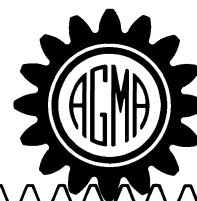


AMERICAN GEAR MANUFACTURERS ASSOCIATION

*Effect of Lubrication on Gear Surface
Distress*

AGMA 925-A03



AGMA INFORMATION SHEET

(This Information Sheet is NOT an AGMA Standard)

American
Gear
Manufacturers
Association

Effect of Lubrication on Gear Surface Distress

AGMA 925-A03

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Approved March 13, 2003

ABSTRACT

AGMA 925-A03 is an enhancement of annex A of ANSI/AGMA 2101-C95. Various methods of gear surface distress are included, such as scuffing and wear, and in addition, micro and macropitting. Lubricant viscometric information has been added, as has Dudley's regimes of lubrication theory. A flow chart is included in annex A, Gaussian theory in annex B, a summary of lubricant test rigs in annex C, and an example calculation in annex D.

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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 Information Sheet 925-A03, *Effect of Lubrication on Gear Surface Distress*.]

The purpose of this information sheet is to provide the user with information pertinent to the lubrication of industrial metal gears for power transmission applications. It is intended that this document serve as a general guideline and source of information about conventional lubricants, their properties, and their general tribological behavior in gear contacts. This information sheet was developed to supplement ANSI/AGMA Standards 2101-C95 and 2001-C95. It has been introduced as an aid to the gear manufacturing and user community. Accumulation of feedback data will serve to enhance future developments and improved methods to evaluate lubricant related wear risks.

It was clear from the work initiated on the revision of AGMA Standards 2001-C95 and 2101-C95 (metric version) that supporting information regarding lubricant properties and general tribological knowledge of contacting surfaces would aid in the understanding of these standards. The information would also provide the user with more tools to help make a more informed decision about the performance of a geared system. This information sheet provides sufficient information about the key lubricant parameters to enable the user to generate reasonable estimates about scuffing and wear based on the collective knowledge of theory available for these modes at this time.

In 1937 Harmon Blok published his theory about the relationship between contact temperature and scuffing. This went largely unnoticed in the U.S. until the early 1950's when Bruce Kelley showed that Blok's method and theories correlated well with experimental data he had generated on scuffing of gear teeth. The Blok flash temperature theory began to receive serious consideration as a predictor of scuffing in gears. The methodology and theories continued to evolve through the 1950's with notable contributions from Dudley, Kelley and Benedict in the areas of application rating factors, surface roughness effects and coefficient of friction. The 1960's saw the evolution of gear calculations and understanding continue with computer analysis and factors addressing load sharing and tip relief issues. The AGMA Aerospace Committee began using all the available information to produce high quality products and help meet its long-term goal of manned space flight. R. Errichello introduced the SCORING+ computer program in 1985, which included all of the advancements made by Blok, Kelley, Dudley and the Aerospace Committee to that time. It became the basis for annex A of ANSI/AGMA 2101-C95 and 2001-C95 which helped predict the risk of scuffing and wear. In the 1990s, this annex formed the basis for AGMA's contribution to ISO 13989-1.

Just as many others took the original Blok theories and expanded them, the Tribology Subcommittee of the Helical Gear Rating Committee has attempted to expand the original annex A of ANSI/AGMA 2001-C95 and 2101-C95. Specifically, the subcommittee targeted the effect lubrication may have on gear surface distress. As discussions evolved, it became clear that this should be a stand alone document which will hopefully serve many other gear types. This should be considered a work in progress as more is learned about the theories and understanding of the various parameters and how they affect the life of the gear. Some of these principles are also mentioned in ISO/TR 13989-1.

AGMA 925-A03 was approved by the AGMA Technical Division Executive Committee on March 13, 2003.

Suggestions for improvement of this document will be welcome. They should be sent to the American Gear Manufacturers Association, 500 Montgomery Street, Suite 350, Alexandria, Virginia 22314.

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Effect of Lubrication on Gear Surface Distress

1 Scope

This information sheet is designed to provide currently available tribological information pertaining to oil lubrication of industrial gears for power transmission applications. It is intended to serve as a general guideline and source of information about gear oils, their properties, and their general tribological behavior in gear contacts. Manufacturers and end-users are encouraged, however, to work with their lubricant suppliers to address specific concerns or special issues that may not be covered here (such as greases).

The equations provided herein allow the user to calculate specific oil film thickness and instantaneous contact (flash) temperature for gears in service. These two parameters are considered critical in defining areas of operation that may lead to unwanted surface distress. Surface distress may be scuffing (adhesive wear), fatigue (micropitting and macropitting), or excessive abrasive wear (scoring). Each of these forms of surface distress may be influenced by the lubricant; the calculations are offered to help assess the potential risk involved with a given lubricant choice. Flow charts are included as aids to using the equations.

This information sheet is a supplement to ANSI/AGMA 2101-C95 and ANSI/AGMA 2001-C95. It has been introduced as an aid to the gear manufacturing and user community. Accumulation of feedback data will serve to enhance future developments and improved methods to evaluate lubricant related surface distress.

It was clear from the work on the revision of standard ANSI/AGMA 2001-C95 (ANSI/AGMA 2101-C95, metric version) that supporting information regarding lubricant properties and general tribological understanding of contacting surfaces would aid in understanding of the standard and provide the user with more tools to make an informed decision about the performance of a geared system. One of the key parameters is the estimated film thickness. This is not a trivial calculation, but one that has significant impact on overall performance of the gear pair. It is considered in performance issues such as scuffing, wear, and surface fatigue. This information sheet provides sufficient information about key lubricant parameters to enable the user to generate reasonable estimates about surface distress based on the collective knowledge available.

Blok [1] published his contact temperature equation in 1937. It went relatively unnoticed in the U.S. until Kelley [2] showed that Blok's method gave good correlation with Kelley's experimental data. Blok's equation requires an accurate coefficient of friction. Kelley found it necessary to couple the coefficient of friction to surface roughness of the gear teeth. Kelley recognized the importance of load sharing by multiple pairs of teeth and gear tooth tip relief, but he did not offer equations to account for those variables.

Dudley [3] modified Kelley's equation by adding derating factors for application, misalignment and dynamics. He emphasized the need for research on effects of tip relief, and recommended applying Blok's method to helical gears.

In 1958, Kelley [4] changed his surface roughness term slightly.

Benedict and Kelley [5] published their equation for variable coefficient of friction derived from disc tests.

The AGMA Aerospace Committee began investigating scuffing in 1960, and Lemanski [6] published results of a computer analysis that contains data for 90 spur and helical gearsets, and formed the terms for AGMA 217.01 [7], which was published in 1965. It used Dudley's modified Blok/Kelley equation and included factors accounting for load sharing and tip relief.

The SCORING+ computer program [8] was released in 1985. It incorporated all advancements made by Blok, Kelley, Dudley and AGMA 217.01. In addition, it added several improvements including:

- Helical gears were analyzed by resolving the load in the normal plane and distributing the normal load over the minimum length of the contact lines. The semi-width of the Hertzian contact band was calculated based on the normal relative radius of curvature;
- Derating factors for application, misalignment and dynamics were explicit input data;
- Options for coefficient of friction were part of input data, including a constant 0.06 (as prescribed by Kelley and AGMA 217.01), a constant under user control, and a variable coefficient based on the Benedict and Kelley equation.

SCORING+ and AGMA 217.01 both use the same value for the thermal contact coefficient of $B_M = 16.5 \text{ N/[mm} \cdot \text{s}^{0.5} \cdot \text{K}]$, and they calculate the same contact temperature for spur gears if all derating factors are set to unity.

Annex A of ANSI/AGMA 2101-C95 and ANSI/AGMA 2001-C95 was based on SCORING+ and included methods for predicting risk of scuffing based on contact temperature and risk of wear based on specific film thickness.

This information sheet expands the information in annex A of ANSI/AGMA 2101-C95 and ANSI/AGMA 2001-C95 to include many aspects of gear tribology.

2 References

The following standards contain provisions which are referenced in the text of this information sheet. At the time of publication, the editions indicated were valid. All standards are subject to revision, and parties to agreements based on this document are encouraged to investigate the possibility of applying the most recent editions of the standards indicated.

ANSI/AGMA 2001-C95, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth*

ANSI/AGMA 2101-C95, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth (Metric Edition)*

ANSI/AGMA 1010-E95, *Appearance of Gear Teeth - Terminology of Wear and Failure*

ISO 10825:1995, *Gears - Wear and Damage to Gear Teeth - Terminology*

3 Symbols and units

The symbols used in this document are shown in table 1.

NOTE: The symbols and definitions used in this document may differ from other AGMA standards.

Table 1 - Symbols and units

Symbol	Description	Units	Where first used
A	Dimensionless constant	--	Eq 61
a_w	Operating center distance	mm	Eq 4
B	Dimensionless constant	--	Eq 61
B_M	Thermal contact coefficient	$\text{N/[mm s}^{0.5}\text{K}]$	6.2.3
B_{M1}, B_{M2}	Thermal contact coefficient (pinion, gear)	$\text{N/[mm s}^{0.5}\text{K}]$	Eq 84
b	Face width	mm	Eq 23
$b_{H(i)}$	Semi-width of Hertzian contact band	mm	Eq 57
$C_A \dots C_F$	Distances along line of action	mm	4.1.2
$C_{R_{avg_x}}$	Surface roughness constant	--	Eq 85
c	Parameter for calculating η_o	--	Eq 69
c_{M1}, c_{M2}	Specific heat per unit mass (pinion, gear)	$\text{J/[kg K}]$	Eq 89, 90
D_i	Internal gear inside diameter	mm	4.1.2
d	Parameter for calculating η_o	--	Eq 69

(continued)

Table 1 (continued)

Symbol	Description	Units	Where first used
E_1, E_2	Modulus of elasticity (pinion, gear)	N/mm ²	Eq 58
E_r	Reduced modulus of elasticity	N/mm ²	Eq 57
F_t	Actual tangential load	N	Eq 42
$(F_t)_{nom}$	Nominal tangential load	N	Eq 40
F_{wn}	Normal operating load	N	Eq 43
G	Materials parameter	--	Eq 65
g	Parameter for calculating η_o	--	Eq 69
$H_{c(i)}$	Dimensionless central film thickness	--	Eq 65
h	Thickness of element measured perpendicular to flow	m	Eq 59
$h_{c(i)}$	Central film thickness	μm	Eq 75
h_{min}	Minimum film thickness	μm	Eq 102
K	Flash temperature constant	--	Eq 84
K_D	Combined derating factor	--	Eq 41
K_m	Load distribution factor	--	Eq 41
K_o	Overload factor	--	Eq 41
K_v	Dynamic factor	--	Eq 41
k	Parameter for calculating α	--	Eq 74
k_{sump}	Parameter for calculating θ_M	--	Eq 91
L_x	Filter cutoff of wavelength x	mm	Eq 77
L_{min}	Minimum contact length	mm	Eq 25
m_n	Normal module	mm	Eq 2
n_1	Pinion speed	rpm	Eq 33
N	Number of load cycles	cycles	Fig 14
n_a	Fractional (non-integer) part of ε_β	--	Eq 25
n_r	Fractional (non-integer) part of ε_α	--	Eq 25
P	Transmitted power	kW	Eq 40
$P_{(x)}$	Probability of survival	--	8.2.2
p	Pressure	N/mm ²	Eq 64
p_{bn}	Normal base pitch	mm	Eq 10
p_{bt}	Transverse base pitch	mm	Eq 9
p_x	Axial pitch	mm	Eq 11
Q	Tail area of the normal probability function	--	Eq B.2
$Q_{(x)}$	Probability of failure	--	8.2.2
R_{avg_x}	Average of the average values of pinion and gear roughness	μm	Eq 87
Ra_{1x}, Ra_{2x}	Average surface roughness (pinion, gear) at L_x	μm	Eq 78
Rq_x	Root mean square roughness at L_x	μm	Eq 79
$Rq_x \text{ avg}$	Arithmetic average of Rq_{1x} and Rq_{2x} at L_x	μm	Eq 99
Rq_{1x}, Rq_{2x}	Root mean square roughness at L_x (pinion, gear)	μm	Eq 99
r_1, r_2	Standard pitch radius (pinion, gear)	mm	Eq 2, 3
r_{a1}, r_{a2}	Outside radius (pinion, gear)	mm	Eq 19, 16
r_{b1}, r_{b2}	Base radius (pinion, gear)	mm	Eq 6, 7
r_{w1}	Operating pitch radius of pinion	mm	Eq 4
S_f	Contact time	ms (sec $\times 10^{-3}$)	Eq 97
s	Parameter for calculating α	--	Eq 74

(continued)

Table 1 (continued)

Symbol	Description	Units	Where first used
T	Absolute temperature	K	Eq 61
$U_{(i)}$	Speed parameter	--	Eq 65
u	Gear ratio (always ≥ 1.0)	--	Eq 1
v	Velocity	m/s	Eq 59
$v_{e(i)}$	Entraining velocity	m/s	Eq 39
$v_{r1(i)}, v_{r2(i)}$	Rolling (tangential) velocity (pinion, gear)	m/s	Eq 36, 37
$v_{s(i)}$	Sliding velocity	m/s	Eq 38
v_t	Operating pitch line velocity	m/s	Eq 35
$W_{(i)}$	Load parameter	--	Eq 65
w_n	Normal unit load	N/mm	Eq 44
X_W	Welding factor	--	Eq 96
$X_{\Gamma(i)}$	Load sharing factor	--	4.3
Z	Active length of line of action	mm	Eq 21
Z_N	Stress cycle factor	--	7.5
Z_Q	Normal probability density function	--	Eq B.3
z_1	Number teeth in pinion	--	Eq 1
z_2	Number teeth in gear (positive)	--	Eq 1
α	Pressure-viscosity coefficient	mm ² /N	Eq 64
α_n	Normal generating pressure angle	degrees	Eq 5
α_t	Transverse generating pressure angle	degrees	Eq 5
α_{wn}	Normal operating pressure angle	degrees	Eq 14
α_{wt}	Transverse operating pressure angle	degrees	Eq 8
β	Helix angle	degrees	Eq 2
β_b	Base helix angle	degrees	Eq 12
β_w	Operating helix angle	degrees	Eq 13
$\xi_{(i)}$	Pinion roll angle at point i along the line of action	radians	Eq 29
$\xi_A \dots \xi_E$	Pinion roll angle at points A ... E	radians	Eq 28
ε_α	Transverse contact ratio	--	Eq 22
ε_β	Axial contact ratio	--	Eq 23
η	Dynamic viscosity	mPa · s	Eq 59
η_{atm}	Viscosity at atmospheric pressure	mPa · s	Eq 64
η_P	Viscosity at pressure P	mPa · s	Eq 64
η_M	Dynamic viscosity at gear tooth temperature θ_M	mPa · s	Eq 67
η_1, η_2	Dynamic viscosity at temperature θ_1, θ_2	mPa · s	Eq 70
η_{40}, η_{100}	Dynamic viscosity at 40°C, 100°C	mPa · s	Eq 71
$\theta_{B(i)}$	Contact temperature	°C	Eq 92
$\theta_{B \max}$	Maximum contact temperature	°C	Eq 93
$\theta_{fl(i)}$	Flash temperature	°C	Eq 84
$\theta_{fl \max}$	Maximum flash temperature	°C	Eq 91
$\theta_{fl \max, \text{test}}$	Maximum flash temperature of test gears	°C	Eq 96
θ_M	Tooth temperature	°C	Eq 69
$\theta_{M, \text{test}}$	Tooth temperature of test gears	°C	Eq 96

(continued)

Table 1 (concluded)

Symbol	Description	Units	Where first used
θ_{oil}	Oil inlet or sump temperature	°C	Eq 91
θ_S	Mean scuffing temperature	°C	Eq 94
θ_{Smet}	Method of calculating scuffing temperature, θ_S	--	Annex A
θ_1, θ_2	Temperature at which η_1, η_2 was measured	°C	Eq 70
λ_{min}	Specific film thickness	--	Eq 104
$\lambda_{2b_{H(i)}}$	Specific film thickness at point i with a filter cutoff wavelength of $2b_H$	--	Eq 76
$\lambda_{M1}, \lambda_{M2}$	Heat conductivity (pinion, gear)	N/[s K]	Eq 89, 90
$\lambda_{W\&H}$	Wellauer and Holloway specific film thickness	--	Eq 102
μ_y	Mean value of random variable y	--	6.5.5
$\mu_{m(i)}$	Mean coefficient of friction	--	Eq 84
μ_{met}	Method for approximating mean coefficient of friction	--	Annex A
$\mu_m \text{ const}$	Mean coefficient of friction, constant	--	Eq 85
$\mu_{\lambda \text{ min}}$	Mean minimum specific film thickness	μm	Eq 109
ν	Kinematic viscosity	mm^2/s	Eq 60
ν_1, ν_2	Poisson's ratio (pinion, gear)	--	Eq 58
ν_{40}, ν_{100}	Kinematic viscosity at 40°C, 100°C	mm^2/s	Eq 62
ρ	Density	kg/m^3	Eq 60
ρ_{M1}, ρ_{M2}	Density (pinion, gear)	kg/m^3	Eq 89, 90
$\rho_{1(i)}, \rho_{2(i)}$	Transverse radius of curvature (pinion, gear)	mm	4.1.5
$\rho_{n(i)}$	Normal relative radius of curvature	mm	Eq 32
$\rho_{r(i)}$	Transverse relative radius of curvature	mm	Eq 31
σ_x	Composite surface roughness for filter cutoff wavelength, L_x	μm	Eq 77
$\sigma_{\lambda \text{ min}}$	Standard deviation of the minimum specific film thickness	μm	Eq 109
$\sigma_{2b_{H(i)}}$	Composite surface roughness adjusted for a cutoff wavelength equal to the Hertzian contact width	μm	Eq 76
τ	Shear stress	N/mm^2	Eq 59
ω_1, ω_2	Angular velocity (pinion, gear)	rad/s	Eq 33, 34

4 Gear information

4.1 Gear geometry

This clause gives equations for gear geometry used to determine flash temperature and elastohydrodynamic (EHL) film thickness. The following equations apply to both spur and helical gears; spur gearing is a particular case with zero helix angle. Where double signs are used (e.g., $\pm$), the upper sign applies to external gears and the lower sign to internal gears.

4.1.1 Basic gear geometry

Gear ratio

$$u = \frac{z_2}{z_1} \quad (1)$$

Standard pitch radii

$$r_1 = \frac{z_1 m_n}{2 \cos \beta} \quad (2)$$

$$r_2 = r_1 u \quad (3)$$

Operating pitch radius of pinion

$$r_{w1} = \frac{a_w}{u \pm 1} \quad (4)$$

Transverse generating pressure angle

$$\alpha_t = \arctan \left(\frac{\tan \alpha_n}{\cos \beta} \right) \quad (5)$$

Base radii

$$r_{b1} = r_1 \cos \alpha_t \quad (6)$$

$$r_{b2} = r_{b1} u \quad (7)$$

Transverse operating pressure angle

$$\alpha_{wt} = \arccos\left(\frac{r_{b1}}{r_{w1}}\right) \quad (8)$$

Transverse base pitch

$$p_{bt} = \frac{2 \pi r_{b1}}{z_1} \quad (9)$$

Normal base pitch

$$p_{bn} = \pi m_n \cos \alpha_n \quad (10)$$

Axial pitch

$$p_x = \frac{\pi m_n}{\sin \beta} \quad (11)$$

Base helix angle

$$\beta_b = \arccos\left(\frac{p_{bn}}{p_{bt}}\right) \quad (12)$$

Operating helix angle

$$\beta_w = \arctan\left(\frac{\tan \beta_b}{\cos \alpha_{wt}}\right) \quad (13)$$

Normal operating pressure angle

$$\alpha_{wn} = \arcsin(\cos \beta_b \sin \alpha_{wt}) \quad (14)$$

4.1.2 Distances along the line of action

Figure 1 is the line of action shown in a transverse plane. Distances C_j are measured from the interference point of the pinion along the line of action. Distance C_A locates the pinion start of active profile (SAP) and distance C_E locates the pinion end of active profile (EAP). The lowest and highest point of single-tooth-pair contact (LPSTC and HPSTC) are located by distances C_B and C_D , respectively. Distance C_C locates the operating pitch point. C_F is the distance between base circles along the line of action.

$$C_F = a_w \sin \alpha_{wt} \quad (15)$$

$$C_A = \pm \left[C_F - (r_{a2}^2 - r_{b2}^2)^{0.5} \right] \quad (16)$$

NOTE: For internal gears $r_{a2} = \frac{D_i}{2}$.

$$C_C = \frac{C_F}{u \pm 1} \quad (17)$$

$$C_D = C_A + p_{bt} \quad (18)$$

$$C_E = (r_{a1}^2 - r_{b1}^2)^{0.5} \quad (19)$$

$$C_B = C_E - p_{bt} \quad (20)$$

$$Z = C_E - C_A \quad (21)$$

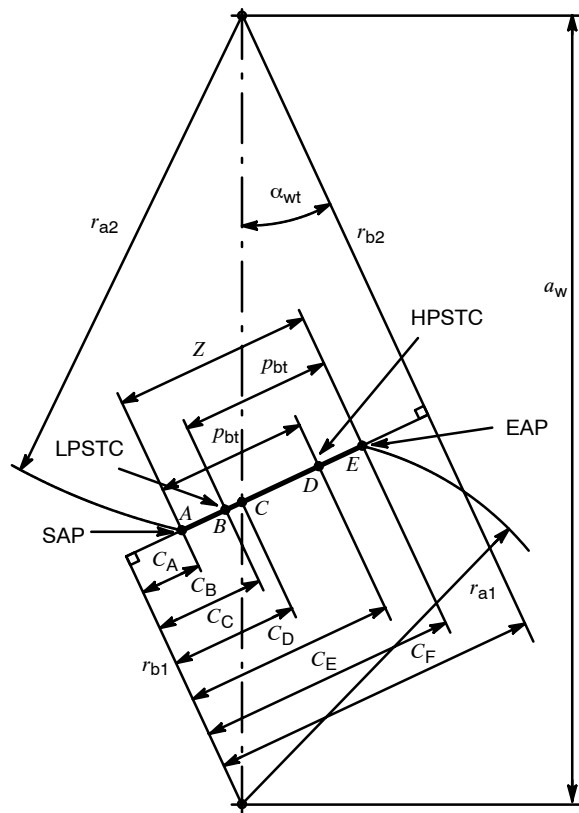


Figure 1 - Distances along the line of action for external gears

4.1.3 Contact ratios

Transverse contact ratio

$$\epsilon_\alpha = \frac{Z}{p_{bt}} \quad (22)$$

n_r is fractional (non-integer) part of ϵ_α .

Axial contact ratio

- for helical gears

$$\epsilon_\beta = \frac{b}{p_x} \quad (23)$$

n_a is fractional (non-integer) part of ϵ_β .

- for spur gears

$$\epsilon_\beta = 0.0 \quad (24)$$

Minimum contact length

- for helical gears, case 1, where $(1 - n_r) \geq n_a$

$$L_{\min} = \frac{\epsilon_\alpha b - n_a n_r p_x}{\cos \beta_b} \quad (25)$$

- for helical gears, case 2, where $(1 - n_r) < n_a$

$$L_{\min} = \frac{\epsilon_\alpha b - (1 - n_a)(1 - n_r) p_x}{\cos \beta_b} \quad (26)$$

- for spur gears

$$L_{\min} = b \quad (27)$$

4.1.4 Roll angles

Pinion roll angles corresponding to the five specific points along the line of action shown in figure 1 are given by:

$$\xi_j = \frac{C_j}{r_{b1}} \quad (28)$$

where

$$j = A, B, C, D, E$$

4.1.5 Profile radii of curvature

Transverse radii of curvature

Figure 2 shows the transverse radii of curvature, $\rho_{1(i)}$ and $\rho_{2(i)}$, of the gear tooth profiles at a general contact point defined by the roll angle, $\xi_{(i)}$, where (i) is any point on the line of action from A to E (see figure 1).

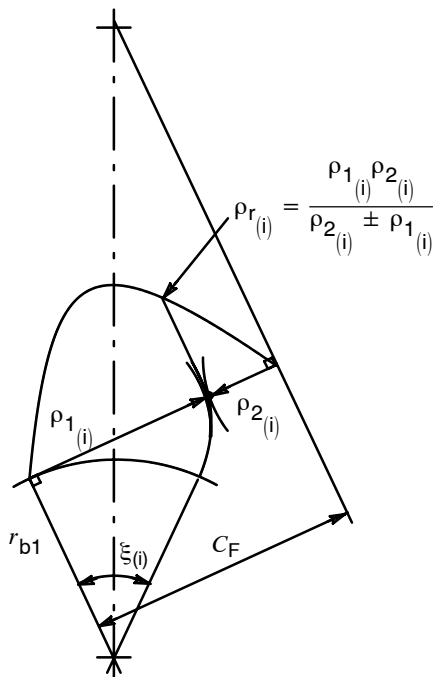


Figure 2 - Transverse relative radius of curvature for external gears

$$\rho_{1(i)} = r_{b1} \xi_{(i)} \quad (29)$$

where

$$\begin{aligned} \xi_A &\leq \xi_{(i)} \leq \xi_E \\ \rho_{2(i)} &= C_F \mp \rho_{1(i)} \end{aligned} \quad (30)$$

Transverse relative radius of curvature

$$\rho_{r(i)} = \frac{\rho_{1(i)} \rho_{2(i)}}{\rho_{2(i)} \pm \rho_{1(i)}} \quad (31)$$

Normal relative radius of curvature

$$\rho_{n(i)} = \frac{\rho_{r(i)}}{\cos \beta_b} \quad (32)$$

$\rho_{r(i)}$ and $\rho_{n(i)}$ are the equivalent radii of cylinders riding on a flat plate that represent the gear pair curvatures in contact along the line of action.

4.2 Gear tooth velocities and loads

Rotational (angular) velocities

$$\omega_1 = \frac{\pi n_1}{30} \quad (33)$$

$$\omega_2 = \frac{\omega_1}{u} \quad (34)$$

Operating pitch line velocity

$$v_t = \frac{\omega_1 r_{w1}}{1000} \quad (35)$$

Rolling (tangential) velocities

$$v_{r1(i)} = \frac{\omega_1 \rho_{1(i)}}{1000} \quad (36)$$

$$v_{r2(i)} = \frac{\omega_2 \rho_{2(i)}}{1000} \quad (37)$$

Sliding velocity (absolute value)

$$v_{s(i)} = |v_{r1(i)} - v_{r2(i)}| \quad (38)$$

Entraining velocity (absolute value)

$$v_{e(i)} = |v_{r1(i)} + v_{r2(i)}| \quad (39)$$

Nominal tangential load

$$(F_t)_{\text{nom}} = \frac{1000 P}{v_t} \quad (40)$$

Combined derating factor

$$K_D = K_o K_m K_v \quad (41)$$

where

K_o is overload factor;

K_m is load distribution factor;

K_v is dynamic factor.

See ANSI/AGMA 2101-C95 for guidance in determining K_o , K_m and K_v factors.

Actual tangential load

$$F_t = (F_t)_{\text{nom}} K_D \quad (42)$$

Normal operating load

$$F_{wn} = \frac{F_t}{\cos \alpha_{wn} \cos \beta_w} \quad (43)$$

Normal unit load

$$w_n = \frac{F_{wn}}{L_{\min}} \quad (44)$$

4.3 Load sharing factor

The load sharing factor accounts for load sharing between succeeding pairs of teeth as influenced by profile modification, and whether the pinion or gear is the driving member. By convention, the load sharing factor is represented by a polygonal function on the line of action with magnitude equal to 1.0 between points B and D (see figure 3).

The load sharing factor is strongly influenced by profile modification of the tooth flanks of both gears. On the other hand, profile modifications are chosen such that load sharing follows a desired function. The following equations give the load sharing factor for unmodified tooth profiles, and for three typical cases of profile modifications.

For unmodified tooth profiles

If there is no tip or root relief (see figure 3):

$$X_{\Gamma(i)} = \frac{1}{3} + \frac{1}{3} \left(\frac{\xi_{(i)} - \xi_A}{\xi_B - \xi_A} \right) \text{ for } \xi_A \leq \xi_{(i)} < \xi_B \quad (45)$$

$$X_{\Gamma(i)} = 1 \text{ for } \xi_B \leq \xi_{(i)} \leq \xi_D \quad (46)$$

$$X_{\Gamma(i)} = \frac{1}{3} + \frac{1}{3} \left(\frac{\xi_E - \xi_{(i)}}{\xi_E - \xi_D} \right) \text{ for } \xi_D < \xi_{(i)} \leq \xi_E \quad (47)$$

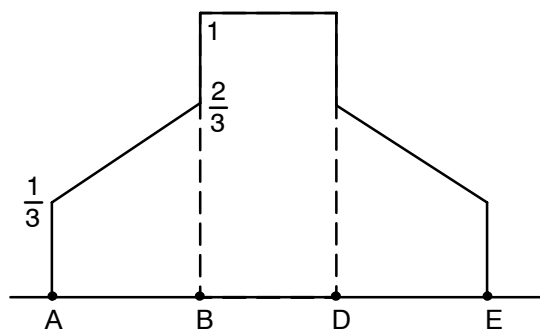


Figure 3 - Load sharing factor - unmodified profiles

For modified tooth profiles

If adequate tip and root relief is designed for high load capacity, and if the pinion drives the gear (see figure 4):

$$X_{\Gamma(i)} = \frac{6}{7} \left(\frac{\xi_{(i)} - \xi_A}{\xi_B - \xi_A} \right) \text{ for } \xi_A \leq \xi_{(i)} < \xi_B \quad (48)$$

$$X_{\Gamma(i)} = 1 \text{ for } \xi_B \leq \xi_{(i)} \leq \xi_D \quad (49)$$

$$X_{\Gamma(i)} = \frac{1}{7} + \frac{6}{7} \left(\frac{\xi_E - \xi_{(i)}}{\xi_E - \xi_D} \right) \text{ for } \xi_D < \xi_{(i)} \leq \xi_E \quad (50)$$

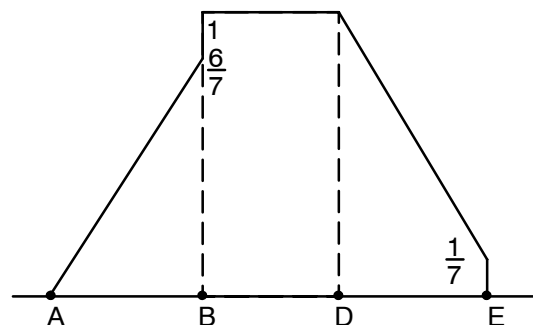


Figure 4 - Load sharing factor - pinion driving

If adequate tip and root relief is designed for high load capacity, and if the pinion is driven by the gear (see figure 5):

$$X_{\Gamma(i)} = \frac{1}{7} + \frac{6}{7} \left(\frac{\xi_{(i)} - \xi_A}{\xi_B - \xi_A} \right) \text{ for } \xi_A \leq \xi_{(i)} < \xi_B \quad (51)$$

$$X_{\Gamma(i)} = 1 \text{ for } \xi_B \leq \xi_{(i)} \leq \xi_D \quad (52)$$

$$X_{\Gamma(i)} = \frac{6}{7} \left(\frac{\xi_E - \xi_{(i)}}{\xi_E - \xi_D} \right) \text{ for } \xi_D < \xi_{(i)} \leq \xi_E \quad (53)$$

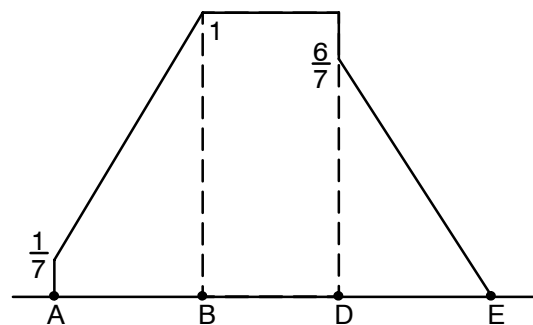


Figure 5 - Load sharing factor - gear driving

For smooth meshing

If adequate tip and root relief is designed for smooth meshing (see figure 6):

$$X_{\Gamma(i)} = \frac{\xi_{(i)} - \xi_A}{\xi_B - \xi_A} \text{ for } \xi_A \leq \xi_{(i)} < \xi_B \quad (54)$$

$$X_{\Gamma(i)} = 1 \text{ for } \xi_B \leq \xi_{(i)} \leq \xi_D \quad (55)$$

$$X_{\Gamma(i)} = \frac{\xi_E - \xi_{(i)}}{\xi_E - \xi_D} \text{ for } \xi_D < \xi_{(i)} \leq \xi_E \quad (56)$$

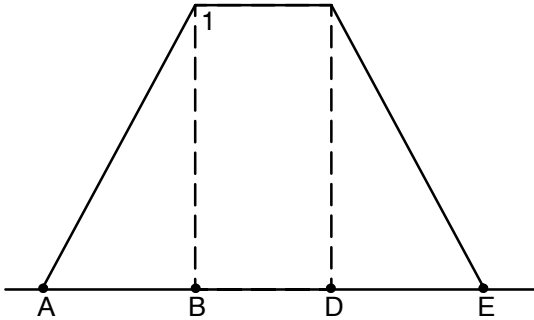


Figure 6 - Load sharing factor - smooth meshing

4.4 Hertzian contact band

The semi-width of the rectangular contact band is given by:

$$b_{H(i)} = \left(\frac{8 X_{\Gamma(i)} w_n \rho_{n(i)}}{\pi E_r} \right)^{0.5} \quad (57)$$

where

$X_{\Gamma(i)}$ is load sharing factor (see 4.3);

w_n is normal unit load, N/mm (see equation 44);

$\rho_{n(i)}$ is normal relative radius of curvature, mm (see equation 32);

E_r is reduced modulus of elasticity given by:

$$E_r = 2 \left(\frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2} \right)^{-1} \quad (58)$$

where

ν_1, ν_2 is Poisson's ratio (pinion, gear);

E_1, E_2 is modulus of elasticity, N/mm² (pinion, gear).

5 Lubrication

5.1 Viscometric information

Lubricants are commonly referred to by their base type, for example mineral or synthetic, and their viscosity, usually in relation to a defined viscosity grade. Viscosity is one of the basic and very important properties of a lubricant and is used extensively in tribological calculations. Viscosity is a bulk property of a fluid, semi-fluid or semi-solid substance that causes it to resist flow. In addition to the basic composition and structure of the material, viscosity decreases with increasing temperature and increases with increasing pressure.

For a liquid under shear, the rate of deformation or shear rate is proportional to the shearing stress. This relationship is Newton's law, which essentially states that the ratio of the stress to the shear rate is a constant. That constant is viscosity. Dynamic viscosity, η , sometimes referred to as absolute viscosity, is defined by equation 59.

$$\eta = 10^9 \frac{\tau}{\left(\frac{dv}{dh} \right)} \quad (59)$$

where:

η is dynamic viscosity, mPa•s;

τ is shear stress, N/mm²;

v is velocity, m/s;

h is thickness of an element measured perpendicular to the flow, m;

$\left(\frac{dv}{dh} \right)$ is known as the rate of shear [s⁻¹] and sometimes listed as γ .

Lubricants used in industry today, however, have their viscosity measured by capillary viscometers which provide a kinematic viscosity. Kinematic viscosity, ν , is the ratio of dynamic viscosity, η , to the density, ρ , at a specified temperature and pressure (see equation 60).

$$\nu = 10^3 \left(\frac{\eta}{\rho} \right) \quad (60)$$

where:

ν is kinematic viscosity, mm²/s;

ρ is density, kg/m³.

ASTM D445 [9] is the most widely used method for measuring the kinematic viscosity of lubricants for many different applications. The most commonly

used temperatures are 40°C and 100°C; these measurements are generally made at atmospheric pressure.

Under Newton's law, viscosity is independent of shear rate. Fluids such as these are referred to as Newtonian fluids. Most conventional, single grade lubricants made with a relatively low molecular weight base stock (non-polymeric) are considered Newtonian fluids. However, there are fluids that do not exhibit this ideal behavior because their viscosity is not independent of shear rate. These are usually finished blends containing higher molecular weight polymers (viscosity modifiers or viscosity index improvers, as well as pour point depressants) that are sensitive to shear rate. Some exhibit shear thinning, whereas others result in shear thickening. It is more common in gear lubricant applications to find shear thinning relationships due to the nature of the polymers typically used in these formulations. This shear thinning translates into lower effective viscosities in the contact region under operation than might be expected from a non-polymer blend of similar viscosity.

5.1.1 Viscosity – temperature relationship

Lubricant viscosity varies inversely with temperature. A truly ideal fluid would have a viscosity that is constant over all temperature. ASTM method D341 [10] can be used to obtain the viscosity-temperature relationship. A simplified form can be used to estimate the kinematic viscosity of a fluid at a given temperature if there is some viscometric information available for the fluid at two other temperatures (see equation 61).

$$\log_{10} \log_{10}(v + 0.7) = A - B \times \log_{10} T \quad (61)$$

where:

T is absolute temperature, K;

v is kinematic viscosity, mm²/s;

A, B are dimensionless constants.

A and B can be determined by solving equation 61 simultaneously with equations 62 and 63, using the kinematic viscosity of the fluid measured at standard temperatures of 40°C and 100°C.

$$B = \frac{\log_{10} \log_{10}(v_{40} + 0.7) - \log_{10} \log_{10}(v_{100} + 0.7)}{\log_{10}(373.15) - \log_{10}(313.15)} \quad (62)$$

then

$$A = \log_{10} \log_{10}(v_{40} + 0.7) + B \times \log_{10}(313.15) \quad (63)$$

Another common numeric designation that provides information about the viscosity-temperature relationship of a fluid is the viscosity index or VI. The viscosity index of a fluid can be calculated by ASTM method D2270 [11]. This arbitrary measure gives a relative viscosity-temperature sensitivity for a given oil. The higher the value the less change in viscosity with temperature.

5.1.2 Viscosity-pressure relationship

Equally important to temperature on the fluid viscosity is the pressure acting on it. This is especially important in highly loaded contacts such as gears and rolling element bearings where pressures can easily exceed 1 GPa. The viscosity of lubricant trapped in a concentrated contact increases exponentially with pressure. In 1893, C. Barus established an empirical equation to describe the isothermal viscosity-pressure relationship for a given liquid as shown in equation 64.

$$\eta_p = \eta_{atm} e^{\alpha p} \quad (64)$$

where

η_p is viscosity at pressure, p , mPa•s;

η_{atm} is viscosity at atmospheric pressure, mPa•s;

α is pressure-viscosity coefficient, mm²/N.

Today the model continues to be refined. So and Klaus [12] provided a comparison of the many models developed since the Barus equation was first introduced. The continued research aided by the development of high pressure rheology techniques to generate empirical information have shown that the viscosity-pressure response of a fluid is also related to its chemical structure [13, 14, 15]. This can have a profound effect on the film forming capabilities of the fluid in question and the overall life of the component involved.

5.2 Film thickness equation

Dowson, Higginson and Toyoda have authored various papers on EHL film thickness [16, 17, 18, 19]. The film thickness equations given in these papers account for the exponential increase of lubricant viscosity with pressure, tooth geometry, velocity of the gear teeth, material elastic properties and the transmitted load. The film thickness determines the operating regime of the gearset and

has been found to be a useful index of wear related distress probability. Wellauer and Holloway [20] also found that specific film thickness could be correlated with the probability of tooth surface distress. The Dowson and Toyoda [19] equation for line contact central EHL film thickness will be used as shown below.

Dimensionless central film thickness:

$$H_{c(i)} = 3.06 \frac{G^{0.56} U_{(i)}^{0.69}}{W_{(i)}^{0.10}} \quad (65)$$

where

(i) (as a subscript) defines a point on the line of action,

and the dimensionless parameters G , $U_{(i)}$ and $W_{(i)}$ are defined below:

materials parameter, G

$$G = \alpha E_r \quad (66)$$

speed parameter, $U_{(i)}$

$$U_{(i)} = \frac{\eta_M v_{e(i)}}{2E_r \rho_{n(i)}} \times 10^{-6} \quad (67)$$

load parameter, $W_{(i)}$

$$W_{(i)} = \frac{X_{\Gamma(i)} w_n}{E_r \rho_{n(i)}} \quad (68)$$

where

η_M is dynamic viscosity at the gear tooth temperature, mPa•s.

$$\eta_M = 10^g - 0.9 \quad (69)$$

where

$$g = 10^c (\theta_M + 273.15)^d$$

θ_M is tooth temperature, °C (see 6.3).

The parameters c and d required for calculating η_M can either be taken from table 2 or calculated with equations 70 and 72, respectively. Equations 70 and 72, derived from a modification of the Walther equation [10], will yield the parameters c and d if two dynamic viscosities, η_1 and η_2 , are known at two corresponding temperatures, θ_1 and θ_2 .

Since dynamic viscosity is generally available at 40°C and 100°C, equations 70 and 72 are modified in equations 71 and 73 to incorporate terms corresponding to those temperatures.

η_1 is dynamic viscosity at temperature θ_1 , mPa•s;

η_2 is dynamic viscosity at temperature θ_2 , mPa•s;

θ_1 is temperature at which η_1 was determined, °C;

θ_2 is temperature at which η_2 was determined, °C.

$$d = \frac{\log_{10} \left[\frac{\log_{10}(\eta_2 + 0.9)}{\log_{10}(\eta_1 + 0.9)} \right]}{\log_{10} \left(\frac{\theta_2 + 273.15}{\theta_1 + 273.15} \right)} \quad (70)$$

when $\theta_1 = 40^\circ\text{C}$ and $\theta_2 = 100^\circ\text{C}$,

$$d = 13.13525 \log_{10} \left[\frac{\log_{10}(\eta_{100} + 0.9)}{\log_{10}(\eta_{40} + 0.9)} \right] \quad (71)$$

$$c = \log_{10} \left[\log_{10}(\eta_1 + 0.9) \right] - d \log_{10}(\theta_1 + 273.15) \quad (72)$$

when $\theta_1 = 40^\circ\text{C}$ and $\theta_2 = 100^\circ\text{C}$,

$$c = \log_{10} \left[\log_{10}(\eta_{40} + 0.9) \right] - 2.495752 d \quad (73)$$

α is pressure-viscosity coefficient, mm²/N. Values range from 0.725×10^{-2} mm²/N to 2.9×10^{-2} mm²/N for typical gear lubricants. Values for pressure-viscosity coefficients vs. dynamic viscosity can be obtained from equation 74.

$$\alpha = k \eta_M^s \quad (74)$$

Table 2 contains viscosity information for mineral oils, MIL-L spec. oils, polyalphaolefin (PAO) based synthetic oils (which contain ester) and polyalkylene glycol (PAG) based synthetic oils, as well as constants c , d , k and s for use in the equations 69 through 74. These values were obtained from the data shown in figures 7 through 11 [22]. It is important that the film thickness is calculated with values of viscosity and pressure-viscosity coefficient for the gear tooth temperature, θ_M , (see 6.3).

The central film thickness at a given point is:

$$h_{c(i)} = H_{c(i)} \rho_{n(i)} \times 10^3 \quad (75)$$

(see clause 4 for $\rho_{n(i)}$).

Table 2 - Data for determining viscosity and pressure-viscosity coefficient

Lubricant	ISO VG ¹⁾	η_{40}	η_{100}	c	d	k	s
Mineral oil	32	27.17816	4.294182	10.20076	-4.02279	0.010471	0.1348
	46	39.35879	5.440514	10.07933	-3.95628	0.010471	0.1348
	68	58.64514	7.059163	9.90355	-3.86833	0.010471	0.1348
	100	86.91484	9.251199	9.65708	-3.75377	0.010471	0.1348
	150	131.4335	12.27588	9.42526	-3.64563	0.010471	0.1348
	220	194.2414	15.98296	9.24059	-3.55832	0.010471	0.1348
	320	284.6312	20.60709	9.09300	-3.48706	0.010471	0.1348
	460	412.0824	26.34104	8.96420	-3.42445	0.010471	0.1348
	680	613.8288	34.24003	8.84572	-3.36585	0.010471	0.1348
	1000	909.4836	38.56783	9.25943	-3.52128	0.010471	0.1348
	1500	1374.931	49.58728	9.19946	-3.48702	0.010471	0.1348
	2200	2031.417	62.69805	9.15646	-3.46064	0.010471	0.1348
	3200	2975.954	78.56109	9.13012	-3.44157	0.010471	0.1348
PAO - based synthetic non-VI improved oil	150	128.5772	16.17971	7.99428	-3.07304	0.010326	0.0507
	220	189.9828	21.60933	7.79927	-2.98154	0.010326	0.0507
	320	278.3370	28.66405	7.63035	-2.90169	0.010326	0.0507
	460	402.8943	37.54020	7.49799	-2.83762	0.010326	0.0507
	680	600.0179	53.20423	7.16434	-2.69277	0.010326	0.0507
	1000	868.1710	68.60767	7.12008	-2.66528	0.010326	0.0507
	1500	1310.350	91.03300	7.07678	-2.63766	0.010326	0.0507
	2200	1933.070	118.0509	7.06113	-2.62221	0.010326	0.0507
	3200	2827.726	151.2132	7.06594	-2.61561	0.010326	0.0507
PAG - based synthetic ²⁾	100	102.630	19.560	6.42534	-2.45259	0.0047	0.1572
	150	153.950	27.380	6.19586	-2.34616	0.0047	0.1572
	220	225.790	40.090	5.76552	-2.16105	0.0047	0.1572
	320	328.430	56.710	5.49394	-2.04065	0.0047	0.1572
	460	472.130	77.250	5.35027	-1.97254	0.0047	0.1572
	680	697.920	113.43	5.06011	-1.84558	0.0047	0.1572
	1000	1026.37	163.30	4.85075	-1.75175	0.0047	0.1572
MIL-L-7808K Grade 3	12	11.35364	2.701402	9.58596	-3.82619	0.005492	0.25472
MIL-L-7808K Grade 4	17	16.09154	3.609883	9.08217	-3.60300	0.005492	0.25472
MIL-L-23699E	23	22.56448	4.591235	8.91638	-3.51779	0.006515	0.16530

NOTES:

1) ν_{40} (mm²/s)

2) Copolymer of ethylene oxide and propylene oxide in 50% weight ratio.

The specific film thickness is the ratio of film thickness divided by the composite roughness of the contacting gear teeth and can be used to assess performance.

To determine this ratio, the cutoff wavelength for the composite surface roughness measurement (σ_x) should be comparable to the width of the Hertzian contact, $2b_{H(i)}$. This results in σ_x becoming $\sigma_{2b_{H(i)}}$ as

shown in equation 76.

$$\lambda_{2b_{H(i)}} = \frac{h_{c(i)}}{\sigma_{2b_{H(i)}}} \quad (76)$$

This may not be practical because many surface measuring instruments have a fixed cutoff wavelength (usually 0.8 mm).