TECHNICAL REPORT

ISO/TR 14179-1

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Gears — Thermal capacity —

Part 1: Rating gear drives with thermal equilibrium at 95 °C sump temperature

Engrenages — Capacité thermique —

Partie 1: Capacité des transmissions par engrenages pour une température de bain d'huile de 95 °C

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Contents

Page

Foreword

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

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is entirely informative in nature and does not have to be reviewed until the data it provides are considered to be no longer valid or useful.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 — Thermal capacity*:

— *Part 1: Rating gear drives with thermal equilibrium at 95 °C sump temperature*

— *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, hypoid and worm gears can be calculated according to theoretical and experimental investigations of these 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 — Thermal capacity

Part 1:
Rating gear drives with thermal equilibrium at 95 °C sump temperature

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, term and definition

For the purposes of this part of ISO TR 14179, the symbols and units given in Table 1, and the following term and definition, 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)	7.12
A_g	Arrangement constant for gearing	—	Eq. (24)	7.9
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)	7.4
C_0	Basic static load rating	N	—	Table 2
C_1	Mesh coefficient of friction constant	—	Eq. (20)	7.4
D	OD of element for gearing windage and churning	mm	Eq. (24)	7.9
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)	7.3.1

Table 1 (*continued*)

Symbol	Meaning	Units	Where first used	Reference
d_m	Bearing mean diameter	mm	Eq. (9)	7.3.1
d_o	Bearing outside diameter	mm	Eq. (10)	7.3.1
E_P	Electric power consumed	kW	Eq. (34)	7.11
e	Bearing factor	—	Eq. (13)	7.3.3
e_m	Electric motor efficiency	—	Eq. (34)	7.11
e_p	Oil pump efficiency	—	Eq. (33)	7.11
F	Total face width of gear or pinion	mm	Eq. (26)	7.9
F_a	Bearing axial load component	N	Eq. (12)	7.3.2
F_r	Bearing radial load component	N	Eq. (13)	7.3.3
f_g	Gear dip factor	—	Eq. (24)	7.9
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 of bearings	N · m	Eq. (27)	7.10
M_1	Bearing load dependent torque	N · m	Eq. (9)	7.3.1
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)	7.3.1

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)	7.1
P_B	Total bearing losses (all bearings)	kW	Eq. (7)	7.2
P_{Bi}	Individual bearing load power loss	kW	Eq. (11)	7.3.1
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)	7.2
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)	7.1
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)	7.1
μ	Coefficient of friction for worm gears	—	—	7.6
ν	Kinematic viscosity of the oil at operating temperature	cSt	Eq. (20)	7.4

2.1
Thermal rating
maximum power that can be continuously transmitted through a gear drive without exceeding a specified oil sump temperature

NOTE 1 The thermal rating must equal or exceed the actual service transmitted power.

NOTE 2 Service factors are not used when determining thermal requirements.

NOTE 3 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 Rating criteria

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.

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).

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 Basis

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 clause 3, P_A is defined as P_T.

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

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