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

Reference number
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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is entirely informative in nature and does not have to be reviewed until the data it provides are considered to be no longer valid or useful.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
$\alpha^*_{sh,face}$	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

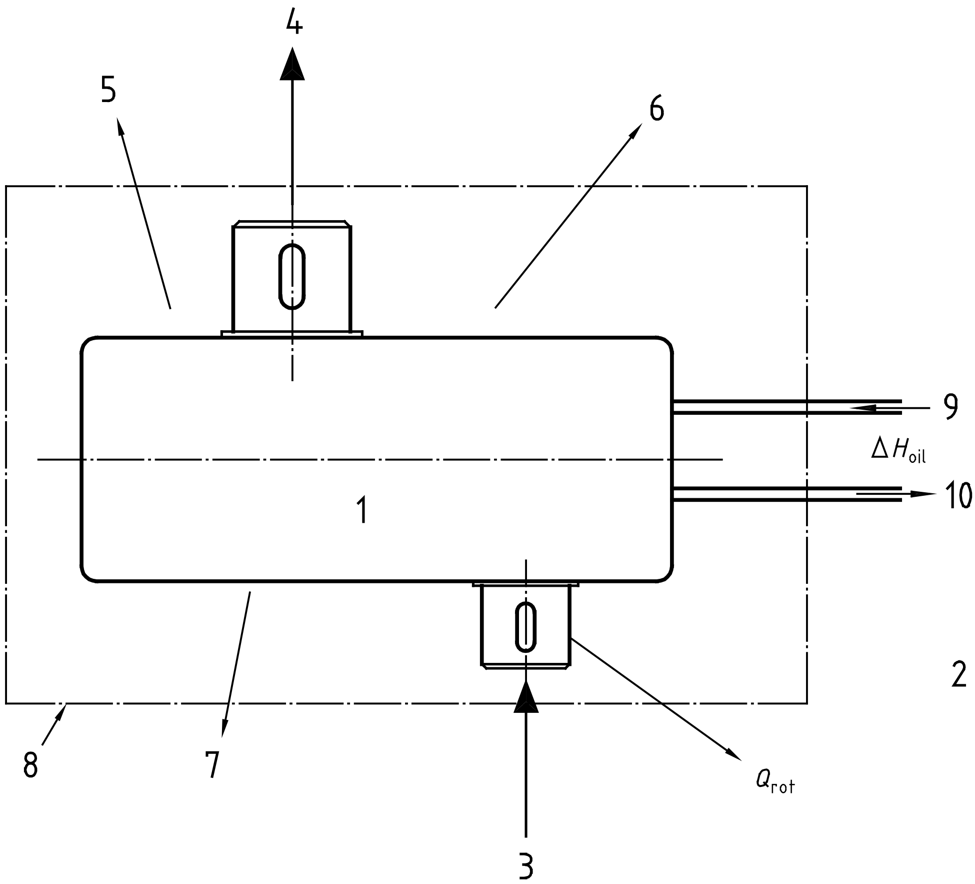
With the calculations given in this part of ISO/TR 14179, it is possible to determine the power loss of gear systems, no-load and load-dependent losses of external and internal cylindrical gears, bevel, hypoid and worm gear systems, the bearing, no-load and load losses of anti-friction and journal bearings, 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.

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

$$P_{\mathrm{Aeq}} = \frac{P_1 t_1 + P_2 t_2 \ \dots \ P_n t_n}{t_1 + t_2 + \dots \ t_n} \tag{1}$$

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

$$P_{\mathrm{Aeq}} = ED \cdot P_{\mathrm{A}} \tag{2}$$

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

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

5 Power loss

5.1 General

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

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

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

$$\eta = \frac{P_A - P_V}{P_A} \tag{4}$$

5.2 Gear losses

5.2.1 General

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

5.2.2 No-load gear losses for cylindrical, bevel and hypoid gears

5.2.2.1 General

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

5.2.2.2 Splash lubrication

The total hydraulic loss torque, T_H, of a gear stage is determined using the following:

$$T_H = C_{Sp} C_1 e^{C_2\left(\frac{v_t}{v_{t0}}\right)} \tag{5}$$

The splash oil factor, C_{sp}, takes into account the effect of the splash oil supply, dependent on the immersion depth (Figures 2 and 3). The factors C_1 and C_2 state the effect of the tooth width and the immersion depth. In the case of low immersion depths, no effect of viscosity was measurable. For high immersion depth, contradictory results for the influence of viscosity were found: In some cases, power loss increased with increasing viscosity, while in others it decreased. Therefore no account was taken of viscosity in the calculation equation.

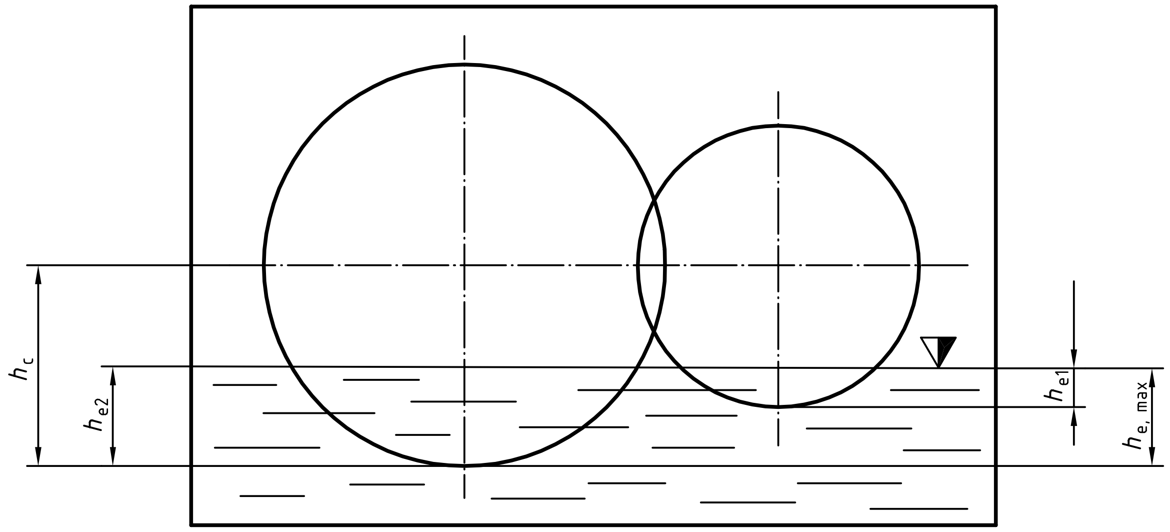


Figure 2 — Splash oil factor according to Mauz

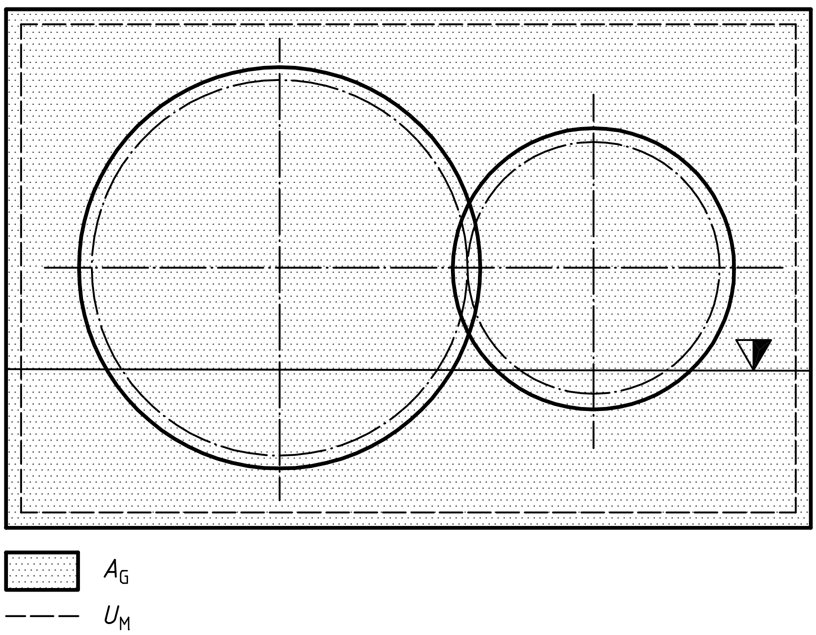


Figure 3 — $l_h = 4A_G/U_M$

$$C_{Sp} = \left(\frac{4h_{e,max}}{3h_c}\right)^{1,5} \frac{2h_c}{l_h} \qquad (6)$$

$$C_1 = 0,063\left(\frac{h_{e1} + h_{e2}}{h_{e0}}\right) + 0,0128\left(\frac{b}{b_0}\right)^3 \qquad (7)$$

$$C_2 = \frac{h_{e1} + h_{e2}}{80h_{e0}} + 0,2$$

with $h_{e0} = 10$ mm, $b_0 = 10$ mm, $v_{t0} = 10$ m/s

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

$$P_{VZ0} = \sum_{i=1}^{\text{stage}} T_{H,i} \frac{\pi n_i}{30} \qquad (8)$$

where $T_{H,i}$ is the loss torque and n_i is the rotational speed of the gear wheel of the gear stage i.

5.2.2.3 Injection lubrication

Evaluation of the experimental results in accordance with Mauz resulted in the following equations, where T_H is the loss torque and d_W is the pitch circle diameter of the gear wheel of the stage.

— Injection into the point of engagement:

$$T_H = 1,67 \cdot 10^{-6} \rho_{oil} \dot{V}_{oil} d_W (v_t - v_S) + 32 \cdot 10^{-9} \rho_{oil} d_W^{1,5} \ \nu_{oil}^{0,065} \ m_n^{0,18} \ b^{0,5} \ v_t^{1,5\left(\dot{V}_{oil}/\dot{V}_0\right)^{0,1}} + 0,1 \qquad (9)$$

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

— Injection into the point of disengagement:

$$T_H = 8,33 \cdot 10^{-6} \rho_{oil} \dot{V}_{oil} d_W (v_t + v_S) \qquad (10)$$

The equations are not dimensionless. The constants have been chosen so that on substitution of the individual variables in the units stated, the loss torque, T_H, in Newton metres, is obtained for both equations. The loss torque thus calculated applies to the mating gear pair. The power loss of a pair of gears is obtained by multiplication of the loss torque, T_H, by the angular velocity, ω, of the gear wheel of the stage. The total power loss of all pairs of gears is obtained by totalling the individual losses, according to (8).

Application of the two equations is restricted by Mauz to the operating and design parameters contained in Table 2. Additionally, the distance between gears and walls has to be big enough to prevent a pumping effect. Sample calculations show that the equations can be usefully applied well in excess of this range.

Table 2 — Range of parameters according to Mauz

Influence variable	Formula	Unit	Range of variation	
			from	to
Reynold's number	$Re = v_t\, d_a / V_{oil}$	—	4125	531 428
Relative immersion depth	$2\, h_e/d_a$	—	0,04	2,0
Tip circle diameter	d_a	mm	132	248
Tooth width	b	mm	10	60
Immersion depth	h_e	mm	5	135
Modulus	m	mm	3	6
Peripheral speed	v_t	m/s	10	60
Kinematic viscosity	v_{oil}	mm^2/s	15	240
Density of oil at 15 °C	ρ_{15}	kg/m^3	855	881

5.2.3 Load-dependent gear losses

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

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

NOTE P_{VZP}: per engagement, not for planetary gear units.

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

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

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

where

Ra = 0,5 ($Ra_{\eta 1} + Ra_2$), and the lubricant factor X_L according to Schlenk [14]:

X_L = 1,0 for mineral oils;

X_L = 0,8 for polyalfaolefins and esters;

X_L = 0,75 $(6/v_{\Sigma})^{0,2}$ for polyglycols;

X_L = 1,3 for phosphoric esters;

X_L = 1,5 for traction fluids.

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

— v_Σ for $v_t \leqslant 50$ m/s

for $v_t > 50$ m/s, v_Σ for v_t is calculated as being = 50 m/s

— $F/b \geqslant 150$ N/mm;

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

In Equation 12, the following should be substituted:

	spur gear helical gear	**bevel gear (equivalent spur gear)**	**hypoid gear**
F	$= F_{bt} = F_t / \cos \alpha_t$	$= F_{bmt} = F_{mt} / \cos \alpha_t$	$= F_n \cdot \cos \beta_{b2}$
b	$= b$	$= b_{eH} = 0{,}85\ b$	$= 0{,}85\ b_2$
v_Σ	$= 2 \cdot v_t \cdot \sin \alpha_{wt}$	$= 2 \cdot v_{mt} \cdot \sin \alpha_{vt}$	$= v_{\Sigma m}$
ρ	$= \rho_{C,\ red} / \cos \beta_b$	$= \rho_{Cn} / \cos \beta_{vb}$	$= \rho_n$

For worm gear units, the coefficient of friction μ_z is calculated separately, as shown on 5.2.6.

5.2.4 Load-dependent gear losses for cylindrical and bevel gears

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

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

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

$$H_v = \frac{\pi(u+1)}{z_1 u \cos \beta_b}\left(1 - \varepsilon_\alpha + \varepsilon_1^2 + \varepsilon_2^2\right) \tag{14}$$

For bevel gear stages, Equation 14 has to be calculated with the equivalent spur gear according to [2] ($u \rightarrow u_v$; $z_1 \rightarrow z_{v1}$; $\beta_b \rightarrow \beta_{vb}$; $\varepsilon_\alpha \rightarrow \varepsilon_{v\alpha}$; $\varepsilon_1 \rightarrow \varepsilon_{v1}$; $\varepsilon_2 \rightarrow \varepsilon_{v2}$).

5.2.5 Load-dependent gear losses for hypoid gears

Calculation of the load-dependent gear power loss, P_{VZP}, for hypoid gears on the equivalent crossed helical gear system in accordance with [2]:

$$P_{VZP} = F_n \cdot v_{gm} \cdot \mu_{mz} \tag{15}$$

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

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

and with the average sum velocity, $v_{\Sigma m}$, to calculate μ_{mz} from:

$$v_{\Sigma m} = \sqrt{v_{\Sigma s}^2 + v_{\Sigma h}^2} \tag{17}$$

$$v_{\Sigma s} = v_{t1}\left(\sin\beta_{m1} + \sin\beta_{m2}\cdot\frac{\cos\beta_{m1}}{\cos\beta_{m2}}\right) \quad (18)$$

$$v_{\Sigma h} = 2\cdot v_{t1}\cdot\cos\beta_{m1}\cdot\sin\alpha_n \quad (19)$$

5.2.6 Gear losses of worm gear units

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

$$P_{VZ} = P_{VZP} + P_{V0} - P_{VL0} \quad (20)$$

with the total no-load losses, P_{V0}, and the bearing no-load losses, P_{VL0}, in accordance with 2.2. The load-dependent gear losses P_{VZ} are obtained from:

$$P_{VZP} = F_n\ \mu_z v_{gm} \quad (21)$$

with the coefficient of friction μ_z from:

$$\mu_z = \mu_{z0}\, Y_W \left(\frac{v_{gm}}{v_\Sigma}\right)^{0,5}\cdot\left(\frac{Rz}{Rz_0}\right)^{0,25} \quad (22)$$

The basic value of the coefficient of friction μ_{z0} can be determined for any material/lubricant combination and standard conditions Rz_0 (~ 3µm), σ_H and $v_{\Sigma m}$ / v_Σ in a twin-disk test rig. For guide values, see Figure 4.

The material factor, Y_W, takes other material combinations into account; for guide values, see Table 3. The values given are valid for a case-hardened and ground worm. For through-hardened, unground worms, the values should be multiplied by 1,2.

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

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

$$P_{V0} = a\left(\frac{n_1}{60}\right)^{\frac{4}{3}}\left(\frac{v_{40}}{1{,}83} + 90\right)\cdot 10^{-4} \quad (23)$$

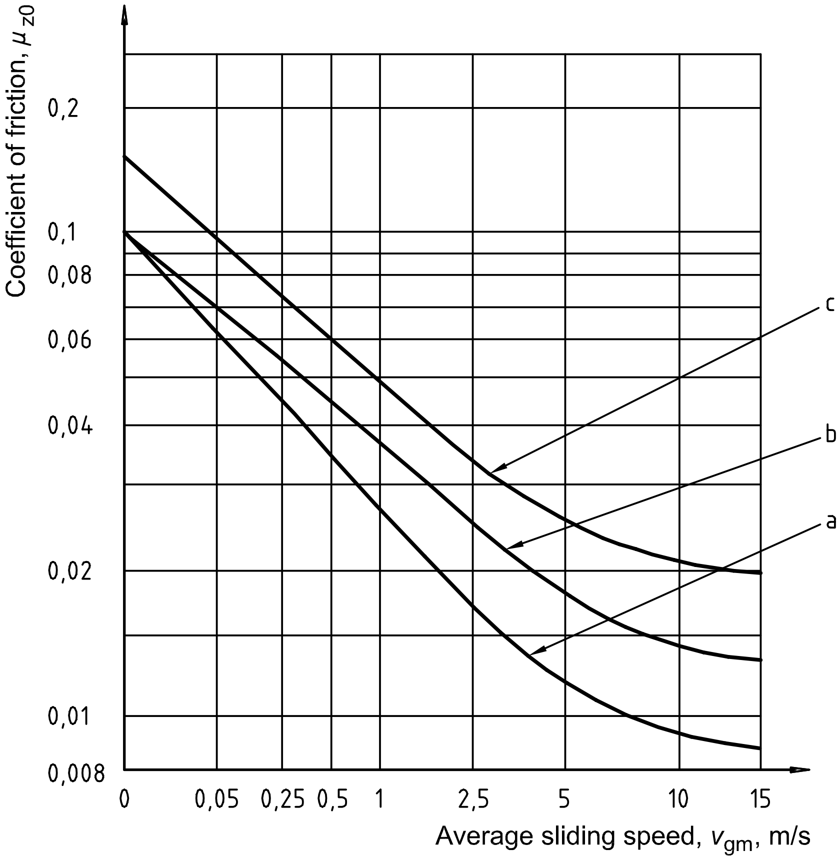
5.2.7 Planetary gear units

Investigations had not been completed at time of publication.

5.2.8 Total load-dependent gear losses

To determine the total load-dependent gear losses each part has to be added:

$$P_{VZP} = \sum_{i=1}^{stage} P_{VZP,i} \quad (24)$$



Average contact stress: $\sigma_H = 300$ N/mm^2

Slide-roll ratio: $v_{gm}/v_{\Sigma} = 1$

Material combination:

— worm: hardened and ground steel ($Rz = Rz_0 = 3$ µm)

— worm gear: centrifugally cast bronze, GZ-CuSn12.

Injection temperature: $\vartheta_{oil} = 80$ °C.

a Polyglycol $v_{40} = 150$ mm^2/s.

b Mineral oil $v_{40} = 180$ mm^2/s.

c Mineral oils to British Standard.

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

Table 3 — Guide values for material factor, Y_W

Worm material	Material factor Y_W
GZ-CuSn12Ni	0,95
GZ-CuSn12, GZ-CuSn10Zn, GZ-CuSn14	1,00
GZ-CuZn25A15, GZ-CuAI10Ni	1,10
G-CuSn12Ni	1,20
G-CuSn12, G-CuSn10Zn, GGG-70	1,30
G-CuZn25Al5, G-CuAI11Ni, GG-25	1,40

5.3 Bearing losses

5.3.1 Rolling bearings

5.3.1.1 General

The bearing loss torque, T_{VL}, in Newton metres, is calculated in accordance with the approximation formulae given in [5]. Here, the loss torque is split into no-load, T_{VL0}, and load-dependent, T_{VLP1}, parts. In the case of axially loaded cylindrical roller bearings and axially loaded needle roller bearings, an additional loss term, T_{VLP2}, occurs, which is dependent on the magnitude of the end thrust.

These components are calculated separately and then added together to give the following for the total loss torque:

$$T_{VL} = T_{VL0} + T_{VLP1} + T_{VLP2} \quad (25)$$

5.3.1.2 No-load bearing power loss

This component depends on the bearing design, the type of lubrication, the viscosity of the lubricant and the bearing speed.

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

$$T_{VLO} = 1{,}6 \cdot 10^{-8} f_0 d_m^3 \quad (26)$$

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

$$T_{VL0} = 10^{-10} f_0 \left(\nu_{oil} n\right)^{2/3} d_m^3 \quad (27)$$

The coefficients, f_0, depend on bearing type and bearing lubrication (see Table 4).

5.3.1.3 Load-dependent bearing power loss

For calculation of the load-dependent bearing loss torques, T_{VLP1} and T_{VLP2}, the following relationship applies according to [5]:

T_{VLP1}:

$$T_{VLP1} = f_1 \cdot P_1{}^a \cdot d_m^b \cdot 10^{-3} \quad (28)$$

where

f_1 is from Table 5;

P_1 is from Table 5;

a and b are according to Table 6.

T_{VLP2}:

— for radial loading:

$T_{VLP2} = 0$

— for cylindrical roller bearings with additional thrust loading:

$$T_{VLP2} = f_2 \cdot F_a \cdot d_m \cdot 10^{-3} \quad (29)$$

where

f_2 is from Table 7

Table 4 — Coefficient, f_0 [a]

Bearing design	Type of lubrication			
	Grease	Oil mist	Oil bath	Oil injection, oil bath with vertical shaft
Deep-groove ball bearing:				
single-row	0,75…2 [b]	1	2	4
double-row	3	2	4	8
Self-aligning ball bearing	1,5…2 [b]	0,7…1 [b]	1,5…2 [b]	3…4 [b]
Angular contact ball bearing:				
single-row	2	1,7	3,3	6,6
double-row	4	3,4	6,5	13
Four-point contact bearing	6	2	6	9
Cylindrical roller bearing (cage):				
Series 10, 2, 3, 4	0,6	1,5	2,2	2,2 [c]
Series 22	0,8	2,1	3	3 [c]
Series 23	1	2,8	4	4 [c]
Cylindrical roller bearing (full roller)				
single-row	5 [d]	—	5	—
double-row	10 [d]	—	10	—
Needle roller bearing	12	6	12	24
Self-aligning roller bearing:				
Series 213	3,5	1,75	3,5	7
Series 222	4	2	4	8
Series 223, 230, 239	4,5	2,25	4,5	9
Series 231	5,5	2,75	5,5	11
Series 232	6	3	6	12
Series 240	6,5	3,25	6,5	13
Series 241	7	3,5	7	14
Taper roller bearing:				
single-row	6	3	6	8…10 [b, c]
double-row	12	6	12	16…20 [b, c]
Deep-groove ball thrust bearing	5,5	0,8	1,5	3
Cylindrical roller thrust bearing	9	—	3,5	7
Needle roller thrust bearing	14	—	5	11
Self-aligning roller thrust bearing:				
Series 292 E	—	—	2,5	5
Series 292	—	—	3,7	7,4
Series 293 E	—	—	3	6
Series 293	—	—	4,5	9
Series 294 E	—	—	3,3	6,6
Series 294	—	—	5	10

[a] The shown values are valid for steady conditions. For lately greased bearings (2...4), f_0 is to be used in the calculation.

[b] The low values apply to the lightweight bearing, and the high values to the heavyweight bearings of a bore series.

[c] Valid for oil injection lubrication. For oil bath lubrication and vertical shaft, the shown value is to be doubled.

[d] Valid for low rotation speed up to 20 % of the reference rotation speed (see bearing tables). At higher rotation speed, the f_0 value is to be doubled for the calculation.

Table 5 — Coefficient, f_1, and equivalent bearing load, P_1

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

[a] If $P_1 < F_r$, P_1 should be calculated as $= F_r$.

[b] The low values apply to the lightweight bearings; the high values to the heavyweight bearings of a bore series.

[c] For additionally thrust-loaded cylindrical roller bearings, T_{VLP2} has to be introduced.

Table 6 — Exponents, *a*, *b*

Bearing series	*a*	*b*
213	1,35	0,2
222	1,35	0,3
223	1,35	0,1
230	1,5	– 0,3
231, 232, 239	1,5	– 0,1
240, 241	1,5	– 0,2
Exponents for self-aligning roller bearing. For other bearings a = b = 1,0.		

Table 7 — Coefficient f_2 for cylindrical roller bearings

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

5.3.1.4 Total bearing power loss

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

$$P_{\mathrm{VL}} = \sum_{i=1}^{\text{bearing}} \left(T_{\mathrm{VL},i}\, \omega_i \right) = \sum_{i=1}^{\text{bearing}} \left(T_{\mathrm{VL},i} \frac{\pi\, n_i}{30} \right) \tag{30}$$

5.3.2 Plain bearings

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

Radial journal bearings as fully and partially surrounding regular cylinder bearings are calculated according to DIN 31652 [6], and as sectioned surface and tilting pad bearings according to DIN 31 657 [7].

Calculation of journal thrust bearings as segmental thrust bearings is given in DIN 31653 [8], and as tilting pad thrust bearings in DIN 31654 [9].

5.4 Shaft seals

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

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

$$P_{\mathrm{VD}} = 7{,}69 \cdot 10^{-6} d_{\mathrm{sh}}^{2} n \tag{31}$$

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

6 Heat dissipation

6.1 General

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

$$Q = Q_{\mathrm{ca}} + Q_{\mathrm{fun}} + Q_{\mathrm{rot}} + \Delta H_{\mathrm{oil}} \tag{32}$$

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

6.2 Heat dissipation through the housing

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

$$Q_{\mathrm{ca}} = k A_{\mathrm{ca}} \left(\vartheta_{\mathrm{oil}} - \vartheta_{\infty} \right) \tag{33}$$

In the area of housing, A_{ca}, the bottom is not included.

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

$$\frac{1}{k} = \frac{1}{\alpha_{\mathrm{oil}}} \frac{A_{\mathrm{ca}}}{A_{\mathrm{oil}}} + \frac{\delta_{\mathrm{wall}}}{\lambda_{\mathrm{wall}}} \frac{A_{\mathrm{ca}}}{A_{\mathrm{oil}}} + \frac{1}{\alpha_{\mathrm{ca}}} \tag{34}$$

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

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

$$\alpha_{\mathrm{ca}} = \alpha_{\mathrm{con}} + \alpha_{\mathrm{rad}} \tag{35}$$

The radiation part can be calculated from:

$$\alpha_{\text{rad}} = 0{,}23 \cdot 10^{-6} \varepsilon \left(\frac{T_{\text{wall}} + T_{\infty}}{2} \right)^3 \tag{36}$$

where the emission ratio, ε, is given in Table 8.

Table 8 — Emission ratio, ε

Material	Condition	Emission ratio ε
Grey cast iron GG	Casting scale	0,60 ... 0,80
	Lathed or hobbed	0,35 ... 0,45
Steel	Rolling skin	0,80 ... 0,90
	Lathed or hobbed	≈ 0,15
	Hobbed and oil covered	≈ 0,35
	Sandblasted	≈ 0,35
	Sandblasted and oil covered	0,50 ... 0,60
Aluminium	Oxide skin	≈ 0,15
	Lathed or hobbed	0,05 ... 0,10
All materials painted	With and without oil or dust cover	0,90 ... 0,95

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

$$\alpha_{\text{con}} = \alpha_{\text{K,free}} \left(1 - \frac{A_{\text{air}}}{A_{\text{ca}}} \right) + \alpha_{\text{K,forced}} \frac{A_{\text{air}}}{A_{\text{ca}}} \eta^{*} \tag{37}$$

where

$$\eta^{*} = \frac{T_{\text{wall}} - T_{\text{air}}}{T_{\text{wall}} - T_{\infty}} \tag{38}$$

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

For free convection ($v_{\text{air}} < 1{,}5$ m/s):

$$\alpha_{\text{K, free}} = 18 h_{\text{ca}}^{-0{,}1} \left(\frac{T_{\text{wall}} - T_{\infty}}{T_{\infty}} \right)^{0{,}3} \tag{39}$$

For forced convection ($v_{\text{air}} > 1{,}5$ m/s):

$$\alpha_{\text{K, forced}} = \frac{0{,}008\,6 \; (Re')^{0{,}64}}{l_{\text{x}}} \tag{40}$$

where

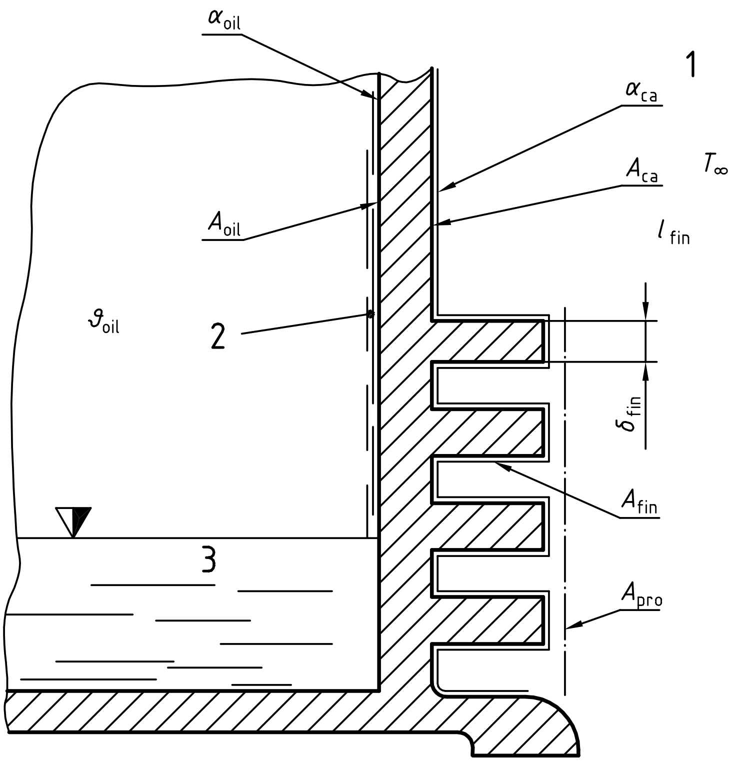
$$Re' = \sqrt{Re^2 + \frac{Gr}{2{,}5}} \tag{41}$$

$$Re = \frac{v_{\text{air}} l_{\text{X}}}{\nu_{\text{air}}} \tag{42}$$

$$Gr = \frac{g\, h_{\text{ca}}^3 \left(T_{\text{wall}} - T_\infty\right)}{T_\infty\, \nu_{\text{air}}^2} \tag{43}$$

with $\nu_{\text{air}} = 15{,}6 \cdot 10^{-6}$ mm²/s.

For housings with thermal finning in accordance with Figure 5, the following is valid.



Key

1 Environment

2 Oil film

3 Oil sump

Figure 5 — Housing with thermal finning

For free convection ($A_{\text{air}} = 0$):

$$\alpha_{\text{ca}} = \frac{A_{\text{fin}}}{A_{\text{ca}}}\left(\alpha_{\text{K,free}} + \alpha_{\text{rad}}\frac{A_{\text{pro}}}{A_{\text{fin}}}\right)\eta_{\text{f}} + \left(1 - \frac{A_{\text{fin}}}{A_{\text{ca}}}\right)\left(\alpha_{\text{K,free}} + \alpha_{\text{rad}}\right) \tag{44}$$

with the fin efficiency, η_{f}:

$$\eta_{\text{f}} = \frac{\tanh\left(m\, l_{\text{fin}}\right)}{\left(m\, l_{\text{fin}}\right)} \tag{45}$$

where

$$m = \sqrt{2\frac{\alpha_{\text{con}} + \alpha_{\text{rad}}\frac{A_{\text{pro}}}{A_{\text{fin}}}}{\delta_{\text{fin}}\ \lambda_{\text{fin}}}} \qquad (46)$$

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

$$\alpha_{\text{ca}} = \frac{A_{\text{air}}}{A_{\text{ca}}}\left(\alpha_{\text{K,forced}}\ \eta^{*} + \alpha_{\text{rad}}\frac{A_{\text{pro}}}{A_{\text{air}}}\right)\eta_{\text{f}} + \left(1 - \frac{A_{\text{air}}}{A_{\text{ca}}}\right)\left(\alpha_{\text{K,free}} + \alpha_{\text{rad}}\right) \qquad (47)$$

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

$$\alpha_{\text{ca}} = \left(1 - \frac{A_{\text{air}}}{A_{\text{ca}}}\right)\left(\alpha_{\text{K,free}} + \alpha_{\text{rad}}\right) + \frac{A_{\text{air}} - A_{\text{fin}}}{A_{\text{ca}}}\left(\alpha_{\text{K,forced}}\ \eta^{*} + \alpha_{\text{rad}}\right) + \frac{A_{\text{fin}}}{A_{\text{ca}}}\left(\alpha_{\text{K,forced}}\ \eta^{*} + \alpha_{\text{rad}}\frac{A_{\text{pro}}}{A_{\text{fin}}}\right)\eta_{\text{f}} \qquad (48)$$

6.3 Heat dissipation via the foundation

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

$$Q_{\text{fun}} = f\ \lambda_{\text{fun}}\ \Delta T_{\text{fun}} \sum_{i=1}^{n} A_{\text{q}i} m_i \frac{\frac{\alpha_{\text{fun}}}{\lambda_{\text{fun}}\ m_i^{*}}\tanh\left(m_i^{*} L_i\right)}{1 + \frac{\alpha_{\text{fun}}}{\lambda_{\text{fun}}\ m_i^{*}}\tanh\left(m_i^{*} L_i\right)} \qquad (49)$$

$$f = 1{,}46\ \left(\frac{A_{\text{foot}}}{A_{\text{bot}}}\right)^{0{,}16} \qquad (50)$$

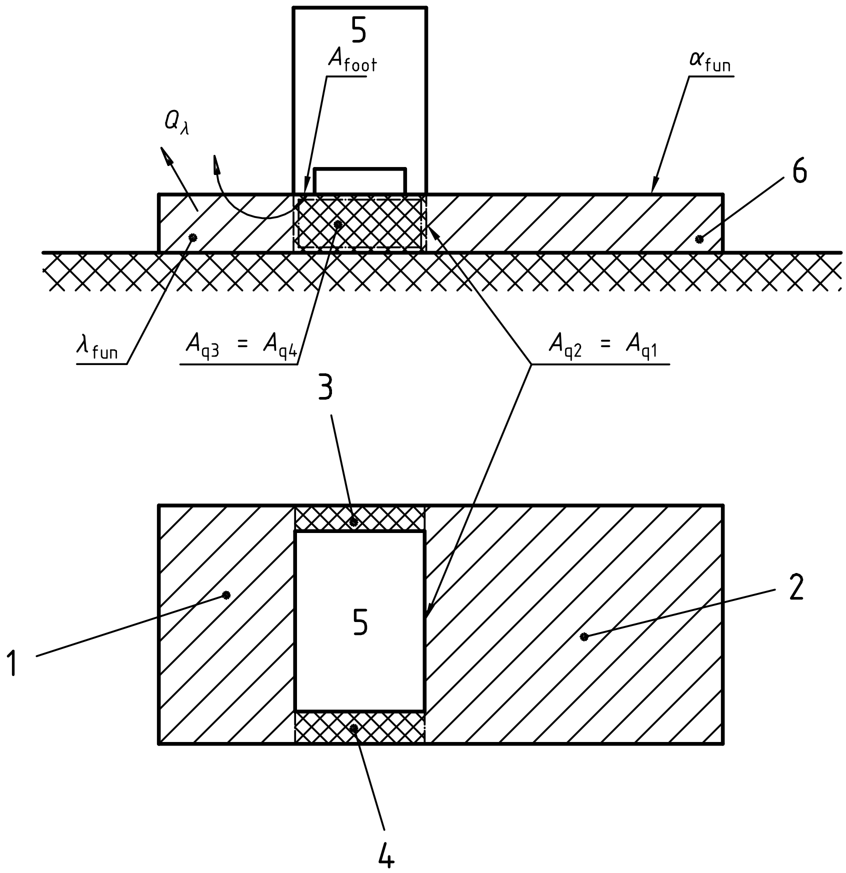
$$\Delta T_{\text{fun}} = 0{,}62\ \left(T_{\text{oil}} - T_{\infty}\right) \qquad (51)$$

$$m_i = \sqrt{\frac{\alpha_{\text{fun}}\ U_i}{\lambda_{\text{fun}} A_{\text{q}i}}} \qquad (52)$$

with

λ_{fun} from Table 9.

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



Key

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

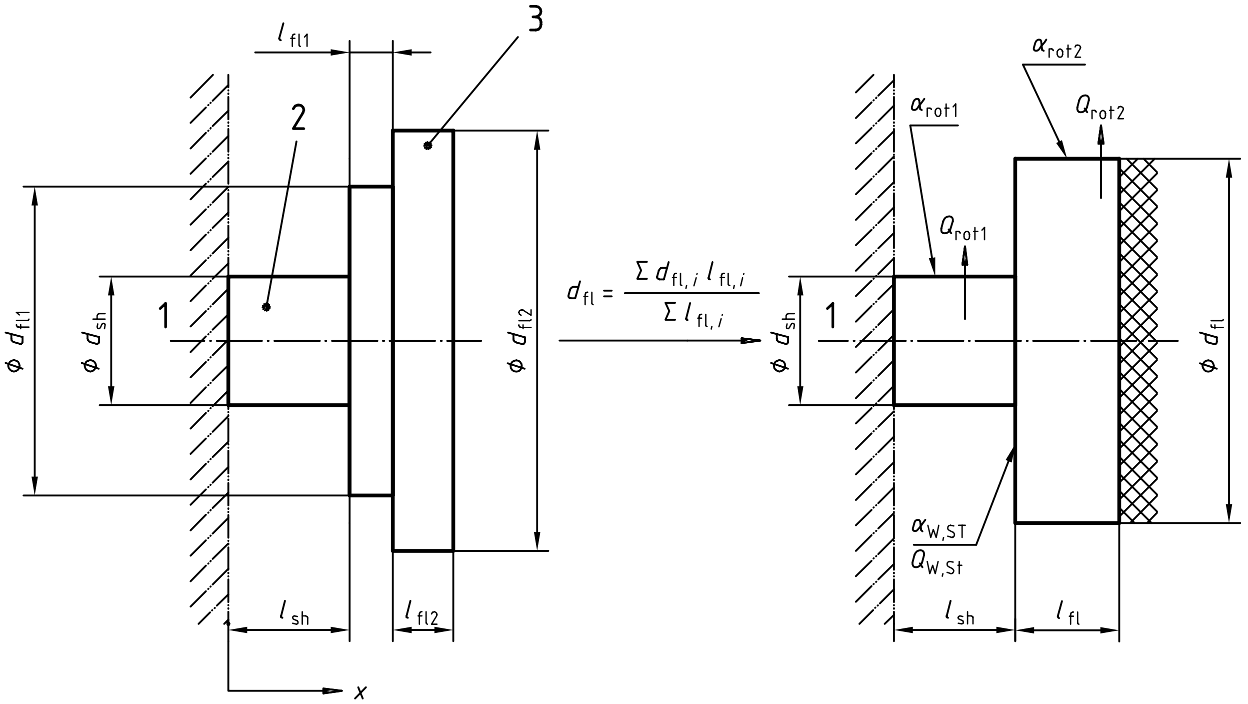
Figure 6 — Heat dissipation through the foundation

Table 9 — Thermal conduction, λ

Material	**Thermal conduction, λ** W/mK
Aluminium	180
Steel	50
Grey cast iron GG	42
Concrete	1
Air	0,027

6.4 Heat dissipation via shafts and couplings

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



Key

1 Gear unit

2 Shaft end

3 Coupling half

Figure 7 — Equivalent system of shaft end and coupling flange

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

$$Q_{\text{rot}} = Q_{\text{rot1}} + Q_{\text{rot2}} \tag{53}$$

with

$$Q_{\text{rot1}} = \lambda_{\text{sh}}\, m_{\text{sh}}\, A_{\text{q,sh}} \left(T_{\text{sh}} - T_{\infty}\right)\Big|_{x=0} \frac{\dfrac{\alpha^{*}_{\text{sh,face}}}{\lambda_{\text{sh}}\, m_{\text{sh}}} + \tanh\left(m_{\text{sh}} l_{\text{sh}}\right)}{1 + \dfrac{\alpha^{*}_{\text{sh,face}}}{\lambda_{\text{sh}}\, m_{\text{sh}}} \tanh\left(m_{\text{sh}} l_{\text{sh}}\right)} \tag{54}$$

and

$$Q_{\text{rot2}} = \lambda_{\text{fl}}\, m_{\text{fl}}\, A_{\text{q,fl}} \left(T_{\text{fl}} - T_{\infty}\right)\Big|_{x=l_{\text{sh}}} \tanh\left(m_{\text{fl}}\, l_{\text{fl}}\right) \tag{55}$$

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

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

$$\alpha^*_{\text{sh,face}} = \frac{\lambda_{\text{fl}} m_{\text{fl}} A_{\text{q,fl}} \tanh\left(m_{\text{fl}} l_{\text{fl}}\right)}{A_{\text{q,sh}}} \tag{56}$$

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

$$m_{\text{sh}} = 2\sqrt{\frac{\alpha_{\text{rot1}}}{\lambda_{\text{sh}} d_{\text{sh}}}} \tag{57}$$

and

$$m_{\text{fl}} = 2\sqrt{\frac{\alpha_{\text{rot2}}}{\lambda_{\text{fl}} d_{\text{fl}}}} \tag{58}$$

with

$\lambda_{\text{sh, fl}}$ from Table 9,

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

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

For $Re \leqslant 2\,500$:

$$Nu = 0{,}40\ Gr^{0{,}25} \tag{59}$$

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

$$Nu = 0{,}095\left(0{,}5 Re^2 + Gr\right)^{0{,}35} \tag{60}$$

For $Re > 15\,000$:

$$Nu = 0{,}073\ Re^{0{,}7} \tag{61}$$

where

$$Re = \frac{n \pi d^2_{\text{sh,fl}}}{60 v_{\text{air}}} \tag{62}$$

$$Nu = \frac{\alpha_{\text{rot1,2}} d_{\text{sh,fl}}}{\lambda_{\text{air}}} \tag{63}$$

$$Gr = g\left(2{,}5 d_{\text{sh,fl}}\right)^3 \frac{T_{\text{sh,fl}} - T_\infty}{T_\infty v_{\text{air}}^2} \tag{64}$$

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

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

$$\left(T_{\text{sh}} - T_{\infty}\right) \Big|_{x=0} = \left(T_{\text{oil}} - T_{\infty}\right) \tag{65}$$

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

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

$$\left(T_{\text{fl}} - T_{\infty}\right) \Big|_{x=l_{\text{sh}}} = \left(T_{\text{sh}} - T_{\infty}\right) \Big|_{x=l_{\text{sh}}} \tag{66}$$

and

$$\left(T_{\text{sh}} - T_{\infty}\right) \Big|_{x=l_{\text{sh}}} = \frac{T_{\text{oil}} - T_{\infty}}{\cosh\left(m_{\text{sh}}\, l_{\text{sh}}\right) + \dfrac{\alpha_{\text{sh,face}}^{*}}{\lambda_{\text{sh}}\, m_{\text{sh}}} \sinh\left(m_{\text{sh}}\, l_{\text{sh}}\right)} \tag{67}$$

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

It is then true to say that:

$$\left(T_{\text{sh}} - T_{\infty}\right) = \frac{1}{2}\left(T_{\text{oil}} - T_{\infty}\right)\left(1 + \frac{1}{\cosh\left(m_{\text{sh}} l_{\text{sh}}\right) + \dfrac{\alpha_{\text{sh,face}}}{\lambda_{\text{sh}} m_{\text{sh}}} \sinh\left(m_{\text{sh}} l_{\text{sh}}\right)}\right) \tag{68}$$

and

$$\left(T_{\text{fl}} - T_{\infty}\right) = \frac{1}{2}\left(T_{\text{sh}} - T_{\infty}\right) \Big|_{x=l_{\text{sh}}} \left(1 + \frac{1}{\cosh\left(m_{\text{fl}}\, l_{\text{fl}}\right)}\right) \tag{69}$$

6.5 Heat dissipation via an external cooler

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

$$\Delta H_{\text{oil}} = 1{,}67 \cdot 10^{-2}\, \dot{V}_{\text{oil}}\, \rho_{\text{oil}} c_{\text{oil}} \Delta \vartheta_{\text{oil}} \tag{70}$$

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

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

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

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

7 Results of calculation

7.1 Splash lubrication

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

$$P_V\left(\vartheta_{oil}\right) = Q\left(\vartheta_{oil}\right) \qquad (71)$$

If Equation (68) is not true, further iterative steps can be made. The calculation has to be modified with an optimized operating temperature, which can be obtained from the following equation:

$$\vartheta_{oil,new} = \frac{\left(\dfrac{P_V}{\alpha_{ca} \cdot A_{ca}} + \vartheta_{\infty}\right) + \vartheta_{oil}}{2} \qquad (72)$$

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

$$P_V\left(\vartheta_{oil\ max}\right) \leqslant Q\left(\vartheta_{oil\ max}\right) \qquad (73)$$

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

7.2 Injection lubrication

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

8 Sample calculation

8.1 General

For the calculation, a three-stage bevel/spur gear unit is chosen.

8.2 Geometry and surrounding conditions

8.2.1 General

In Figure 8, a schematic view of the gear unit is shown. It consists of the shafts (I), (II), (III), (IV), the stages A (bevel gear), B, C (spur gears) and the bearings 1 to 8. The gear unit is splash-lubricated with mineral oil.

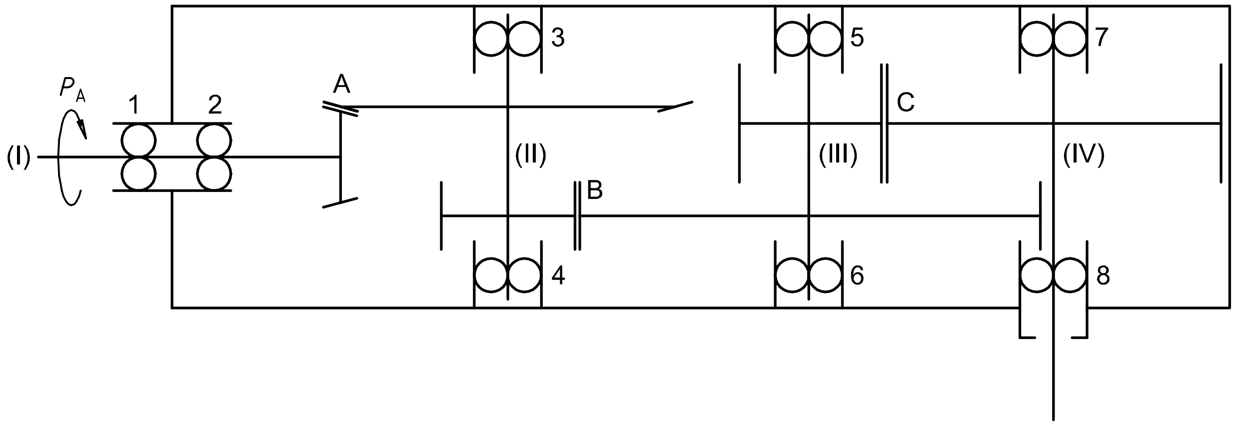


Figure 8 — Schematic view of the examined gear unit

8.2.2 Data of the gear unit

a) Dimensions of the gear unit:

— height: 662 mm

— length: 925 mm

— width: 370 mm.

b) Oil level: 75 mm below axis.

c) Air velocity: $v_{air} = 1$ m/s.

d) Ambient temperature: $\vartheta_{\infty} = 25$ °C.

e) Input speed: $n_A = 1\,000$ rpm.

f) Input torque: $T_A = 200$ N · m.

g) Input power: $P_A = 20\,944$ W.

h) Oil type: mineral oil, ISO VG 320.

i) Oil lubricant factor: $X_L = 1{,}0$.

j) Oil viscosity at 40 °C: $\nu_{40} = 320$ mm^2/s.

k) Oil viscosity at 100 °C: $\nu_{100} = 22$ mm^2/s.

l) Wall thickness: $\delta_{wall} = 10$ mm.

To start the calculating process, the oil temperature has first to been assumed. The temperature is needed to specify the operating oil viscosity for the calculation of losses and for the determination the heat dissipation. The calculation starts with an estimated oil temperature:

— estimated oil temperature: $\vartheta_{oil} = 60$ °C;

— operating oil viscosity: $\nu_{oil} = 103{,}6$ mm^2/s according to Ubbleohde-Walter [1].

8.2.3 Gear stage data

See Table 10.

Table 10 — Gear stages

Parameter	Unit	Stage A		Stage B		Stage C	
		pinion	gear	pinion	gear	pinion	gear
b	mm	34		80		120	
F_t	N	12 133		27 178		77 051	
h_e	mm	0	25	0	81	0	145
Ra	µm	0,4	0,4	0,4	0,4	0,4	0,4
u	—	4,45		4,05		4,05	
v_t	m/s	2,23		0,844		0,298	
v_Σ	m/s	1,79		0,684		0,244	
z	—	11	49	20	81	20	81
α_n	°	20		20		20	
β	°	34,1		13		13	
ε	—	0,810 (ε_{v1})	0,326 (ε_{v2})	0,831	0,570	0,914	0,430
ρ_C	mm	9,83		11,9		17,3	

8.2.4 Bearing data

See Table 11.

Table 11 — Taper roller bearings

Bearing No.	Bearing type	Shaft No.	n (rpm)	d_m (mm)	F_r (N)	F_a (N)	Y (–)
1	Taper roller bearing	I	1 000	87,5	3470	1577	1,1
2	Taper roller bearing	I	1 000	87,5	13 110	8 710	1,1
3	Taper roller bearing	II	224,5	95	12 530	18 310	1,1
4	Taper roller bearing	II	224,5	95	21 870	9 940	1,1
5	Taper roller bearing	III	55,43	132,5	55 870	22 960	1,4
6	Taper roller bearing	III	55,43	132,5	42 350	12 560	1,4
7	Taper roller bearing	IV	13,69	180	46 070	16 450	1,7
8	Taper roller bearing	IV	13,69	180	32 920	32 690	1,7

8.3 Power loss

8.3.1 Gear losses

8.3.1.1 No-load gear losses

$l_h = 4A_G/U_M = 4 \cdot (662 \cdot 925)/(2(662 + 925)) = \underline{772}$ mm — from Figure 3

Stage A:

$C_{Sp,1} = \left((4 \cdot 25)/[3 \cdot (75 + 25)]\right)^{1,5} \cdot 2(75 + 25)/772 = \underline{0,049\ 9}$ — from (6)

$C_{1,1} = 0,063(25 + 0)/10 + 0,012\ 8(34/10)^3 = \underline{0,661}$ — from (7)

$C_{2,1} = (25 + 0)/(80 \cdot 10) + 0,2 = \underline{0,231}$ — from (7)

$T_{H,1} = 0,049\ 9 \cdot 0,946 \cdot e^{0,231 \cdot 2,23/10} = \underline{0,034\ 7}$ N · m — from (5)

$P_{H,1} = 0,034\ 7 \cdot \pi \cdot 224,5/30 = \underline{0,815\ \text{W}}$ — from (8)

Stage B:	Stage C:	
$C_{Sp,2} = \ldots = \underline{0,23}$	$C_{Sp,3} = \ldots = \underline{0,470}$	from (6)
$C_{1,2} = \ldots = \underline{7,06}$	$C_{1,3} = \ldots = \underline{23,0}$	from (7)
$C_{2,2} = \ldots = \underline{0,301}$	$C_{2,3} = \ldots = \underline{0,381}$	from (7)
$T_{H,2} = \ldots = \underline{1,69}$ N · m	$T_{H,3} = \ldots = \underline{10,9}$ N · m	from (5)
$P_{H,2} = \ldots = \underline{9,79\ \text{W}}$	$P_{H,3} = \ldots = \underline{15,7\ \text{W}}$	from (8)

Total no-load gear losses:

$P_H = P_{H,1} + P_{H,2} + P_{H,3} = 0,81 + 9,79 + 15,7 = \underline{\mathbf{26,3\ W}}$ — from (8)

8.3.1.2 Load-dependent gear losses

Stage A:

NOTE This stage is a bevel gear stage, so the calculation will be made with the equivalent spur gear according to [2].

$\sin(\beta_{vb}) = \sin(\beta_m) \cos(\alpha_n) = \sin(34,1) \cdot \cos(20) => \beta_{vb} = 31,79°$ — [2]

$u_v = u^2 = 4,45^2 = 19,80$ — [2]

$z_{v1} = z_1 \cdot \cos[\arctan(1/u)] = 11 \cdot \cos[(\arctan(1/4.45)] = 11,27$ — [2]

$\mu_{mz} = 0,048\ [(12\ 133/34,1)/(1,79 \cdot 9,83)]^{0,2} \cdot 103,6^{-0,05} \cdot 0,4^{0,25} \cdot 1,0 = \underline{0,055\ 2}$ — from (12)

$H_V = \pi(19,80 + 1)/[11,27 \cdot 19,80 \cdot \cos(31,79)]$
$[1 - (0,810 + 0,326) + 0,810^2 + 0,326^2] = \underline{0,216}$ — from (14)

$P_{VZP,1} = 20\ 944 \cdot 0{,}0552 \cdot 0{,}215 = \underline{250\ W}$ from (13)

Stage B:

$\mu_{mz} = \ldots = 0{,}063\ 8$

$H_V = \ldots = 0{,}124$

$P_{VZP,2} = \ldots = \underline{165\ W}$

Stage C:

$\mu_{mz} = \ldots = 0{,}082\ 7$ from (12)

$H_V = \ldots = 0{,}136$ from (14)

$P_{VZP,3} = \ldots = \underline{235\ W}$ from (13)

Total load-dependent gear losses:

$P_{VZP} = 250 + 165 + 235 = \underline{\mathbf{650\ W}}$ from (21)

8.3.2 Bearing losses

8.3.2.1 Shaft I

$\nu_{oil} \cdot n = 103{,}6 \cdot 1\ 000 = 103\ 600 > 2\ 000$

8.3.2.2 Bearing 1

Taper roller bearing:

$f_0 = 6$ from Table 4

$f_1 = 0{,}000\ 4$ from Table 5

$P_1 = 2\ Y \cdot F_a = 2 \cdot 1{,}1 \cdot 1\ 577 = 3\ 470$ from Table 5

$T_{VL0,1} = 10^{-10} \cdot 6 \cdot (103{,}6 \cdot 1\,000)^{2/3} \cdot 87{,}5^3 = 0{,}886\ \ N \cdot m$ from (27)

$T_{VLP1,1} = 0{,}000\ 4 \cdot 3\ 470 \cdot 87{,}5 \cdot 10^{-3} = \underline{0{,}121\ N \cdot m}$ from (28)

$T_{VLP2,1} = \underline{0\ N \cdot m}$

$T_{VL,1} = 0{,}886 + 0{,}121 + 0 = \underline{1{,}007\ N \cdot m}$ from (25)

$P_{VL,1} = 1{,}007 \cdot 1\ 000 \cdot \pi/30 = \underline{105\ W}$ from (30)

8.3.2.3 Bearing 2

$P_1 = 2 \cdot 1{,}1 \cdot 8710 = 19\ 160$ from Table 5

$T_{VL0,2} = \ldots = \underline{0{,}886\ N \cdot m}$ from (27)

$T_{VLP1,2} = \ldots = 0{,}668\ N \cdot m$ from (28)

$T_{VLP2,2} = \underline{0\ N \cdot m}$

$T_{VL,2} = \ldots = \underline{1{,}554\ N \cdot m}$ from (25)

$P_{VLP,2}$ = … = 163 W from (30)

8.3.2.4 Shaft IV

$\nu_{oil} \cdot n = 103{,}6 \cdot 13{,}7 = 1\,420 < 2\,000$

8.3.2.5 Bearing 7

$T_{VL0,7} = 1{,}6 \cdot 10^{-8} \cdot 6 \cdot 180^3 = \underline{0{,}560}$ N · m from (26)

See Table 12.

Table 12 — Sum of load-dependent and non-load-dependent bearing losses

Bearing	$\nu_{oil} \cdot n$	T_{VL0}	T_{VLP1}	T_{VL}	P_{VL}
1	103 600	0,89	0,12	1,01	105,6
2	103 600	0,89	0,67	1,56	163,1
3	23 260	0,42	1,53	1,95	45,84
4	23 260	0,42	0,83	1,25	29,39
5	5 740	0,45	3,41	3,85	22,38
6	5 740	0,45	1,86	2,31	13,42
7	1 420	0,56	4,03	4,59	6,58
8	1 420	0,56	8	8,56	12,28

Total bearing losses:

$P_{VL} = 105{,}6 + 163{,}1 + 45{,}8 + 29{,}4 + 22{,}4 + 13{,}4 + 6{,}6 + 12{,}3 =$ **398,6 W** from (30)

8.3.3 Shaft seal losses

Non-contacting seals = > P_{VD} = 0 W

8.3.4 Total gear unit losses

$P_V = 26{,}3 + 650 + 399 =$ **1 075 W** from (3)

NOTE The measured total power loss was $P_V = 1\,165$ W.

8.4 Heat dissipation

ε = 0,9 (steel, rolling skin) from Table 8

$\alpha_{rad} = 0{,}23 \cdot 10^{-6} \cdot 0{,}9 \left\{ \left[(25+273) + (60+273) \right] / 2 \right\}^3 =$ **6,5 W/m²K** from (36)

$\alpha_{k,free} = 18 \cdot 0{,}662^{-0{,}1} \cdot \left\{ \left[(60+273) - (25+273) \right] / (25+273) \right\}^{0{,}3} =$ **9,9 W/m²K** from (39)

$\alpha_{k,force} = 0$ from (40)

α_{ca} = 6,5 + 9,9 + 0 = **16,4 W/m²K** from (35), (37)

A_{ca} = A_{oil}

λ_{wall} = 50 W/mK from Table 9

α_{wall} = 200 W/(m²K) from clause 6

1/k = 1/200 + 0,01/50 + 1/16,4 = **1/15,1 m²K/W** from (34)

A_{ca} = 2 · (0,662·0,925 + 0,662·0,370) + 0,925·0,370 = 2,06 m² (without bottom)

Q_{ca} = 15,1·2,06·(60-25) = **1 085 W** from (33)

8.5 Comparison

P_V (60°) < Q (60°) from (71)

The results are nearly equal, so the estimated temperature was right.

If a better accuracy is needed, a second iterative step has to be calculated. The same procedure has to be started with an optimized oil temperature:

$\vartheta_{oil, new}$ = [1 075/(15,1·2,06) + 25 + 60]/2 = **59,8 °C** from (72)

With further iterative steps, the equilibrium will appear at 59,7 °C.

NOTE The measured temperature with this gear unit was ϑ_{oil} = 70 °C.

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