan cooling, the air velocity distribution over the housing surface can be measured.

While no-load testing cannot yield a thermal rating, it can be used to establish the heat transfer coefficient if the power required to operate the drive at no-load is measured.

7.4 Method B — Calculations for determining the thermal rating, $P_{T\text{-}B}$

7.4.1 Heat balance calculation

The calculation of thermal rating is an iterative process due to the load dependency of the coefficient of friction for the gear mesh and the bearing power loss.

The basis of the thermal rating is when the power losses, P_V at P_A, are equal to the heat dissipation, P_Q, of the gear drive.

$$P_Q = P_V \tag{32}$$

When this is satisfied under the conditions of 7.1, $P_{T\text{-}B}$ is P_A.

The heat generation in a gear drive comes from both load-dependent, P_L, and non-load-dependent power losses, P_N.

$$P_V = P_L + P_N \tag{33}$$

P_L is a function of the input power, P_A.

$$P_L = f(P_A) \tag{34}$$

Using equation 32 and rearranging terms, we can write the basic heat balance equation as follows:

$$P_Q - P_N - f(P_A) = 0 \tag{35}$$

To determine the basic thermal rating, $P_{T\text{-}B}$, vary P_A until equation 35 is satisfied. This can be done by recalculating the load-dependent power losses, P_L, at different input powers, P_A. If $P_Q \leqslant P_N$, the gear drive does not have any thermal capacity. The design shall be changed to increase P_Q or auxiliary cooling methods used.

When equation 35 is satisfied, the overall drive efficiency, η, is calculated as follows:

$$\eta = 100 - \frac{P_L - P_N}{P_A} \times 100 \tag{36}$$

The thermal rating of the drive is as follows:

$$P_{T-B} = \frac{P_Q}{1 - \frac{\eta}{100}} \tag{37}$$

7.4.2 Heat generation

Heat generation comes from two sources, load and non-load-dependent. The load-dependent power losses consist of the bearing power losses and the gear mesh power losses.

$$P_L = \sum_{i=1}^{n} P_{Bi} + \sum_{i=1}^{n} P_{Mi} \tag{38}$$

The non-load-dependent power losses consist of the oil seal power losses, the internal gear windage and oil churning power losses, the internal bearing windage and oil churning power losses, and the oil pump power consumed.

$$P_N = \sum_{i=1}^{n} P_{Si} + \sum_{i=1}^{n} P_{WGi} + \sum_{i=1}^{n} P_{WBi} + \sum_{i=1}^{n} P_{Pi} \tag{39}$$

These power losses shall be summed for each component in the gear drive.

The calculations for values of the bearing power loss, P_B, mesh power loss, P_M, oil seal power loss, P_S, gear windage and churning power loss, P_{WG}, bearing windage and churning power loss, P_{WB}, and oil pump power loss, P_P, shall be made by an acceptable method such as given in annexes C and D.

7.4.3 Heat dissipation, P_Q

The heat dissipated from a gear drive is influenced by the surface area of the gear drive, the air velocity across the surface, the temperature differential, ΔT, between the oil sump and the ambient air, the heat transfer rate from the oil to the gear case and the heat transfer rate between the gear case and the ambient air. The heat dissipation is given by equation 40.

$$P_Q = A_C\, k\, \Delta T \tag{40}$$

The calculation of heat dissipation, P_Q, shall be made by an acceptable method such as given in annexes C and D.

7.5 Corrections for non-standard operating conditions

When the actual operating conditions for a specific application are different from the standard conditions defined in 7.1 and the thermal rating is calculated for the conditions of 7.1, the thermal rating may be modified for the application as follows:

$$P_{\mathrm{Thm}} = P_{\mathrm{T}} \, B_{\mathrm{réf}} \, B_{\mathrm{V}} \, B_{\mathrm{A}} \, B_{\mathrm{T}} \, B_{\mathrm{D}} \quad (41)$$

B_{ref} and B_{A} can be applied to natural or shaft fan cooling. B_{V} shall be applied only to natural cooling.

The gear drive manufacturer should be consulted when the conditions exceed the limits given in annex C, Tables C.5 through C.9, or when correction factors are required for any type of cooling other than natural or shaft fan.

When the ambient air temperature is below 25 °C, B_{ref} allows an increase in the thermal rating. Conversely, with an ambient air temperature above 25 °C, the thermal rating is reduced, see annex C, Table C.5.

When the surrounding air has a steady velocity in excess of 1,4 m/s due to natural or operational wind fields, the increased convection heat transfer allows the thermal rating to be increased by applying B_{V} . Conversely, with an ambient air velocity of $\leqslant$ 0,50 m/s, the thermal rating is reduced. See annex C, Table C.6.

At high altitudes the decrease in air density results in the derating factor, B_{A}. See annex C, Table C.7.

The standard maximum allowable oil sump temperature is 95 °C. A lower sump temperature requires a reduction in the thermal rating using B_{T}, see annex C, Table C.8. A maximum allowable sump temperature in excess of 95 °C will increase the thermal rating and can provide acceptable gear drive performance in some applications. However, it must be recognized that operating above 95 °C can reduce lubricant and contact seal life and increase the surface deterioration on the gears and bearings with a subsequent increase in the frequency of maintenance. The gear manufacturer should be consulted when a maximum allowable oil sump temperature in excess of 95 °C is being considered.

When a gear drive sees less than continuous operation with periods of zero speed, the resulting “cool-off” time allows the thermal rating to be increased by B_{D}. See annex C, Table C.9.

8 Measurement of sound and vibration

Certain frequencies and levels of sound or vibrations can be objectionable and damaging. Acceptable levels of sound and vibration are often specified according to the manufacturer or by national or local regulations.

The gear drive is only part of the total acoustic system which includes, in addition to the gear drive, the prime mover, driven equipment, gear drive mountings, foundations and environment. Each of these can affect the measured level of sound or vibration emitted by the gear drive. Unless otherwise agreed, the gear manufacturer's responsibility is to ensure that the level emitted by a gear drive under test conditions in his factory are within contractually specified or negotiated limits.

Due to system response and environment, it is difficult to predict from test data recorded at the manufacturer's factory the levels from a gear drive in its installed environment. Unless otherwise specified, this is not usually the gear drive manufacturer's responsibility.

The measuring methods and test procedures necessary for the determination of a gear drive's sound or vibration level for acceptance testing, at the gear drive manufacturer's factory, should be in accordance with ISO 8579-1, for sound, or ISO 8579-2, for vibration.

9 Selection factor, K_{sf}

9.1 Selection factor definition

The selection factor accounts for the influence of a specific application on the performance of a gear drive. This factor covers the following influences which are typical for each application:

— influence of external dynamic loads which appear in normal operating conditions, covering normal starting conditions and reversing applications when they occur;

— influence of life duration which characterizes the typical application;

— influence of reliability required and covering the percentage of failures allowed for the considered application;

— safety as regards random overloads which cannot be predicted and which can occur in normal operation.

Before an enclosed gear drive can be selected for an application, a "modified power" rating for the drive shall be determined. This is done by multiplying the specified transmitted power by the selection factor. Since a selection factor represents the normal relationship between gear drive design power rating and the maximum potential transmitted power, it is suggested that the selection factor be applied to the nameplate rating of the driven machine or prime mover, as applicable.

Manufacturer and user shall agree upon which power, prime mover rating or driven machine requirements, should dictate the selection of the gear drive. It is necessary that the gear drive selected have a rated load capacity equal to or in excess of this "modified power".

9.2 Factors affecting external dynamic loads

To determine the selection factor, consideration should be given to the fact that many prime movers can develop momentary peak torques appreciably greater than those determined by the nominal ratings of either the prime mover or of the driven equipment. There are many other possible sources of overload which should be considered:

— system vibration;

— critical speeds;

— acceleration torques;

— overspeeds;

— sudden variations in system operation;

— braking;

— negative torques, such as those produced by retarders on vehicles, which result in loading on the reverse side of the gear teeth.

Analysis within the operating range of the drive is essential. If critical speeds are present, changes in the design of the overall drive system should be made either to eliminate or to provide system damping so that gear and shaft vibrations are eliminated. The enclosed gear drive designer or manufacturer is not responsible for the system analysis, unless agreed to in the purchase contract.

9.2.1 Operational characteristics

Some of the operational characteristics that could affect an increase or decrease in selection factors are:

— type of prime mover: different types of prime movers are electric motors, hydraulic motors, steam or gas turbines, and internal combustion engines having single or multiple cylinders;

starting conditions: starting conditions where peak loads exceed 200 % of the rated load or applications with frequent starts and stops require special load analysis. The rated load is defined as the gear unit rating with a selection factor of 1,0. When a soft start coupling is used between the prime mover and the gear drive, the choice of selection factors can be based on the gear drive manufacturer's analysis for the application;

— overloads: loads which are in excess of the rated load are considered overloads. Overload can be of momentary duration, periodic, quasi-steady state, or vibratory in nature. The magnitude and the number of stress cycles require special analysis to prevent low cycle fatigue or yield stress failure. Applications such as high torque motors, extreme repetitive shock, or where high energy loads are absorbed, as when stalling, require special consideration;

— overspeeds: overspeeds contributing to external transmitted loads and dynamic loads require special analysis;

— brake equipped applications: when a gear drive is equipped with a "working" brake that is used to decelerate the motion of the system, select the drive based on the brake rating or the transmitted power, whichever is greater. If the brake is used for holding only, and is applied after the motion of the system has come to rest, the brake rating should be less than 200 % of the gear unit rating. If the brake rating is greater than 200 % of the gear unit rating, or the brake is located on the output shaft of the gear drive, special analysis is required;

— reliability and life requirement: applications requiring a high degree of dependability or unusually long life should be given careful consideration by the user and the gear manufacturer before assigning a selection factor.

9.2.2 System conditions

An essential phase in the design of a system of rotating machinery is the analysis of the dynamic (vibratory) response of a system to excitation forces.

9.2.2.1 Vibration analysis

Any vibration analysis shall consider the complete system including prime mover, gear drive, driven equipment, couplings, and foundations. The dynamic loads imposed upon a gear drive are the result of the dynamic behaviour of the total system and not of the gear drive alone.

9.2.2.2 Dynamic response

The dynamic response of a system results in additional loads imposed on the system and relative motion between adjacent elements in the system.

The vibratory loads are superimposed upon the mean running load in the system and, depending upon the dynamic behaviour of the system, could lead to failure of the system components.

9.2.2.3 System induced failure

In a gear drive, system induced failures could occur as tooth breakage or severe surface deterioration of the gear elements, shaft breakage, bearing failure, or failure of other component parts.

9.2.2.4 Special system considerations

It should be pointed out that synchronous motors, certain types of high torque induction motors and generator drives require special care in system design.

Synchronous motors have high transient torques during starting and when they momentarily trip-out and restart.

Induction motors of special high slip design can produce extremely high starting torques. Also, high torques are produced when the motor trips out for a very short time and then the trip re-closes, or when the motor is started with a star-delta starting.

Generators have extremely high loads when they are out of phase with the main system. Also, across-the-line shorts can produce torque loads up to twenty times the normal running torque.

All special torque conditions should be considered when determining a selection factor.

9.2.3 Special considerations

Adjustments to the gear drive selection may be necessary when one or more of the following conditions exist:

— ambient conditions: extremes of temperature and environment;

— lubrication: any lubricant not in accordance with manufacturer's recommendations;

— misalignment and distortions;

— reversing applications;

— high risk applications involving human safety.

9.3 Determination of K_{sf}

The selection of K_{sf} is made for any application by taking the corresponding value in the tables given in annex A (Tables A.1 and A.2) which also take into account the degree of shock due to the choice of motor.

9.4 Limitations of selection factor

K_{sf} covers typical starting conditions for the application.

If exceptional overtorques occur, they can be tolerated without damage if the maximum torque, T_{max}, is less than $2T_n \times K_{sf}$ and if they are not applied for a number of cycles exceeding those defined in 4.6.

If these values are exceeded, special devices should be installed to protect the gearbox, or a gearbox with a greater rating selected.

9.5 Selection factor for drive components

9.5.1 Relationship of application, life, reliability and safety

When a selection factor equal to unity is used for a gear set calculation, it corresponds to:

— an application factor K_A equal to 1 for pitting resistance and bending;

— the life factors, Y_{NT} and Z_{NT}, calculated for 10 000 h, at the design speed for pitting and bending resistance ratings for the material and heat treatment;

— a reliability corresponding to 99 % for pitting and bending resistance;

— a minimum safety factor of 1,2 for bending ($S_{F\,min}$ = 1,2) and of 1,0 for pitting resistance ($S_{H\,min}$ = 1,0).

9.5.2 Establishment of an enclosed drive selection factor

Selection factors can better be established from a thorough analysis of service experience with a particular application.

A selection factor is used to include the combined effects of reliability, acceptable life, application conditions and safety factors in an empirically determined single factor. The individual numerical influence of each of these quantities may not be specifically established, but taken only as a whole.

9.6 Normal working conditions

The selection factor, K_{sf}, covers the overload and dynamic conditions which are generated only in normal working conditions. Such conditions are considered as normal if there are no excitations of natural torsional frequencies within the connected power transmission system where the gearbox is installed.

To be sure that the working conditions are normal, it is necessary to make a torsional analysis of the complete installation including the gearbox. The torsional analysis is the responsibility of the supplier of the complete system. For high input speeds particular care should be taken to verify that there is no risk of gear meshing resonance.

To perform this analysis, the gear manufacturer shall provide all the relevant gear data to the designer of the complete installation.

10 Marking

A suitable nameplate should be attached to the gear drive with the following minimum information in addition to the manufacturer's name:

— manufacturer's size and type designation;

— ratio;

— manufacturing or serial number.

Optional information may be included such as:

— nominal power rating;

— high speed shaft rotational frequency (r/min);

— selection factor;

— lubrication requirements.

11 Customer responsibility, transportation, installation and storage

Gear drives which are catalogue items will normally be purchased through a distributor. The gear drive manufacturer will have ensured that the drive has been tested, drained of oil and packed suitable for transportation and storage, and will provide details for subsequent installation. Further action is beyond his control.

With gear drives produced to a customer's specification, the agreement should also specify any requirements the customer has for the following:

— protection against corrosion;

— transportation;

— extended storage;

— site installation;

— inspection before start up;

— no load testing of the gear drive.

A detailed listing of customer responsibilities is given in annex E.

12 Operation and maintenance

12.1 Preparation for operation

After any maintenance period, when the original integrity of the gear drive and auxiliary systems may have been changed, it is necessary to conduct a number of checks prior to starting the gear drive, as follows:

— the mounting bolts and coupling bolts have been correctly torqued;

— the integrity and levels of the lubrication and cooling system;

— the direction of rotation of any electric pumps.

12.2 Operation

Before starting the gear drive, perform the following operations:

— run the lubrication system under pressure for 30 min;

— confirm the proper flow of lubricant to each gear mesh and bearing.

On starting the gear drive, check for abnormal vibration.

12.3 Maintenance

Satisfactory operation requires correct maintenance. Negligence of maintenance and inspection for long periods can result in unexpected problems. Examples of inspection intervals and the items to be inspected are shown in Table 7.

Table 7 — Maintenance

Maintenance interval	Items
Daily	Check the following: — oil level in the tank or the sump is within operating range; — oil pressure is within operating range; — bearing temperatures are within operating range; — vibration levels are normal; — noise levels are normal; — oil leakage is within acceptable limits.
Weekly	Remove any sediments from the oil strainer.
Monthly	Sample-check the lubricant for water and metallic contamination. During periods when the gear drive is stopped, check the hold-down bolts and the coupling bolts for tightness.
Annually	Change the lubricant charge. If the drive has low numbers of operating hours, a lubricant sample should be analyzed; if it is found to be acceptable, a full change of lubricant is not required. Check the following items: — oil seals and seal running surfaces; — tooth contact markings; — degree of wear in the bearings; — condition of consumables (e.g., filters), replace as required; — alignment of input and output couplings.

13 Test and inspection

The testing and inspection procedures for assembled gear drives are normally at the discretion of the manufacturer. Individual component inspection and process control are beyond the scope of this Technical Report.

When testing of the gear unit is performed, the information given in annex F should be used. Test loads and speeds are normally at the gear manufacturer's discretion unless specific test loads are agreed upon and included as a part of the purchase contract. In individual cases, especially where unusually high speeds or power are involved, alternate operating conditions may be negotiated.

CAUTION — It is recommended that gear drives not be tested with loads in excess of gear unit rating, since such practice will reduce the design life of the drive.

The duration of the running test shall be decided by the drive manufacturer unless a specific time has been contractually agreed upon between manufacturer and purchaser.

Annex A
(informative)

Selection factors

A.1 Purpose

This annex provides a detailed guide for determining selection factors for enclosed gear drives.

A.2 Selection of selection factors

Before an enclosed speed reducer or increaser can be selected for any application, a modified power rating of the drive (selection factor = 1,0) to be transmitted shall be determined. This is done by multiplying the specified power by the selection factor. Since selection factors represent the normal relationship between gear unit design power rating and the maximum potential transmitted power, it is suggested that the selection factor be applied to the nameplate rating of the prime mover or driven machine rating, as applicable.

Manufacturer and user shall agree upon which power, prime mover rating or driven machine requirements should dictate the selection of the gear drive. It is necessary that the gear drive selected have a rated drive capacity equal to or in excess of this "modified power" rating.

All selection factors listed are 1,0 or greater. Selection factors less than 1,0 can be used in some applications when specified by the user and agreed to by the manufacturer.

Table A.2 should be used with caution, since much higher values have been used in some applications. Values as high as ten have been observed. On some applications up to six times nominal torque can occur, such as turbine/generator drives, heavy plate and billet rolling mills.

A.3 Listing of selection factors

The table of selection factors has been developed from the experience of manufacturers and users of gear drives for use in common applications. It is suggested that selection factors for special applications be agreed upon by the user and the gear manufacturer when variations of the values in the table may be required.

A.4 Determining selection factors

In addition to the tables, K_{sf} can also be determined by the approximate relationship:

For pitting resistance,

$$K_{sf} = \frac{K_A \left(\frac{S_H}{S_{H\,min}} \right)^2}{Z_{NT}^2} \tag{A.1}$$

For bending,

$$K_{sf} = \frac{K_A \left(\frac{S_F}{S_{F\,min}} \right)}{Y_{NT}} \tag{A.2}$$

In general, the same value of K_{sf} is used for bending and pitting. All factors affecting external dynamic loads should be considered.

A.5 Selection factor tables

A.5.1 General considerations

Selection factors have served industry well when the application has been identified by knowledgeable and experienced gear design engineers. The tables are provided for information purposes only and should only be used after taking into account all of the external influences which may affect the operation of the enclosed gear drive.

A.5.2 Use of tables

Selection factors shown in Table A.2 are for gear drives driven by motors (electric or hydraulic) and turbines (steam or gas).

A.5.3 Driver influence

When the driver is a single cylinder or multi-cylinder engine, the selection factors from Table A.2 shall be modified by the values from Table A.1 for the appropriate type of prime mover.

Table A.1 — Conversion table for single or multi-cylinder engines to find equivalent single or multi-cylinder selection factor

Steam and gas turbines, hydraulic or electric motor	Single cylinder engines	Multi-cylinder engines
1,00	1,50	1,25
1,25	1,75	1,50
1,50	2,00	1,75
1,75	2,25	2,00
2,00	2,50	2,25
2,25	2,75	2,50
2,50	3,00	2,75
2,75	3,25	3,00
3,00	3,50	3,25

A.6 Example

If the application is a centrifugal blower, the selection factor from Table A.2 is 1,25 for a motor or turbine. Table A.1 converts this value to 1,50 for a multi-cylinder engine and 1,75 for a single cylinder engine.

CAUTION — Any user of enclosed gear drives should make sure he has the latest available data on the factors affecting the selection of a gear drive. When better load intensity information is available on the drive or driven equipment, this should be considered when a selection factor is selected.

Table A.2 — Selection factors, K_{sf}, for enclosed gear drives, driven by motors (hydraulic and electric) and turbines (steam or gas)

Application	Up to 3 h per day	Load duration 3 h to 10 h per day	Over 10 h per day
Agitators (mixers)			
Pure liquids	1,00	1,00	1,25
Liquids and solids	1,00	1,00	1,50
Liquids — variable density	1,00	1,25	1,50
Blowers			
Centrifugal	1,00	1,00	1,25
Lobe	1,00	1,25	1,50
Vane	1,00	1,25	1,50
Brewing and distilling			
Bottling machinery	1,00	1,25	1,25
Brew kettles — continuous duty	1,25	1,25	1,25
Cookers — continuous duty	1,25	1,25	1,25
Mash tubs — continuous duty	1,25	1,25	1,25
Scale hopper — frequent starts	1,25	1,25	1,25
Can filling machines	1,00	1,00	1,25
Car dumpers	1,50	1,75	2,00
Car pullers	1,00	1,25	1,50
Clarifiers	1,00	1,00	1,25
Classifiers	1,00	1,25	1,50
Clay working machinery			
Brick press	1,50	1,75	2,00
Briquette machine	1,50	1,75	2,00
Pug mill	1,00	1,25	1,50
Compactors	2,00	2,00	2,00
Compressors			
Centrifugal	1,00	1,00	1,25
Lobe	1,00	1,25	1,50
Reciprocating, multi-cylinder	1,50	1,50	1,75
Reciprocating, single-cylinder	1,75	1,75	2,00
Conveyors — general purpose			
Uniformly loaded or fed	1,00	1,00	1,25
— Heavy duty			
Not uniformly fed	1,00	1,25	1,50
— Reciprocating or shaker	1,50	1,75	2,00
Crusher			
Stone or ore	1,75	1,75	2,00
Dredges			
Cable reels	1,25	1,25	1,50
Conveyors	1,25	1,25	1,50
Cutter head drives	2,00	2,00	2,00
Pumps	2,00	2,00	2,00
Screen drives	1,75	1,75	2,00
Stackers	1,25	1,25	1,50
Winches	1,25	1,25	1,50
Elevators			
Bucket	1,00	1,25	1,50
Centrifugal discharge	1,00	1,00	1,25
Escalators	1,00	1,00	1,25
Freight	1,00	1,25	1,50
Gravity discharge	1,00	1,00	1,25
Extruders			
General	1,50	1,50	1,50
Plastics			
Variable speed drive	1,50	1,50	1,50
Fixed speed drive	1,75	1,75	1,75
Rubber			
Continuous screw operation	1,75	1,75	1,75
Intermittent screw operation	1,75	1,75	1,75

Table A.2 *(continued)*

Application	Up to 3 h per day	Load duration	
		3 h to 10 h per day	Over 10 h per day
Fans			
Centrifugal	1,00	1,00	1,25
Cooling towers	2,00	2,00	2,00
Forced draft	1,25	1,25	1,25
Induced draft	1,50	1,50	1,50
Industrial and mine	1,50	1,50	1,50
Feeders			
Apron	1,00	1,25	1,50
Belt	1,00	1,15	1,50
Disc	1,00	1,00	1,25
Reciprocating	1,50	1,75	2,00
Screw	1,00	1,25	1,50
Food industry			
Cereal cooker	1,00	1,00	1,25
Dough mixer	1,25	1,25	1,50
Meat grinders	1,25	1,25	1,50
Slicers	1,25	1,25	1,50
Generators and exciters	1,00	1,00	1,25
Hammer mills	1,75	1,75	2,00
Hoists			
Heavy duty	1,75	1,75	2,00
Medium duty	1,25	1,25	1,50
Skip hoist	1,25	1,25	1,50
Laundry			
Tumblers	1,25	1,25	1,50
Washers	1,50	1,50	2,00
Lumber industry			
Barkers — spindle feed	1,25	1,25	1,50
Main drive	1,75	1,75	1,75
Conveyors — burner	1,25	1,25	1,50
Main or heavy duty	1,50	1,50	1,50
Main log	1,75	1,75	2,00
Re-saw, merry-go-round	1,25	1,25	1,50
Conveyors			
Slab	1,75	1,75	2,00
Transfer	1,25	1,25	1,50
Chains			
Floor	1,50	1,50	1,50
Green	1,50	1,50	1,75
Cut-off saws			
Chain	1,50	1,50	1,75
Drag	1,50	1,50	1,75
Debarking drums	1,75	1,75	2,00
Feeds			
Edger	1,25	1,25	1,50
Gang	1,75	1,75	1,75
Trimmer	1,25	1,25	1,50
Log deck	1,75	1,75	1,75
Log hauls — incline — well type	1,75	1,75	1,75
Log turning devices	1,75	1,75	1,75
Planer feed	1,25	1,25	1,50
Planer tilting hoists	1,50	1,50	1,50
Rolls — live-off brg. — roll cases	1,75	1,75	1,75
Sorting table	1,25	1,25	1,50
Tipple hoist	1,25	1,25	1,50
Transfers			
Chain	1,50	1,50	1,75
Craneway	1,50	1,50	1,75
Tray drives	1,25	1,25	1,50
Veneer lathe drives	1,25	1,25	1,50

Table A.2 *(continued)*

Application	Up to 3 h per day	Load duration 3 h to 10 h per day	Over 10 h per day
Metal mills			
Draw bench carriage and main drive	1,25	1,25	1,50
Runout table			
Non-reversing			
Group drives	1,50	1,50	1,50
Individual drives	2,00	2,00	2,00
Reversing	2,00	2,00	2,00
Slab pushers	1,50	1,50	1,50
Shears	2,00	2,00	2,00
Wire drawing	1,25	1,25	1,50
Wire winding machine	1,25	1,50	1,50
Metal strip processing machinery			
Bridles	1,25	1,25	1,50
Coilers and uncoilers	1,00	1,00	1,25
Edge trimmers	1,00	1,25	1,50
Flatteners	1,25	1,25	1,50
Loopers (accumulators)	1,00	1,00	1,25
Pinch rolls	1,25	1,25	1,50
Scrap choppers	1,25	1,25	1,50
Shears	2,00	2,00	2,00
Slitters	1,00	1,25	1,50
Mills, rotary type			
Ball and rod	2,00	2,00	2,00
Spur ring gear	2,00	2,00	2,00
Helical ring gear	1,50	1,50	1,50
Direct connected	2,00	2,00	2,00
Cement kilns	1,50	1,50	1,50
Dryers and coolers	1,50	1,50	1,50
Mixers			
Concrete	1,25	1,25	1,50
Paper mills[a]			
Agitator (mixer)	1,50	1,50	1,50
Agitator for pure liquors	1,25	1,25	1,25
Barking drums	2,00	2,00	2,00
Barkers - mechanical	2,00	2,00	2,00
Beater	1,50	1,50	1,50
Breaker stack	1,25	1,25	1,25
Calendar[b]	1,25	1,25	1,25
Chipper	2,00	2,00	2,00
Chip feeder	1,50	1,50	1,50
Coating rolls	1,25	1,25	1,25
Conveyors			
Chip, bark, chemical	1,25	1,25	1,25
Log (including slab)	2,00	2,00	2,00
Couch rolls	1,25	1,25	1,25
Cutter	2,00	2,00	2,00
Cylinder molds	1,25	1,25	1,25
Dryers[b]			
Paper machine	1,25	1,25	1,25
Conveyor type	1,25	1,25	1,25
Embosser	1,25	1,25	1,25
Extruder	1,50	1,50	1,50
Fourdrinier rolls (includes lump breaker, dandy roll, wire turning, and return rolls)	1,25	1,25	1,25
Jordan	1,50	1,50	1,50
Kiln drive	1,50	1,50	1,50
Mt. Hope roll	1,25	1,25	1,25

Table A.2 *(continued)*

Application	Up to 3 h per day	Load duration: 3 h to 10 h per day	Load duration: Over 10 h per day
Paper mills[a] (cont.)			
Paper rolls	1,25	1,25	1,25
Platter	1,50	1,50	1,50
Presses — felt and suction	1,25	1,25	1,25
Pulper	2,00	2,00	2,00
Pumps — vacuum	1,50	1,50	1,50
Reel (surface type)	1,25	1,25	1,25
Screens			
Chip	1,50	1,50	1,50
Rotary	1,50	1,50	1,50
Vibrating	2,00	2,00	2,00
Size press	1,25	1,25	1,25
Super calendar[c]	1,25	1,25	1,25
Thickener (AC motor)	1,50	1,50	1,50
(DC motor)	1,25	1,25	1,25
Washer (AC motor)	1,50	1,50	1,50
(DC motor)	1,25	1,25	1,25
Wind and unwind stand	1,00	1,00	1,25
Winders (surface type)	1,25	1,25	1,25
Yankee dryers[b]	1,25	1,25	1,25
Plastics industry			
Primary processing			
Intensive internal mixers			
Batch mixers	1,75	1,75	1,75
Continuous mixers	1,50	1,50	1,50
Batch drop mill — two smooth rolls	1,25	1,25	1,25
Continuous feed, holding and blend mill	1,25	1,25	1,25
Compounding mill	1,25	1,25	1,25
Calendars	1,50	1,50	1,50
Secondary processing			
Blow molders	1,50	1,50	1,50
Coating	1,25	1,25	1,25
Film	1,25	1,25	1,25
Pipe	1,25	1,25	1,25
Pre-plasticizers	1,50	1,50	1,50
Rods	1,25	1,25	1,25
Sheet	1,25	1,25	1,25
Tubing	1,25	1,25	1,50
Pullers — barge haul	1,25	1,25	1,50
Pumps			
Centrifugal	1,00	1,00	1,25
Proportioning	1,25	1,25	1,50
Reciprocating			
Single acting, three or more cylinders	1,25	1,25	1,50
Double acting, two or more cylinders	1,25	1,25	1,50
Rotary			
Gear type	1,00	1,00	1,25
Lobe	1,00	1,00	1,25
Vane	1,00	1,00	1,25
Rubber industry			
Intensive internal mixers			
Batch mixers	1,75	1,75	1,75
Continuous mixers	1,50	1,50	1,50
Mixing mill — two smooth rolls (if corrugated rolls are used, then use the same selection factors that are used for a cracker-warmer).	1,50	1,50	1,50
Batch drop mill — two smooth rolls	1,50	1,50	1,50
Cracker warmer — two rolls; one corrugated roll	1,75	1,75	1,75
Cracker — two corrugated rolls	2,00	2,00	2,00
Holding, feed and blend mill — two rolls	1,25	1,25	1,25
Refiner — two rolls	1,50	1,50	1,50
Calendars	1,50	1,50	1,50

Table A.2 *(concluded)*

Application	Up to 3 h per day	Load duration 3 h to 10 h per day	Over 10 h per day
Sand muller	1,25	1,25	1,50
Sewage disposal equipment			
Bar screens	1,25	1,25	1,25
Chemical feeders	1,25	1,25	1,25
Dewatering screens	1,50	1,50	1,50
Scum breakers	1,50	1,50	1,50
Slow or rapid mixers	1,50	1,50	1,50
Sludge collectors	1,25	1,25	1,25
Thickeners	1,50	1,50	1,50
Vacuum filters	1,50	1,50	1,50
Screens			
Air washing	1,00	1,00	1,25
Rotary — stone or gravel	1,25	1,25	1,50
Travelling water intake	1,00	1,00	1,25
Sugar industry			
Beet slicer	2,00	2,00	2,00
Cane knives	1,50	1,50	1,50
Crushers	1,50	1,50	1,50
Mills (low speed end)	1,75	1,75	1,75
Textile industry			
Batchers	1,25	1,25	1,50
Calendars	1,25	1,25	1,50
Cards	1,25	1,25	1,50
Dry cans	1,25	1,25	1,50
Dryers	1,25	1,25	1,50
Dyeing machinery	1,25	1,25	1,50
Looms	1,25	1,25	1,50
Mangles	1,25	1,25	1,50
Nappers	1,25	1,25	1,50
Pads	1,25	1,25	1,50
Slashers	1,25	1,25	1,50
Soapers	1,25	1,25	1,50
Spinners	1,25	1,25	1,50
Tenter frames	1,25	1,25	1,50
Washers	1,25	1,25	1,50
Winders	1,25	1,25	1,50

[a] Selection factors for paper mill applications are applied to the nameplate rating of the electric drive motor at the motor rated based speed.
[b] Anti-friction bearings only. Use 1,5 for sleeve bearings.
[c] A selection factor of 1,00 may be applied at base speed of a super calendar operating overspeed range of part range constant horsepower, constant torque where the constant horsepower speed range is greater than 1,5 to 1. A selection factor of 1,25 is applicable to super calendars operating over the entire speed range at constant torque or where the constant horsepower speed range is less than 1,5 to 1.

Annex B
(informative)

Other enclosed gear drive components

B.1 Purpose

This annex provides additional information on other enclosed gear drive components.

B.2 Seals

The basic functions of the seals are to seal in oil, grease, or other fluids; to seal out dirt, fluids and contaminants; maintain applied pressure or vacuum; or perform a combination of the above.

Standard design lip type fluid seals are not intended to withstand pressure. An internal pressure materially reduces seal life, and means should be provided to relieve internal pressure build-up. Special seal design is required for high pressure sealing.

B.2.1 Shaft finish

Proper shaft finishes assure maximum sealing efficiency. The coarser the finish, the greater the risk of leakage and wear on the seal. Surface finish value is not, however, the only criterion for shaft finishes. Type of finish, direction of finishing marks, and spiral lead are all factors. Plunge ground finishes with concentric finish marks are preferred. When a finish lead is present, its direction should lead the fluid inward.

CAUTION — Oil seals should be selected in accordance with the seal manufacturer's recommendations.

B.2.2 Special seals and breathers

It is recognized that gear drives applied in certain industries and under certain atmospheric conditions should be equipped with special seals and breathers. Drives installed in dusty and corrosive atmospheres such as chemical plants, cement mills, or taconite processing plants should be equipped with special seals and breathers designed for these conditions. It is also recommended that drives which are to be exposed to severe moisture and vapour-laden atmospheres be equipped with moisture barrier seals and breathers. Some applications in wet locations, subject to direct or indirect wash down, such as in the paper and food industries, may preclude the use of breathers. In these cases expansion chambers may be used.

B.2.3 Seal retainers

Seal retainers are generally used to locate the seal in proper relationship to a shaft, or to lock a split type seal in place.

B.3 Bearing retainers

A bearing retainer is any device that supports a bearing. All retainers shall be designed to locate and maintain dimensional stability for the bearing and gearing in accordance with the bearing and gearing manufacturer's specifications. Types of bearing retainers include:

— locknuts: provide axial support for bearings and include a device to lock them in place;

— keeper plates: provide axial support for bearings;

- end caps: provide axial support for bearings mounted in the housing;
- cartridges or carriers: provide axial and radial support for bearings mounted in the cartridge or carrier;
- retaining rings: provide axial support for bearings;
- load rings: provide axial support for bearings.

B.4 Grease retainers

Grease retainers are generally located between the bearing cavity and oil sump area to retain grease in the bearings.

B.5 Dowels/pins

Many different types of dowels/pins are used to provide positive location between two or more parts or to prevent movement between parts under load.

These devices are often installed after two or more parts have been bolted together and are used to return these parts to the exact position required if disassembly is necessary. Care should be taken to see that the required holes have the proper size to provide a tight fit.

CAUTION — In all of the above cases the dowel pin manufacturer's recommendations for fit and strength should be followed.

B.6 Spacers

Spacers are generally used to position bearings, gears, and other components. The spacer construction and material shall be of sufficient strength, stiffness and diameter to provide proper support to adjacent components under maximum operating loads (internal and external) and shall withstand all loads imposed at assembly.

B.7 Oil level indicators

Oil level indicators are generally used to identify the proper oil level in a drive at its specified mounting position when it is not operating. Oil level indicators may also identify high and low level when the drive is not operating, or when the drive is in operation. The manufacturer shall specify under which conditions the oil level shall be checked. Typical oil level indicators include:

- pipe plug;
- bull's eye sight gage;
- standpipe;
- dipstick.

CAUTION — Vented oil gages should not be used on reducers without breathers, because a pressure build-up inside the reducer can give a false reading.

Annex C
(informative)

Thermal calculations

C.1 Purpose

This annex gives recommended thermal calculations to determine the bearing power loss, P_B, mesh power loss, P_M, oil seal power loss, P_S, gear windage and churning power loss, P_{WG}, bearing windage and churning power loss, P_{WB}, oil pump power loss, P_P, and the heat dissipated from a gear drive, P_Q.

The calculation methods of this annex must be used collectively together as a means of determining a thermal capacity. The methods of annex D shall also be used together without use of this annex. Mixing calculations from this annex with those of annex D may give a false indication of capacity.

C.2 Heat generation

Heat generation comes from two sources, load and non-load-dependent. The load-dependent power losses consist of the bearing power losses and the gear mesh power losses.

$$P_L = \sum_{i=1}^{n} P_{Bi} + \sum_{i=1}^{n} P_{Mi} \tag{C.1}$$

The non-load-dependent power losses consist of the oil seal power losses, the internal gear windage and oil churning power losses, the internal bearing windage and oil churning power losses, and the oil pump power consumed.

$$P_N = \sum_{i=1}^{n} P_{Si} + \sum_{i=1}^{n} P_{WGi} + \sum_{i=1}^{n} P_{WBi} + \sum_{i=1}^{n} P_{Pi} \tag{C.2}$$

These power losses shall be summed for each component in the gear drive.

C.2.1 Bearing power loss, P_B

Rolling contact bearing power loss, P_B, may be estimated by using equations C.3 and C.4. Both clockwise and counterclockwise rotations must be calculated. Values for the bearing coefficient of friction, f_b, may be approximated using the values from Table C.1. When more exact values are known, they should be used. For more detailed information, see [1], [2], [3] and [4].

$$P_B = \frac{T_b n_b}{9\,549} \tag{C.3}$$

where

T_b is the rolling bearing friction torque, in Nm;

$$T_b = \frac{f_b W (d_o + d_i)}{4\,000} \tag{C.4}$$

where

n_b is bearing shaft speed, in r/min;

f_b is bearing coefficient of friction (Table C.1);

W is bearing load, in N;

d_o is bearing outside diameter, in mm;

d_i is bearing bore, in mm.

The power losses for hydrodynamic plane bearings are to be calculated by acceptable methods, but are not given in this annex.

Table C.1 — Coefficient of friction f_b - bearings [1]

Type of bearing	Coefficient of friction[a], f_b
Radial ball bearing (single row deep groove)	0,001 5
Self-aligning ball bearing	0,001 0
Angular-contact ball bearing	0,001 3
Thrust ball bearing	0,001 3
Cylindrical roller bearing	0,001 1
Spherical roller bearing[b]	0,001 8
Tapered roller bearing[b]	0,001 8

[a] Variation in f_b depends on speed and load.

[b] f_b is greater on tapered and spherical roller bearings due to rubbing on the roller ends.

C.2.2 Mesh power loss, P_M

Mesh power losses are a function of the mechanics of tooth action and the coefficient of friction. Tooth action involves some sliding with the meshing teeth separated by an oil film.

The mesh efficiency is expressed as a function of the specific sliding velocities and the coefficient of friction.

The coefficient of friction is difficult to assess. Reliable published data is rather limited, especially at high pitch line velocities. In the past, windage and churning effects have often been combined with the assumed friction values. Ideally, the coefficient of friction depends on the lubricant properties, surface conditions and sliding velocity. It also changes with contact load factor, K.

For spur and helical gears, the following equation can be used to estimate the gear tooth mesh power losses [4]:

$$P_M = \frac{f_m \, T_1 \, n_1 \cos^2 \beta_w}{9\,549 M} \tag{C.5}$$

where

f_m is the mesh coefficient of friction at mesh oil temperature.

If the pitch line velocity, v, is 2 m/s < v < 25 m/s and K is 1,4 N/mm^2 < K < 14 N/mm^2, then f_m can be estimated by equation C.6. Outside these limits the mesh coefficient should be determined experimentally.

$$f_m = \frac{K^{0,35}}{L v^{0,23}} \tag{C.6}$$

where

- T_1 is the torque on the pinion, in Nm;
- n_1 is the rotational frequency of the pinion, in r/min;
- β_w is the helix angle at working diameter, in degrees;
- M is the mesh mechanical advantage;
- L is lubricant constant (see Table C.2);
- v is pitch line velocity, in m/s.

Table C.2 — Lubrication constant, L [6]

ISO VG	L
46	60,2
68	56,3
150	50,0
220	47,3
320	45,1
460	42,9

K is given by the equation:

$$K = \frac{1000 T_1 (z_1 + z_2)}{2b\, r_{w1}^2\, z_2} \tag{C.7}$$

where

- z_1 is the number of pinion teeth;
- z_2 is the number of gear teeth;
- b is the face width in contact with mating element, in mm;
- r_{w1} is the pinion operating pitch radius, in mm.

The equation for mesh mechanical advantage is:

$$M = \frac{2 \cos \alpha_w (H_s + H_t)}{H_s^2 + H_t^2} \tag{C.8}$$

where

- α_w is the operating transverse pressure angle, in degrees;
- H_s is the sliding ratio at start of approach;
- H_t is the sliding ratio at end of recess.

The values for H_s and H_t are:

$$H_s = (u+1)\left[\left(\frac{r_{o2}^2}{r_{w2}^2} - \cos^2 \alpha_w\right)^{0,5} - \sin \alpha_w\right] \tag{C.9}$$

$$H_S = \left(\frac{u+1}{u}\right)\left[\left(\frac{r_{o1}^2}{r_{w1}^2} - \cos^2 \alpha_w\right)^{0,5} - \sin \alpha_w\right] \qquad \text{(C.10)}$$

where

u is the gear ratio, z_2 / z_1;

r_{o2} is the gear outside radius, in mm;

r_{w2} is the gear operating pitch radius, in mm;

r_{o1} is the pinion outside radius, in mm.

C.2.3 Oil seal power loss, P_S

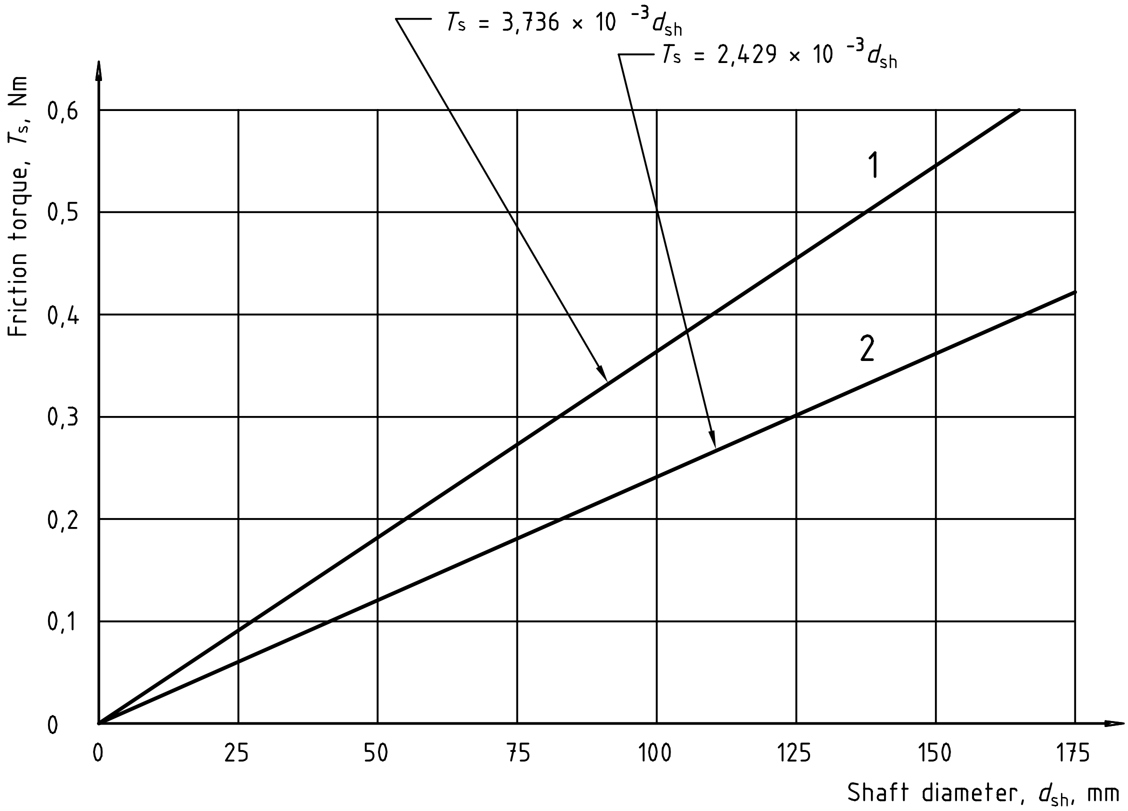
Contact lip oil seal power losses are a function of shaft speed, shaft size, oil sump temperature, oil viscosity, depth of submersion of the oil seal in the oil and oil seal design. Oil seal power losses can be estimated from equation C.11. Figure C.1 can be used to estimate oil seal frictional torque as a function of shaft diameter for oil seals typically used in gear drives, see [7].

$$P_S = \frac{T_S n}{9\ 549} \qquad \text{(C.11)}$$

where

T_S is the oil seal torque, in Nm (Figure C.1);

n is the shaft speed, in r/min.



Key

1 Viton

2 Buna N

Figure C.1 — Seal friction torque [5]

C.2.4 Gear windage and churning power loss, P_{WG}

For gear drives covered by this Technical Report, gear windage and churning power losses are generally combined into a single loss, This loss, P_{WG}, is estimated for each gear or pinion element, individually, from equation C.12. See [8].

$$P_{WG} = \frac{1{,}42 \times 10^{-11} \, d_w^2 \, n^2 \, b_t \cos^3 \beta_w \, m_n}{A} \tag{C.12}$$

where

d_w is the operating pitch diameter, in mm;

n is the shaft speed, in r/min;

b_t is the total face width, in mm;

m_n is the normal module, in mm.

The empirical arrangement constant is a function of the lubricant absolute viscosity, ξ, at 95 °C [9].

$$A = \frac{22\,440}{\xi} \tag{C.13}$$

C.2.5 Bearing windage and churning power loss, P_{WB}

For gear drives covered by this Technical Report, bearing windage and churning power losses are generally combined into a single loss. For other than tapered roller bearings, the windage and churning power losses are included in P_B.

For tapered roller bearings, the windage and churning power loss, P_{WB}, can be estimated for each bearing from equation C.14. See [9].

For tapered roller bearings only:

$$P_{WB} = \frac{1{,}42 \times 10^{-11} \, d_m^2 \, n^2 \, B \cos^3 \alpha_B \, D_R}{\pi \, 0{,}78 \, A} \tag{C.14}$$

where

d_m is the mean bearing diameter [1/2 (bearing cup outer diameter + bearing cone bore diameter)], of the tapered roller bearing, in mm;

B is the length thru bore of bearing, in mm;

D_R is the mean roller diameter, in mm;

α_B is the cup angle of a tapered roller bearing and can be determined by:

$$\tan \alpha_B = \frac{e}{1{,}5} \tag{C.15}$$

The value e is determined from the bearing manufacturer for the specific bearing number; or when e is not provided:

$$\tan \alpha_B = \frac{0{,}389}{K_5} \tag{C.16}$$

where

K_5 is the ratio of the basic dynamic radial load rating to basic dynamic thrust load rating.

The value of K_5 is available from the bearing manufacturer for the specific bearing number.

C.2.6 Oil pump power loss, P_P

The required power and capacity of most lubrication oil pumps varies directly with the speed. Thus the required power is a function of the oil flow and oil pressure at a given pump speed. For an oil pump driven by one of the reducer shafts, the oil pump power losses can be estimated by equation C.17 [9].

$$P_P = \frac{Qp}{60 e_P} \tag{C.17}$$

where

Q is oil flow, in l/min;

p is operating oil pressure, in N/mm^2;

e_P is oil pump efficiency.

For an oil pump driven by an electric motor, the power losses may be estimated using equation C.18, which considers the electric power consumed and the efficiencies of both the electric motor, e_m, and the oil pump, e_p (usually around 85 %).

$$P_P = E_P \left(\frac{e_m}{e_P} \right) \tag{C.18}$$

where

E_P is electric power consumed, in kW;

e_m is electric motor efficiency.

C.3 Heat dissipation, P_Q

The heat dissipated from a gear drive is influenced by the surface area of the gear drive, the air velocity across the surface, the temperature differential, ΔT, between the oil sump and the ambient air, the heat transfer rate from the oil to the gear case and the heat transfer rate between the gear case and the ambient air. The heat dissipation is given by equation C.19.

$$P_Q = A_C \, k \, \Delta T \tag{C.19}$$

NOTE A_C is the gear case surface area exposed to ambient air, not including fins, bolts or bosses.

The lubricant must be selected to accommodate the extreme conditions of the temperature differentials.

The heat transfer coefficient, k, is defined as the average value over the entire gear drive outer surface. The heat transfer coefficient will vary depending upon the material of the gear case, the cleanliness of the external surface, the extent of wetting of the internal surfaces by the hot oil, the configuration of the gear drive and the air velocity across the external surface.

For gear drives covered by this Technical Report, typical values for k can be found in Table C.3.

Table C.3 — Heat transfer coefficient, k, without auxiliary cooling

Condition	Air velocity m/s	Heat transfer coefficient[a] k kW/(m²·K)
Small confined space	< 1,40	0,010 to 0,014
Large indoor space	≤ 1,40	0,016 to 0,020
Large indoor space	> 1,40	0,018 to 0,022
Outdoors	> 3,70	0,020 to 0,025

[a] The choice of k values within each range is affected by the items listed in 7.1. Use of the high values in each range should be justified by test.

The heat transfer coefficient for a shaft fan cooled gear drive is a function of fan design, shroud design and fan speed. It will vary substantially depending upon the effectiveness of the fan and the proportion of the exterior surface cooled by the resulting air flow. The air velocity is defined to be the average air velocity over 60 % of the surface area, A_C, of the gear drive. The effect of using multiple fans on a gear drive could increase the average air velocity thereby resulting in a higher heat transfer coefficient. Table C.4 provides values of k for fan-cooled drives.[10]

Table C.4 — k for fan-cooled gear drives

Air velocity m/s	k kW/(m²·K)
2,5	0,015
5,0	0,024
10,0	0,042
15,0	0,058

C.4 Corrections for non-standard operating conditions

When the actual operating conditions for a specific application is different from the standard conditions defined in 7.1 and the thermal rating is calculated for the conditions of 7.1, the thermal rating may be modified for the application as follows:

$$P_{Thm} = P_T \; B_{réf} \; B_V \; B_A \; B_T \; B_D \qquad (C.20)$$

B_{ref} and B_A can be applied to natural or shaft fan cooling. B_V is to be applied only to natural cooling.

The gear drive manufacturer should be consulted when the conditions exceed the limits given in Tables C.5 through C.9 or when correction factors are required for any type of cooling other than natural or shaft fan.

When the ambient air temperature is below 25 °C, B_{ref} allows an increase in the thermal rating. Conversely, with an ambient air temperature above 25 °C, the thermal rating is reduced, see Table C.5.

Table C.5 — Ambient temperature vs. B_{ref}

Ambient temperature °C	B_{ref}
10	1,17
15	1,12
20	1,06
25	1,00
30	0,94
35	0,88
40	0,81
45	0,74
50	0,66

When the surrounding air has a steady velocity in excess of 1,4 m/s due to natural or operational wind fields, the increased convection heat transfer allows the thermal rating to be increased by applying B_V. Conversely, with an ambient air velocity of $\leqslant$ 0,50 m/s, the thermal rating is reduced. See Table C.6.

Table C.6 — Ambient air velocity, V_{ref} vs. B_V

V_{ref} m/s	B_V
$V_{ref} \leqslant 0,5$	0,75
$0,5 < V_{ref} \leqslant 1,4$	1,00
$1,4 < V_{ref} < 3,7$	1,40
$V_{ref} \geqslant 3,7$	1,90

At high altitudes the decrease in air density results in the derating factor B_A, see Table C.7.

Table C.7 — Altitude vs. B_A

Altitude m	B_A
0 – sea level	1,00
750	0,95
1 500	0,90
2 250	0,85
3 000	0,81
3 750	0,76
4 500	0,72
5 250	0,68

The standard maximum allowable oil sump temperature is 95 °C. A lower sump temperature requires a reduction in the thermal rating using B_T, see Table C.8. A maximum allowable sump temperature in excess of 95 °C will increase the thermal rating and can provide acceptable gear drive performance in some applications. However, it must be recognized that operating above 95 °C can reduce lubricant and contact seal life and increase the surface deterioration on the gears and bearings with a subsequent increase in the frequency of maintenance. The gear manufacturer should be consulted when a maximum allowable oil sump temperature in excess of 95 °C is being considered.

When a gear drive sees less than continuous operation with periods of zero speed, the resulting “cool-off” time allows the thermal rating to be increased by B_D, see Table C.9.

Table C.8 — Maximum oil sump temperature vs. B_T

Maximum oil sump temperature °C	B_T
65	0,60
85	0,81
95	1,00
105	1,13

Table C.9 — Operation time vs. B_D

Operation time per each hour	B_D
100 %	1,00
80 %	1,05
60 %	1,15
40 %	1,35
20 %	1,80

Annex D
(informative)

Alternate thermal calculations

D.1 Introduction

When power is transmitted by a gear unit, losses occur at the various components which are converted into heat. The losses, together with the drive power, determine the efficiency of the gear unit. Dependent on the heat dissipation via the lubricant to the housing and from there to the environment or via oil cooler to the coolant, in quasi-stationary state, a gear unit temperature occurs which in the case of high values causes rapid oil ageing, results in low oil film thicknesses in contact surfaces and reduces the load carrying capacity with pitting, wear and scuffing of tooth systems and bearings, as well as the service life of the seals. From calculation of the thermal balance, it is possible for splash lubricated gear units to determine the anticipated steady-state temperature, and for injection lubricated gear units the quantity of heat to be dissipated via the oil flow and the oil cooler.

Method A of determining the thermal load carrying capacity includes measurement on original gear units under practical conditions. This takes the form of either measurement of the power loss or of the heat dissipation or both, or, in the case of splash-lubricated gear units, determination of the quasi-stationary temperature in the oil sump. The methods of calculation for all individual components of power loss and heat dissipation described in this annex are to be regarded as an alternate, Method B.

The calculation methods of this annex shall be used collectively together as a means of determining a thermal capacity. The methods of annex C shall also be used together without use of this annex. Mixing calculations from annex C with those of annex D may give a false indication of capacity.

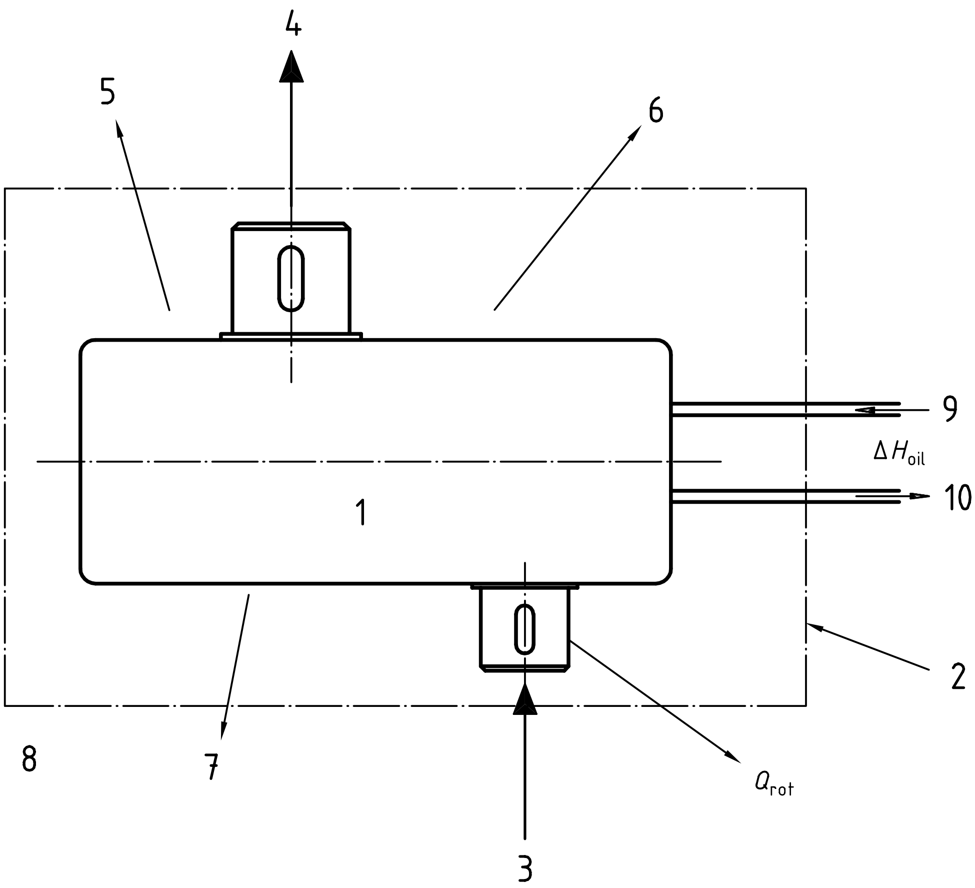
D.1.1 Purpose

With these calculations, it is possible to determine the power loss of the gear system: no-load and load-dependent losses of external and internal cylindrical gears, bevel, hypoid and worm gear systems; of the no-load and load losses of anti-friction and journal bearings; and of radial shaft seals. The calculations can be applied to single and multi-speed gear units, torque dividing gear units and to planetary gear units. The heat dissipation is calculated as free and/or forced convection as well as radiation from the housing, as forced convection and radiation from shafts and couplings, as heat conduction into the foundation and as heat dissipation via the lubricant and an external cooler when using injection lubrication.

The calculation is valid for quasi-stationary conditions; non-stationary conditions taking account of the heat capacity are not covered. In the case of gear units with intermittent duty (duty factor less than 100 %) and in the case of variable loads and speeds, calculation can be carried out, introducing a quasi-stationary equivalent input power.

The system limits are to be defined by the user such that all components of the heat input are recorded in the same way (see Figure D.1). In particular, it should be taken into account at the connection points with driving and driven machine whether heat flows can be dissipated from the gear unit at the coupling points or whether heat flows are passing from the machines connected into the gear unit.

For calculation of power losses and heat dissipation, the oil temperature is required. This must either be known or estimated as set point, or it can be determined from iteration taking account the heat dissipation.



Key

1 Gear unit
2 System boundary
3 Input power, P_a
4 Output power, P_b
5 Convection, Q_{ca}
6 Radiation, $Q_{\alpha Rad}$
7 Conduction, Q_{fun}
8 Environment
9 Oil inlet
10 Oil outlet

Figure D.1 — Individual heat flows on a gear unit (diagrammatic)

D.1.2 Application

The range of operating conditions assured by test rig trials is, where applicable, stated in the individual section of calculation. Extrapolation past the stated range increases the uncertainty factor, but has proved to be an adequate approximation in wide ranges.

D.2 Symbols and units

Symbol	Units	Description
a	mm	centre distance
A_{bot}	m^2	gear unit bottom area
A_{ca}	m^2	overall housing area (external)
A_{foot}	m^2	footprint of gear unit
A_{oil}	m^2	overall housing area (internal)
A_{pro}	m^2	projected fin area (housing external)
A_q	m^2	cross-sectional area
A_{fin}	m^2	total fin area (housing external)
A_{air}	m^2	ventilated housing area
b	mm	tooth width, bearing width
b_{eff}	mm	tooth contact width
b_0	mm	reference value of tooth width, b_0 = 10 mm
C_{lub}	—	lubrication factor
C_{Sp}	—	splash oil factor
C_0	N	static load rating of anti-friction bearing
$C_{1,2}$	—	factors
d_a	mm	tip circle diameter
d_{fl}	m	equivalent flange diameter
d_w	mm	pitch circle diameter
d_m	mm	mean bearing diameter
d_s	mm	pitch circle diameter of equivalent crossed helical gears
d_{sh}	m	shaft diameter
e	—	base of natural logarithm, e = 2,718
e	mm	tip circle immersion depth with oil level stationary
e_0	mm	reference value of immersion depth, e_0 = 10 mm
ED	—	duty factor
$f_{0,1,2}$	—	coefficients for bearing losses
F_a	N	bearing thrust load
F_{bt}	N	tooth normal force, transverse section
F_n	N	tooth normal force, normal section
F_r	N	radial bearing load
g	m/s^2	g = 9.81 m/s^2
Gr	—	Grashof number
h_c	mm	height of point of contact above the lowest point of the immersing gear
h_{ca}	mm	overall height of gear unit housing
H_v	—	tooth loss factor
ΔH_{oil}	W	enthalpic flow with oil
$h_{0,1}$	mm	lubrication gap heights
k	$W/(m^2 \cdot K)$	heat transmission coefficient
l_{fl}	m	equivalent length of coupling flange
l_h	mm	hydraulic length = 4 A_G/U_M
l_{fin}	m	depth of one fin
l_{sh}	m	length of free shaft end
m, m^*	—	fin factor
m	mm	module
n	1/min	speed
Nu	—	Nusselt number
P_A	W	input power
P_{Aeq}	W	equivalent input power
Pr	—	Prandtl number
P_V	W	power loss

Symbol	Units	Description
P_{VD}	W	seal power loss
P_{VL}	W	bearing power loss
P_{VX}	W	auxiliary power losses
P_{VZ}	W	gear power loss
P_0	N	equivalent static bearing load
P_1	N	equivalent bearing load
Q	W	total heat flow
Q_{ca}	W	heat flow across housing surface
Q_{fun}	W	heat flow across foundation
Q_{rot}	W	heat flow across shafts and couplings
Re	—	Reynolds number
$Ra_{1,2}$	µm	arithmetic average roughness of pinion and gear wheel
s	—	size factor of bearing
t	min	duration
T_H	Nm	hydraulic loss torque
T_{fl}	K	temperature of flange
T_{sh}	K	temperature of shaft
T_{VL}	Nm	total bearing loss torque
T_{VL0}	Nm	no-load bearing loss torque
$T_{VLP1,2}$	Nm	load-dependent bearing loss torque
T_{wall}	K	temperature of housing wall
T_{air}	K	cooling air temperature
T_{perm}	K	maximum permissible gear unit temperature
T_{∞}	K	ambient temperature
u	—	gear ratio
v, U	m/s	mean peripheral speed
$\dot{V}_{oil}$	l/min	oil injection rate
$\dot{V}_0$	l/min	reference oil injection rate, V_0 = 2 l/min
v_{gm}	m/s	mean sliding speed
v_{gs}	m/s	helical speed
$v_{g\tau 1,2}$	m/s	total surface speed at tooth tip
v_S	m/s	oil jet velocity
v_t	m/s	peripheral speed at pitch circle
v_{t0}	m/s	reference speed, v_{t0} = 10 m/s
v_{air}	m/s	impingement velocity
$V_{\Sigma C}$	m/s	sum velocity at pitch point
$V_{\Sigma h}$	m/s	sum velocity in direction of tooth depth
$V_{\Sigma m}$	m/s	mean resultant sum velocity
$V_{\Sigma s}$	m/s	sum velocity in direction of tooth length
x	—	addendum modification
X_L	—	lubricant factor
X_R	—	roughness factor
Y	—	axial factor from bearing tables, Y for $F_a/F_r > e$
Y_W	—	material factor
α_{fun}	W/(m²·K)	heat transfer coefficient at gear unit foundation
α_{ca}	W/(m²·K)	air-side transfer coefficient at housing
α_{con}	W/(m²·K)	heat transfer due to convection
$\alpha_{K,free}$	W/(m²·K)	heat transfer due to free convection
$\alpha_{K, forced}$	W/(m²·K)	heat transfer due to forced convection
α_{oil}	W/(m²·K)	oil-side heat transfer
α_{rad}	W/(m²·K)	heat transfer due to radiation
β	°	helix angle

Symbol	Units	Description
β_b	°	helix angle at base circle
σ_H	N/mm^2	contact stress
δ_{fin}	m	thickness of one fin
δ_{wall}	m	mean housing wall thickness
ε	—	emission ratio of gear unit housing
ε_α	—	profile contact ratio
$\varepsilon_{1,2}$	—	addendum contact ratio, pinion/gear wheel
λ_{fin}	W/(mK)	thermal conductivity of fin
λ_{fl}	W/(mK)	thermal conductivity of flange
λ_{fun}	W/(mK)	thermal conductivity of foundation
λ_{wall}	W/(mK)	thermal conductivity of housing
λ_{sh}	W/(mK)	thermal conductivity of shaft
μ	—	coefficient of friction
μ_{mz}	—	mean coefficient of friction of the gear mesh
$\nu_{40,100}$	mm^2/s	kinematic viscosity of oil at 40, 100°C
ν_{oil}	mm^2/s	kinematic viscosity of oil at operating temperature
ν_{air}	m^2/s	kinematic viscosity of air
ρ_C	mm	equivalent radius of curvature at pitch point of contact
ρ_n	mm	equivalent radius of curvature, normal section
ρ_{15}	kg/dm^3	density of oil at 15°C
ρ_{oil}	kg/dm^3	density of oil at operating temperature
ω	rad/s	angular velocity
η	—	efficiency
η_f	—	fin efficiency
η_{oil}	mPas	dynamic viscosity of oil at operating temperature
ϑ_{oil}	°C	oil temperature
ϑ_∞	°C	ambient temperature

Indices

0 Load-independent
1 Pinion
2 Gear wheel
C referred to the pitch point
n normal
P Load-dependent

D.3 Equivalent transmitted power

The mean equivalent transmitted power, P_{Aeq}, definitive for heat calculation is determined for gear units in continuous service with constant nominal loading from the rated power, P_A. As brief external or internal overloads do not play any part for the thermal balance and the internal heat distribution is not taken into account either, in every case all derating factors, such as for example in the case of gear calculation K_A, K_V, $K_{H\beta}$ and $K_{H\alpha}$, should be assumed to be 1,0. As with increasing load and decreasing speed the coefficient of friction increases, under operating conditions with equal transmitted power the most unfavourable conditions are present for slow speed.

In the case of variable load conditions as a function of time or in the case of gear units with a duty factor of less than 100 %, the equivalent transmitted power should be based on the power which assumes a maximum value averaged over the period recognized for quasi-stationary conditions.

In the case of splash lubricated gear units, a quasi-stationary condition is obtained in respect of oil temperature after 1 h to 3 h, depending on gear unit design. As a guide, one can assume the period until a largely quasi-stationary temperature is reached as being 1 h.

As an approximation therefore, the maximum possible mean power in this period can be substituted as the thermo-equivalent transmitted power. The following will apply:

$$P_{\text{Aeq}} = \frac{P_1 t_1 + P_2 t_2 \ldots + P_n t_n}{t_1 + t_2 + \ldots + t_n} \tag{D.1}$$

In the case of gear units with a duty factor of less than 100 %, the thermally equivalent power, P_{Aeq}, is determined from:

$$P_{\text{Aeq}} = \text{ED} P_{\text{A}} \tag{D.2}$$

with the duty factor ED as the operating time related to the total time. Here it is assumed that stationary and operating times are distributed uniformly over the operating period. When specifying the duty factor of electric motors, the reference period is usually based on t = 10 min.

NOTE As an aid to decision for equation D.2 in the determination of the thermally equivalent power for journal bearings, the duty factor is assumed as linear in the standards, as in equation D.2. For electric motors, the square root of the duty factor is substituted. For gear units, in one manufacturer's catalogue the cube root of the duty factor is used. In these cases the input power P_{A} has to be substituted by P_{Aeq} in the following chapters.

D.4 Power loss

The total power loss, P_{V}, produced in a gear unit consists of the load-dependent and the no-load losses of the tooth systems, P_{VZ}, and of the bearings, P_{VL}, as well as the load-independent losses of the seals, P_{VD}, and other gear unit components, P_{VX}:

$$P_{\text{V}} = P_{\text{VZ0}} + P_{\text{VZP}} + P_{\text{VL0}} + P_{\text{VLP}} + P_{\text{VD}} + P_{\text{VX}} \tag{D.3}$$

The efficiency, η, is then determined with the transmitted power, P_{A}, from:

$$\eta = \frac{P_{\text{A}} - P_{\text{V}}}{P_{\text{A}}} \tag{D.4}$$

D.4.1 Gear losses

The total gear losses consist of the no-load-dependent component, P_{VZ0}, and the load-dependent component, P_{VZP}. For cylindrical gears, bevel gears and hypoid gears, these are determined separately according to Niemann/Winter [1] [2], and jointly for worm gears. The losses of bevel gears are calculated on the equivalent cylindrical gear system, that of hypoid gears on the equivalent crossed helical gear system [2].

D.4.1.1 No-load gear losses for cylindrical, bevel and hypoid gears

The no-load gear system losses are determined according to Mauz [3]. In the case of the arithmetic formulations derived by Mauz, no distinction is made between splash and squeeze losses, as according to his investigations, the squeeze component is negligible.

a) **Splash lubrication:** The total hydraulic loss torque, T_{H}, is determined by the following formulation:

$$T_{\text{H}} = C_{\text{Sp}}\ C_1\ e^{C_2 \left(\frac{v_{\text{t}}}{v_{\text{to}}} \right)} \tag{D.5}$$

The splash oil factor, C_{Sp}, takes into account the effect of the splash oil supply dependent on the immersion depth, Figure D.2. The factors C_1 and C_2 state the effect of the tooth width and the immersion depth. In the case of low immersion depths, no effect of viscosity was measurable. For high immersion depth contradictory results for the influence of viscosity were found: in some cases power loss increased, in some decreased, with increasing viscosity. Therefore no account was taken of viscosity in the calculation equation.

$$C_{\text{Sp}} = \left(\frac{4\ e_{\text{max}}}{3\ h_{\text{c}}} \right)^{1,5} \frac{2\ h_{\text{c}}}{l_{\text{h}}} \tag{D.6}$$

$$C_1 = 0{,}063\left(\frac{e_1 + e_2}{e_0}\right) + 0{,}012\,8\left(\frac{b}{b_0}\right)^3 \quad \text{(D.7)}$$

$$C_2 = \frac{e_1 + e_2}{80\ e_0} + 0{,}2$$

where e_0 = 10 mm, b_0 = 10 mm.

The no-load power loss of each stage can be calculated by multiplying the no-load-torque with the angular velocity of the gear.

$$P_{\text{VZ0}} = \sum_{i=1}^{n} T_{\text{H}i}\,\frac{\pi\,n_i}{30} \quad \text{(D.8)}$$

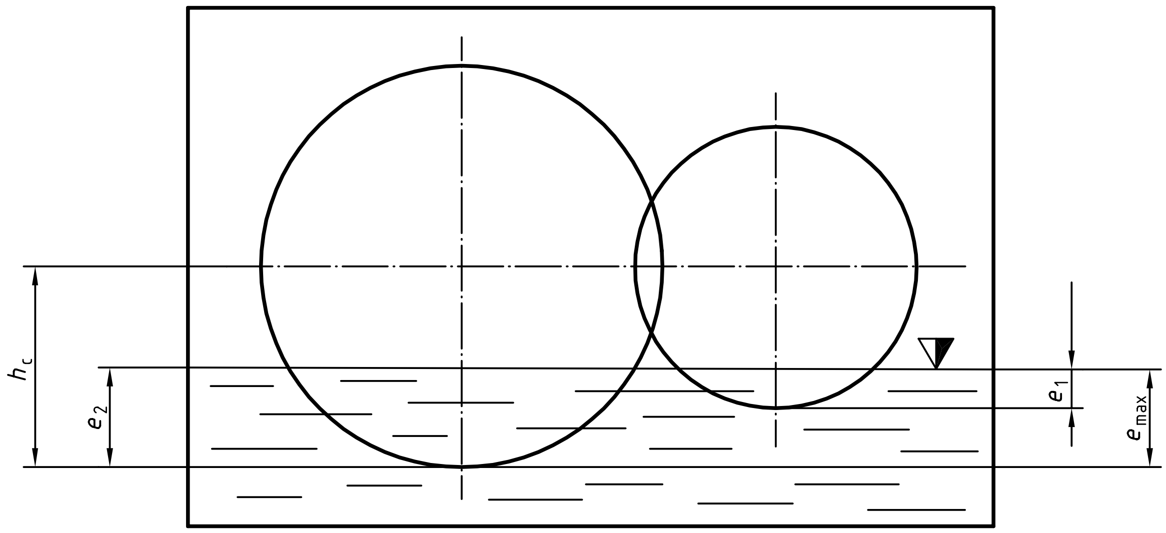


Figure D.2 — Splash oil factor according to Mauz [3]

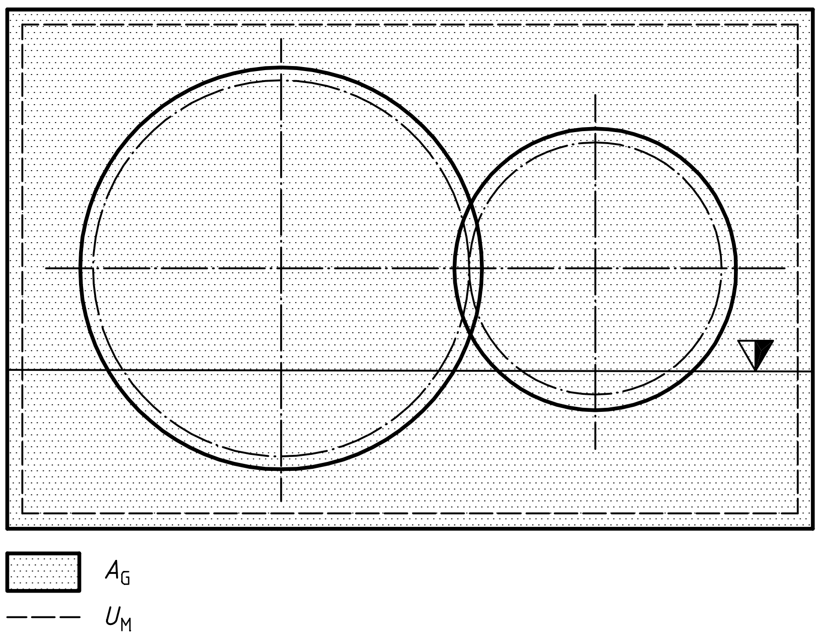


Figure D.3 — $l_h = 4A_G/U_M$

b) **Injection lubrication:** Evaluation of the experimental results in accordance with [3] resulted in the following equations:

— injection into the point of engagement:

$$T_H = 1{,}67 \times 10^{-6}\ \rho_{oil} \dot{V}_{oil} d_w (v_t - v_S) + 32 \times 10^{-9}\ \rho_{oil}\ d_w^{1{,}5}\ \nu_{oil}^{0{,}065}\ m_n^{0{,}18}\ b^{0{,}5}\ v_t^{1{,}5\left(\frac{\dot{V}_{oil}}{\dot{V}_0}\right)^{0{,}1} + 0{,}1} \tag{D.9}$$

with the reference oil injection volume $\dot{V}_0$ = 2 l/min;

— injection into the point of disengagement:

$$T_H = 8{,}33 \times 10^{-6}\ \rho_{oil}\ \dot{V}_{oil}\ d_w (v_t + v_S) \tag{D.10}$$

The equations are not dimensionless. The constants have been chosen so that on substitution of the individual variables in the units stated, the loss torque, T_H, in Nm is obtained for both equations. The loss torque thus calculated applies to the mating gear pair. The power loss of a pair of gears is obtained by multiplication by the angular velocity, ω, applicable to the pitch diameter, d_w, used. The total power loss of all pairs of gears is obtained by totalling the individual losses.

Application of the two equations is restricted by Mauz [3] to the operating and design parameters contained in Table D.1. Sample calculations show that the equations can usefully be applied well in excess of this range.

Table D.1 — Range of parameters examined according to Mauz [3]

Influence variable	Formula	Unit	Range of variation	
			from	to
Reynolds number	$Re = v_t d_a / \nu_{oil}$	—	4 125	531 428
Relative immersion depth	$2\ e / d_a$	—	0,04	2,0
Relative wall distance	s_r / d_a	—	0,03	3,15
Tip circle diameter	D_a	mm	132	248
Tooth width	b	mm	10	60
Immersion depth	e	mm	5	135
Modulus	m	mm	3	6
Peripheral speed	v_t	m/s	10	60
Kinematic viscosity	ν_{oil}	mm²/s	15	240
Density of oil	ρ_{15}	kg/dm³	855	881

D.4.1.2 Load-dependent gear losses

Generally, the Coulomb law is applicable to local power loss:

$$P_{VZP} = F_n(x)\ \mu(x)\ v_g(x) \tag{D.11}$$

with the local values of the tooth normal force $F_n(x)$, the coefficient of friction $\mu(x)$ and the sliding speed $v_g(x)$ at each point x of the path of contact.

Equation D.11 is evaluated per engagement, and is not for applicable planetary gear units.

As the coefficient of friction only changes slightly with the variable operating conditions on the path of contact, it is possible, for the purpose of approximation, to assume an average coefficient of friction. This can be determined for spur, bevel and hypoid gears according to the following equation:

$$\mu_{mz} = 0{,}048 \left(\frac{F/b}{V_\Sigma \rho} \right)^{0{,}2} \nu_{oil}^{-0{,}05} Ra^{0{,}25} X_L \tag{D.12}$$

where

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

and the lubricant factor X_L:

$X_L = 1{,}0$ for mineral oils;

$X_L = 0{,}8$ for polyalfaolefins and esters;

$X_L = 0{,}75\ (b/v_\Sigma)^{0{,}2}$ for polyglycols;

$X_L = 1{,}3$ for phosphoric esters;

$X_L = 1{,}5$ for traction fluids.

When calculating μ_{mz}, the following limits shall be observed:

— V_Σ for $v_t \leqslant 50$ m/s,
for $v_t > 50$ m/s, V_Σ for v_t is calculated as being = 50 m/s;

— $F/b \geqslant 150$ N/mm,
for $F/b \leqslant 150$ N/mm, μ_{mz} for F/b is calculated as being = 150 N/mm.

In equation D.12, the following should be substituted:

— for cylindrical and bevel gears:

$F = F_{btm}$;
$b = b$;
$V_\Sigma = V_{\Sigma C}$;
$\rho = \rho_{Cm}$;
$d_1 = d_{w1m}$;

— for hypoid gears:

$F = F_n \cos\beta_{b2}$;
$b = b_{eff} = 0{,}85\ b_2$;
$V_\Sigma = V_{\Sigma m}$;
$\rho = \rho_n$;
$d_1 = d_{s1}$.

For worm gear units, the coefficient of friction, μ_z, is calculated separately, as shown in D.4.1.5.

D.4.1.3 Load-dependent gear losses for cylindrical and bevel gears

Calculation of the load-dependent gear power loss, P_{VZP}, in accordance with [1]:

$$P_{VZP} = P_A\ \mu_{mz}\ H_v \tag{D.13}$$

with the average coefficient of friction, μ_{mz}, in accordance with equation D.11 and the tooth loss factor, H_v:

$$H_{\mathrm{v}} = \frac{\pi(u+1)}{z_1 \, u \cos\beta_{\mathrm{b}}}\left(1 - \varepsilon_\alpha + \varepsilon_1^2 + \varepsilon_2^2\right) \tag{D.14}$$

D.4.1.4 Load-dependent gear losses for hypoid gears

The load-dependent gear loss, P_{VZP}, of hypoid gears is calculated on the equivalent crossed helical gear system according to [2] in accordance with equation D.10 with the coefficient of friction, μ_{mz}, in accordance with equation D.11 as well as the average sum velocity, $V_{\Sigma\mathrm{m}}$, from:

$$V_{\Sigma m} = \sqrt{V_{\Sigma s}^2 + V_{\Sigma h}^2} \tag{D.15}$$

and the average sliding speed, v_{gm}, according to [2] from:

$$v_{\mathrm{gm}} = v_{\mathrm{gs}} + \frac{\left(v_{\mathrm{g}\gamma} - v_{\mathrm{gs}}\right)^2 + \left(v_{\mathrm{g}\gamma 2} - v_{\mathrm{gs}}\right)^2}{2\left(v_{\mathrm{g}\gamma 1} + v_{\mathrm{g}\gamma 2} - 2v_{\mathrm{gs}}\right)} \tag{D.16}$$

D.4.1.5 Gear losses of worm gear units

The gear losses of worm gear units are calculated according to [2] from:

$$P_{\mathrm{VZ}} = P_{\mathrm{VZP}} + P_{\mathrm{V0}} - P_{\mathrm{VL0}} \tag{D.17}$$

with the total no-load losses, P_{V0}, and the bearing no-load losses, P_{VL0}, in accordance with D.3.2. The load-dependent gear losses, P_{VZP}, are obtained from:

$$P_{\mathrm{VZP}} = F_{\mathrm{n}} \, \mu_{\mathrm{z}} \, v_{\mathrm{gm}} \tag{D.18}$$

with the coefficient of friction, μ_{z}, from:

$$\mu_{\mathrm{z}} = \mu_{\mathrm{z0}} Y_{\mathrm{W}} \sqrt{\frac{v_{\mathrm{gm}}}{V_\Sigma}} \sqrt[4]{\frac{R_{\mathrm{z}}}{R_{\mathrm{z0}}}} \tag{D.19}$$

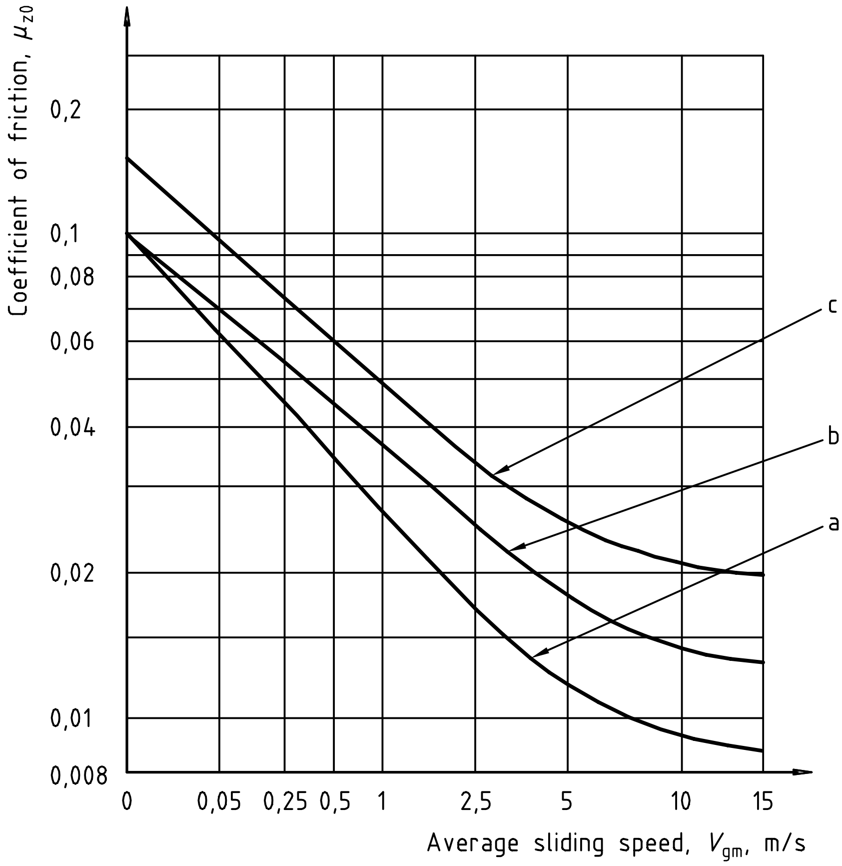
The basic value of the coefficient of friction, μ_{z0}, can be determined for any material/lubricant combination and standard conditions R_{z0}, σ_{H} and v_{gm}/V_Σ in a twin-disk test rig. For guide values, see Figure D.4.

The material factor, Y_{w}, takes other material combinations into account; for guide values, see Table D.2. The values given are valid for a case-hardened and ground worm. For through hardened, unground worms, the values should be multiplied by 1,2.

The ratio of average sliding speed, v_{gm}, to sum velocity, V_Σ, can be taken from EDP programs, for example in accordance with [4]. Guide values for *ZI, ZA, ZN* and *ZK* worms where $x \approx 0$: $v_{\mathrm{gm}}/V_\Sigma = 2{,}7$, for *ZH* worms where $x \approx +0{,}5$: $v_{\mathrm{gm}}/V_\Sigma = 2{,}2$.

If no measurements are available, the following can be assumed for gear units with anti-friction bearings, bottom mounted worm and oil splash lubrication for the total no-load power loss, P_{V0} [2]:

$$P_{V0} = a\left(\frac{n_1}{60}\right)^{\frac{3}{4}}\left(\frac{v_{40}}{1{,}83} + 90\right) \times 10^{-4} \tag{D.20}$$



Average contact stress σ_H = 300 N/mm^2

Slide roll ratio v_{gm}/V_{Σ} = 1

Material combination: hardened and ground steel (worm) against centrifugally cast bronze (worm gear)

Injection temperature 80 °C

[a] Polyglycol ν_{40} = 150 mm^2/s

[b] Mineral oil ν_{40} = 180 mm^2/s

[c] Mineral oil according to British Standard

Figure D.4 — Coefficients of friction according to tests on the twin-disk rig [4]

Table D.2 — Guide values for the material factor, Y_w

Worm material	Material factor Y_w
GZ-CuSn12Ni	0,95
GZ-CuSn12, GZ-CuSn10Zn, GZ-CuSn 14	1,00
GZ-CuZn25A15, GZ-CuAl10Ni	1,10
G-CuSn12Ni	1,20
G-CuSn12, G-CuSn10Zn, GGG-70	1,30
G-CuZn25Al5, G-CuAl11Ni, GG-25	1,40

D.4.2 Bearing losses

D.4.2.1 Rolling bearings

The bearing loss torque, T_{VL}, (in Nm) is calculated in accordance with the approximation formulae given in [5]. Here, the loss torque is split into a no-load, T_{VL0}, and a load-dependent, T_{VLP1}, part. In the case of axially loaded cylindrical roller bearings and axially loaded needle roller bearings, an additional loss term, T_{VLP2}, occurs which is dependent on the magnitude of the end thrust. These components are calculated separately and then added together.

This gives the following for the total loss torque:

$$T_{VL} = T_{VL0} + T_{VLP1} + T_{VLP2} \tag{D.21}$$

a) **No-load bearing power loss:** this component depends on the bearing design, the type of lubrication, the viscosity of the lubricant and the bearing speed.

— For the range $v_{oil}\, n < 2\,000$ mm²/s min, the following is valid:

$$T_{VL0} = 1{,}6 \times 10^{-8} f_0\, d_m^3 \tag{D.22}$$

— For the range $v_{oil}\, n \geqslant 2\,000$ mm²/s min, the following is valid:

$$T_{VL0} = 10^{-10} f_0 \left(v_{oil} n\right)^{2/3} d_m^3 \tag{D.23}$$

The coefficients, f_0, depends on bearing type and bearing lubrication (see Table D.3).

b) **Load-dependent bearing power loss:** for calculation of the load-dependent bearing loss torques, T_{VLP1} and T_{VLP2}, the following relationship applies according to [5]:

$$T_{VLP} = f\, P_1\, d_m\, 10^{-3} \tag{D.24}$$

where

— for radial loading:

$T_{VLP} = T_{VLP1}$;

$f = f_1$ from Table D.4;

P_1 from Table D.4;

— for cylindrical roller bearings with additional thrust loading:

$T_{VLP} = T_{VLP1} + T_{VLP2}$;

T_{VLP2}: $f = f_2$ from Table D.5;

$P_1 = F_a$

From the calculated loss torque, T_{VL}, it is possible to calculate the bearing power loss, P_{VL}, as follows:

$$P_{VL} = T_{VL}\varpi = T_{VL}\frac{\pi n}{30} \tag{D.25}$$

Table D.3 — Coefficient, f_0

Bearing design	Type of lubrication			
	Grease[a]	Oil mist	Oil bath	Oil injection, oil bath with vertical shaft
Deep-groove ball bearing				
single-row	0,75 ... 2[b]	1	2	4
double-row	3	2	4	8
Self-aligning ball bearing	1,5 ... 2[b]	0,7 ... 1[b]	1,5 ... 2[b]	3 ... 4[b]
Angular contact ball bearing				
single-row	2	1,7	3,3	6,6
double-row	4	3,4	6,5	13
Four-point contact bearing	6	2	6	9
Cylindrical roller bearing (cage)				
Series 10, 2, 3, 4	0,6	1,5	2,2	2,2[c]
Series 22	0,8	2,1	3	3[c]
Series 23	1	2,8	4	4[c]
Cylindrical roller bearing (full roller)				
single-row	5[d]	—	5	—
double-row	10[d]	—	10	—
Needle roller bearing	12	6	12	24
Self-aligning roller bearing				
Series 213	3,5	1,75	3,5	7
Series 222	4	2	4	8
Series 223, 230, 239	4,5	2,25	4,5	9
Series 231	5,5	2,75	5,5	11
Series 232	6	3	6	12
Series 240	6,5	3,25	6,5	13
Series 241	7	3,5	7	14
Taper roller bearing				
single-row	6	3	6	8 ... 10[b,c]
Deep-groove ball thrust bearing	5,5	0,8	1,5	3
Cylindrical roller thrust bearing	9	—	3,5	7
Needle roller thrust bearing	14	—	5	11
Self-aligning roller thrust bearing				
Series 292 E	—	—	2,5	5
Series 292	—	—	3,7	7,4
Series 293 E	—	—	3	6
Series 293	—	—	4,5	9
Series 294 E	—	—	3,3	6,6
Series 294	—	—	5	10

[a] The values shown are valid for steady conditions. For lately greased bearings (2...4) f_0 is to be used in the calculation.
[b] The low values apply to the lightweight bearings, and the high values to the heavyweight bearings of a bore series.
[c] Valid for oil injection lubrication. For oil bath lubrication and vertical shaft, the value shown is to be doubled.
[d] Valid for low rotation speed up to 20 % of the reference rotation speed (see bearing tables). At higher rotation speed the f_0 value is to be doubled for the calculation.

Table D.4 — Coefficient, f_1, and equivalent bearing load, P_1

Bearing design	f_1	P_1 [a]
Deep-groove ball bearing	$(0{,}000\,6 \ldots 0{,}000\,9)\,(P_0/C_0)^{0{,}5}$ [b]	$3\,F_a - 0{,}1\,F_r$
Self-aligning ball bearing	$0{,}000\,3\,(P_0/C_0)^{0{,}4}$	$1{,}4\,Y_2F_a - 0{,}1\,F_r$
Angular contact ball bearing		
single-row	$0{,}001\,(P_0/C_0)^{0{,}33}$	$F_a - 0{,}1\,F_r$
double-row	$0{,}001\,(P_0/C_0)^{0{,}33}$	$1{,}4\,F_a - 0{,}1\,F_r$
Four-point contact bearing	$0{,}001\,(P_0/C_0)^{0{,}33}$	$1{,}5\,F_a + 3{,}6\,F_r$
Cylindrical roller bearing (cage)		
Series 10	0,000 2	F_r [c]
Series 2	0,000 3	F_r [c]
Series 3	0,000 35	F_r [c]
Series 4, 22, 23	0,000 4	F_r [c]
Cylindrical roller bearing (full roller)	0,000 55	F_r [c]
Needle roller bearing	0,002	F_r
Self-aligning roller bearing		
Series 213	0,000 22	$1{,}35\,Y_2F_a$, if $F_r/F_a < Y_2$
Series 222	0,000 15	
Series 223	0,000 65	$F_r\,[1 + 0{,}35\,(Y_2\,F_a/F_r)^3]$,
Series 230,241	0,001	
Series 231	0,000 35	if $F_r/F_a \geqslant Y_2$
Series 232	0,000 45	
Series 239	0,000 25	(valid for all series)
Series 240	0,000 8	
Taper roller bearing		
single-row	0,000 4	$2\,YF_a$
single-row, doubled	0,000 4	$1{,}2\,Y_2F_a$
Deep-groove ball thrust bearing	$0{,}000\,8\,(F_a/C_0)^{0{,}33}$	F_a
Cylindrical roller thrust bearing, needle roller thrust bearing	0,001 5	F_a
Self-aligning roller thrust bearing		
Series 292 E	0,000 23	$F_a\,(F_{r\,max} \leqslant 0{,}55\,F_a)$
Series 292	0,000 3	
Series 293 E	0,000 3	(valid for all series)
Series 293	0,000 4	
Series 294 E	0,000 33	
Series 294	0,000 5	

[a] If $P_1 < F_r$, P_1 should be calculated as $= F_r$.
[b] The low values apply to the lightweight bearings, and the high values to the heavyweight bearings of a bore series.
[c] For additionally thrust-loaded cylindrical roller bearings, T_{VLP2} has to be introduced.

Table D.5 — Coefficient, f_2, for cylindrical roller bearings

Type of bearing	Type of lubrication	
	grease	oil
bearing with cage		
design EC	0,003	0,002
all others	0,009	0,006
full roller bearing		
single-row	0,006	0,003
double-row	0,015	0,009

D.4.2.2 Plain bearings

The power loss of hydrodynamically lubricated radial and thrust bearings is calculated in accordance with the statements in the relevant DIN standards.

Radial journal bearings as fully and partially surrounding regular cylinder bearings are calculated according to DIN 31 652 [6] as sectioned surface, and tilting pad bearings to DIN 31 657 [7].

Calculation of journal thrust bearings as segmental thrust bearings is laid down in DIN 31 653 [8], and as tilting pad thrust bearings in DIN 31 654 [9].

D.4.3 Shaft seals

For non-contacting seals, it can be assumed as an approximation that no contribution to power loss occurs.

A calculation statement for radial shaft seals is stated in [10]:

$$P_{VD} = 7{,}69 \times 10^{-6}\ d_{sh}^2\ n \qquad \text{(D.26)}$$

Other types of seals, such as mechanical seals, are not covered here.

D.5 Heat dissipation

The quantity of heat, P_V, generated in the gear unit by the power loss is balanced by the quantity of heat, Q, dissipated at the temperature level, ϑ_{oil}, ensuing. The latter consists of the heat dissipation via the housing Q_{ca}, via the foundation, Q_{fun}, via connected shafts and coupling, Q_{rot}, and, in the case of injection lubrication, via the heat transport of the cooling oil flow, ΔH_{oil}:

$$Q = Q_{ca} + Q_{fun} + Q_{rot} + \Delta H_{oil} \qquad \text{(D.27)}$$

From the equilibrium of heat quantity supplied and dissipated, it is possible by iteration to calculate the mean gear oil temperature, ϑ_{oil}, occurring. In the case of injection lubrication, it is additionally possible for the given gear oil temperature, ϑ_{oil}, to calculate necessary heat dissipation via the cooling oil flow and thus to obtain data for the oil flow rate required and the cooler design.

D.5.1 Heat dissipation through the housing

The quantity of heat dissipated through the housing by convection is calculated from:

$$Q_{ca} = kA_{ca}\,(\vartheta_{oil} - \vartheta_{\infty}) \qquad \text{(D.28)}$$

The heat transmission coefficient, k, includes the internal heat transfer between oil and housing, the heat conduction through the housing wall and the external heat transfer to the environment:

$$\frac{1}{k} = \frac{1}{\alpha_{oil}}\frac{A_{ca}}{A_{oil}} + \frac{\delta_{wall}}{\lambda_{wall}}\frac{A_{ca}}{A_{oil}} + \frac{1}{\alpha_{ca}} \qquad \text{(D.29)}$$

As a rule, the heat dissipation via the housing is determined by the larger value air-side thermal resistance at the housing surface. The first two terms in the above equation can then be neglected. For high air velocities and thus good external heat transfer, it will possibly be necessary to take account of the oil-side heat transfer as well. As a reference value, $\alpha_{oil} > 200$ W/(m^2·K) can be assumed. The heat conduction through the housing should only be taken into account in special cases, for example in the case of double-walled housings, housings with sound insulation and non-metallic housings. The appropriate coefficient of thermal conduction, λ_{wall}, has to be introduced for the housing material in question.

The air-side heat transmission, α_{ca}, incorporates a convection part, α_{con}, and a radiation part, α_{rad}:

$$\alpha_{ca} = \alpha_{con} + \alpha_{rad} \qquad \text{(D.30)}$$

The radiation part can be calculated from:

$$\alpha_{\text{rad}} = 0{,}23 \times 10^{-6} \varepsilon \left(\frac{T_{\text{wall}} + T_\infty}{2} \right)^3 \tag{D.31}$$

The convection part can originate from free or forced convection. According to investigations by Funck [11], the following can be stated:

$$\alpha_{\text{con}} = \alpha_{\text{K, free}} \left(1 - \frac{A_{\text{air}}}{A_{\text{ca}}} \right) + \alpha_{\text{K, forced}} \frac{A_{\text{air}}}{A_{\text{ca}}} \eta^* \tag{D.32}$$

where

$$\eta^* = \frac{T_{\text{wall}} - T_{\text{air}}}{T_{\text{wall}} - T_\infty} \tag{D.33}$$

For housings without thermal finning, the following can be stated:

for free convection:

$$\alpha_{\text{K, free}} = 18\, h_{\text{ca}}^{-0{,}1} \left(\frac{T_{\text{wall}} - T_\infty}{T_\infty} \right)^{0{,}3} \tag{D.34}$$

for forced convection:

$$\alpha_{\text{K, forced}} = \frac{0{,}008\,6 (Re')^{0{,}64}}{l_x} \tag{D.35}$$

where

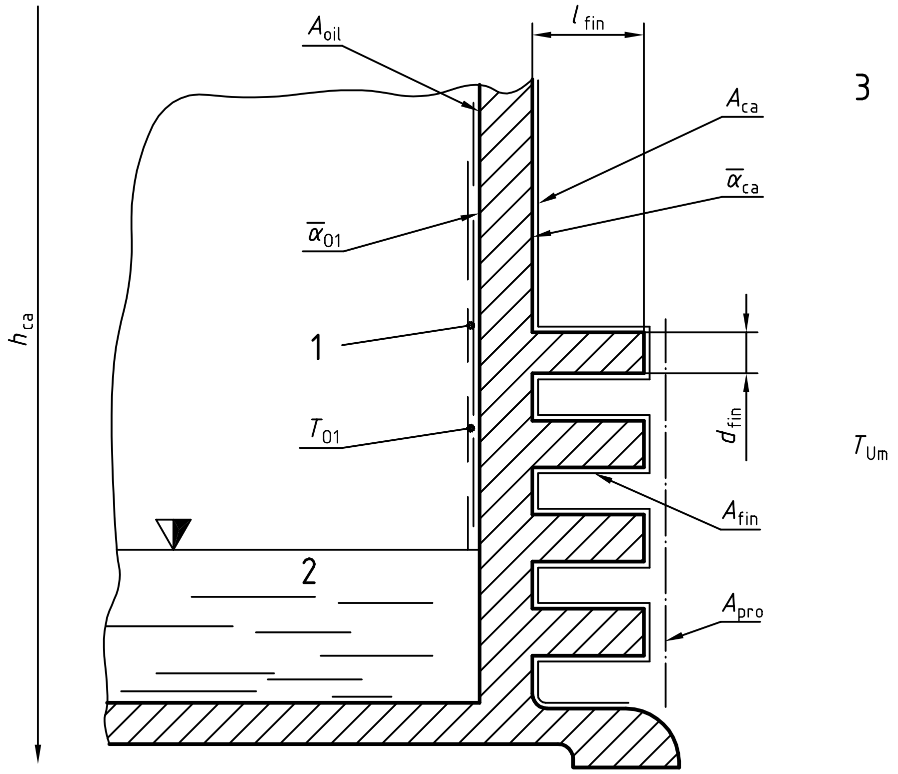
$$Re' = \sqrt{Re^2 + \frac{Gr}{2{,}5}} \tag{D.36}$$

$$Re = \frac{v_{\text{air}} l_x}{\nu_{\text{air}}} \tag{D.37}$$

$$Gr = \frac{g h_{\text{ca}}^3 (T_{\text{wall}} - T_\infty)}{T_\infty \nu_{\text{air}}^2} \tag{D.38}$$

Table D.6 — Emission ratio, ε

Material	Condition	Emission ratio, ε
grey cast iron GG	casting scale lathed or hobbed	0,60 – 0,80 0,35 – 0,45
steel	rolling skin lathed or hobbed hobbed and oil covered sandblasted sandblasted and oil covered	0,80 – 0,90 ≈ 0,15 ≈ 0,35 ≈ 0,35 0,50 – 0,60
aluminium	oxide skin lathed or hobbed	≈ 0,15 0,05 – 0,10
all materials painted	with and without oil or dust cover	0,90 – 0,95



Key

1 Oil film

2 Oil sump

3 Environment

Figure D.5 — Housing with thermal finning

For housings with thermal finning in accordance with Figure D.5, the following is valid: for free convection ($A_{air} = 0$):

$$\alpha_{ca} = \frac{A_{fin}}{A_{ca}}\left(\alpha_{K,\,free} + \alpha_{rad}\frac{A_{pro}}{A_{fin}}\right)\eta_f + \left(1 - \frac{A_{fin}}{A_{ca}}\right)\left(\alpha_{K,\,free} + \alpha_{rad}\right) \quad \text{(D.39)}$$

with the fin efficiency, η_f:

$$\eta_f = \frac{\tanh(m\, l_{fin})}{(m\, l_{fin})} \quad \text{(D.40)}$$

where

$$m = \sqrt{2\frac{\alpha_{con} + \alpha_{rad}\frac{A_{pro}}{A_{fin}}}{\delta_{fin}\,\lambda_{fin}}} \quad \text{(D.41)}$$

For free convection and ventilated fin surface ($A_{fin} = A_{air}$):

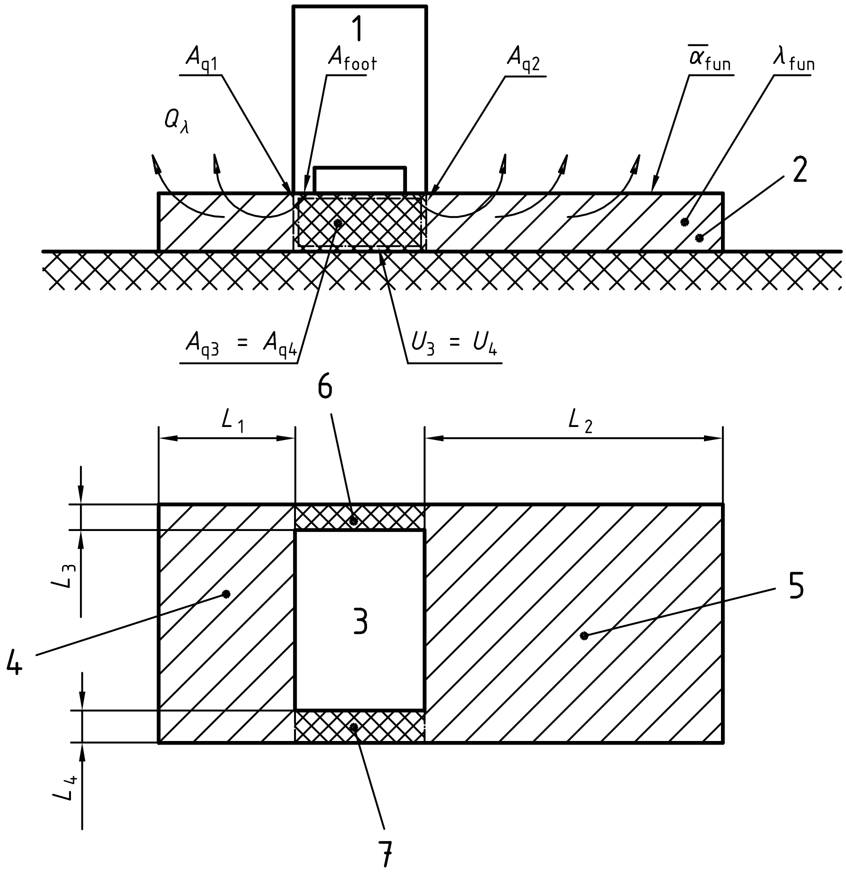
$$\alpha_{ca} = \frac{A_{air}}{A_{ca}}\left(\alpha_{K,\,forced}\,\eta^* + \alpha_{rad}\frac{A_{pro}}{A_{air}}\right)\eta_f + \left(1 - \frac{A_{air}}{A_{ca}}\right)\left(\alpha_{K,\,free} + \alpha_{rad}\right) \quad \text{(D.42)}$$

For free and forced convection ($A_{air} > A_{fin}$):

$$\alpha_{ca} = \left(1 - \frac{A_{air}}{A_{ca}}\right)\left(\alpha_{K,\,free} + \alpha_{rad}\right) + \frac{A_{air} - A_{fin}}{A_{ca}}\left(\alpha_{K,\,forced}\,\eta^* + \alpha_{rad}\right) + \frac{A_{fin}}{A_{ca}}\left(\alpha_{K,\,forced}\,\eta^* + \alpha_{rad}\frac{A_{pro}}{A_{fin}}\right)\eta_f \qquad (D.43)$$

D.5.2 Heat dissipation via the foundation

Calculation of the foundation conduction is based on division of the gear unit foundation into several single fins and uses the fin equation known from thermodynamics. The component heat flows along the surfaces, A_{qi}, are added to the overall foundation conduction (Figure D.6).



Key

1 Gear unit
2 Foundation
3 Gear unit
4 Fin 1
5 Fin 2
6 Fin 3
7 Fin 4

Figure D.6 — Heat dissipation through the foundation

$$Q_{fun} = f\lambda_{fun}\Delta T_{fun}\sum_{i=1}^{n} A_{qi}\, m_i^* \frac{\dfrac{\alpha_{fun}}{\lambda_{fun}\, m_i^*} + \tanh\left(m_i^*\, L_i\right)}{1 + \dfrac{\alpha_{fun}}{\lambda_{fun}\, m_i^*}\tanh\left(m_i^*\, L_i\right)} \qquad (D.44)$$

$$f = 1{,}46\left(\frac{A_{foot}}{A_{bot}}\right)^{0{,}16} \qquad (D.45)$$

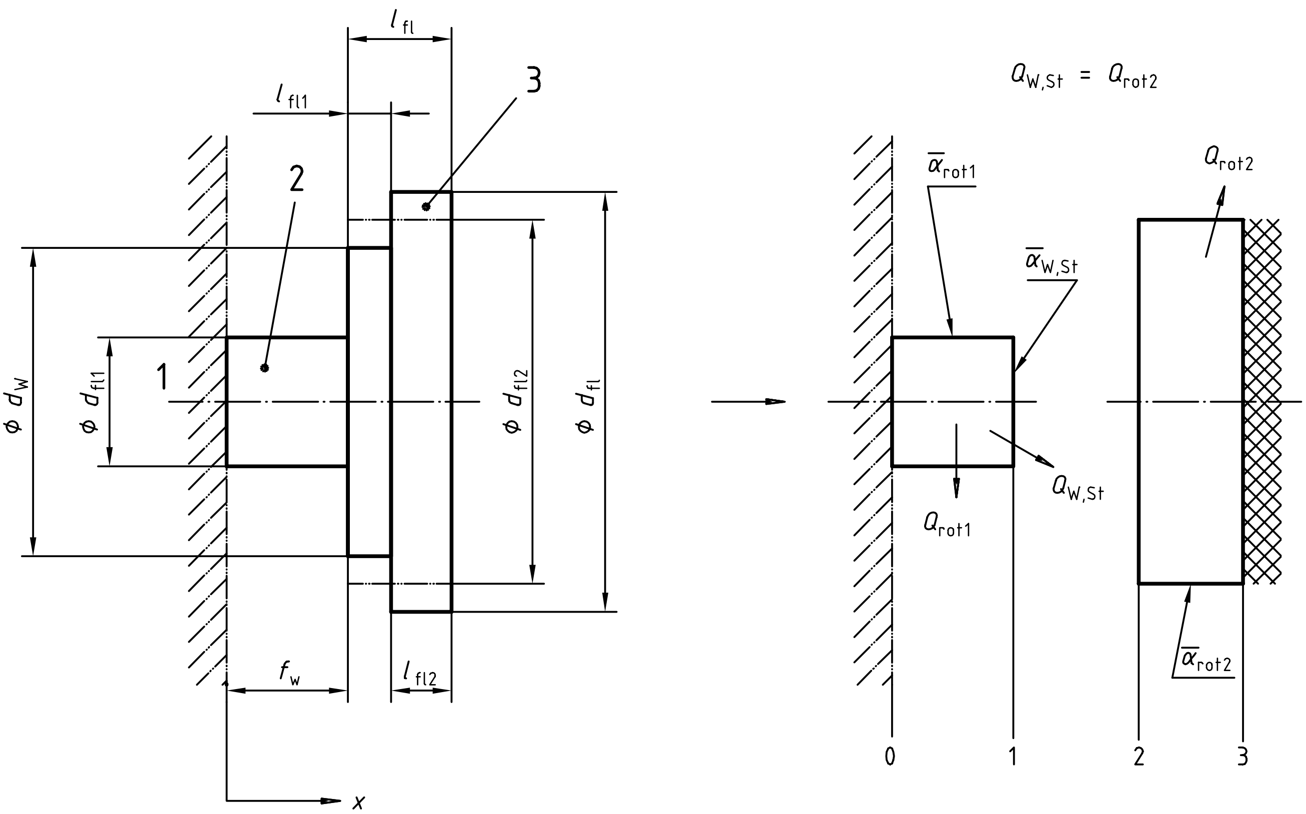
$$\Delta T_{\text{fun}} = 0{,}62\left(T_{\text{oil}} - T_{\infty}\right) \tag{D.46}$$

$$m_i = \sqrt{\frac{\alpha_{\text{fun}} U_i}{\lambda_{\text{fun}} A_{\text{q}i}}} \tag{D.47}$$

In the case of heat dissipation of the foundation in the upward direction only (insulated underneath), $m_i^* = 0{,}75 m_i$ should be assumed; in the case of heat dissipation of the foundation upwards and downwards, $m_i^* = m_i$ should be assumed.

D.5.3 Heat dissipation via shafts and couplings

Calculation of the heat dissipation via shafts and couplings also uses the fin equation. The heat transfer coefficient, $\alpha_{\text{rot 1,2}}$, effective at the shaft and coupling surface is calculated iterating as a function of the shaft speed according to [12] (Figure D.7).



Key

1 Gear unit

2 Shaft end

3 Coupling half

$$d_{\text{fl}} = \frac{d_{\text{fl1}}\ l_{\text{fl1}} + d_{\text{fl2}}\ l_{\text{fl2}}}{l_{\text{fl}}}$$

Figure D.7 — Equivalent system of shaft end and coupling flange

According to Figure D.7, the following is valid for the shaft/coupling system divided into two equivalent systems:

$$Q_{\text{rot}} = Q_{\text{rot1}} + Q_{\text{rot2}} \tag{D.48}$$

but

$$Q_{\text{rot1}} = \lambda_{\text{sh}}\, m_{\text{sh}}\, A_{\text{q,sh}} \left(T_{\text{sh}} - T_{\infty} \right) \Big|_{x=0} \frac{\dfrac{\alpha^{*}_{\text{sh, face}}}{\lambda_{\text{sh}}\, m_{\text{sh}}} + \tanh\left(m_{\text{sh}}\, l_{\text{sh}} \right)}{1 + \dfrac{\alpha^{*}_{\text{sh, face}}}{\lambda_{\text{sh}}\, m_{\text{sh}}} \tanh\left(m_{\text{sh}}\, l_{\text{sh}} \right)} \quad \text{(D.49)}$$

and

$$Q_{\text{rot2}} = \lambda_{\text{fl}}\, m_{\text{fl}}\, A_{\text{q,fl}} \left(T_{\text{fl}} - T_{\infty} \right) \Big|_{x = l_{\text{sh}}} \tanh\left(m_{\text{fl}}\, l_{\text{fl}} \right) \quad \text{(D.50)}$$

The cross-sectional areas, $A_{\text{q,sh}}$ and $A_{\text{q,fl}}$, are each calculated from the equivalent diameter, d_{sh} and d_{fl}, see Figure D.7.

The heat transfer coefficient, $\alpha^{*}_{\text{sh, face}}$, face equivalent to the heat flow from the shaft to the coupling is calculated from the relationship:

$$\alpha_{\text{sh, face}} = \frac{\lambda_{\text{fl}}\, m_{\text{fl}}\, A_{\text{q,fl}} \tanh\left(m_{\text{fl}}\, l_{\text{fl}} \right)}{A_{\text{q,sh}}} \quad \text{(D.51)}$$

The variables m_{sh} and m_{fl} follow from:

$$m_{\text{sh}} = 2 \sqrt{\frac{\alpha_{\text{rot1}}}{\lambda_{\text{sh}}\, d_{\text{sh}}}} \quad \text{(D.52)}$$

and

$$m_{\text{fl}} = 2 \sqrt{\frac{\alpha_{\text{rot2}}}{\lambda_{\text{fl}}\, d_{\text{fl}}}} \quad \text{(D.53)}$$

where the heat transfer coefficients at shaft and coupling — related to the equivalent diameter — are calculated according to Dropkin [12].

Dropkin [12] states equations for calculation of heat transfer coefficient at rotating shafts as a function of the Reynolds numbers for three different ranges:

For $Re \leqslant 2\,500$:

$$Nu = 0{,}40 Gr^{0{,}25} \quad \text{(D.54)}$$

For $2\,500 < Re \leqslant 15\,000$:

$$Nu = 0{,}095 \left(0{,}5 Re^{2} + Gr \right)^{0{,}35} \quad \text{(D.55)}$$

For $Re > 15\,000$:

$$Nu = 0{,}073 Re^{0{,}7} \quad \text{(D.56)}$$

where

$$Re = \frac{n \pi\, d^{2}_{\text{sh,fl}}}{60 \nu_{\text{L}}} \quad \text{(D.57)}$$

$$Nu = \frac{\alpha_{\text{rot1,2}}\, d_{\text{sh,fl}}}{\lambda_{\text{L}}} \quad \text{(D.58)}$$

$$Gr = g\left(2{,}5 d_{\text{sh,fl}}\right)^3 \frac{T_{\text{sh,fl}} - T_\infty}{T_\infty \nu_{\text{L}}^2} \tag{D.59}$$

The average temperatures of shaft and coupling, T_{sh} and T_{fl}, are obtained from integration of the temperature variation along the length of the shaft, l_{sh}, and the coupling, l_{fl}, where the relationships known from thermodynamics for the rod of finite length can be used for calculation of the temperature variation.

For the overtemperature of the shaft at the point $x = 0$, the following is substituted as an approximation:

$$\left(T_{\text{sh}} - T_\infty\right)\Big|_{x=0} = \left(T_{\text{oil}} - T_\infty\right) \tag{D.60}$$

In the experiments with actual gear units, the overtemperature at the beginning of the shaft was approximately up to 20 % below the oil overtemperature.

The overtemperature at the beginning of the coupling equivalent cylinder must be determined by iterating, it being assumed that:

$$\left(T_{\text{fl}} - T_\infty\right)\Big|_{x=l_{\text{sh}}} = \left(T_{\text{sh}} - T_\infty\right)\Big|_{x=l_{\text{sh}}} \tag{D.61}$$

and

$$\left(T_{\text{sh}} - T_\infty\right)\Big|_{x=l_{\text{sh}}} = \frac{T_{\text{oil}} - T_\infty}{\cosh\left(m_{\text{sh}}\, l_{\text{sh}}\right) + \dfrac{\alpha^*_{\text{sh,face}}}{\lambda_{\text{sh}}\, m_{\text{sh}}} \sinh\left(m_{\text{sh}}\, l_{\text{sh}}\right)} \tag{D.62}$$

Calculation is simplified if, instead of the integration, the mean temperature differences ($T_{\text{sh,fl}} - T_\infty$) are averaged arithmetically from the overtemperatures at the beginning and end of shaft and coupling. As the proportion of heat dissipation via shafts and couplings is only approximately 10 % of the total heat dissipation, this simplification is generally permissible for practical calculations.

It is then true that:

$$\left(T_{\text{sh}} - T_\infty\right) = \frac{1}{2}\left(T_{\text{oil}} - T_\infty\right)\left(1 + \frac{1}{\cosh\left(m_{\text{sh}}\, l_{\text{sh}}\right) + \dfrac{\alpha^*_{\text{sh,face}}}{\lambda_{\text{sh}}\, m_{\text{sh}}} \sinh\left(m_{\text{sh}}\, l_{\text{sh}}\right)}\right) \tag{D.63}$$

and

$$\left(T_{\text{fl}} - T_\infty\right) = \frac{1}{2}\left(T_{\text{sh}} - T_\infty\right)\Big|_{x=l_{\text{sh}}}\left(1 + \frac{1}{\cosh\left(m_{\text{fl}}\, l_{\text{fl}}\right)}\right) \tag{D.64}$$

D.5.4 Heat dissipation via an external cooler

The enthalpic flow, ΔH_{oil}, via the lubricant to an external cooler is calculated according to the following:

$$\Delta H_{\text{oil}} = 1{,}67 \times 10^{-2}\ \dot{V}_{\text{oil}}\, \rho_{\text{oil}}\, c_{\text{oil}}\, \Delta\vartheta_{\text{oil}} \tag{D.65}$$

Here, $c_{\text{oil}} = (1{,}7 \ldots 2{,}1) \times 10^3$ can be substituted as an approximation for the thermal capacity of the oil irrespective of the type of oil. As approximate values for the temperature difference in the cooler, $\Delta\vartheta_{\text{oil}}$, the following can be assumed:

— without cooler (only lines and pump outside the housing): 3 K ... 5 K

— with cooler on large gear units, continuous operation usually at rated power: 10 K ... 15 K

— with cooler on small gear units, periodic duty usually below 70 % rated power: 15 K ... 20 K

D.6 Results of calculation

D.6.1 Splash lubrication

In the case of splash lubricated gear units, the oil temperature occurring can be calculated by iterating from the thermal equilibrium of supplied power loss and dissipated quantity of heat:

$$P_{\mathrm{V}}(\vartheta_{\mathrm{oil}}) = Q(\vartheta_{\mathrm{oil}}) \tag{D.66}$$

When a maximum permissible oil temperature is specified, it can be checked whether the quantity of heat occurring for these conditions can be dissipated:

$$P_{\mathrm{V}}(\vartheta_{\mathrm{oil\ max}}) \leqslant Q(\vartheta_{\mathrm{oil\ max}}) \tag{D.67}$$

If this is not the case, the effectiveness of any modifications for reducing the power loss (for example oil viscosity, oil type, etc.) or to increase the heat dissipation (for example fins, fan, etc.) can be estimated. If such modifications are not adequate, external cooling should be provided by changeover to injection lubrication.

D.6.2 Injection lubrication

For injection lubricated gear units, with specified desired oil injection temperature, the enthalpic flow can be calculated which must be dissipated via the oil and an external cooler. For the possible temperature difference in the cooler, it is possible to estimate the inject flow rate which will be required for heat dissipation from the individual friction points.

Annex E
(informative)

Customer responsibility, storage, transportation, installation and testing

E.1 Purpose

This annex provides a detailed guide to the responsibilities and requirements for the procurement, transport, installation and commissioning of specified gear drives (non-catalogue drives).

E.2 Storage and protection against corrosion

Gear drives are normally shipped without protection against corrosion which is suitable for more than three months, indoors under cover, from the time of shipment.

The customer should specifically request any protection against corrosion to extend beyond this period and/or to withstand more extreme environments. Any protection specified shall be applied after all testing and inspection has been completed. If such protection is not contracted for, it is the responsibility of the customer.

In general, protection measures have a limited life. The manufacturer is responsible only for the protection he applies.

The customer can use any protection which may be applied without disassembling the gear drive. The customer shall consult with the manufacturer regarding recommended procedures to be followed.

E.3 Transportation

The conditions of transportation shall be indicated by the customer for the transport of the gear drive to his site:

— transport by train: particular care shall be taken to guard the gear drive against repeated shocks;

— transport by road or by water: it is recommended that shaft rotation be prevented;

— where a particular packing for transport is necessary, the customer shall specify the particular standards or regulations to be applied by the manufacturer to the gear drive.

E.4 Site installation

E.4.1 Premises

The gear drive shall preferably be placed in a closed area. It is recommended to have a separating wall from polluting machines, dust and radiating heat.

In cases of very hot or cold environments, the premises should be adapted to:

— permit normal cooling of the gear drive (high temperatures);

— avoid difficult starting conditions (low temperature).

In cases where such conditions cannot be met, the customer shall inform the manufacturer so that he can offer accessories necessary for the operation of the gear drive (heat exchanger, radiators, heaters, protective devices).

E.4.2 Handling

The manufacturer of the gear drive shall inform the customer of the type of spreaders or slings to be used for movement of the gear drive by travelling or cable crane.

In the case where handling is executed by rollers, the surfaces supporting the transport of the gear drive shall be protected by metal plates to avoid damage to the surface from the rolling contact with the rollers.

The handling is the responsibility of the installer who, in the case of large installations, shall inform and obtain the agreement of the gear drive manufacturer for the intended installation procedure.

E.4.3 Foundations

To assure that the gear drive has sufficient stability during its operation, the manufacturer shall, on receipt of the order for a gear drive, prepare and supply to the customer an installation drawing showing the position and magnitude of the forces at each point of attachment.

The customer shall design and use a concrete base or a frame with enough stability and rigidity to guarantee good operation and to withstand these loads.

In the case of large installations with a concrete base, the customer may give to the manufacturer a justification of the calculations of the base reinforcement and the results of measurement of the compression of test pieces taken from the concrete base.

The drawings of fasteners to be used for the mounting of the gear drive and the values of the torques to be exerted should be submitted to the gear drive manufacturer for his approval.

E.4.4 Levelling and alignment

In the majority of cases these two operations are done separately.

During levelling the tolerances given by the manufacturer of the gear drive shall be followed.

The measurements will be effected on the levelling extensions provided on the housing with a level graduated in steps of 0,02 mm/m to 0,05 mm/m.

The shims placed under the gear drive shall in no case have a thickness of less than 1 mm. When possible the shims should be made thicker or should be made of resin.

To ensure that shims are parallel to the bottom flange of the gear drive, shims should be put on metallic surfaces which are made an integral part of the concrete base or be held in resin that has been levelled beforehand.

The alignment of the input shaft and the output shaft shall be made to the appropriate tolerances for the coupling to be used.

The vertical displacement may be accomplished using shims. The lateral displacement to accomplish this operation shall be carried out with the help of jacks or screws or screw bolts on stops; in each case shocks to the gear drive should not occur.

The inspection of the alignment shall be carried out with the aid of a compairator or LASER by measuring:

— the parallel offset;

— angular misalignment.

These measurements shall be performed by simultaneous rotation of the coupling hubs.

At the time of installation the measurements should be made cold. The values obtained will then take into account the relative variation in the positions (angular and height) between the gear drive and the driving or driven machine. These values shall be compared with the limits supplied by the gear drive manufacturer.

It is recommended that the same measurements be taken again when the machines are at the normal operating temperature.

E.5 Inspection of the gear drive before start-up

After mounting of the gear drive and its alignment, the following operations shall be conducted:

— cleaning the teeth;

— measurement of the contact pattern on the teeth, made with the aid of blue marking ink. The traces obtained are to be “taken off" for record purposes with the aid of cellophane tape.

The first inspection shall be made before the drive is filled with oil. On drives which are difficult to turn by hand, the driving or driven units which are difficult to turn should be disconnected. They shall be reconnected with the coupling alignment as before.

NOTE Gears with corrected flanks may not have a full contact pattern without load.

In all cases the contact patterns obtained shall be compared to those taken when the gear was checked at the manufacturer's plant.

E.6 Corrosion inspection

There should be an inspection of the surface of the teeth and the general aspect of the bearings in order to detect any traces of corrosion or foreign matter.

Only small traces of corrosion located outside of the active surfaces of the teeth and bearings shall be accepted when they are thoroughly wiped off by hand repair, with the customer's agreement. Corrosion traces located on the active surfaces shall be carefully studied to determine if the corroded layer can be ground off within acceptable dimensions.

E.7 Seal surface inspection

If possible, the cover supporting the seal should be displaced to permit inspection of the actual running surface.

No corrosion under the seal shall be allowed. In the cases were corrosion is found, the seal shall be changed and the shaft surface re-finished.

When rubber is present on the shaft, the shaft shall be cleaned and oiled. The seal shall be renewed before running.

E.8 Inspection of the auxiliaries

The recommendations of the manufacturer should be followed.

After connecting the gear drive lubrication system, cooling fluids and the control system, the following checks should be made prior to starting the system:

— the correctness of connections for pipes made on site;

— the cleanliness of pipes made on site;

— the tightness of all connections;

— the direction of rotation of electric pumps, if fitted;

— the settings of pressure switches.

When the above are complete, run the gear drive lubrication system under pressure for 30 min and record flow values and check them against the manufacturer's specification. Use the inspection panels to check the flow of lubricant to each bearing and gear mesh.

E.9 No load testing of the gear drive

Whenever possible it is recommended to undertake a no load test of the gear drive at operating speed for a few hours.

Prior to running the gear drive, the following checks should be performed:

— check that the mounting bolts and the coupling bolts have been correctly torqued;

— check the lubrication system for integrity and levels.

During the run, the following checks should be performed:

— check for abnormal vibration;

— the stabilized values of temperature at the bearings and the oil sump shall be recorded and compared with those taken during the acceptance testing of the drive at the manufacturer's site.

At the end of the test, the filters shall be removed, cleaned and refitted.

In the case where the gear drive is heavily corroded, this test is necessary to remove the majority of the oxidation particles from the active surfaces of the gear drive. After this test the filters should be changed and, if necessary, the lubricating oil should be changed.

Annex F
(informative)

Testing and inspection

F.1 Purpose

This annex covers the testing and inspection procedures for assembled gear drives. Individual component inspection and process control are beyond the scope of this Technical Report.

When testing of the gear drive is required, the drive should be properly mounted for running test in the intended operating position to ensure that all facets of the assembly are correct. Under normal test conditions the gear drive is connected by coupling or belt drive to an electric motor that is available for the purpose at the manufacturer's test facility. The following applies to only those gear drives which are lubricated in accordance with manufacturer's recommendations and tested in a system of connected rotating parts. During testing, the system should be free from critical speeds, torsional vibrations, and overloads as tested at the gear drive manufacturer's facility.

F.2 Inspection of the assembled gear drive

The correct mating of a gear set not only depends on the accuracy of the gear teeth, but also on the position and the alignment of the gears' axes relative to each other. The components, having been fully approved prior to assembly, are assembled and proper tooth contact, backlash and bearing settings are verified.

F.3 Tooth contact inspection

Checking the tooth contact pattern (tooth bearing area) is an important test of the gear drive and is of value when gears have been mounted in a housing. The test will indicate whether the helix and pressure angle and the resultant base pitch of the mating gears meet the specified requirements to achieve optimal gear performance. The pinion profiles are generally coated with a marking compound and then rotated in mesh with the mating gear, and the resulting tooth pattern can be documented, see F.3.1.

The percentage of tooth contact will vary depending upon the loading of the gears, but the pattern obtained even under a no load condition shall provide the manufacturer with important information.

F.3.1 Procedure for tooth contact check

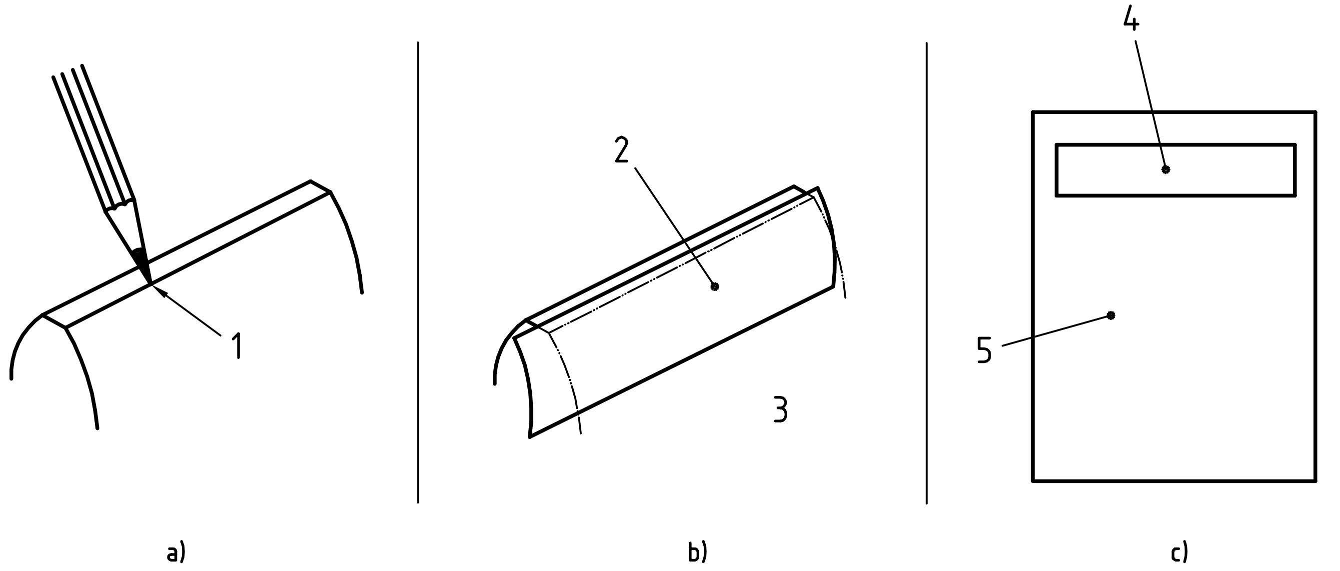
See Figure F.1.

a) Paint four or five tooth surfaces of the pinion uniformly with paint such as red lead dissolved with oil or oil-soluble quick drying ink to a thickness of about 5 µm.

b) Rotate the pinion manually. The tooth contact is then obtained from the paint transferred to the corresponding teeth of the gear.

c) To make the record of the tooth contact, apply carbon to the corner of each tooth so that the actual figure of the tooth will be identified. Stick the adhesive side of a plastic tape that is a little wider than the face width evenly to the surfaces. Peel off the tape and stick it on pasteboard.

NOTE See ISO/TR 10064-4 for tooth contact inspection.



Key

1 carbon
2 plastic tape
3 stick and peel
4 tape
5 pasteboard

Figure F.1 — Procedure for tooth contact check

F.4 Backlash

Backlash in gears is the clearance or play between mating tooth surfaces. The theoretical backlash of a gear set is based on the tooth thickness of each member in mesh as well as the centre distance at which the gears are operated. The actual backlash will be a function of the tolerances on tooth thickness, runout, lead, profile, centre distance, and by the temperature differences between the housing and the gears.

Functional backlash is the backlash at the tightest point of mesh on the pitch circle in a direction normal to the tooth surfaces when the gears are mounted in their assembled positions. Functional backlash is typically measured with feeler gauges or dial indicators normal to the gear tooth for a given mesh.

Circumferential backlash of the assembled drive is the reference circle arc length by which one gear can be rotated backwards and forwards, while the other gear, mounted at the prescribed centre distance is stopped. It is determined by using a linear compairator, and locating the stylus against one flank and at a tangent to the gear.

F.5 Rolling element bearings

When rolling element bearings are used, the manufacturer, based on his experience, the application and the recommendations of his bearing supplier shall determine the type of bearings and their settings. Assembly procedures normally require a tolerance to be established for the desired setting. An incorrectly set bearing can be a source of damage for the gear drive. Bearing end play may be set one shaft at a time and finally checked when both end cover plates are bolted in place with the required shims. End play should be checked to ensure compliance with the specification. Full end play is typically measured with the shaft moved all the way in one direction and then moved fully in the other direction. Total movement is the end play.

F.6 Testing procedure

For the purpose of a running test the following conditions apply.

F.6.1 Speed

A gear drive intended for service at a single speed shall be tested at that speed unless otherwise agreed upon between gear manufacturer and purchaser. The testing speed of a gear drive intended for service over a range of speeds should be negotiated between the manufacturer and the purchaser. A single test speed which is the arithmetic mean of the design speed range shall be the speed for testing in the absence of an agreement.

F.6.2 Loading

Gear drives may be operated with or without load at the gear manufacturer's discretion unless specific test loads are agreed upon and included as a part of the purchase contract. In individual cases, especially where unusually high speeds or power are involved, alternate operating conditions may be negotiated.

CAUTION — It is recommended that gear drives not be tested with loads in excess of gear unit rating, since such practice will reduce the design life of the drive.

F.6.3 Test requirements

The duration of the running test shall be decided by the gear drive manufacturer unless a specific time has been contractually agreed upon between manufacturer and purchaser.

Features such as oil tightness, noise level, temperature rise, axial and radial play of input and output shafts, contact pattern of the gear meshes, and lubrication system may be checked and recorded at this time.

F.6.4 Lubrication system performance

The lubrication system shall be checked for adequacy at certified speed or at both ends of the speed range if the speed is variable:

- on splash systems, the oil level shall be high enough to lubricate all components. It shall not be unnecessarily high, because sound and heat will be generated;
- on pressure lubrication systems, oil lines, troughs, gauges, pumps, filters, etc., shall be checked for performance and any leakage. Flow, pressure and temperature are to be recorded at regular intervals.

F.6.5 Other general test requirements

Any deviations from any applicable specifications on the certified print shall be noted on the test report.

All deficiencies such as oil leaks, excessive sound level, vibration, abnormal temperature rise, and insufficient tooth contact shall be corrected before the gear drive is shipped.

The ratio should be verified along with the assembly, shaft extension details, and direction of rotation.

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