The Lubrication of Gears - Part II

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Introduction

What follows is Part 2 of a three-part article covering the principles of gear lubrication. Part 2 gives an equation for calculating the lubricant film thickness, which determines whether the gears operate in the boundary, elastohydrodynamic, or full-film lubrication regime. An equation for Blok's flash temperature, which is used for predicting the risk of scuffing, is also given.

Elastohydrodynamic Lubrication

Gear teeth are subjected to enormous contact pressures on the order of the ultimate tensile strength of hardened steel, yet they are quite successfully lubricated with oil films that are less than one micrometer thick. This is possible because a fortuitous property of lubricants causes their viscosity to increase dramatically with increased pressure. Fig. 1 depicts the region of contact between mating gear teeth. It shows the shape of the elastically deformed teeth and the pressure distribution developed within the contact zone. The molecular adsorption of the lubricant onto the gear tooth surfaces causes it to be dragged into the inlet region of the contact, where its pressure is increased due to the convergence of the tooth surfaces. The viscosity increase of the lubricant caused by the increasing pressure helps to entrain the lubricant into the contact zone. Once it is within the high pressure, Hertzian region of the contact, the lubricant cannot escape because its viscosity has increased to the extent where the lubricant is virtually a rigid solid.

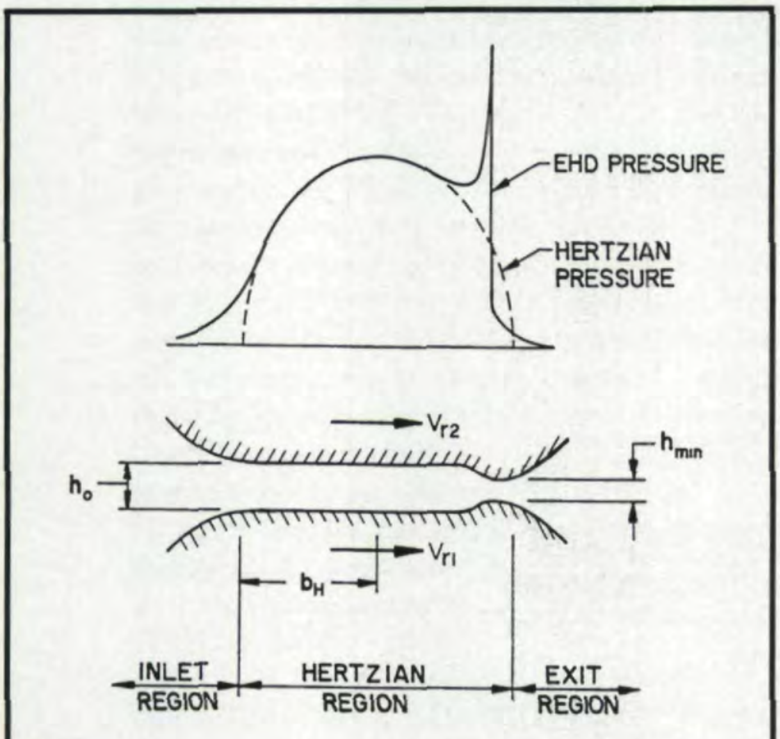

Fig. 1

The following equation, from Dowson and Higginson[1] gives the miniumum film thickness that occurs near the exit of the contact.

Minimum film thickness:

$$hmin = \frac{1.63\alpha^{0.54}(\mu_o V_e)^{0.7} \rho n^{0.43}}{(X_r w_{Nr})^{0.13} Er^{0.03}}$$

The specific film thickness is given by

$$\lambda = \frac{hmin}{\sigma}$$

where

σ = composite surface roughness

$$\sigma = (\sigma_1^2 + \sigma_2^2)^{1/2}$$

σ_1, σ_2 = surface roughness, rms (pinion, gear)

μ_0 = absolute viscosity, Reyns (lb sec/in^2) Fig. 2 gives average values of viscosity versus temperature for typical mineral gear lubricants with viscosity index of 95.

α = pressure-viscosity coefficient, (in^2/lb). The pressure-viscosity coefficient ranges from $\alpha = 0.5 \times 10^{-4}$ to $\alpha = 2 \times 10^{-4}$ in^2/lb for typical gear lubricants. Data for pressure-viscosity coefficients versus temperture for typical gear lubricants are given in Fig. 3.

Er = reduced modulus of elasticity given by

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

where

ν_1, ν_2 = Poisson's ratio (pinion, gear)

E_1, E_2 = modulus of elasticity (pinion, gear)

ρn = normal relative radius of curvature

$$\rho n = \frac{\rho_1 \rho_2}{(\rho_2 \pm \rho_1)\cos\psi_b}$$

ρ_1, ρ_2 = transverse radius of curvature (pinion, gear)

ψ_b = base helix angle

Ve = entraining velocity given by

$Ve = Vr_1 + Vr_2$

where

Vr_1, Vr_2 = rolling velocities given by

$Vr_1 = \omega_1 \rho_1$

$Vr_2 = \omega_2 \rho_2$

ω_1, ω_2 = angular velocities (pinion, gear)

W_{Nr} = normal unit load given by

$$W_{Nr} = \frac{W_{Nr}}{Lmin}$$

where

W_{Nr} = normal operating load

Lmin = minimum contact length

Load Sharing Factor, X_r

The load sharing factor accounts for load sharing between succeeding pairs of teeth as influenced by profile modification (tip and/or root relief) and whether the pinion or gear is the driver. Fig. 4 gives plots of the load sharing factors for unmodified and modified tooth profiles.

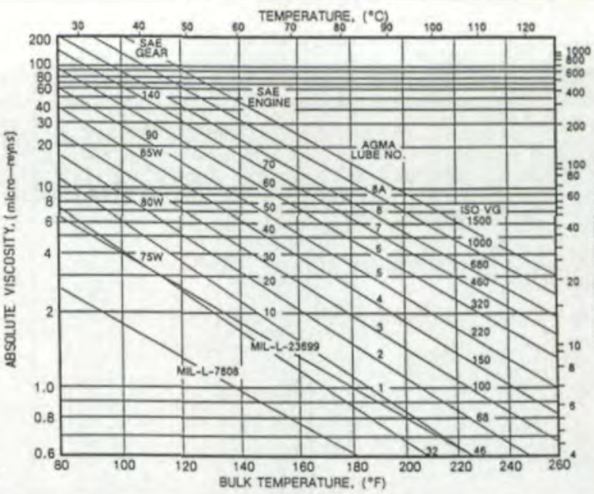

Fig. 2

As shown by the exponents in the Dowson and Higginson equation, the film thickness is essentially determined by the entraining velocity, lubricant viscosity, and pressure-viscosity coefficient, while the elastic properties of the gear teeth and the load have relatively small influences. In effect, the relatively high stiffness of the oil film makes it insensitive to load, and an increase in load simply increases the elastic deformation of the tooth surfaces and widens the contact area, rather than decreasing the film thickness.

Blok's Contact Temperature

Blok's[2] contact temperature theory states that scuffing will occur in gear teeth that are sliding under boundary-lubricated conditions when the maximum contact temperature of the gear teeth reaches a critical magnitude. The contact temperature is the sum of two components, the bulk

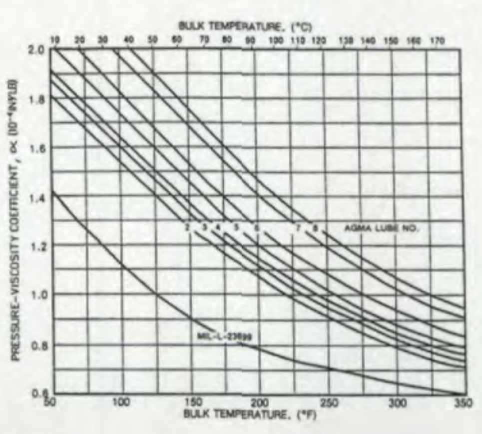

Fig. 3

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Nomenclature Table		
Symbol	**Description**	**Units**
B_M	-thermal contact coefficient	1bf/[in$s^{0.5}$°F]
b_H	-semi-width of Hertzian contact band	in
c	-constant (See Table 3.)	hp/gpm
c_M	-specific heat per unit mass	1bf in/[lb °F]
d	-operating pitch diameter of pinion	in
E_1,E_2	-modulus of elasticity (pinion, gear)	lbf/in^2
Er	-reduced modulus of elasticity	lbf/in^2
h_{min}	-minimum film thickness	in
L_{min}	-minimum contact length	in
n	-pinion speed	rpm
P	-transmitted power	hp
q	-oil flow rate	gpm
S	-average surface roughness, rms	μin
Tb	-bulk temperature	°F
Tb,test	-bulk temperature of test gears	°F
Tc	-contact temperature	°F
Tf	-flash temperature	°F
Tf, test	-maximum flash temperature of test gears	°F
Ts	-scuffing temperature	°F
V	-operating pitch line velocity	ft/min
Ve	-entraining velocity	in/s
Vr_1,Vr_2	-rolling velocity (pinion,gear)	in/s
W_{Nr}	-normal operating load	lbf
w_{Nr}	-normal unit load	lbf/in
Xw	-welding factor	--
X_r	-load sharing factor	--
α	-pressure-viscosity coefficient	in^2/1b^2
λ	-specific film thickness	--
λ_M	-heat conductivity	lbf/[s °F]
μ_m	-mean coefficient of friction	--
μ_o	-absolute viscosity	Reyns (lbs/in^2)
$\nu_1\nu_2$	-Poisson's ratio (pinion, gear)	--
ν_{40}	-kinematic viscosity of 40°C	cSt
$\rho_1\rho_2$	-transverse radius of curvature (pinion, gear)	in
ρ_M	-density	lb/in^3
ρ_n	-normal relative radius of curvature	in
σ	-composite surface roughness, rms	μin
σ_1,σ_2	-surface roughness, rms (pinion, gear)	μin
ψ_b	-base helix angle	deg
ω_1,ω_2	-angular velocity (pinion, gear)	rad/s

perature of the gear teeth reaches a critical magnitude. The contact temperature is the sum of two components, the bulk temperature and the flash temperature; i.e., Tc = Tb + Tf.

Blok's flash temperature equation as formulated in AGMA 2001-B88, Appendix A[3] for spur and helical gears is

$$Tf = \frac{0.8\mu_m X_r w_{Nr} \left| (V_{r1})^{0.5} - (V_{r2})^{0.5} \right|}{B_M (b_H)^{0.5}}$$

where

μ_m = mean coefficient of friction
X_r = load sharing factor
w_{Nr} = normal unit load
V_{r1} = rolling velocity of the pinion
V_{r2} = rolling velocity of the gear
B_M = thermal contact coefficient
b_H = semi-width of Hertzian contact band

Mean Coefficient of Friction, μ_m

The following equation gives a typical value of $0.06 < \mu_m < 0.18$ for the mean coefficient of friction for gears operating the the partial EHD regime ($\lambda < 1$). It may give values too low for boundary-lubricated gears where μ_m may be greater than 0.2, or too high for gears in the full-film regime ($\lambda > 2$), where μ_m may be less than 0.01.

$$\mu_m = 0.06 \left(\frac{50}{50 - S} \right)$$

where

$$\left(\frac{50}{50 - S} \right) \leq 3.0$$

S = average surface roughness, rms

$$S = \frac{\sigma_1 + \sigma_2}{2}$$

Thermal Contact Coefficient, B_M

The thermal contact coefficient is given by

$$B_M = (\lambda_M \rho_M C_M)^{0.5}$$

where

λ_M = heat conductivity
ρ_M = density
C_M = specific heat per unit mass
For typical gear steels, $B_M \approx 43$ Lbf/[in $s^{0.5}$ °F]

Table 1[5] - Welding Factor X_w

Material	X_w
Through-hardened steel	1.00
Phosphated steel	1.25
Copper-plated steel	1.50
Nitrided steel	1.50
Carburized steel	
Content of austenite < average	1.15
Content of austenite average	1.00
Content of austenite > average	0.85
Stainless steel	0.45

Semi-Width of Hertzian Contact Band, b_H

$$b_H = \left(\frac{8 X_r w_{Nr} \rho_n}{\pi E_r} \right)^{0.5}$$

Bulk Temperature, T_b

The gear bulk temperature is the equilibrium bulk temperature of the gear teeth before they enter the meshing zone. In some cases, the bulk temperature may be significantly higher than the temperature of the oil supplied to the gear mesh. In a test with ultra high-speed gears [4], the pinion bulk temperature was 275°F (171°F hotter than the oil inlet temperature). For turbine gears at lower speeds, the bulk temperature rise of the gear teeth over the inlet oil temperature may range from 20F at 12,000 fpm pitch line velocity to 40°F at 16,000 fpm. At similar speeds, the bulk temperature rise of aircraft gears with less oil flow may range from 40°F to 60°F.

Scuffing Temperature, Ts

The scuffing temperature is the contact temperature at which scuffing is likely to occur with the chosen combination of lubricant and gear materials.

For mineral oils without anti-scuff additives or for mineral oils with low concentrations of anti-scuff additives, the scuffing temperature is independent of the operating conditions for a fairly wide range. For these oils, the scuffing temperature may be correlated with the composition of the oil. The viscosity grade is a convenient index of the composition and, thus, of the scuffing temperature.

For non-anti-scuff mineral oils, the mean scuffing temperature (50% chance of scuffing) is given by

Table 2[6]
Synthetic Lubricant Mean Scuffing Temperature, Ts

Lubricant	Mean Scuffing Temp. Ts (°F)
MIL-L-6081 (grade 1005)	264
MIL-L-7808	400
MIL-L-23699	425
DERD2487	440
DERD2497	465
DOD-L-85734	500
MOBIL SHC624	540
DEXRON II	550

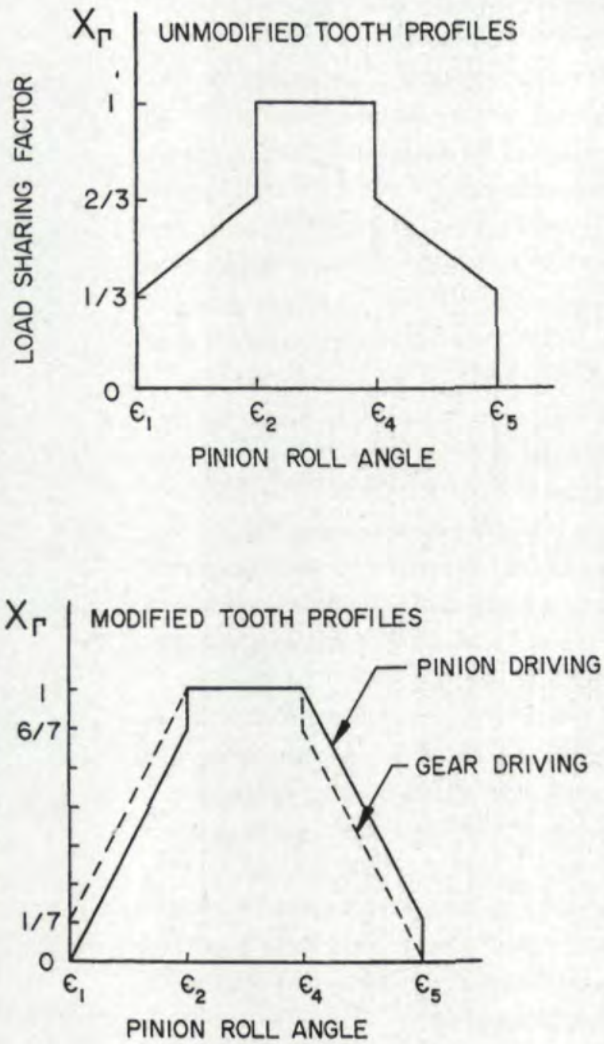

Fig. 4

$$Ts = 146 + 591n\nu_{40}\,°F$$

For mineral oils with low concentrations of anti-scuff additives, the mean scuffing temperature is given by

$$Ts = 245 + 59 \ln \nu_{40}\,°F$$

where

ν_{40} = kinematic viscosity at 40°C, cSt

The scuffing temperature determined from FZG test gears for mineral oils without anti-scuff additives or with low concentrations of anti-scuff additives may be extended to different gear steels, heat treatments, or surface treatments by introducing an empirical welding factor:

$$Ts = T_{b'test} + X_w\, T_{f'test}$$

where

X_w = welding factor (See Table 1.)
$T_{b'test}$ = bulk temperature of test gears
$T_{f'test}$ = maximum flash temperature of test gears.

For synthetic lubricants and carburized gears typical of the aerospace industry, the scuffing temperatures are shown in Table 2.

For mineral oils with high concentrations of anti-scuff additives, such as hypoid gear oils, research is still needed to determine whether the scuffing temperature is dependent on the materials and/or operating conditions. Special attention has to be paid to the correlation between test conditions and actual or design conditions.

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