TECHNICAL REPORT

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Gears —

Part 1: Thermal rating

Engrenages —

Partie 1: Capacité thermique

PROOF/ÉPREUVE

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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.

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 3.

The main task of technical committees is to prepare International Standards. Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

In exceptional circumstances, 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), it may decide by a simple majority vote of its participating members to publish a Technical Report. A Technical Report is entirely informative in nature and does not have to be reviewed until the data it provides are considered to be no longer valid or useful.

Attention is drawn to the possibility that some of the elements of this part of ISO/TR 14179 may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights.

ISO/TR 14179-1 was prepared by Technical Committee ISO/TC 60, *Gears*, Subcommittee SC 2, *Gear capacity calculation*.

ISO/TR 14179 consists of the following parts, under the general title *Gears*:

— *Part 1: Thermal rating*

— *Part 2: Thermal load-carrying capacity*

Introduction

ISO/TR 14179 consists of two parts.

This part of ISO/TR 14179 is the American proposal. It utilizes an analytical heat balance model to calculate the thermal transmittable power for a single or multiple stage gear drive lubricated with mineral oil. Many of the factors in the analytical model can trace their roots to published works of various authors.

The procedure is based on the calculation method presented in AGMA (American Gear Manufacturers Association) Technical Paper 96FTM9 [1]. The bearing losses are calculated from catalogue information supplied by bearing manufacturers, which in turn can be traced to the work of Palmgren. The gear windage and churning loss formulations originally appeared in work presented by Dudley, and have been modified to account for the effects of changes in lubricant viscosity and amount of gear submergence. The gear load losses are derived from the early investigators of rolling and sliding friction who approximated gear tooth action by means of disk testers. The coefficients in the load loss equation were then developed from a multiple parameter regression analysis of experimental data from a large population of tests in typical industrial gear drives. These gear drives were subjected to testing which varied operating conditions over a wide range. Operating condition parameters in the test matrix included speed, power, direction of rotation and amount of lubricant. The formulation has been verified by cross checking predicted results to experimental data for various gear drive configurations from several manufacturers.

ISO/TR 14179-2 is based on a German proposal whereby the thermal equilibrium between power loss and dissipated heat is calculated. From this equilibrium, the expected gear oil sump temperature for a given transmitted power, as well as the maximum transmittable power for a given maximum oil sump temperature, can be calculated. For spray lubrication, it is also possible to calculate the amount of external cooling necessary for maintaining a given oil inlet temperature. The calculation is an iterative method.

The power loss of cylindrical, bevel and hypoid and worm gears can be calculated according to theoretical and experimental investigations into those different gear types undertaken at the Technical University in Munich. The load dependent gear power loss results in the calculation of the coefficient of mesh friction. The influence of the main parameters of load, speed, viscosity and surface roughness on the coefficient of friction were measured individually in twin disk tests and verified in gear experiments. The same equations for the coefficient of friction are used in ISO/TR 13989 for the calculation of the scuffing load capacity of gears, and are used in German standard methods for the calculation of the relevant temperature for oil film thickness to evaluate the risk of wear and micropitting. The no-load power loss of gears is derived from systematic experiments with various parameters from published research projects. The power loss calculation of the anti-friction bearings was taken from the experience of the bearing manufacturers, as published in their most recent catalogues.

The equations for heat dissipation are based on theoretical considerations combined with experimental investigations on model gear cases using different gear wall configurations in natural and forced convection. Radiation from the housing is based on the Stefan-Boltzman law, with measured values of the relative radiation coefficient measured for different surface finish and coatings of the gear case surface. Also included are equations for the calculation of the heat transfer from rotating parts and to the foundation. The results were verified with heat dissipation measurements in practical gear drives. A computer programme, "WAEPRO", with the proposed thermal calculation method, was developed within a research project of the FVA (Forschungsvereinigung Antriebstechnik e.V., Frankfurt) and is widely used in the German gear industry.

Gears —

Part 1: Thermal rating

1 Scope

This part of ISO/TR 14179 utilizes an analytical heat balance model to provide a means of calculating the thermal transmittable power of a single- or multiple-stage gear drive lubricated with mineral oil. The calculation is based on standard conditions of 25 °C maximum ambient temperature and 95 °C maximum oil sump temperature in a large indoor space, but provides modifiers for other conditions.

2 Symbols and units

For the purposes of this part of ISO TR 14179, the symbols and units given in Table 1 apply.

Table 1 — Symbols and units

Symbol	Meaning	Units	Where first used	Reference
A_c	Gear case surface area exposed to ambient air	m^2	Eq. (35)	7.11
A_g	Arrangement constant for gearing	—	Eq. (24)	7.8
a	Load modifying exponent	—	Eq. (9)	Table 3
B_A	Altitude modifier	—	Eq. (36)	Table 10
B_D	Operating time modifier	—	Eq. (36)	Table 12
B_{ref}	Ambient temperature modifier	—	Eq. (36)	Table 8
B_T	Sump temperature modifier	—	Eq. (36)	Table 11
B_V	Ambient air velocity modifier	—	Eq. (36)	Table 9
b	Diameter modifying exponent	—	Eq. (9)	Table 3
b_w	Face width in contact with mating element	mm	Eq. (21)	7.3
C_0	Basic static load rating	N	—	Table 2
C_1	Mesh coefficient of friction constant	—	Eq. (20)	7.3
D	OD of element for gearing windage and churning	mm	Eq. (24)	7.8
D_{OR}	Bearing diameter over rolling elements	mm	Eq. (29)	Figure 3
D_s	Shaft diameter	mm	—	Figure 2
d_i	Bearing bore diameter	mm	Eq. (10)	7.2

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Table 1 (*continued*)

Symbol	Meaning	Units	Where first used	Reference
d_m	Bearing mean diameter	mm	Eq. (9)	7.2
d_o	Bearing outside diameter	mm	Eq. (10)	7.2
E_P	Electric power consumed	kW	Eq. (34)	7.10
e	Bearing factor	—	Eq. (13)	7.2.2
e_m	Electric motor efficiency	—	Eq. (34)	7.10
e_p	Oil pump efficiency	—	Eq. (33)	7.10
F	Total face width of gear or pinion	mm	Eq. (26)	7.8
F_a	Bearing axial load component	N	Eq. (12)	7.2.1
F_r	Bearing radial load component	N	Eq. (13)	7.2.2
f_g	Gear dip factor	—	Eq. (24)	7.8
f_m	Mesh coefficient of friction	—	Eq. (15)	Eq. (20)
f_0	Bearing dip factor	—	Eq. (27)	Table 5
f_1	Coefficient of friction for bearings	—	Eq. (9)	Table 2
f_2	Cylindrical roller bearing factor	—	Eq. (12)	Table 4
f_3	Bearing seal factor	—	Eq. (30)	Table 6
f_4	Bearing seal factor	—	Eq. (30)	Table 6
g	Load intensity modifying exponent	—	Eq. (20)	7.4
H	Depth that bearing rolling element dips in oil	mm	Eq. (29)	Figure 3
H_s	Sliding ratio at start of approach	—	Eq. (16)	Eq. (17)
H_t	Sliding ratio at end of recess	—	Eq. (16)	Eq. (18)
h	Pitch line velocity modifying exponent	—	Eq. (20)	7.4
j	Viscosity modifying exponent	—	Eq. (20)	7.4
K	Load intensity	N/mm^2	Eq. (20)	Eq. (21)
K_a	External axial force	N	—	7.3.3
k	Heat transfer coefficient	kW/(m^2 °C)	Eq. (35)	Table 7
L	Length of element for gearing windage and churning	mm	Eq. (24)	7.9
M	Mesh mechanical advantage	—	Eq. (15)	Eq. (16)
M_0	No-load torque moment for bearings	N · m	Eq. (27)	7.10
M_1	Bearing load dependent torque	N · m	Eq. (9)	7.3
M_2	Cylindrical roller bearing axial load dependent moment	N · m	Eq. (11)	7.3.2
M_3	Frictional moment of bearing seal	N · m	Eq. (30)	7.10
m_t	Transverse tooth module	—	Eq. (23)	7.9
n	Rotational shaft speed	rpm	Eq. (11)	7.3

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Table 1 (*continued*)

Symbol	Meaning	Units	Where first used	Reference
n_1	Pinion rotational speed	rpm	Eq. (15)	7.4
P	Bearing load	N	Eq. (13)	7.3.3
P_A	Transmitted power	kW	Eq. (3)	7
P_B	Total bearing losses (all bearings)	kW	Eq. (7)	7.2
P_{Bi}	Individual bearing load power loss	kW	Eq. (11)	7.3
P_{GWi}	Individual gear windage and churning loss	kW	Eq. (24)	7.9
P_L	Load dependent losses	kW	Eq. (2)	Eq. (3)
P_M	Total gear mesh losses (all meshes)	kW	Eq. (7)	7.4
P_{Mi}	Individual loaded mesh power loss	kW	Eq. (15)	7.4
P_N	Non-load dependent losses	kW	Eq. (2)	Eq. (8)
P_P	Total oil pump power required (all pumps)	kW	Eq. (8)	7.11
P_{Pm}	Motor driven oil pump power	kW	Eq. (32)	Eq. (34)
P_{Ps}	Shaft driven oil pump power	kW	Eq. (32)	Eq. (33)
P_Q	Heat dissipated	kW	Eq. (1)	7.12
P_S	Total oil seal losses (all seals)	kW	Eq. (8)	7.8
P_{Si}	Individual oil seal power loss	kW	Eq. (22)	7.8
P_T	Basic thermal power rating	kW	Eq. (6)	7
P_{THm}	Adjusted thermal power rating	kW	Eq. (36)	8
P_V	Heat generated	kW	Eq. (1)	Eq. (2)
P_W	Total combined windage and churning losses (of all meshes)	kW	Eq. (8)	7.9
P_{WB}	Oil churning losses, bearings (all bearings)	kW	Eq. (8)	7.10
P_{WBi}	Individual bearing churning power loss	kW	Eq. (31)	7.10
P_0	Equivalent static bearing load	N	—	Table 2
P_1	Bearing dynamic load	N	Eq. (9)	Table 2
p	Operating oil pressure	N/mm^2	Eq. (33)	7.11
Q	Oil flow	l/min	Eq. (33)	7.11
R_f	Roughness factor for gear teeth	—	Eq. (23)	7.9
r_{o1}	Pinion outside radius	mm	Eq. (18)	7.4
r_{o2}	Gear outside radius	mm	Eq. (17)	7.4
r_{w1}	Pinion operating pitch radius	mm	Eq. (18)	7.4
r_{w2}	Gear operating pitch radius	mm	Eq. (17)	7.4
T_S	Oil seal torque	N · m	Eq. (22)	Figure 2

Table 1 (*continued*)

Symbol	Meaning	Units	Where first used	Reference
T_1	Torque on the pinion	N · m	Eq. (15)	7.4
u	Gear ratio	—	Eq. (17)	7.4
V	Pitch line velocity	m/s	Eq. (20)	7.4
v_s	Sliding velocity at mean worm diameter	m/s	—	7.6
W_t	Tangential tooth load on worm gear	N	—	7.6
Y, Y_2	Bearing factors	—	Eq. (14)	7.3.3
z_1	Number of pinion teeth	—	Eq. (19)	7.4
z_2	Number of gear teeth	—	Eq. (19)	7.4
α_w	Operating transverse pressure angle	degrees	Eq. (16)	7.4
β	Generated helix angle	degrees	Eq. (26)	7.9
β_w	Operating helix angle at operating pitch diameter	degrees	Eq. (15)	7.4
ΔT	Temperature differential	°C	Eq. (35)	7.12
K	Viscosity ratio	—	—	7.3.2
η	Efficiency	%	Eq. (5)	7
μ	Coefficient of friction for worm gears	—	—	7.6
v	Kinematic viscosity of the oil at operating temperature	cSt	Eq. (20)	7.4

3 Principle

3.1 General

Maintaining an acceptable temperature in the oil sump of a gear drive is critical to its life. Therefore, in the selection of a gear drive, not only the mechanical rating but also the thermal rating must be considered.

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

3.2 Rating criteria

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 and may reduce the life of these elements. To achieve the required life and performance of a gear drive, the operating oil sump temperatures must be evaluated and limited.

Thermal ratings of gear drives rated by this method 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 clause 8).

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Additional criteria that must 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;

— continuous operation.

Modifying factors for deviation from these criteria are given in clause 8.

4 Service conditions

4.1 Intermittent service

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

4.2 Adverse conditions

The ability of a gear drive to operate within its thermal power rating may 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.

4.3 Favourable conditions

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

4.4 Auxiliary cooling

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

— fan cooling, in which case the fan shall maintain the fan cooled thermal power rating;

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

5 Methods for determining the thermal rating

Thermal rating may be determined by one of two methods: method A, testing, or method B, calculation.

Method A, a test of full scale gear drives at operating conditions, is the most accurate means of establishing the thermal rating of the gear drive. See clause 6.

When method B is used, the thermal rating of a gear drive can be calculated using the heat balance equation, which equates heat generated with heat dissipated. See clause 7 (the means of calculating heat generation is discussed in 7.2 to 7.11; for heat dissipation, in 7.12).

6 Method A — Test

Testing a specific gear drive at its design operating conditions is the most reliable means of establishing the thermal rating. Thermal testing involves measuring the steady-state bulk 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.

While no-load testing cannot yield a thermal rating, it may be used to approximate the heat transfer coefficient for comparison purposes, provided the power required to operate the drive at no-load is measured.

The following are some guidelines for acceptable thermal testing.

— The ambient air temperature and velocity must 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 $\leqslant$ 1 °C/h.

The oil temperature in the sump at various locations can vary by as much as 15 °C. The location of the temperature measurement should represent the bulk oil temperature. 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 fan cooling, the air velocity distribution over the housing surface can be measured.

7 Method B — Calculations for determining the thermal power rating, P_T

7.1 General

The calculation of thermal rating, P_T, 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 losses, P_V, at P_A are equal to the heat dissipation, P_Q, of the gear drive.

$$P_Q = P_V \tag{1}$$

When this is satisfied under the conditions of 3.2, P_T is P_A.

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

$$P_V = P_L + P_N \tag{2}$$

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P_L is a function of the input power, P_A.

$$P_L = f(P_A) \tag{3}$$

Using Equation 1 and rearranging terms, we can write the basic heat balance equation as follows:

$$P_Q - P_N - f(P_A) = 0 \tag{4}$$

To determine the basic thermal rating, P_T, vary P_A until Equation 4 is satisfied. This can be done by recalculating the load dependent losses, P_L, at different input powers, P_A. If $P_Q \leqslant P_N$ at no-load, the gear drive does not have any thermal capacity. The design must be changed to increase P_Q or auxiliary cooling methods must be used.

When Equation 4 is satisfied, the overall unit efficiency, η, is calculated as follows:

$$\eta = 100 - \frac{P_L + P_N}{P_A} \times 100 \tag{5}$$

The thermal rating of the gear drive is as follows:

$$P_T = \frac{P_Q}{1 - \frac{\eta}{100}} \tag{6}$$

The following thermal model has been established using empirical factors. It is based on the experience of several gear manufacturers. The model has been validated by extensive testing of concentric-shaft, base-mounted reducers with shafts mounted in a horizontal orientation. Limited testing of some parallel shaft gear units has also been performed in order to spot check the adequacy (validity) of the model. Values of some variables such as arrangement constant, heat transfer coefficient and coefficient of friction may not adequately address other enclosed drive configurations and operating conditions. These configurations or conditions may necessitate modifications of the particular variables. Changing any variable requires care and testing to ensure that the principles of the heat balance formulation are not violated.

7.2 Heat generation

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

The load dependent losses are comprised of the sum of all the individual bearing losses, P_B, and the sum of all the individual gear mesh losses, P_M:

$$P_L = \sum P_B + \sum P_M \tag{7}$$

The non-load dependent losses consist of the sum of all the individual oil seal losses, P_S, the sum of all the individual internal windage and oil churning losses for the gears and bearings, P_W and P_{WB}, respectively, and the sum of all the individual oil pump powers, P_P, consumed.

$$P_N = \sum P_S + \sum P_W + \sum P_{WB} + \sum P_P \tag{8}$$

These losses must be summed for each occurrence in the gear drive.

7.3 Bearing power loss, P_B

7.3.1 General

Page 49 of [2] provides a method of calculating the load dependent losses for bearings. Equation 9 gives the value for the torque on each bearing as a function of the applied load. The coefficient of friction, f_1, and the equations for calculating the load, P_1, are given in Table 2. The exponents, a and b, which modify P_1 and d_m are given in Table 3. Further information on the power loss in bearings can be found in bibliographical references [3] and [4]. The values presented in Tables 2, 3 and 4 are based on SKF[1)] bearings and may vary depending on the manufacturer.

$$M_1 = \frac{f_1 (P_1)^a (d_m)^b}{1\,000} \tag{9}$$

where

M_1 is the bearing load dependent torque, in Newton metres;

f_1 is the coefficient of friction (Table 2);

P_1 is the bearing dynamic load, in Newtons (Table 2);

d_m is the bearing mean diameter, in millimetres (Equation 10).

$$d_m = \frac{(d_i + d_o)}{2} \tag{10}$$

where

d_i is the bearing bore diameter, in millimetres;

d_o is the bearing outside diameter, in millimetres.

$$P_{Bi} = \frac{(M_1 + M_2)n}{9\,549} \tag{11}$$

where

P_{Bi} is the power loss for the individual bearing, in kilowatts;

n is the bearing rotational speed, in revolutions per minute;

M_2 is the cylindrical roller bearing axial load dependent moment, in Newton metres (Equation 12).

1) These are examples of products available commercially. This information is given for the convenience of users of this part of ISO/TR 14179 and does not constitute an endorsement by ISO of these products.

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Table 2 — Factors for calculating M_1

Bearing type	f_1	P_1 [a]
Deep groove ball bearings	$(0{,}0006 \ldots 0{,}0009)\ (P_0/C_0)^{0{,}55}$ [b]	$3\ F_a - 0{,}1\ F_r$
Self-aligning ball bearings	$0{,}0003\ (P_0/C_0)^{0{,}4}$	$1{,}4\ Y_2F_a - 0{,}1\ F_r$
Angular contact ball bearings: single-row double-row, paired single-row	 $0{,}001\ (P_0/C_0)^{0{,}33}$ $0{,}001\ (P_0/C_0)^{0{,}33}$	 $F_a - 0{,}1\ F_r$ $1{,}4\ F_a - 0{,}1\ F_r$
Four-point contact ball bearings	$0{,}001\ (P_0/C_0)^{0{,}33}$	$1{,}5\ F_a + 3{,}6\ F_r$
Cylindrical roller bearings, with cage: series 10 series 2 series 3 series 4, 22, 23	 0,000 2 0,000 3 0,000 35 0,000 4	 F_r [c] F_r [c] F_r [c] F_r [c]
Cylindrical roller bearings, full complement	0,000 55	F_r [c]
Needle roller bearings	0,002	F_r
Spherical roller bearings: series 213 series 222 series 223 series 230, 241 series 231 series 232 series 239 series 240	 0,000 22 0,000 15 0,000 65 0,001 0,000 35 0,000 45 0,000 25 0,000 8	$1{,}35\ Y_2F_a$ if $F_r/F_a < Y_2$ $F_r\ [1 + 0{,}35\ (Y_2F_a/F_r)^3]$ if $F_r/F_a \leqslant Y_2$ (valid for all series)
Taper roller bearings: single-row paired single-row	 0,000 4 0,000 4	 $2\ YF_a$ [d] $1{,}2\ Y_2F_a$ [d]
Thrust ball bearings	$0{,}000\ 8\ (F_a/C_0)^{0{,}33}$	F_a
Cylindrical roller thrust bearings, needle roller thrust bearings	0,001 5	F_a
Spherical roller thrust bearings: series 292E series 292 series 293E series 293 series 294E series 294	 0,000 23 0,000 3 0,000 3 0,000 4 0,000 33 0,000 5	F_a (F_r max $\leqslant 0{,}55\ F_a$) (valid for all series)

Symbols:

P_0 = equivalent static bearing load, in Newtons (see manufacturer's bearing tables);

C_0 = basic static load rating, in Newtons (see manufacturer's bearing tables);

F_a = axial component of dynamic bearing load, in Newtons;

F_r = radial component of dynamic bearing load, in Newtons;

Y, Y_2 = axial load factors (see manufacturer's bearing tables).

[a] If $P_1 < F_r$, then $P_1 = F_r$ should be used.

[b] The small values are for light series bearings, the large values for heavy series bearings.

[c] For bearings subjected to additional axial loads, refer to 7.3.2.

[d] Refer to 7.3.3.

Table 3 — Exponents for calculation of M_1

Bearing type	Exponent a	Exponent b
All (except spherical roller bearings)	1	1
Spherical roller bearings:		
series 213	1,35	0,2
series 222	1,35	0,3
series 223	1,35	0,1
series 230	1,5	− 0,3
series 231, 232, 239	1,5	− 0,1
series 240, 241	1,5	− 0,2

Table 4 — Factor f_2 for cylindrical roller bearings

Bearing	f_2 Lubrication grease	f_2 Lubrication oil
Bearings with cage:		
EC design	0,003	0,002
other bearings	0,009	0,006
Full complement bearings:		
single-row	0,006	0,003
double-row	0,015	0,009

7.3.2 Axially loaded cylindrical roller bearings

For cylindrical roller bearings that have to support an additional axial load, the equation given earlier for the total frictional moment must be expanded to include the frictional moment, M_2, which depends on the axial load:

$$M_2 = \frac{f_2 F_a d_m}{1\,000} \tag{12}$$

where

M_2 is the axial load dependent moment, in Newton metres;

f_2 is a factor (Table 4) depending on bearing design and lubrication;

F_a is the axial bearing load, in Newtons.

The values quoted for f_2 assume that the viscosity ratio $K \geqslant 1{,}5$. Additionally, the ratio of the axial load to the simultaneously acting radial bearing load (F_a/F_r) must not exceed 0,5 for EC design and single-row full complement bearings, 0,4 for the other bearings with cage, or 0,25 for double-row full complement bearings.

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7.3.3 Tapered roller bearings

Equivalent dynamic bearing load for single-row taper roller bearings:

$$P = F_r \text{ where } F_a / F_r \leqslant e \tag{13}$$

$$P = 0{,}4\, F_r + YF_a \text{ where } F_a / F_r \leqslant e \tag{14}$$

The values of factors e and Y will be found in the bearing tables for each individual bearing.

Because the raceways are at an angle to the bearing axis, when taper roller bearings are subjected to a radial load, an axial force is induced within the bearing. This must be considered when calculating the equivalent dynamic bearing load. All the requisite equations for the various bearing arrangements and load cases are given in Figure 1. These are only valid if the bearings are adjusted against each other to give zero clearance in operation, but are without preload. In the bearing arrangements shown, bearing A is always subjected to radial load F_{rA}, and bearing B to radial load F_{rB}. F_{rA} and F_{rB} are always considered as positive, even in cases when both act in the direction opposite to that shown in the figures. The radial loads act at the pressure centres of the bearings (see dimension a in the bearing manufacturer tables). Additionally, an external axial force, K_a, acts on the shaft (or the housing). Cases 1c and 2c in Figure 1 are also valid for $K_a = 0$. Values for the axial load factor, Y, for bearings A and B can be found in the bearing tables or can be approximated by using the bearing K factor.

7.4 Mesh power loss, P_M, spur and helical gears

Mesh 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 sliding ratios and the mesh coefficient of friction in [5] and [6].

The gear tooth mesh losses can be expressed by Equation 15. This equation contains the mesh coefficient of friction, f_m, which is a function of the applied load and the mesh mechanical advantage, M, which describes the mechanics of the tooth action. These functions must be solved before solving Equation 15.

$$P_{Mi} = \frac{f_m T_1 n_1 \cos^2 \beta_w}{9\,549 M} \tag{15}$$

where

P_{Mi} is the mesh power loss, in kilowatts;

f_m is the mesh coefficient of friction (Equation 20);

T_1 is the pinion torque, in Newton metres;

n_1 is the pinion rotational speed, in revolutions per minute;

β_w is the operating helix angle, in degrees;

M is the mesh mechanical advantage (Equation 16).

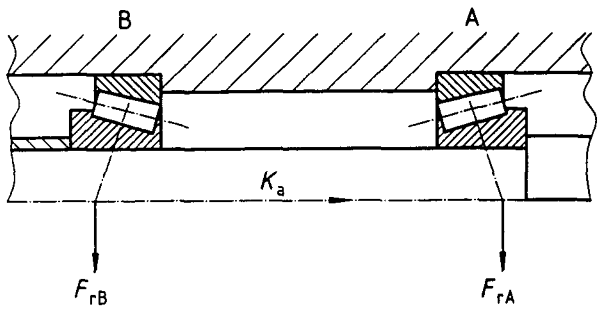
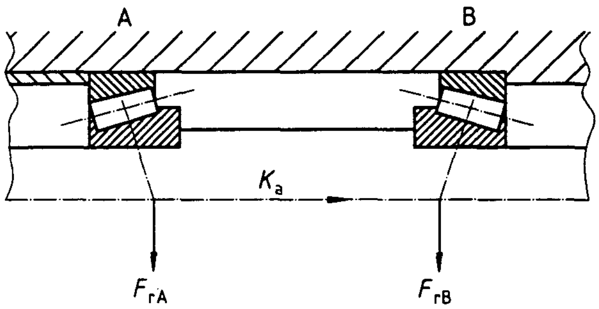
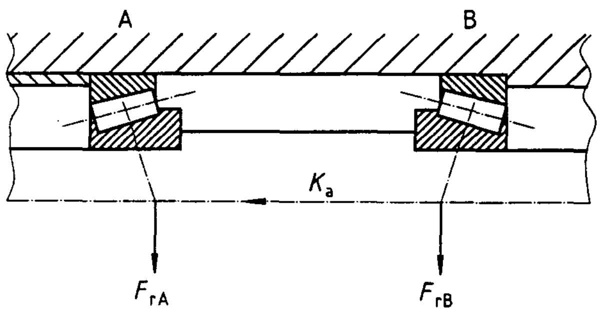
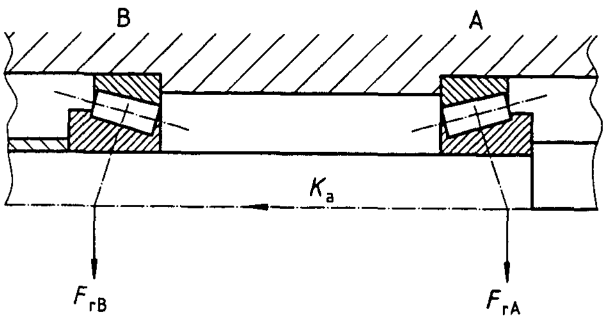
<table>
<tr><th>Arrangement</th><th colspan="2">Load case</th><th>Axial loads</th></tr>
<tr><td rowspan="3">Back-to-back

Face-to-face
</td>
<td>1a)</td><td>$\frac{F_{rA}}{Y_A} \geqslant \frac{F_{rB}}{Y_B}$
$K_a \geqslant 0$</td><td>$F_{aA} = \frac{0,5F_{rA}}{Y_A}$
$F_{aB} = F_{aA} + K_a$</td></tr>
<tr><td>1b)</td><td>$\frac{F_{rA}}{Y_A} < \frac{F_{rB}}{Y_B}$
$K_a \geqslant 0,5\left(\frac{F_{rB}}{Y_B} - \frac{F_{rA}}{Y_A}\right)$</td><td>$F_{aA} = \frac{0,5F_{rA}}{Y_A}$
$F_{aB} = F_{aA} + K_a$</td></tr>
<tr><td>1c)</td><td>$\frac{F_{rA}}{Y_A} < \frac{F_{rB}}{Y_B}$
$K_a < 0,5\left(\frac{F_{rB}}{Y_B} - \frac{F_{rA}}{Y_A}\right)$</td><td>$F_{aA} = F_{aB} - K_a$
$F_{aB} = \frac{0,5F_{rB}}{Y_B}$</td></tr>
<tr><td rowspan="3">Face-to-face

Back-to-back
</td>
<td>2a)</td><td>$\frac{F_{rA}}{Y_A} \leqslant \frac{F_{rB}}{Y_B}$
$K_a \geqslant 0$</td><td>$F_{aA} = F_{aB} + K_a$
$F_{aB} = \frac{0,5F_{rB}}{Y_B}$</td></tr>
<tr><td>2b)</td><td>$\frac{F_{rA}}{Y_A} > \frac{F_{rB}}{Y_B}$
$K_a \geqslant 0,5\left(\frac{F_{rA}}{Y_A} - \frac{F_{rB}}{Y_B}\right)$</td><td>$F_{aA} = F_{aB} + K_a$
$F_{aB} = \frac{0,5F_{rB}}{Y_B}$</td></tr>
<tr><td>2c)</td><td>$\frac{F_{rA}}{Y_A} > \frac{F_{rB}}{Y_B}$
$K_a < 0,5\left(\frac{F_{rA}}{Y_A} - \frac{F_{rB}}{Y_B}\right)$</td><td>$F_{aA} = \frac{0,5F_{rA}}{Y_A}$
$F_{aB} = F_{aA} - K_a$</td></tr>
</table>

Figure 1 — Tapered roller bearing load equations

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The mesh mechanical advantage can be calculated using Equation 16. This equation is a function of the sliding ratios. For external gears, the sliding ratio at the start of approach action, H_s, is calculated using Equation 17, and the sliding ratio at the end of recess action, H_t, is calculated with Equation 18. Equation 19 gives the gear ratio, u.

$$M = \frac{2\cos\alpha_w (H_s + H_t)}{H_s^2 + H_t^2} \tag{16}$$

where

α_w is the transverse operating pressure angle, in degrees;

H_s is the sliding ratio at start of approach (Equation 17);

H_t is the sliding ratio at the end of recess (Equation 18).

$$H_s = (u+1)\left[\left(\frac{r_{o2}^2}{r_{w2}^2} - \cos^2\alpha_w\right)^{0,5} - \sin\alpha_w\right] \tag{17}$$

$$H_t = \left(\frac{u+1}{u}\right)\left[\left(\frac{r_{o1}^2}{r_{w1}^2} - \cos^2\alpha_w\right)^{0,5} - \sin\alpha_w\right] \tag{18}$$

where

u is the gear ratio (Equation 19);

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

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

r_{o1} is the pinion outside radius, in millimetres;

r_{w1} is the pinion operating pitch radius, in millimetres.

$$u = \frac{z_2}{z_1} \tag{19}$$

where

z_2 is number of gear teeth;

z_1 is the number of pinion teeth.

If the pitch line velocity, V, is 2 m/s < $V \leqslant$ 25 m/s and the K-factor is 1,4 N/mm² < $K \leqslant$14 N/mm², then the mesh coefficient of friction, f_m, can be expressed by Equation 20. Outside these limits, the values for f_m must be determined by experience. Load intensity, K, can be calculated using Equation 21. The exponents, j, g and h modify the viscosity, ν, the load intensity, K, and the tangential pitch line velocity, V, respectively.

$$f_m = \frac{\nu^j K^g}{C_1 V^h} \tag{20}$$

where

ν is the kinematic oil viscosity at operating sump temperature, in centistokes (square millimetres per second);

K is the load intensity, in Newtons per square millimetre;

C_1 is a constant;

V is the tangential pitch line velocity, in metres per second.

$$K = \frac{1\,000 T_1 \left(z_1 + z_2\right)}{2 b_\text{w} \left(r_{\text{w}1}\right)^2 z_2} \tag{21}$$

where

b_w is the face width in contact, in millimetres.

Values to be used for the exponents j, g and h and the constant C_1 are as follows:

$j = -\ 0{,}223$

$g = -\ 0{,}40$

$h = 0{,}70$

$C_1 = 3{,}239$

7.5 Mesh power loss, P_M, bevel gears

The mesh power loss, P_M, for bevel gears is based on the tooth geometry, pitch line velocity and tooth contact pressure. Refer to annex A for a proposed calculation method.

7.6 Mesh power loss, P_M, cylindrical worm gears

The mesh power loss, P_M, for worm gearing is dependent on the worm gear tangential tooth load, W_t, sliding velocity, v_s, and the coefficient of friction, μ, where the tangential tooth load is also a function of transmitted power. Refer to annex B for calculation methods.

7.7 Mesh power loss, P_M, double enveloping worm gears

Refer to annex B for a suggested calculation method.

7.8 Oil seal power loss, P_S

Contact lip oil seal 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 22. Figure 2 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].

$$P_{\text{S}i} = \frac{T_\text{S} n}{9\,549} \tag{22}$$

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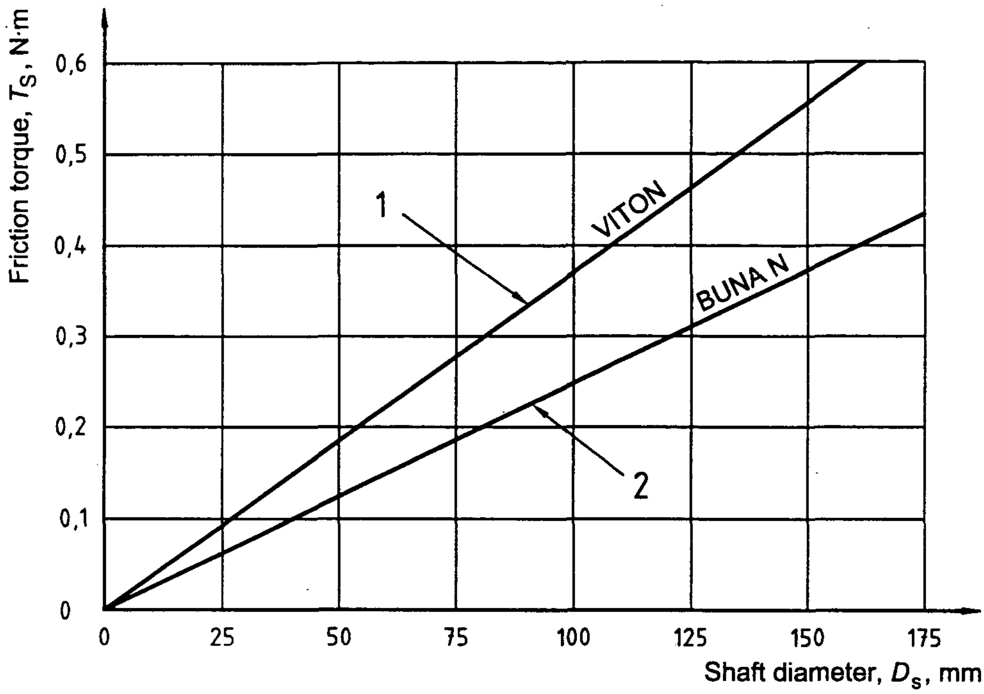
where

P_{Si} is the power loss for each individual oil seal, in kilowatts;

T_S is the oil seal torque, in Newton metres (Figure 2);

n is the shaft speed, in revolutions per minute.



Key

1 $T_S = 3{,}737 \times 10^{-3} D_s$

2 $T_S = 2{,}429 \times 10^{-3} D_s$

Figure 2 — Seal friction torque

7.9 Gear windage and churning power loss, P_W

The equations for gear windage and churning power loss that follow are derived from the equations that appear in section 12.5.2 of Dudley[6]. They have been modified to include the oil viscosity, v, the gear dip factor, f_g, and an arrangement constant, A_g. In addition, the exponent for the diameter, D, was adjusted.

Before calculating gear windage and friction losses, the gear dip factor, f_g, must be determined. This factor is based on the amount of dip that the element has in the oil. Since windage effects for typical industrial gear reducers are negligible with respect to the other losses, $f_g = 0$ when the element does not dip in the oil. When the element is fully submerged in the oil, $f_g = 1$. When the element is partly submerged in the oil, linearly interpolate between $f_g = 0$ and $f_g = 1$. For example, for a gear that has the oil level at the centre line of its shaft, $f_g = 0{,}50$.

Use a value of 0,200 for the arrangement constant, A_g, in Equations 24, 25 and 26.

The power loss equation for the tooth surface calls for a roughness factor, R_f. Table 12.5 of Dudley[6] provides some values based on tooth size. Equation 23 is a reasonable approximation of the values from Dudley.

$$R_f = 7{,}93 - \frac{4{,}648}{m_t} \quad (23)$$

where

R_f is the roughness factor;

m_t is the transverse tooth module.

Gear windage and churning losses encompass three types of loss. For those losses associated with a smooth outside diameter, such as the outside diameter of a shaft, use Equation 24. For those losses associated with the smooth sides of a disc, such as the faces of a gear, use Equation 25. It should be pointed out that Equation 25 includes both sides of the gear, so do not double the value. For those losses associated with the tooth surfaces, such as the outside diameter of a gear or pinion, use Equation 26.

For smooth outside diameters,

$$P_{\mathrm{GWi}} = \frac{7{,}37 f_g \nu n^3 D^{4{,}7} L}{A_g 10^{26}} \tag{24}$$

For smooth sides of discs,

$$P_{\mathrm{GWi}} = \frac{1{,}474 f_g \nu n^3 D^{5{,}7}}{A_g 10^{26}} \tag{25}$$

For tooth surfaces,

$$P_{\mathrm{GWi}} = \frac{7{,}37 f_g \nu n^3 D^{4{,}7} F \left(\frac{R_f}{\sqrt{\tan\beta}} \right)}{A_g 10^{26}} \tag{26}$$

where

P_{GWi} is the power loss for each individual element, in kilowatts;

f_g is the gear dip factor;

D is the outside diameter of the element, in millimetres;

A_g is the arrangement constant;

F is the total face width, in millimetres;

L is the length of the element, in millimetres;

β is the generated helix angle, in degrees. For helix angles less than 10°, use 10° in Equation 26.

After calculating the individual elements for each shaft assembly in a reducer, they must be added together for the total loss. For example, an output shaft assembly, Equation 24 would be used for the OD of the shaft outside of the gear between the bearings, Equation 25 for the smooth sides of the gear and Equation 26 for the tooth surfaces.

7.10 Bearing windage and churning power loss, P_{WB}

Bearing windage and churning losses are based on the methods described in [2]. The load-independent frictional moment, M_0, is given by Equations 27 and 28. The kinematic viscosity, ν, is a function of sump temperature.

If $vn < 2000$

$$M_0 = 1{,}6 \times 10^{-8} f_0 d_m^3 \quad (27)$$

If $vn \geqslant 2000$

$$M_0 = 10^{-10} f_0 (vn)^{2/3} d_m^3 \quad (28)$$

where

M_0 is the no-load torque moment on the bearing, in Newton metres;

f_0 is the bearing dip factor.

Factor f_0 adjusts the torque based on the amount that the bearing dips in the oil and varies from $f_{0(\min)}$ to $f_{0(\max)}$. Use $f_{0(\min)}$ if the rolling elements do not dip into the oil and $f_{0(\max)}$ if the rolling elements are completely submerged in the oil. When the rolling elements are partly submerged in the oil, linearly interpolate between $f_{0(\min)}$ and $f_{0(\max)}$. Values for $f_{0(\min)}$ and $f_{0(\max)}$ can be found in Table 5. See Equation 29 and Figure 3. For sealed bearings, use $f_{0(\min)}$ to calculate the moment M_0, however, the moment M_3 must also be calculated.

$$f_0 = f_{0(\min)} + \left(f_{0(\max)} - f_{0(\min)}\right)\frac{H}{D_{OR}} \quad (29)$$

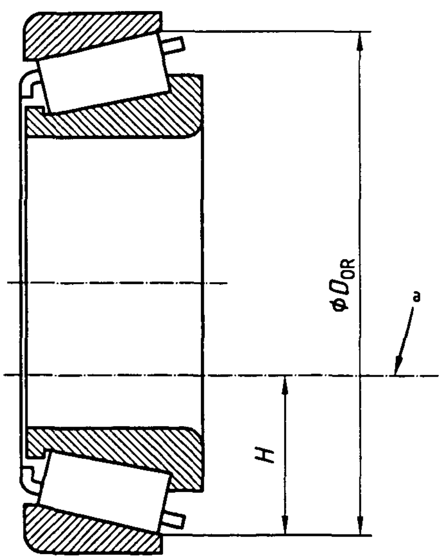
Where bearings are fitted with rubbing seals, the frictional losses arising from the seal may exceed those arising from the bearing itself.

Table 5 — Bearing dip factor, f_0

Bearing type	$f_{0(\min)}$	$f_{0(\max)}$
Deep groove ball bearings:		
single-row	2	4
double-row	4	8
Self-aligning ball bearings	2	4
Angular contact ball bearings:		
single-row	3,3	6,6
double-row, paired single-row	6,5	13
Four-point contact ball bearings	6	12
Cylindrical roller bearings, with cage:		
series 10, 2, 3, 4	2	4
series 22	3	6
series 23	4	8
Cylindrical roller bearings, full complement:		
single-row	5	10
double-row	10	20
Needle roller bearings	12	24

Table 5 (*continued*)

Bearing type	$f_{0(min)}$	$f_{0(max)}$
Spherical roller bearings:		
series 213	3,5	7
series 222	4	8
series 223, 230, 239	4,5	9
series 231	5,5	11
series 232	6	12
series 240	6,5	13
series 241	7	14
Taper roller bearings:		
single-row	4	8
paired single-row	8	16
Thrust ball bearings	1,5	3
Cylindrical roller thrust bearings	3,5	7
Needle roller thrust bearings	5	11
Spherical roller thrust bearings:		
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] Oil level.

Figure 3 — Bearing dip

The frictional moment, M_3, of the seals for a bearing which is sealed at both sides can be estimated using the following empirical equation:

$$M_3 = \frac{\left(\frac{d_m}{f_3}\right)^2 + f_4}{1\,000} \tag{30}$$

where

M_3 is the frictional moment of seals, in Newton metres;

f_3 is a bearing seal factor (Table 6);

f_4 is a bearing seal factor (Table 6).

Table 6 — Factors f_3 and f_4

Bearing (design)	Factors	
	f_3	f_4
Deep groove ball bearings (2RS1), self-aligning ball bearings (2RS1), angular contact ball bearings (2RS), Y-bearings (series 17262(00)-2RS1 and 17263(00)-2RS1)	20	10
Y-bearings (all other series), needle roller bearings (2RS)	20 [a]	25 [a]
Cylindrical roller bearings, full complement (2LS)	10	50

[a] Does not apply to Y-bearings with special flingers.

The power loss for each bearing can then be calculated using Equation 31.

$$P_{WBi} = \frac{(M_0 + M_3)n}{9\,549} \tag{31}$$

7.11 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.

$$P_P = P_{Ps} + P_{Pm} \tag{32}$$

For an oil pump driven by one of the reducer shafts, the oil pump loss, P_{Ps}, can be estimated using Equation 33.

$$P_{Ps} = \frac{Qp}{60 e_p} \tag{33}$$

where

Q is the oil flow, in litres per minute;

p is the operating oil pressure, in Newtons per square millimetre;

e_p is the oil pump efficiency.

For an oil pump driven by an electric motor, the oil pump losses, P_{Pm}, can be estimated using Equation 34, which considers the electric power consumed and the efficiency of both the electric motor and the oil pump.

$$P_{Pm} = E_P \left(\frac{e_m}{e_p} \right) \tag{34}$$

where

E_P is electric power consumed, in kilowatts;

e_m is electric motor efficiency.

7.12 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 35.

$$P_Q = A_c k \Delta T \tag{35}$$

where

A_c is the gear case surface area, in square metres;

k is the heat transfer coefficient, in kilowatts per square metre degrees Celsius;

ΔT is the temperature differential, in degrees Celsius.

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

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 part of ISO/TR 14179, typical values for k range from 0,017 kW/(m^2 °C) to 0,020 kW/(m^2 °C). This range is typical for gear drives applied in a large, indoor space. See [8].

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 70 % 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 7 provides values for k for fan-cooled gear drives.

Table 7 — Heat transfer coefficient, k, for gear drives with fan cooling

Air velocity m/s	Heat transfer coefficient kW/(m^2 °C)
2,5	0,015
5,0	0,024
10,0	0,042
15,0	0,058

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8 Modifications for non-standard operating conditions

When the actual operating conditions for a specific application are different from the standard conditions defined in 3.2, and the thermal rating is calculated for the conditions 3.2 defines, the thermal rating may be modified for the application as follows:

$$P_{THm} = P_T B_{ref} B_V B_A B_T B_D \tag{36}$$

where

P_{THm} is the modified thermal power rating, in kilowatts;

P_T is the basic thermal power rating, in kilowatts;

B_{ref} is the ambient temperature modifier;

B_V is the ambient air velocity modifier;

B_A is the altitude modifier;

B_T is the allowable oil sump temperature modifier;

B_D is the operation time modifier.

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

The gear drive manufacturer should be consulted when the conditions exceed the limits given in Tables 8 to 12 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 8.

Table 8 — Ambient temperature modifier, B_{ref}

Ambient temperature °C	B_{ref}
10	1,15
18	1,07
25	1,00
30	0,93
40	0,83
43	0,75
50	0,67

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 9.

Table 9 — Ambient air velocity modifier, B_V

Ambient air velocity m/s	B_V
$\leqslant$ 0,5	0,75
> 0,5 $\leqslant$ 1,4	1,00
> 1,4 < 3,7	1,40
$\geqslant$ 3,7	1,90

At high altitudes the decrease in air density results in the derating factor B_A. See Table 10.

Table 10 — Altitude modifier, 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	0,77
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 11). 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 may 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.

Table 11 — Maximum allowable oil sump temperature modifier, B_T

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

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 12.

Table 12 — Operation time modifier, B_D

Operation time per each hour %	B_D
100 (continuous)	1,00
80	1,05
70	1,15
40	1,35
20	1,80

Annex A
(informative)

Bevel gear mesh and gear windage power losses

A.1 Purpose

The purpose of this annex is to provide a method of calculating the power loss for bevel gear meshes, and the windage and churning losses for bevel gears and pinions, in order to determine thermal rating.

These procedures are offered in an annex because at the time of publication there was insufficient data available for confirmation. Actual thermal ratings may differ substantially from thermal ratings calculated using the following procedures.

A.2 Mesh power loss, P_M

The following equations are proposed for estimating the gear tooth mesh losses for bevel gears (see Figures A.1 and A.2).

$$P_M = \frac{f_m \ T_1 \, n_1 \ \cos^2 \beta_m}{9\,549\,M} \tag{A.1}$$

where

f_m is the coefficient of friction (see Equation 20);

T_1 is the pinion torque, in Newton metres;

n_1 is the pinion speed, in revolutions per minute;

β_m is the mean spiral angle, in degrees;

M is the mesh mechanical advantage, (see Equation 16).

If the pitch line velocity, V, is 2 m/s < $V \leqslant$ 25 m/s and the K-factor is 1,4 < K < 14 N/mm^2, then f_m can be estimated using Equation 20. Outside these limits, the values for f_m must be determined by experience.

For bevel gearing, the pitch line velocity is calculated at the larger end of the tooth.

The K-factor is given by the equation:

$$K = \frac{1\,000 T_1 \left(z_1 + z_2\right)}{2 b_w \, r_{m1}^2 z_2} \tag{A.2}$$

where

z_1 is the number of pinion teeth;

z_2 is the number of gear teeth;

b_w is the face width in contact with mating element, in millimetres;

r_{m1} is the mean reference radius, in millimetres.

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The equation for mesh mechanical advantage is:

$$M = \frac{2\cos\alpha_{tm}(H_s + H_t)}{H_s^2 + H_t^2} \quad \text{(A.3)}$$

where

α_{tm} is the 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.

$$\alpha_{tm} = \arctan\left[\frac{\tan\alpha_n}{\cos\beta_m}\right] \quad \text{(A.4)}$$

where

α_n is the normal pressure angle at pitch surface, in degrees.

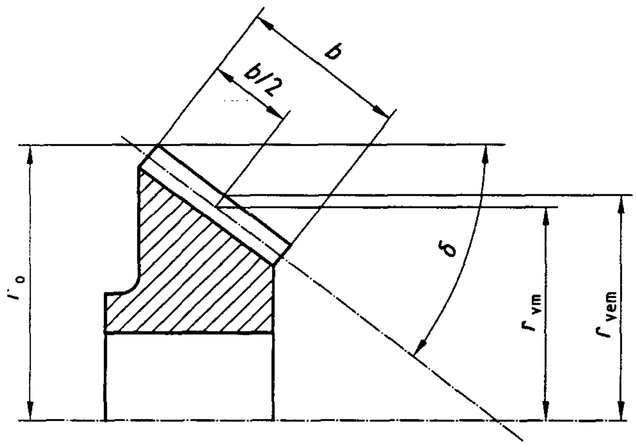

Figure A.1 — Uniform depth tooth

The values for H_s and H_t are:

$$H_s = (u_v + 1)\left[\left(\frac{r_{vem2}^2}{r_{vm2}^2} - \cos^2\alpha_{tm}\right)^{0,5} - \sin\alpha_{tm}\right] \quad \text{(A.5)}$$

$$H_t = \left(\frac{u_v + 1}{u_v}\right)\left[\left(\frac{r_{vem1}^2}{r_{vm1}^2} - \cos^2\alpha_{tm}\right)^{0,5} - \sin\alpha_{tm}\right] \quad \text{(A.6)}$$

where

u_v is the equivalent gear ratio;

r_{vm1} is the equivalent mean reference radius, pinion, in millimetres;

r_{vm2} is the equivalent mean reference radius, gear, in millimetres;

r_{vem1} is the equivalent tip radius at mid-face width, pinion, in millimetres;

r_{vem2} is the equivalent tip radius at mid-face width, gear, in millimetres;

α_{tm} is the transverse pressure angle, in degrees.

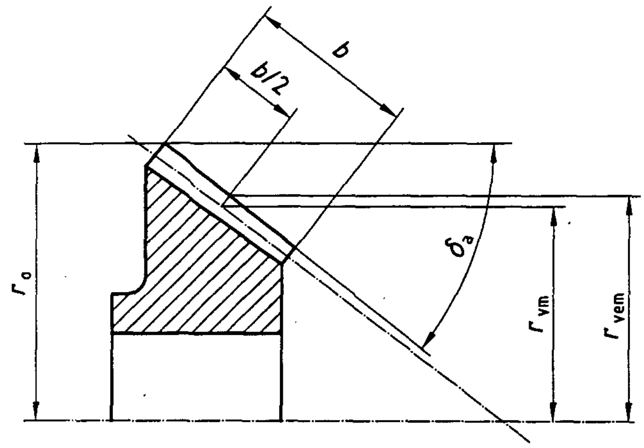

Figure A.2 — Taper depth tooth

The equation for equivalent mean reference radius is given by:

$$r_{vm} = \frac{R_m r}{R_e \cos\delta} \tag{A.7}$$

where

r is the pitch radius, in millimetres;

R_m is the mean cone distance, in millimetres;

R_e is the outer cone distance, in millimetres;

δ is the reference cone angle, in degrees.

The equivalent gear ratio can be calculated as:

$$u_v = \frac{r_{vm2}}{r_{vm1}} \tag{A.8}$$

The equivalent tip radius at mid-face can be calculated as:

$$r_{vem} = r_{vm} + h_{am} \tag{A.9}$$

where

h_{am} is the addendum at mid-face, in millimetres.

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If the addendum at the outer end and the face angle are known, the addendum at mid-face can be calculated as:

$$h_{am} = h_{ae} - \frac{b \tan(\delta_a - \delta)}{2} \tag{A.10}$$

where

b is the face width, in millimetres;

h_{ae} is the addendum at outer end, in millimetres;

δ_a is the face angle, in degrees;

δ is the reference cone angle, in degrees.

A.3 Gear windage and churning power loss, P_{WG}

Subclause 7.9 and Equations 23 to 26 can be used for the windage and churning power losses of bevel gears. The use of the dimensions and tooth geometry based on the large end of the teeth will result in conservative values.

Annex B
(informative)

Worm gear mesh power losses

B.1 Purpose

The purpose of this annex is to provide a method that can be used to calculate the power loss for worm gear meshes to determine thermal rating.

These procedures are offered in an annex because at the time of publication there was insufficient data available for confirmation. Actual thermal ratings may differ substantially from thermal ratings calculated using the following procedures.

The worm gear mesh power loss is usually the largest single component of the total loss for a worm gear drive and is directly related to the operating coefficient of friction. AGMA provides one general value for this factor based only on pitch line velocity, but the operating coefficient of friction will also be influenced by factors such as load, materials, contact (assembly), surface finish, surface accuracy, hardness, geometry, lubrication and temperature. Tests at various speeds on a series of small centre distance units, 35 mm to 75 mm, with various ratios, did show a significant scatter in the operating coefficient of friction. When using the coefficient of friction from Tables 5 and 7 of [9], in the following procedure, the actual mesh power loss and resulting thermal power capacity may vary significantly from the calculated value. However, when using a coefficient of friction more representative of the operating conditions, the calculated mesh power loss should closely match the actual mesh loss.

B.2 Mesh power loss, P_M, cylindrical worm gear

The mesh power loss, P_M, for cylindrical worm gearing is dependent on the worm gear tangential tooth load, W_t, the sliding velocity, v_s, and the coefficient of friction, μ, where the tangential tooth load is also a function of the transmitted power, P_A.

The input power can be used to calculate the tangential tooth load per Equation B.1.

$$W_t = \frac{P_A - P_N}{\left(\frac{n_w D_m}{1{,}91 \times 10^7 u} + \frac{v_s \mu}{1\,000 \cos\lambda \cos\alpha_n} \right)} \tag{B.1}$$

$$v_s = \frac{n_w d_m}{19\,098 \cos\lambda} \tag{B.2}$$

where

W_t is the tangential tooth load on worm gear, in Newtons;

P_A is the transmitted power, in kilowatts;

P_N is the no-load power loss, in kilowatts;

n_w is the revolutions per minute of worm shaft;

D_m is the mean wormgear diameter, in millimetres;

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d_m is the mean worm diameter, in millimetres;

v_s is the sliding velocity at mean worm diameter, in metres per second;

λ is the lead angle at mean worm diameter, in degrees;

α_n is the normal pressure angle of worm thread at mean diameter, in degrees;

u is the gear ratio;

μ is the coefficient of friction.

NOTE When using a synthetic lubricant, values of the coefficient of friction, μ, from ANSI/AGMA 6034-B92, can be reduced by 30 % for establishing mesh power loss.

The mesh power loss, P_M, for this operating condition can then be calculated using Equation B.3.

$$P_M = \frac{v_s \mu W_t}{1000 \cos\lambda \cos\alpha_n} \tag{B.3}$$

B.3 Mesh power loss, P_M, double enveloping worm gear

The AGMA standard for double enveloping wormgearing, ANSI/AGMA 6017-E86 [10], does not provide a direct method for establishing a mesh power loss, but does provide typical efficiency curves established by testing. Based on this data, the cylindrical wormgear method can be used for calculating double enveloping mesh losses by reducing the value for μ by 30 %. This will provide a reasonable approximation for double enveloping mesh power loss.

The mean worm diameter, d_m, is calculated as follows:

$$d_m = 2C - D_G \tag{B.4}$$

where

C is the centre distance, in millimetres.

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