



American
Gear Manufacturers
Association®

AGMA 947-A23

AGMA Information Sheet

Gear Reducers – Thermal Capacity

AGMA 947-A23

American Gear Manufacturers Association ***Gear Reducers – Thermal Capacity***
AGMA 947-A23

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ABSTRACT

This information sheet 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.

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Foreword

[The foreword, footnotes, and annexes, if any, in this document are provided for informational purposes only and are not to be construed as a part of AGMA 947-A23, *Gear Reducers – Thermal Capacity*.]

This thermal rating method was the American proposal to ISO/TR 14179. 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 Technical Paper 96FTM9 by A.E. Phillips [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.

AGMA 947-A23 was renamed from AGMA ISO 14179-1 because it is not identical to ISO/TR 14179-1:2001, *Gears – Thermal capacity – Part 1: Rating gear drives with thermal equilibrium at 95°C sump temperature*. During the revision, several additions were made to the document:

- Power variables and subsequent equations were converted to Watts.
- Mesh coefficient of friction calculations are based on ISO 14179-2 and were expanded based on gearing type and to include Hohn's modification. ISO 10300-1 methods were included for calculating mesh coefficient of friction for bevel gears.
- The oil seal power loss formula was updated based on updated manufacturer information.
- The gear windage and churning losses clauses were expanded to include the alternate method presented in ISO 14179-2:2001, *Gears – Thermal capacity – Part 2: Thermal load-carrying capacity*.
- Bearing windage and churning loss calculations were updated based on updated manufacturer information.
- The heat dissipation clauses were greatly expanded to include calculations for the heat transfer coefficient under various operating and configuration conditions.

The first draft of AGMA ISO 14179-1 was made in December 2002. It was approved by the AGMA membership in March 2004.

The first draft of AGMA 947-A23 was created in April 2019. It was approved by the Technical Division Executive Committee in June 2023.

Suggestions for improvement of this information sheet will be welcome. They may be submitted to tech@agma.org.

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American Gear Manufacturers Association – Gear Reducers – Thermal Capacity

1 Scope

This information sheet 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 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.

Mesh power loss for planetary gear units is not covered in this information sheet.

2 Normative references

The following documents contain provisions which, through reference in this text, constitute provisions of this information sheet. All publications below are subject to revision. When publication date does not appear after the document, it is understood that the most recent editions of the documents apply.

AGMA 925, *Effect of Tribology and Lubrication on Gear Surface Distress*

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

ISO 10300-1:2014, *Calculation of load capacity of bevel gears – Part 1: Introduction and general influence factors*

ISO 10300-20:2021, *Calculation of load capacity of bevel gears – Part 20: Calculation of scuffing load capacity – Flash temperature method*

3 Symbols and units, terms, and definitions

For the purposes of this information sheet, the symbols and units given in Table 1, and the following terms and definitions, apply.

3.1 Sump

Interior space at the bottom of the gear housing that collects lubricant that drains from the gearbox internal components. Alternatively it may be an external reservoir separate from the gear housing.

3.2 Thermal power rating

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

NOTE 1: Thermal power rating needs to 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 power rating depends upon the specifics of the drive, operating conditions, maximum allowable sump temperature, as well as the type of cooling employed.

Table 1 – Symbols and terms

Symbol	Definition	Units	Where first used
A_F	Surface area exposed to forced convection	m ²	Eq. 71
A_g	Arrangement constant for gearing	- -	Eq. 40
A_G	Surface area of the side wall	mm ²	Eq. 46
A_T	Total surface area exposed to ambient air	m ²	Eq. 66
a_w	Operating center distance	mm	Eq. 26
B	Bearing width	mm	Eq. 50
b	Face width	mm	Eq. 14
b_v	Face width of virtual cylindrical gear set	mm	Eq. C.3
b_0	Reference face width	mm	Eq. 47
C_A	Locates the pinion start of active profile	mm	Eq. 20
C_C	Equivalent radius of curvature at pitch point of contact	mm	Eq. 17
C_E	Locates the pinion end of active profile	mm	Eq. 19
C_F	Distance between base circles along the line of action	mm	Eq. 23
C_{rel}	Radius of relative curvature vertical to the contact line	mm	Eq. 30
C_{Sp}	Splash oil factor	- -	Eq. 44
C_w	Bearing width constant	- -	Eq. 50
C_0	Basic static load rating	N	Table 2
C_1	Tooth width factor	- -	Eq. 44
C_2	Immersion depth factor	- -	Eq. 44
D	Outside diameter of the element	mm	Eq. 40
D_g	Largest gear outside diameter	mm	Eq. 45
D_i	Internal gear inside diameter	mm	Eq. 23
D_s	Shaft diameter	mm	Eq. 38
D_w	Mean wormgear diameter	mm	Eq. B.1
d_i	Bearing bore diameter	mm	Eq. 11
d_m	Bearing mean diameter	mm	Eq. 10
d_o	Bearing outside diameter	mm	Eq. 11
d_{oi}	Bearing outside ring inside diameter	mm	8.2.5
d_s	Seal counterface diameter	mm	Eq. 61
d_{va1}	Tip diameter of the virtual cylindrical pinion	mm	Eq. 34
d_{va2}	Tip diameter of the virtual cylindrical gear	mm	Eq. 35
d_{vb1}	Base diameter of the virtual cylindrical pinion	mm	Eq. 34
d_{vb2}	Base diameter of the virtual cylindrical gear	mm	Eq. 35
d_{v1}	Reference diameter of the virtual cylindrical pinion	mm	Eq. 34

Symbol	Definition	Units	Where first used
d_{v2}	Reference diameter of the virtual cylindrical gear	mm	Eq. 35
d_w	Mean worm diameter	mm	Eq. B.2
E_p	Electric power consumed	W	Eq. 65
e	Base of natural logarithm	- -	Eq. 44
e_m	Electric motor efficiency	- -	Eq. 65
e_p	Oil pump efficiency	- -	Eq. 64
F	Force normal to the tooth	N	Eq. 14
F_a	Axial component of dynamic bearing load	N	Eq. 12
F_{aA}	Axial component of dynamic bearing load on bearing A	N	Figure 3
F_{aB}	Axial component of dynamic bearing load on bearing B	N	Figure 3
F_n	Tooth normal force, normal section	N	Eq. 13
F_r	Radial component of dynamic bearing load	N	Table 2
F_{rA}	Radial component of dynamic bearing load on bearing A	N	Figure 3
F_{rB}	Radial component of dynamic bearing load on bearing B	N	Figure 3
F_t	Tangential load	N	Eq. 17
F_{vmt}	Nominal tangential force of virtual cylindrical gear	N	Eq. 30
f_A	Bearing geometry factor	- -	Eq. 57
f_{bA}	Load modifying exponent	- -	Eq. 10
f_{bB}	Diameter modifying exponent	- -	Eq. 10
f_g	Gear dip factor	- -	Eq. 40
f_s	Bearing seal exponent	- -	Eq. 61
f_t	Bearing dip factor	- -	Eq. 49
f_1	Coefficient of friction for bearings	- -	Eq. 10
f_2	Cylindrical roller bearing factor	- -	Eq. 12
g_{va1}	Length of tip path of contact of the virtual cylindrical pinion	mm	Eq. 32
g_{va2}	Length of tip path of contact of the virtual cylindrical gear	mm	Eq. 33
H	Depth that the bearing rolling element dips in oil	mm	Eq. 56
H_g	Immersion depth of the gear	mm	Eq. 47
H_{max}	Largest immersion depth of the gear pair	mm	Eq. 45
H_p	Immersion depth of the pinion	mm	Eq. 47
H_s	Sliding ratio at start of approach	- -	Eq. A.3
H_t	Sliding ratio at end of recess	- -	Eq. A.3
H_v	Tooth loss factor	- -	Eq. 16
H_0	Reference immersion depth	mm	Eq. 47
h_{ae}	Addendum at outer end	mm	Eq. A.10

Symbol	Definition	Units	Where first used
h_{am}	Mean addendum at mid-face	mm	Eq. A.9
h_F	Forced convection heat transfer coefficient	W/[m ² K]	Eq. 68
h_N	Natural convection heat transfer coefficient	W/[m ² K]	Eq. 67
h_R	Radiation heat transfer coefficient	W/[m ² K]	Eq. 69
h_T	Total heat transfer coefficient	W/[m ² K]	Eq. 66
i_{rw}	Number of ball rows	- -	Eq. 51
K	Load intensity (K-factor)	N/mm ²	Eq. A.2
K_a	External axial force	N	Figure 3
K_{ball}	Rolling element constant for ball bearings	- -	Eq. 49
K_L	Roller bearing type related geometric constant	- -	Eq. 52
K_{roll}	Rolling element constant for roller bearings	- -	Eq. 50
K_{S1}	Seal and bearing type and size factor	- -	Eq. 61
K_{S2}	Seal and bearing type and size factor	- -	Eq. 61
K_Z	Bearing type related geometric constant	- -	Table 7
L	Length of element	mm	Eq. 40
L_f	Portion of gear case exposed to forced convection	mm	Eq. 75
L_h	Portion of gear case exposed to partial forced convection	mm	Eq. 75
L_H	Height of gearbox	m	Eq. 67
L_L	Length of gearbox	m	Eq. 72
L_p	Portion of gear case exposed to partial forced convection	mm	Eq. 75
L_w	Width of gearbox	m	Eq. 72
L_x	The length of flow of the air	m	Eq. 68
l_D	Roller bearing aspect ratio factor	- -	Eq. 53
l_h	Hydraulic length	mm	Eq. 45
M	Mesh mechanical advantage	- -	Eq. A.1
M_0	No-load torque of bearing	Nmm	Eq. 49
M_1	Bearing load dependent torque	Nmm	Eq. 9
M_2	Cylindrical roller bearing axial load dependent torque	Nmm	Eq. 9
M_3	Frictional torque of bearing seals	Nmm	Eq. 61
m_t	Transverse tooth module	mm	Eq. 39
n	Rotational speed	rpm	Eq. 9
n_w	Rotational speed of the worm shaft	rpm	Eq. B.1
n_1	Rotational speed of the pinion	rpm	Eq. 43
P_A	Transmitted power	kW	Figure 1
P_B	Bearing power loss	W	Eq. 7

Symbol	Definition	Units	Where first used
P_{GWi}	Power loss for each individual element	W	Eq. 40
P_L	Load dependent losses	W	Eq. 2
P_M	Mesh power loss	W	Eq. 7
P_N	Non-load dependent losses	W	Eq. 2
P_P	Total oil pump power	W	Eq. 8
P_{Pm}	Motor driven oil pump power	W	Eq. 63
P_{Ps}	Shaft driven oil pump power	W	Eq. 63
P_Q	Heat dissipated	W	Eq. 1
P_S	Oil seal power loss	W	Eq. 8
P_T	Thermal power rating	kW	Clause 4
P_V	Heat generated	W	Eq. 1
P_{VZ}	No-load power loss of a gear pair	W	Eq. 43
P_W	Gear windage and churning power losses	W	Eq. 8
P_{WB}	Bearing windage and churning power losses	W	Eq. 8
P_0	Equivalent static bearing load	N	Table 2
P_1	Bearing dynamic load	N	Eq. 10
p	Operating oil pressure	N/mm ²	Eq. 64
p_{bt}	Transverse base pitch	mm	Eq. 19
p_{vet}	Transverse base pitch of the virtual cylindrical gear	mm	Eq. 32
Q	Oil flow	l/min	Eq. 64
Ra	Arithmetic average roughness of pinion and gear	μm	Eq. 14
Ra_1	Average roughness of pinion	μm	Eq. 14
Ra_2	Average roughness of gear	μm	Eq. 14
R_e	Outer cone distance	mm	Eq. A.7
R_f	Roughness factor for gear teeth	- -	Eq. 39
R_m	Mean cone distance	mm	Eq. A.7
R_s	Resistance area	mm ²	Eq. 49
Rz	Average roughness depth	μm	Eq. 37
Rz_0	Reference roughness depth for worm gear sets	μm	Eq. 37
r	Pitch radius	mm	Eq. A.7
r_{a1}	Outside radius of pinion	mm	Eq. 22
r_{a2}	Outside radius of gear	mm	Eq. 23
r_{b1}	Base radius of pinion	mm	Eq. 22
r_{b2}	Base radius of gear	mm	Eq. 23
r_{m1}	Mean reference radius, pinion	mm	Eq. A.2

Symbol	Definition	Units	Where first used
r_{vem1}	Equivalent tip radius at mid-face width, pinion	mm	Eq. A.6
r_{vem2}	Equivalent tip radius at mid-face width, gear	mm	Eq. A.5
r_{vm1}	Equivalent mean reference radius, pinion	mm	Eq. A.6
r_{vm2}	Equivalent mean reference radius, gear	mm	Eq. A.5
r_1	Pinion pitch radius	mm	Eq. 27
T_A	Ambient air temperature	°C	Figure 1
T_H	No-load torque	Nm	Eq. 43
T_{SS}	Steady-state oil sump temperature	°C	Figure 1
ΔT_{SS}	Steady-state oil sump temperature rise	°C	Figure 1
T_1	Pinion torque	Nm	Eq. A.1
t	Included dip angle	degrees	Eq. 55
U_M	Perimeter length of the side wall	mm	Eq. 46
u	Gear ratio	- -	Eq. 18
u_v	Virtual cylindrical gear ratio	- -	Eq. 31
V_M	Drag loss factor	- -	Eq. 49
v_{air}	The average cooling fan air velocity	m/s	Eq. 68
v_g	Sliding speed	m/s	Eq. 13
v_{gm}	Mean sliding speed	m/s	Eq. 36
v_{gs}	Helical speed	m/s	Eq. C.2
$v_{\text{gy1,2}}$	Total surface speed at tooth tip	m/s	Eq. C.2
v_s	Sliding velocity at mean worm diameter	m/s	Eq. B.1
v_t	Pitch line velocity	m/s	Eq. 17
v_{tm}	Pitch line velocity at mean pitch diameter	m/s	Eq. 30
v_{tm1}	Pitch line velocity of pinion at mean pitch diameter	m/s	Eq. C.5
v_{t0}	Reference pitch line velocity	m/s	Eq. 44
v_{Σ}	Sum velocity at the pitch diameter	m/s	Eq. 14
$v_{\Sigma h}$	Sum velocity in direction of tooth depth	m/s	Eq. C.4
$v_{\Sigma m}$	Mean resultant sum velocity	m/s	Eq. C.3
$v_{\Sigma s}$	Sum velocity in direction of tooth length	m/s	Eq. C.4
W_t	Tangential tooth load on worm gear	N	Eq. B.1
X_L	Lubricant factor	- -	Eq. 14
x	is each point in the path of contact	- -	Eq. 13
Y, Y_2	Axial load factors	- -	Table 2
Y_A, Y_B	Axial load factors	- -	Figure 3
Y_W	Material factor	- -	Eq. 37

Symbol	Definition	Units	Where first used
z_{v1}	Number of teeth of the virtual cylindrical pinion	- -	Eq. 31
z_1	Number of pinion teeth	- -	Eq. 18
z_2	Number of gear teeth	- -	Eq. A.2
α_n	Normal pressure angle	degrees	Eq. A.4
α_{nw}	Normal pressure angle of worm thread at mean diameter	degrees	Eq. B.1
α_t	Transverse pressure angle	degrees	Eq. 17
α_{vet}	Transverse pressure angle of virtual cylindrical gear	degrees	Eq. 30
α_{wt}	Operating transverse pressure angle	degrees	Eq. 17
β	Helix angle	degrees	Eq. 42
β_b	Helix angle at base circle	degrees	Eq. 17
β_m	Mean spiral angle	degrees	Eq. A.1
β_{m1}	Mean spiral angle of the pinion	degrees	Eq. C.5
β_{m2}	Mean spiral angle of the gear	degrees	Eq. C.5
β_{vb}	Helix angle at base circle of the virtual cylindrical gear	degrees	Eq. 30
δ	Reference cone angle	degrees	Eq. A.7
δ_a	Face angle	degrees	Eq. A.10
ε	The emissivity of the gear drive housing surface	- -	Eq. 69
ε_{v1}	Addendum contact ratio of the virtual cylindrical pinion	- -	Eq. 31
ε_{v2}	Addendum contact ratio of the virtual cylindrical gear	- -	Eq. 31
$\varepsilon_{v\alpha}$	Transverse contact ratio of the virtual cylindrical gear	- -	Eq. 31
ε_1	Addendum contact ratio of the pinion	- -	Eq. 18
ε_2	Addendum contact ratio of the gear	- -	Eq. 18
ε_α	Transverse contact ratio	- -	Eq. 18
η	Unit efficiency	%	Eq. 5
η_{oil}	Dynamic viscosity of oil at operating temperature	mPa s	Eq. 14
κ	Viscosity ratio as defined in ISO 281 [2]	- -	Table 4
λ	Lead angle at mean worm diameter	degrees	Eq. 36
μ	Coefficient of friction	- -	Eq. 13
μ_{mz}	Average coefficient of friction of the gear mesh	- -	Eq. 14
μ_z	Coefficient of friction of a worm gear set	- -	Eq. 36
μ_{z0}	Basic value of the coefficient of friction of a worm gear set	- -	Eq. 37
ν_{oil}	Kinematic viscosity of the oil at operating temperature	cSt	Eq. 40
ρ	Radius of curvature	mm	Eq. 14

4 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 power rating has to be considered.

The primary thermal power 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, operating oil sump temperatures need to be evaluated and limited.

Thermal power 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 the application requirements, selection can be made for oil sump temperatures above or below 95°C, see Clause 9.

Additional criteria that need to be applied in establishing the thermal power rating for a specific gear drive with a given type of cooling are related to the operating conditions of the drive. The thermal power 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 less than or equal to 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 9.

5 Service conditions

5.1 Intermittent service

For intermittent service, the transmitted power may exceed the manufacturer's thermal power rating, provided the oil sump temperature does not exceed 95°C. See Clause 9.

5.2 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;
- build-up of material that may cover the gear drive and reduce heat dissipation;
- high ambient temperature, such as boiler or turbine rooms, or in conjunction with hot processing equipment;
- high altitudes;
- presence of solar energy or radiant heat.

5.3 Favorable conditions

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

5.4 Auxiliary cooling

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

- fan cooling, in which case the fan needs to maintain the fan cooled thermal power rating;
- heat exchanger, which when used needs to be capable of absorbing generated heat that cannot be dissipated by the gear drive by convection and radiation.

6 Methods for determining the thermal power rating

Thermal power 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 power rating of the gear drive. See Clause 7.

When Method B is used, the thermal power rating of a gear drive can be calculated using the heat balance equation, which equates heat generated with heat dissipated. See Clause 8 (the means of calculating heat generation is discussed in 8.2 to 8.2.6, and for heat dissipation, in 8.3).

7 Method A – Test

Testing a specific gear drive at its design operating conditions is the most reliable means of establishing the thermal power rating. Thermal testing involves measuring the steady-state oil sump temperature of the gear drive operating at its rated speed at no-load and at least one or two increments of load. While no-load testing cannot yield a thermal power rating directly, 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.

- ambient air temperature and velocity need to be stabilized and measured for the duration of the test;
- 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 assumed when the change in oil sump temperature is less than or equal to 1°C/h;
- both directions of rotation need to be tested since different oil sump temperatures can result;
- if at all possible, the measured steady state oil sump temperature for at least one test point should be close to the rating temperature, 95°C;
- measured oil sump temperature should represent the bulk oil temperature. A survey of various sump measurement locations can help define the proper temperature as temperatures can vary;
- housing surface temperature can vary substantially from the oil sump temperature and should not be used in lieu of sump temperature measurement;
- additional data taken during thermal testing might include housing surface temperatures or air velocity distribution over the housing for heat transfer analysis;
- while a thermal rating cannot be established with no load testing, no load testing can be of value in heat transfer studies and definition of non-load dependent losses.

Using thermal test results to determine the thermal power rating by test is illustrated in Figure 1. To minimize the effect of small variations ($\pm 3^\circ\text{C}$) in the ambient air temperature, the measured steady-state oil sump temperature rise (sump temperature – ambient air temperature) is plotted versus the transmitted power. Figure 1 illustrates the thermal power rating determined by test to be at the intersection of the maximum allowable oil sump temperature rise and the curve fit to the test data. Thermal testing can also be used to verify a calculated thermal power rating, or to improve the correlation and empirical factors used in the thermal power rating calculation.

The procedure to use Figure 1 is:

1. Plot test data (ΔT_{SS} versus P_A).
2. Establish curve fit to test data.
3. Establish maximum allowable ΔT_{SS} based on maximum allowable oil sump temperature and average ambient air temperature from test data.
4. The intersection of the maximum allowable ΔT_{SS} and the test data curve fit identifies the thermal power rating, P_T .

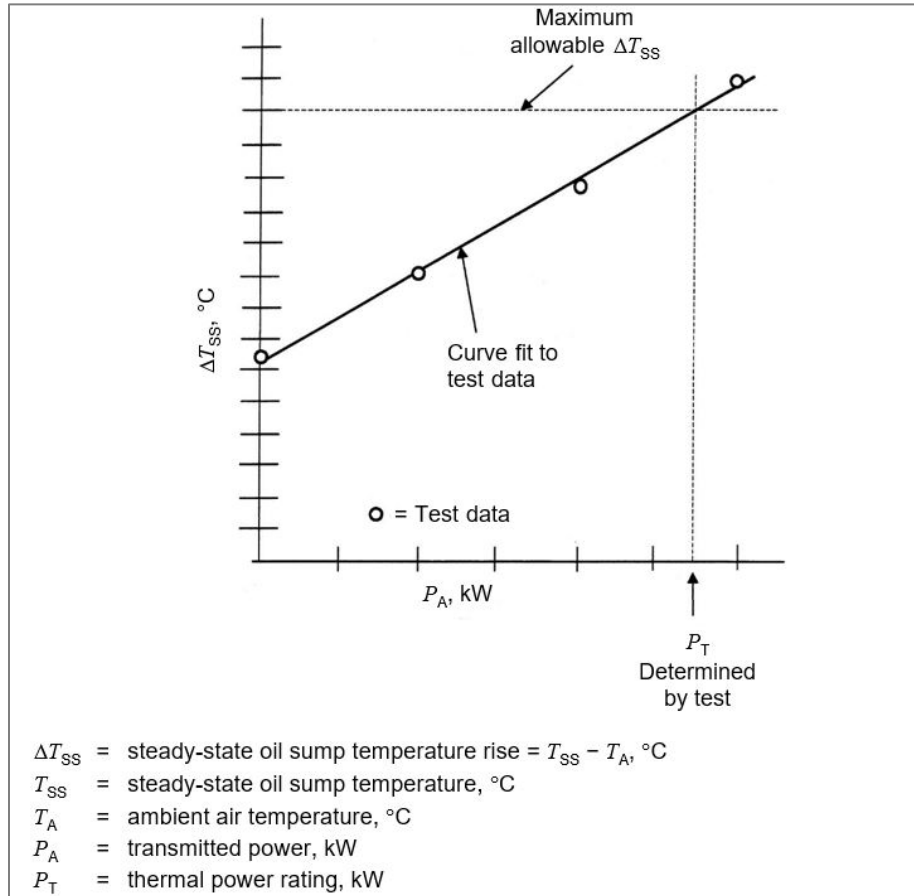


Figure 1 – Determination of thermal power rating by test

8 Method B – Calculations for determining the thermal power rating, P_T

Method B provides an analytical approach to calculate the thermal power rating of a gear drive using the heat balance equation; however, using this method may result in thermal ratings that deviate from actual thermal ratings derived through testing.

The following thermal model has been established using empirical factors. It is based on the experience of several gear manufacturers. Values of some variables such as arrangement constant, heat transfer coefficient and coefficient of friction may not adequately address all enclosed drive configurations and operating conditions. Configurations or conditions not explicitly defined below may necessitate modifications of the particular variables. Changing any variable requires care and testing to ensure that the principles of the heat balance formulation are not violated.

8.1 Basis

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

$$P_Q = P_V \quad (1)$$

where

P_Q is heat dissipated, W;

P_V is heat generated, W.

When this is satisfied under the conditions of Clause 4, P_A is defined as P_T .

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

$$P_V = P_L + P_N \quad (2)$$

where

P_L is load dependent losses, W;

P_N is non-load dependent losses, W.

P_L is a function of the transmitted power, P_A .

$$P_L = f(P_A) \quad (3)$$

where

P_A is transmitted power, kW.

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

$$P_Q - P_N - P_L = 0 \quad (4)$$

where

P_Q is heat dissipated, W.

P_L and P_Q depend on the applied power, P_A . As a result, the calculation of thermal rating does not lend itself to a simple closed form solution. Therefore, to determine the thermal power rating, P_T , vary P_A until Equation 4 is satisfied. This can be done by recalculating the load dependent losses, P_L , at different transmitted powers, P_A . With appropriate heat dissipation and heat generation terms included in Equation 4, values of P_A are tried until a heat balance is obtained (iterative solution) and the solution value of P_A becomes the thermal power rating P_T . A graphical representation of this procedure is illustrated in Figure 2 where calculated P_V and P_Q are plotted versus oil sump temperature rise. Calculated P_V curves are illustrated for various iterative values of P_A . The thermal power rating is defined by the thermal equilibrium point where $P_V = P_Q$ at the maximum allowable oil sump temperature rise, with the corresponding value of P_A equal to P_T .

The procedure to use Figure 2 is:

1. Establish maximum allowable ΔT_{SS} based on maximum allowable oil sump temperature and rating ambient air temperature.
2. Establish math model for calculation of P_V and P_Q versus ΔT_{SS} .
3. By iteration on the values of P_A used in the calculation of P_V , find the value of P_A that results in $P_V = P_Q$ at the maximum allowable ΔT_{SS} (thermal equilibrium point). The value of P_A is then the thermal rating P_T .

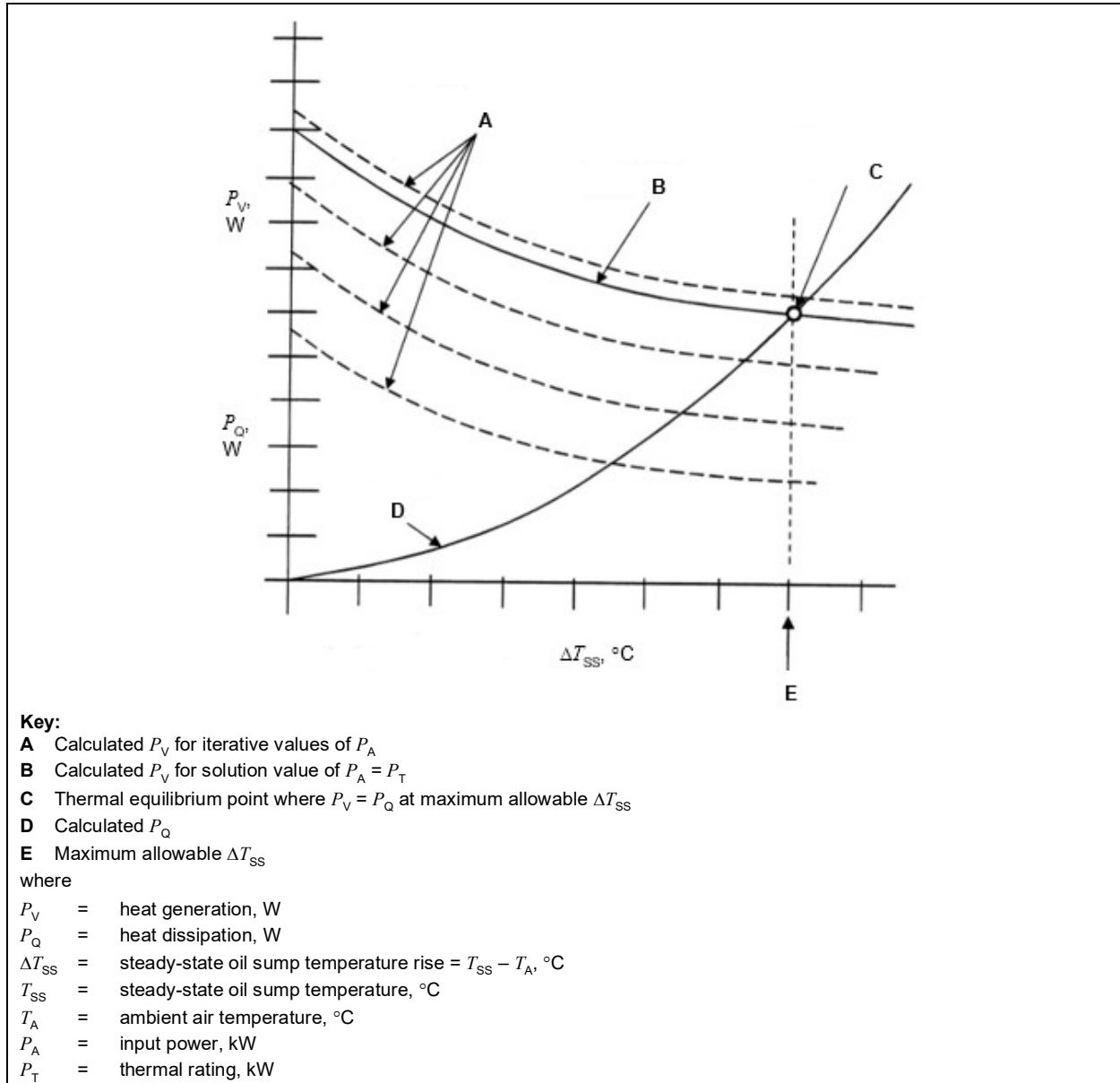


Figure 2 – Graphical representation of calculation of thermal power rating

If P_Q is less than or equal to P_N at no-load, the gear drive does not have any thermal capacity. The design should be changed to increase P_Q , or auxiliary cooling methods are used.

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

$$\eta = 100 - \frac{P_L + P_N}{P_A} \times 100 \quad (5)$$

where

η is unit efficiency, %.

The thermal power rating of the gear drive is as follows:

$$P_T = \frac{P_Q}{1000 \left(1 - \frac{\eta}{100} \right)} \quad (6)$$

where

P_T is thermal power rating, kW.

8.2 Heat generation

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

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

$$P_L = \sum P_B + \sum P_M \quad (7)$$

where

P_B is bearing power loss, W;

P_L is load dependent losses, W;

P_M is mesh power loss, W.

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

$$P_N = \sum P_S + \sum P_W + \sum P_{WB} + \sum P_P \quad (8)$$

where

P_N is non-load dependent losses, W;

P_P is total oil pump power, W;

P_S is oil seal power loss, W;

P_W is gear windage and churning power losses, W;

P_{WB} is bearing windage and churning power losses, W.

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

8.2.1 Bearing power loss, P_B

8.2.1.1 Radially loaded bearings

The SKF Catalog [3] provides one possible method of calculating the load dependent losses for bearings. Equation 10 is a simplified equation that gives an approximation for the torque on each bearing as a function of applied load. The coefficient of friction, f_1 , and the equations for calculating the load, P_1 , are given in Table 2. Exponents, f_{bA} and f_{bB} , which modify P_1 and d_m , are given in Table 3. Further information on the power loss in bearings can be found in bibliographical references Eschmann [4] and Harris [5] or by contacting appropriate bearing manufacturers. The values presented in Table 2, Table 3, and Table 4 are based on a specific bearing manufacturer and may vary depending on the manufacturer.

Table 2 – Factors for calculating M_1

Bearing type	f_1	P_1 ¹⁾
Deep groove ball bearings	$(0.0006 \dots 0.0009) (P_0 / C_0)^{0.55}$ ²⁾	$3 F_a - 0.1 F_r$
Self-aligning ball bearings	$0.0003 (P_0 / C_0)^{0.4}$	$1.4 Y_2 F_a - 0.1 F_r$
Angular contact ball bearings: single-row double-row, paired single-row	$0.001 (P_0 / C_0)^{0.33}$ $0.001 (P_0 / C_0)^{0.33}$	$F_a - 0.1 F_r$ $1.4 F_a - 0.1 F_r$
Four-point contact ball bearings	$0.001 (P_0 / C_0)^{0.33}$	$1.5 F_a + 3.6 F_r$
Cylindrical roller bearings, with cage: series 10 series 2 series 3 series 4, 22, 23	0.0002 0.0003 0.00035 0.0004	F_r ³⁾ F_r ³⁾ F_r ³⁾ F_r ³⁾
Cylindrical roller bearings, full complement	0.00055	F_r ³⁾
Needle roller bearings	0.002	F_r
Spherical roller bearings: series 213 series 222 series 223 series 230, 241 series 231 series 232 series 239 series 240	0.00022 0.00015 0.00065 0.001 0.00035 0.00045 0.00025 0.0008	$\text{If } F_r / F_a < Y_2: 1.35 Y_2 F_a$ $\text{If } F_r / F_a \geq Y_2: F_r [1 + 0.35 (Y_2 F_a / F_r)^3]$ (valid for all series)
Taper roller bearings: single-row paired single-row	0.0004 0.0004	$2 Y F_a$ ⁴⁾ $1.2 Y_2 F_a$ ⁴⁾
Thrust ball bearings	$0.0008 (F_a / C_0)^{0.33}$	F_a
Cylindrical roller thrust bearings, needle roller thrust bearings	0.0015	F_a
Spherical roller thrust bearings: series 292E series 292 series 293E series 293 series 294E series 294	0.00023 0.0003 0.0003 0.0004 0.00033 0.0005	$F_a (F_r \text{ max} \leq 0.55 F_a)$ (valid for all series)
Symbols: f_1 = coefficient of friction for bearings P_1 = bearing dynamic load, N P_0 = equivalent static bearing load, N (see manufacturer's bearing tables); C_0 = basic static load rating, N (see manufacturer's bearing tables); F_a = axial component of dynamic bearing load, N; F_r = radial component of dynamic bearing load, N; Y_1, Y_2 = axial load factors (see manufacturer's bearing tables).		
1) If $P_1 < F_r$, then $P_1 = F_r$ should be used. 2) Small values are for light series bearings; large values for heavy series bearings. 3) For bearings subjected to additional axial loads, refer to 8.2.1.2. 4) Refer to 8.2.1.3.		
NOTE: These are examples of products available commercially. This information is given for the convenience of users of this information sheet and does not constitute an endorsement by AGMA of these products.		

$$P_B = \frac{(M_1 + M_2)n}{9549} \quad (9)$$

where

P_B is bearing power loss, W;

n is rotational speed, rpm;

M_1 is bearing load dependent torque, Nmm;

M_2 is cylindrical roller bearing axial load dependent torque, Nmm, see Equation 12.

$$M_1 = f_1 (P_1)^{f_{bA}} (d_m)^{f_{bB}} \quad (10)$$

where

f_{bA} is load modifying exponent;

f_{bB} is diameter modifying exponent;

f_1 is coefficient of friction for bearings, see Table 2;

P_1 is bearing dynamic load, N, see Table 2;

d_m is bearing mean diameter, mm, see Equation 11.

$$d_m = \frac{(d_i + d_o)}{2} \quad (11)$$

where

d_i is bearing bore diameter, mm;

d_o is bearing outside diameter, mm.

Table 3 – Exponents for calculation of M_1

Bearing type	Exponent	
	f_{bA}	f_{bB}
All (except spherical roller bearings)	1	1
Spherical roller bearings:		
series 213	1.35	0.2
series 222	1.35	0.3
series 223	1.35	0.1
series 230	1.5	-0.3
series 231, 232, 239	1.5	-0.1
series 240, 241	1.5	-0.2

8.2.1.2 Axially loaded cylindrical roller bearings

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

$$M_2 = f_2 F_a d_m \quad (12)$$

where

d_m is bearing mean diameter, mm;

f_2 is cylindrical roller bearing factor, see Table 4;

F_a is axial component of dynamic bearing load, N;

M_2 is cylindrical roller bearing axial load dependent torque, Nmm.

The ratio of axial load to the simultaneously acting radial bearing load (F_a/F_r) cannot exceed 0.5 for EC design and single-row full complement bearings, 0.4 for the other bearings with cage, or 0.25 for double-row full complement bearings.

Table 4 – Factor f_2 for cylindrical roller bearings

Bearing	$f_2^{1)}$	
	Lubrication	
	grease	oil
Bearings with cage:		
EC design	0.003	0.002
other bearings	0.009	0.006
Full complement bearings:		
single-row	0.006	0.003
double-row	0.015	0.009
NOTE: ¹⁾ Values quoted for f_2 assume that the viscosity ratio, κ as defined in ISO 281 [2], is greater than or equal to 1.5.		

8.2.1.3 Tapered roller bearings

Because the raceways are at an angle to the bearing axis, when taper roller bearings are subjected to a radial load, an axial force is induced within the bearing. This is considered when calculating the equivalent dynamic bearing load. All the requisite equations for the various bearing arrangements and load cases are given in Figure 3. These are only valid if the bearings are adjusted against each other to give zero clearance in operation but are without preload. In the bearing arrangements shown, bearing A is always subjected to radial load F_{rA} , and bearing B to radial load F_{rB} . F_{rA} and F_{rB} are always considered as positive, even in cases when both act in the direction opposite to that shown in the figures. Radial loads act at the pressure centers of the bearings (see dimension a in the bearing manufacturer's tables). Additionally, an external axial force, K_a , acts on the shaft (or housing). Cases 1c and 2c in Figure 3 are also valid for $K_a = 0$. Values for the axial load factor, Y , for bearings A and B can be found in bearing manufacturers catalogs.