TECHNICAL REPORT

ISO/TR 14179-2

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Gears — Thermal capacity

Part 2: Thermal load-carrying capacity

Engrenages — Capacité thermique

Partie 2: Capacité de charge thermique

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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is entirely informative in nature and does not have to be reviewed until the data it provides are considered to be no longer valid or useful.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 14179-2 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 14179-1 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. 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.

This part of ISO/TR 14179 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 program, “WTplus”, 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 2:
Thermal load-carrying capacity

1 Scope

This part of ISO/TR 14179 presents a means for determining the thermal load carrying capacity of gears that includes measurement on original gear units under practical conditions. This takes the form of either measurement of the power loss, heat dissipation or both, or, in the case of splash-lubricated gear units, the 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 part of ISO/TR 14179 are to be regarded as an alternative method.

2 Symbols, units and indices

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

Table 1 — Symbols, units and indices

Symbol	Meaning	Units
a	Centre distance	mm
A_{bot}	Gear unit bottom area	m^2
A_{ca}	Overall housing area (external)	m^2
A_{foot}	Footprint of gear unit	m^2
A_{oil}	Overall housing area (internal)	m^2
A_{pro}	Projected fin area (housing external)	m^2
A_q	Cross-sectional area	m^2
A_{fin}	Total fin area (housing external)	m^2
A_{air}	Ventilated housing area	m^2
b	Tooth width, bearing width	mm
b_{eH}	Tooth contact width	mm
b_0	Reference value of tooth width, $b_0 = 10$ mm	mm
C_{lub}	Lubrication factor	—
C_{Sp}	Splash oil factor	—

Table 1 (*continued*)

Symbol	Meaning	Units
C_0	Static load rating of anti-friction bearing	N
$C_{1,2}$	Factors	—
d_a	Tip circle diameter	mm
d_{fl}	Equivalent flange diameter	m
d_w	Pitch circle diameter	mm
d_m	Mean bearing diameter	mm
d_s	Pitch circle diameter of equivalent crossed helical gears	mm
d_{sh}	Shaft diameter	m
e	Base of natural logarithm, $e = 2{,}718$	—
$f_{0,\,1,\,2}$	Coefficients for bearing losses	—
ED	Duty factor	—
F_a	Bearing thrust load	N
F_t	Force at pitch circle	N
F_{bt}	Tooth normal force, transverse section	N
F_n	Tooth normal force, normal section	N
F_r	Radial bearing load	N
g	$g = 9{,}81$ m/s^2	m/s^2
Gr	Grashoff number	—
h_c	Height of point of contact above the lowest point of the immersing gear	mm
h_{ca}	Overall height of gear unit housing	m
H_v	Tooth loss factor	—
$h_{e1,e2}$	Tip circle immersion depth with oil level stationary	mm
h_{e0}	Reference value of immersion depth, $h_{e0} = 10$ mm	mm
$h_{e,\,max}$	Max. Tip circle immersion depth with oil level stationary	mm
ΔH_{oil}	Enthalpic flow with oil	W
$h_{0,\,1}$	Lubrication gap heights	mm
k	Heat transmission coefficient	W/(m^{2}K)
l_{fl}	Equivalent length of coupling flange	m
l_h	Hydraulic length $= 4\,A_G/U_M$	mm
l_{fin}	Depth of one fin	m
l_x	Flow length (path of flow filament along housing wall)	m
l_{sh}	Length of free shaft end	m

Table 1 (*continued*)

Symbol	Meaning	Units
m, m^*	Fin factors	—
m	Module	mm
n	Rotational speed	1/min
Nu	Nusselt number	—
P_A	Input power	W
P_{Aeq}	Equivalent input power	W
Pr	Prandtl number	—
P_V	Power loss	W
P_{VD}	Seal power loss	W
P_{VL}	Bearing power loss	W
P_{Vx}	Auxiliary power losses	W
P_{VZ}	Gear power loss	W
P_0	Equivalent static bearing load	N
P_1	Equivalent bearing load	N
Q	Total heat flow	W
Q_{ca}	Heat flow across housing surface	W
Q_{fun}	Heat flow across foundation	W
Q_{rot}	Heat flow across shafts and couplings	W
Re	Reynold's number	—
$Ra_{1,2}$	Arithmetic average roughness of pinion and gear wheel	µm
Rz	Average roughness depth	µm
Rz_0	Reference roughness depth for worm gear units ($Rz_0 = 3$ µm)	µm
s	Size factor of bearing	—
t	Duration	min
T_H	Hydraulic loss torque	N · m
T_{VL}	Total bearing loss torque	N · m
T_{VL0}	No-load bearing loss torque	N · m
$T_{VLP1,2}$	Load dependent bearing loss torque	N · m
T_{wall}	Temperature of housing wall	K
T_{air}	Cooling air temperature	K
T_{perm}	Maximum permissible gear unit temperature	K
T_{∞}	Ambient temperature	K
u	Gear ratio	—

Table 1 (*continued*)

Symbol	Meaning	Units
U	Circumference of the foundation	m
v	Mean peripheral speed	m/s
v_t	Tangential speed	m/s
v_{t0}	Reference tangential speed	m/s
$\dot{V}_{oil}$	Oil injection rate	l/min
$\dot{V}_0$	Reference oil injection rate, $\dot{V}_0 = 2$ l/min	l/min
v_{gm}	Mean sliding speed	m/s
v_{gs}	Helical speed	m/s
$v_{gy1,2}$	Total surface speed at tooth tip	m/s
v_S	Oil jet velocity	m/s
v_t	Peripheral speed at pitch circle	m/s
v_{t0}	Reference speed, $v_{t0} = 10$ m/s	m/s
v_{air}	Impingement velocity	m/s
$v_{\Sigma C}$	Sum velocity at pitch point	m/s
$v_{\Sigma h}$	Sum velocity in direction of tooth depth	m/s
$v_{\Sigma m}$	Mean resultant sum velocity	m/s
$v_{\Sigma s}$	Sum velocity in direction of tooth length	m/s
x	Addendum modification factor	—
X_L	Oil Lubricant factor	—
X_R	Roughness factor	—
Y	Axial factor from bearing tables, Y for $F_a/F_r > e$	—
Y_W	Material factor	—
z	Number of teeth	—
α_{fun}	Heat transfer coefficient at gear unit foundation	W/(m^{2}K)
α_{ca}	Air-side heat transfer coefficient at housing	W/(m^{2}K)
α_{con}	Heat transfer coefficient due to convection	W/(m^{2}K)
$\alpha_{K,free}$	Heat transfer coefficient due to free convection	W/(m^{2}K)
$\alpha_{K,forced}$	Heat transfer coefficient due to forced convection	W/(m^{2}K)
α_{oil}	Oil-side heat transfer coefficient	W/(m^{2}K)
α_{rad}	Heat transfer coefficient due to radiation	W/(m^{2}K)
α_{rot}	Heat transfer coefficient at rotating shafts	W/(m^{2}K)

Table 1 (*continued*)

Symbol	Meaning	Units
$\alpha^*_{\mathrm{sh,face}}$	Heat transfer coefficient at the face of a shaft	W/(m^{2}K)
α_{t}	Transverse pressure angle	°
α_{wt}	Working pressure angle	°
β	Helix angle	°
β_{b}	Helix angle at base circle	°
δ_{fin}	Thickness of one fin	m
δ_{wall}	Mean housing wall thickness	m
ε	Emission ratio of gear unit housing	—
ε_{α}	Profile contact ratio	—
$\varepsilon_{1,2}$	Addendum contact ratio, pinion/gear wheel	—
λ_{fun}	Thermal conductivity of foundation	W/(mK)
λ_{wall}	Thermal conductivity of housing	W/(mK)
λ_{sh}	Thermal conductivity of shaft	W/(mK)
μ	Coefficient of friction	—
μ_{mz}	Mean coefficient of friction of the gear mesh	—
$\nu_{40,100}$	Kinematic viscosity of oil at 40 °C, 100 °C	mm^2/s
ν_{oil}	Kinematic viscosity of oil at operating temperature	mm^2/s
ν_{air}	Kinematic viscosity of air	m^2/s
ρ_{c}	Equivalent radius of curvature at pitch point of contact	mm
ρ_{n}	Equivalent radius of curvature, normal section	mm
ρ_{15}	Density of oil at 15 °C	kg/m^3
ρ_{oil}	Density of oil at operating temperature	kg/m^3
ω	Angular velocity	rad/s
η	Efficiency	—
η_{f}	Fin efficiency	—
η_{oil}	Dynamic viscosity of oil at operating temperature	mPa · s
ϑ_{oil}	Oil temperature	°C
ϑ_{∞}	Ambient temperature	°C
η^*	Temperature ratio	—
μ_{z}	Coefficient of friction of a warm gear unit	—
μ_{z0}	Basic value of the coefficient of friction of a warm gear unit	—

Table 1 (*continued*)

Indices	Meaning
0	Load independent
1	Pinion
2	Gear wheel
C	Referred to the pitch point
m	Medium circle for bevel and hypoid gears
n	Normal
v	Equivalent spur gear for bevel and hypoid gears
P	Load-dependent

3 Principle

3.1 General

When power is transmitted by a gear unit, losses occur at the various components which are converted into heat. These losses, together with the drive power, determine the efficiency of the gear unit. Depending 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 can be reached which, in the case of high values, results in rapid oil ageing, low oil film thicknesses in contact surfaces and reduced load carrying capacity with pitting, wear and scuffing of tooth systems and bearings, as well as a reduction in the service life of the seals.

From calculation of the thermal balance, it is possible to determine the anticipated steady-state temperature for splash-lubricated gear units, and the quantity of heat to be dissipated via the oil flow and the oil cooler in the case of injection-lubricated gear units.

3.2 Purpose and applicability

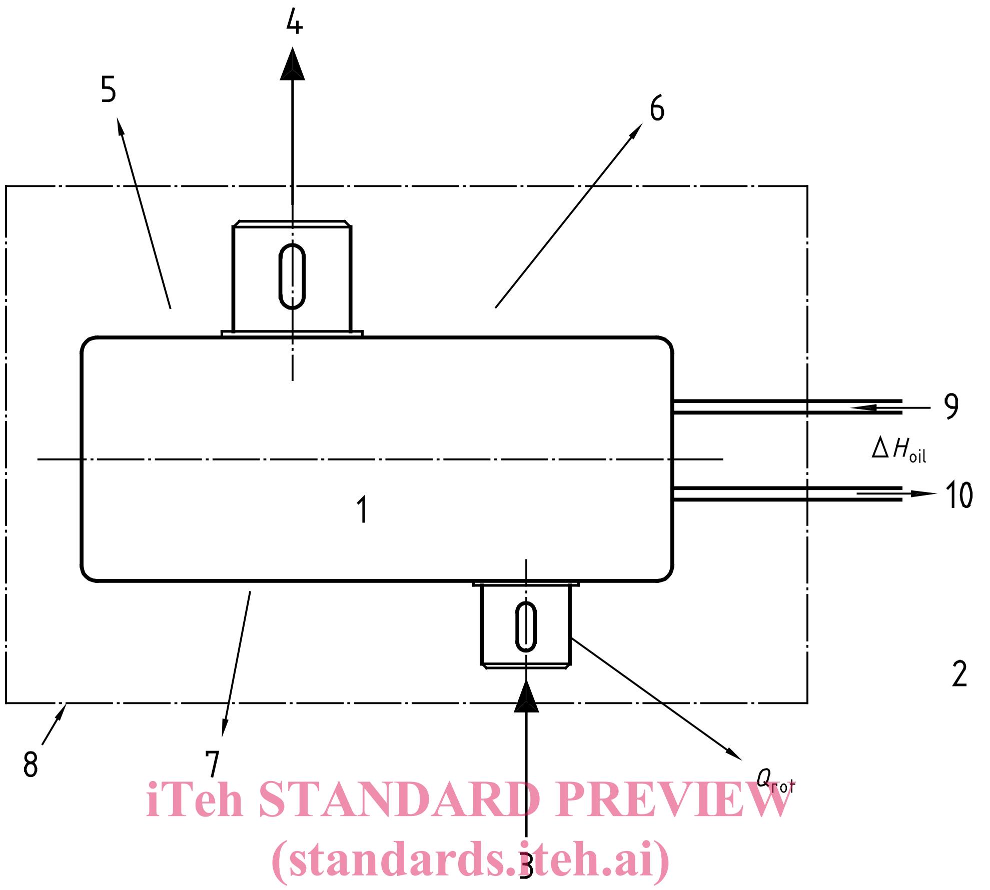
With the calculations given in this part of ISO/TR 14179, it is possible to determine the power loss of gear systems, no-load and load-dependent losses of external and internal cylindrical gears, bevel, hypoid and worm gear systems, the bearing, no-load and load losses of anti-friction and journal bearings, and of radial shaft seals. The calculations can be applied to single and multispeed gear units, torque-dividing gear units and planetary gear units. The heat dissipation is calculated as free or forced convection, or both, 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. Calculation can be carried out in the case of gear units with intermittent duty (duty factor of less than 100 %) and in the case of variable loads and speeds, 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 1). In particular, the fact of whether heat flows can be dissipated from the gear unit at the coupling points or passing from the machines connected into the gear unit should be taken into account at the connection points of driving and driven machines.

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

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.



Key

1 Gear unit
2 Environment
3 Input power, P_A
4 Output power
5 Convection, Q_{ca}
6 Radiation
7 Conduction
8 System boundary
9 Oil inlet
10 Oil outlet

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

4 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 in the thermal balance, and neither is the internal heat distribution taken into account, in every case, all derating factors (e.g. in the case of gear calculation K_A, K_V, $K_{H\beta}$ and $K_{H\alpha}$) should be assumed as being 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 at slow speeds.

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 that 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.