The Lubrication of Gears - Part III

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Introduction

This is the final part of a three-part series on the basics of gear lubrication. It covers selection of lubricant types and viscosities, the application of lubricants, and a case history.

Selecting Lubricant Type

The choice of lubricant depends on the type of gearing and enclosure, operating speed and load, ambient temperature, and method of lubricant application. Most gears are lubricated with one of the following types: oil, grease, adhesive open-gear lubricant, or solid lubricant. The optimum lubricant for any application is the least expensive, considering both initial cost and maintenance costs, that meets the requirements.

Oil is the most widely used lubricant because it is readily distributed to gears and bearings and has both good lubricating and cooling properties. Also, contamination may be readily removed by filtering or draining and replacing the oil. However, it requires an oil-tight enclosure provided with adequate shaft seals.

Grease is suitable only for low-speed, low-load applications because it does not circulate well, and it is a relatively poor coolant. Grease lubricated gears are generally boundary lubricated because the grease is either pushed aside or thrown from the gear teeth. Contamination from wear particles or other debris is usually trapped in the grease and requires costly maintenance to eliminate. Grease is often used to avoid leakage from enclosures that are not oil-tight. However, if all the factors are considered, it is usually found that an oil lubricant is more economical and reliable than a grease for gear lubrication.

Open-gear lubricants are viscous, adhesive semi-fluids used on large, low-speed, open gears, such as those used in iron ore and cement mills, antenna drives, bridge drives, cranes, etc. Gears in these applications run slowly, and they are therefore boundary lubricated. The lubricant must bond strongly to resist being thrown off the gear teeth. However, the squeezing and sliding action of gear teeth tends to push the lubricant into the roots of the gear teeth where it is relatively ineffective. These lubricants are applied by hand brushing or by automatic systems which deliver an intermittent spray. Some open gear lubricants are thinned with a quick-evaporating solvent/diluent to make them easier to apply. Open-gear lubricants share the disadvantages of grease lubrication, and they are especially costly (and messy) to maintain. For these reasons, the trend is away from open gears toward enclosed, oil-lubricated gearboxes whenever possible.

Solid lubricants, usually in the form of bonded, dry films, are used where the temperature is too high or too low for an oil or grease; where leakage cannot be tolerated; or where the gears must operate in a vacuum. These lubricants are usually molybdenum disulfide (MoS_2) or graphite in an inorganic binder, which is applied to the gear teeth and cured to form a dry film coating. Polytetrafluoroethylene (PTFE) and tungsten disulfide (WS_2) coatings are also used. Solid lubricants are expensive to apply and have limited wear lives. However, in many applications, such as spacecraft, they are the only alternative and can provide excellent service.

Only oil lubricants will be discussed in greater detail. Oil should be used as the lubricant unless the operating conditions preclude its use. Generally, the simplest and least expensive lubrication system for gears is a totally enclosed, oil-bath of mineral oil.

The lubrication requirements of spur, helical, straight-bevel, and spiral-bevel gears are essentially the same. For this class of gears, the magnitudes of the loads and sliding speeds are similar, and requirements for viscosity and anti-scuff properties are virtually identical. Many industrial spur and helical gear units are lubricated with rust and oxidation-inhibited (R&O) mineral oils. The low viscosity R&O oils, commonly called turbine oils, are used in many high-speed gear units where the gear tooth loads are relatively low. Mineral oils without anti-scuff additives are suitable for high-speed, lightly loaded gears where the high entraining velocity of the gear teeth develops thick EHD oil films. In these cases the most important property of the lubricant is viscosity. Anti-scuff/EP additives are unnecessary because the gear teeth are separated, eliminating metal-to-metal contact and the scuffing mode of failure. Slower speed gears, especially carburized gears, tend to be more heavily loaded. These gears generally require higher viscosity lubricants with anti-scuff additives.

Hypoid gears, such as those used for automotive axles, are especially prone to scuffing because they are heavily loaded and they have high sliding velocities. For these reasons, hypoid gear oils have the higher concentrations of anti-scuff additives.

For critical applications, the contact temperature should be calculated with Blok's[1] equation and compared to the scuffing temperature of the lubricant. This quantitative method is effective for selecting a lubricant with adequate scuffing resistance.

Worm gears have high sliding velocity which generates significant frictional losses. Fortunately, their tooth loads are relatively light, and they are successfully lubricated with mineral oils that are compounded with lubricity additives. These oils contain 3% to 10% fatty oil or low acid tallow. The polar molecules of the additive form surface films by physical adsorption or by reaction with the surface oxide to form a metallic soap which acts as a low shear strength film, improving the "lubricity" or friction-reducing property.

Synthetic lubricants are used for applications, such as aircraft gas turbines, where the oil must operate over a wide temperature range and have good oxidation stability at high temperature. Ester and hydrocarbon synthetic lubricants have high viscosity indices, giving them good fluidity or low viscosities at very low temperatures and acceptable viscosities at high temperatures. The volatility of esters is lower than that of mineral oils of the same viscosity, thus reducing oil loss at high temperature. Despite their long service life, the extra cost of synthetic lubricants generally cannot be justified for oil-bath systems unless there are extreme temperatures involved, because the oil must be changed frequently to remove contamination.

Selecting Gear Lubricant Viscosity

The recommendations of AGMA 250.04[2] should be followed when selecting lubricants for enclosed gear drives that operate at pitch line velocities up to 5,000 fpm. AGMA 421.06[3] should be consulted for high-speed drives (> 5,000 fpm).

In our discussion of gear failure modes, we found that viscosity is one of the most important lubricant properties, and the higher the viscosity, the greater the protection against the various gear tooth failures. However, the viscosity must be limited to avoid excessive heat generation and power loss from churning and shearing of the lubricant by high-speed gears or bearings. The operating temperature of the gear drive determines the operating viscosity of the lubricant. If the lubricant is too viscous, excessive heat is generated. The heat raises the lubricant temperature and reduces its viscosity, reaching a point of diminishing returns where increasing the starting viscosity of the lubricant leads to a higher operating temperature and a higher oxidation rate, without a significant gain in operating viscosity.

Gear drives operating in cold climates must have a lubricant that circulates freely and does not cause high starting torques. A candidate gear lubricant should have a pour point at least 5°C (9°F) lower than the expected minimum ambient start-up temperature. Typical pour points for mineral gear oils are 20°F while synthetic gear lubricants have significantly lower pour points of about -40°F. Pour point depressants are used to tailor pour points of mineral lubricants for automotive hypoid gears to be as low as -40°F.

The pitch line speed of the gears is a good index of the required viscosity. An empirical equation for determining required viscosity is

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$$\nu_{40} = \frac{7000}{(V)^{0.5}} \tag{1}$$

where

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

V = operating pitch line velocity, ft/min

V = 0.262 d n

d = operating pitch diameter of pinion, in.

n = pinion speed, rpm

Caution must be used when using AGMA recommendations for viscosity. The author knows of an application where two gear drives were considered to be high-speed. The pinion speed was 3,625 rpm, qualifying the gear units as high-speed gear drives per AGMA 421.06. The gear drives were supplied with oil having the recommended viscosity per AGMA 421.06 of ISO 68. However, because the pinion was relatively small, its pitch line velocity was only 3,000 fpm. This qualifies the gear drives as slow-speed per AGMA 250.04, which recommends a viscosity of ISO 150. Both gear drives failed within weeks of start-up by pitting fatigue. The empirical equation for this application give

$$\nu_{40} = \frac{7000}{(3000)^{0.5}} = 128 \text{ cSt} \tag{2}$$

This indicates that the viscosity per AGMA 421.06 (68 cSt) is much too low, and the viscosity per AGMA 250.04 (150 cSt) is appropriate. Hence, definitions of high-speed versus slow-speed must be carefully considered, and pitch line velocity is generally a better index than shaft speed. The gear drives were rebuilt with new gearsets and the ISO VG 68 oil was replaced with ISO VG 150. The gear drives now operate without overheating, and the pitting has been eliminated.

For critical applications, the specific film thickness should be calculated with Dowson and Higginson's[4] equation. The specific film thickness is a useful measure of the lubrication regime. It can be used with Fig. 1 as an approximate guide to the probability of wear-related surface distress.

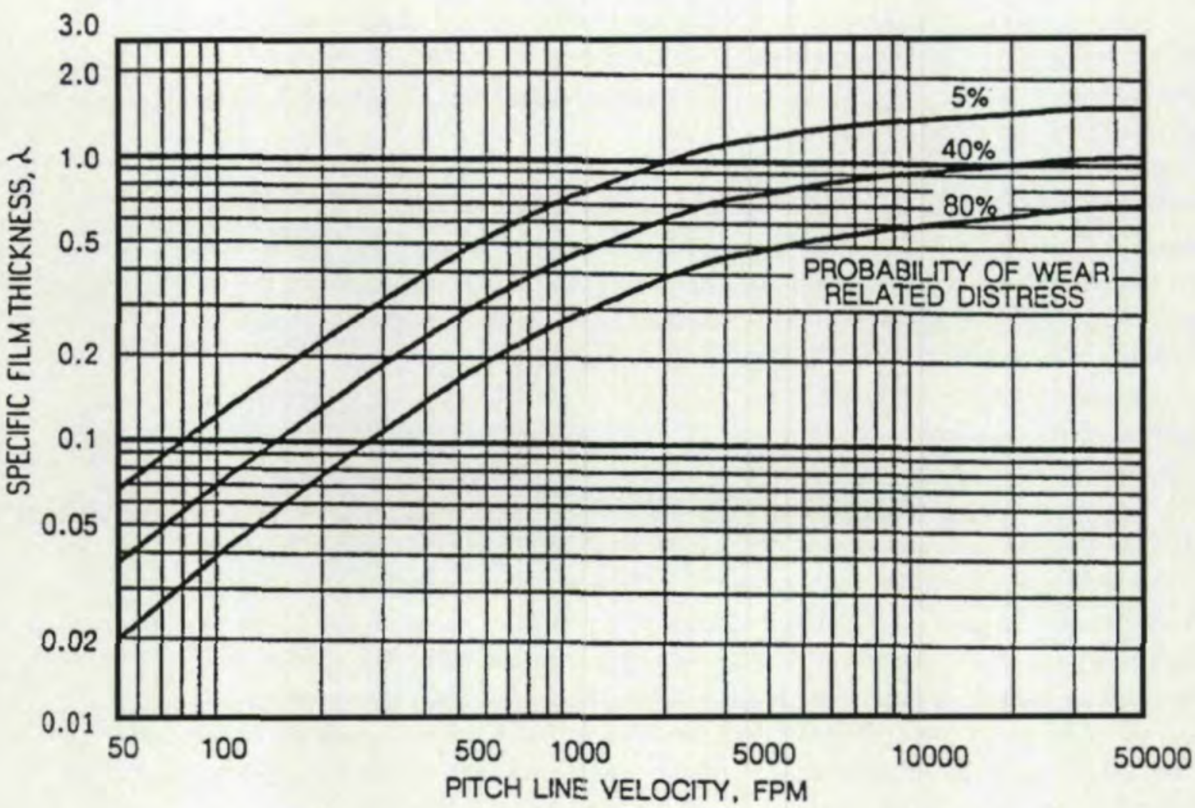

Fig. 1 - Probability of wear distress, percent. (From AGMA 2001-B88, 1988.)

Fig. 1 is based on the data of Wellauer and Holloway,[5] which were obtained from several hundred laboratory tests and field applications of gear drives.

Applying Gear Lubricants

The method of applying the lubricant to the gear teeth depends for the most part primarily on the pitch line velocity.

Splash lubrication systems are the simplest, but they are limited to a pitch line velocity of about 3,000 fpm. The gears should dip into the oil bath for about twice the tooth depth to provide adequate splash for pinions and bearings and to reduce losses due to churning. The gear housing should have troughs to capture the oil flowing down the housing walls, channeling it to the bearings.

The range of splash lubrication can be extended to about 5,000 fpm by using baffles and oil pans to reduce churning. However, above 3,000 fpm, providing auxiliary cooling with fans and improving heat transfer by adding fins to the housing is usually necessary.

Above 5,000 fpm, most gears are lubricated by a pressure-fed system. For gearboxes with antifriction bearings, spraying the oil at the gear mesh only and relying on splash to lubricate the bearings is permissible up to a pitch line velocity of 7,000 fpm maximum. Above this speed, and for gear drives with journal bearings, both the gears and bearings should be pressure-fed.

The oil jets should be placed on the incoming side of the gear mesh for pitch line velocities up to 8,000 fpm. Above 8,000 fpm, more oil is needed for cooling than for lubricating, and the oil flow removes heat best by being directed at the outgoing side of the gear mesh where the oil jets can strike the hot, drive-side of the gear teeth.

For very high-speed gears[6] (above 16,000 fpm), there is a danger that the amount of oil carried to the incoming side of the gear mesh may be inadequate, and it is prudent to add a supplementary flow at the incoming side of the gear mesh. Generally, about 2/3 of the oil flow should be supplied to the outgoing side of the mesh for cooling, and 1/3 of the flow directed at the incoming side for lubrication. The placement of the oil jets is a crucial factor when pitch line velocities exceed 20,000 fpm. At speeds this high, experiments are required to find the optimum number and location for the oil jets.

In pressure-fed systems, the following parameters must be considered to ensure adequate lubrication and cooling of the gear mesh: Quantity of flow, jet size, feed pressure, and number of jets. There are general guidelines, based on experience and experimentation, for specifying these parameters, but each application must be evaluated independently based on its particular operating conditions and requirements.

An empirical equation used to calculate the quantity of oil flow in gallons per minute is

$$q = P/c$$

where c is taken from Table 1

P = transmitted power, hp

q = oil flow rate, gpm

For a typical industrial application transmitting 200 hp, where weight is not critical, the designer might choose the constant c = 200hp/gpm, resulting in a copious flow of 1 gpm. On the other hand, for a high efficiency aviation application transmitting 200 hp, where weight is critical, c = 800 might be chosen, resulting in a lean flow of 0.25 gpm. Some applications may require different flow rates than those given by Table 1. For instance, wide-face, high-speed

Table 1 - Typical Oil Flows Per Gear Mesh

c	Flow	Comment
(hp/gpm)	Conditions	General industrial
200	Copious	Typical aviation
400	Adequate	Light-weight, high-
800	Lean	efficiency aviation
1000	Starved	Only for unusual
		conditions

gearing may require a higher flow rate to ensure uniform cooling and full-face coverage.

The proper jet size, feed pressure, and number of jets must be determined to maintain the proper flow rate, jet velocity, and full-face coverage. The diameter of a jet can be calculated for a given flow rate and pressure based on the viscosity of the oil at the operating temperature.[7] There are practical limitations on jet size, and the minimum recommended size is 0.03". If a jet smaller than this is used, contaminants in the oil may clog it. Typical jet diameters range from 0.03"- 0.12".

The feed pressure determines the jet velocity, which in turn determines the amount of oil that penetrates the gear mesh. Typical feed pressures range from 20-100 psig. Industrial application feed pressures are typically 30 psig, and high-speed aerospace applications are typically 100 psig. In general, the higher the pressure, the greater the cooling,[8] but the higher the pressure, the smaller the jet diameter. Therefore, pressure is limited by the minimum recommended jet diameter of 0.03".

The number of jets should be sufficient to provide complete lubrication coverage of the face width. More than one jet for each gear mesh is advisable because of the possibility of clogging. The upper limit on the number of jets is determined by the flow rate and jet diameter; too many jets for a given flow rate will result in a jet diameter less than the minimum recommended.

Case History

In an industrial application, 24 speed-increaser gearboxes were used to transmit 346 horsepower and increase speed from 55 rpm to 375 rpm. The gears were parallel-shaft, single helical, carburized, and ground. The splash lubrication system used a mineral oil without anti-scuff additives with ISO 100 viscosity. After about 250 hours of operation, two gearboxes failed by bending fatigue. The gear tooth profiles were so badly worn determining the primary failure mode was impossible. Three other gearboxes with less service were selected for inspection. One had logged 15 hours, while the other two had operated for 65 hours each. Upon disassembly, no broken teeth were found, but all three

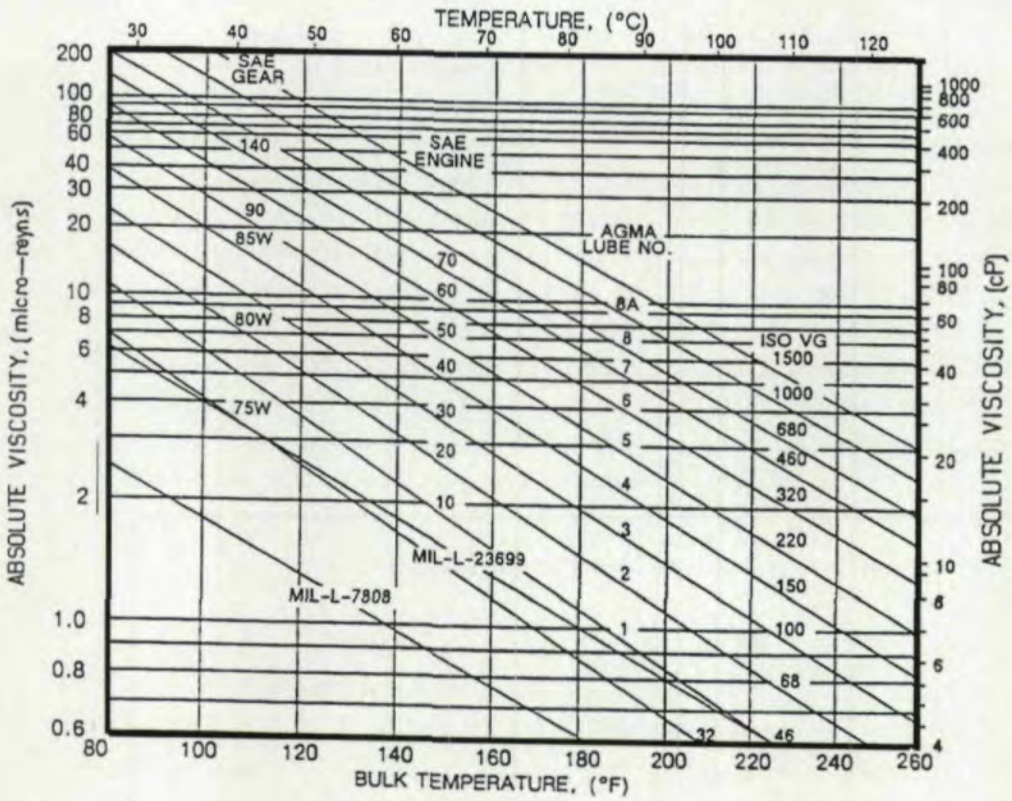

Fig. 2 - Absolute viscosity versus temperature for mineral gear lubricants with a viscosity index of 95. (From AGMA 2001-B88, 1988.)

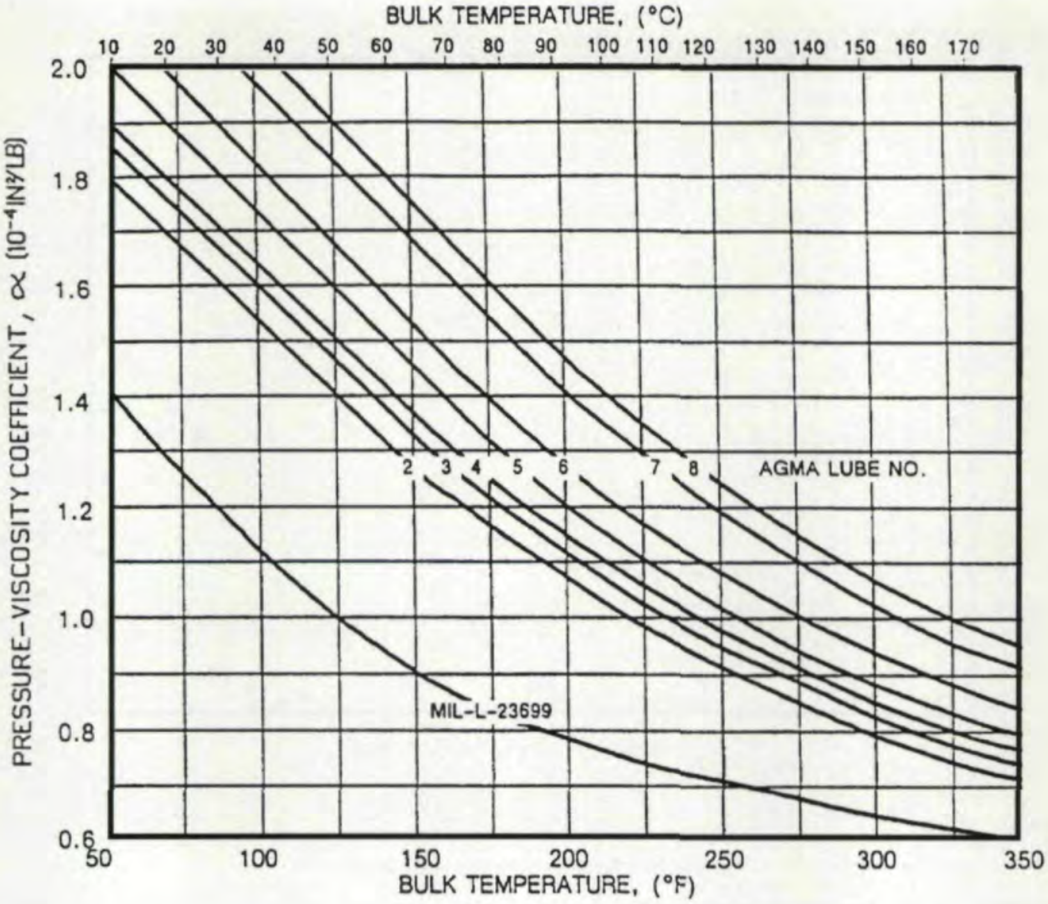

Fig. 3 - Pressure-viscosity coefficient versus temperature for mineral gear lubricants. (From AGMA 2001-B88, 1988.)

gearboxes had scuffed gear teeth. The primary failure mode was scuffing, and the earlier bending fatigue failures were caused by dynamic loads generated by the worn gear teeth. Subsequent inspection of the remaining gearboxes revealed that all had scuffing damage, which probably had occurred immediately upon start-up because the loads were not reduced during run-in.

Fortunately, a prototype gearbox had been run at 1/2 load for about 50 hours. When these gears were inspected, no signs of distress were seen on any of the gear teeth. The tooth profiles were smooth, with surface roughness estimated to be 20 μin rms, and the contact pattern indicated 100% face contact. This gearbox was reassembled and run under 1/2 load until its oil sump temperature reached equilibrium at 200°F. For this application, the ambient temperature was in the range of 50°F to 125°F. The center distance of the gears was 16 inches and the pitch line velocity was 400 fpm. Referring to AGMA 250.04, the recommended viscosity for these conditions is ISO 150 or ISO 220.

Using the empirical equation we get:

$$\nu_{40} = \frac{7000}{(400)^{0.5}} = 350 \text{ cSt} \qquad (3)$$

Hence, the empirical equation recommends a viscosity close to ISO 320. It is apparent that the viscosity that was originally supplied (ISO VG 100) was too low.

The EHD film thickness was calculated with a special computer program.[9] The gear bulk temperature was assumed to be 230°F (30 degrees hotter than the measured oil sump temperature). The following data for the ISO VG 100 lubricant was obtained from Figs. 2 and 3:

$\mu_0 = 6.6$ cP$(0.96 \times 10^{-6}$ Reyns)
$\alpha = 1.02 \times 10^{-4}\ in^2 lb$

Fig. 4 shows a plot of the film thickness versus position on the pinion tooth. The minimum film thickness occurs low on the pinion

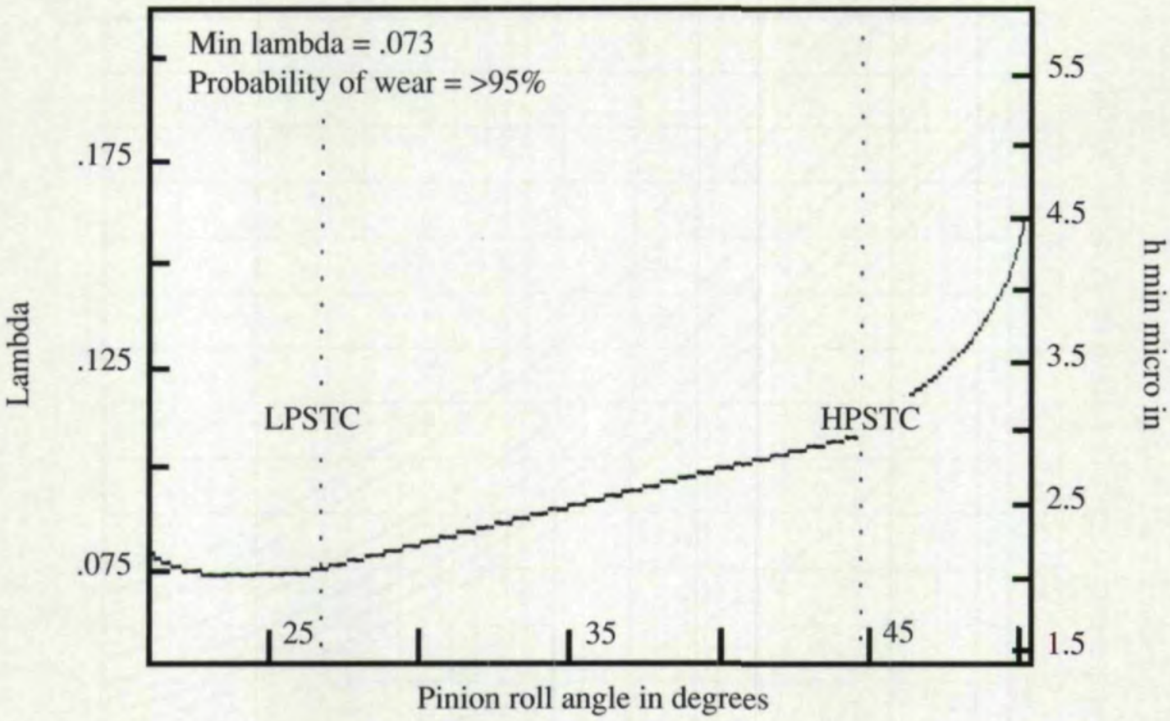

Fig. 4 - Film thickness versus pinion roll angle for gear tooth geometry of scuffed gearset.

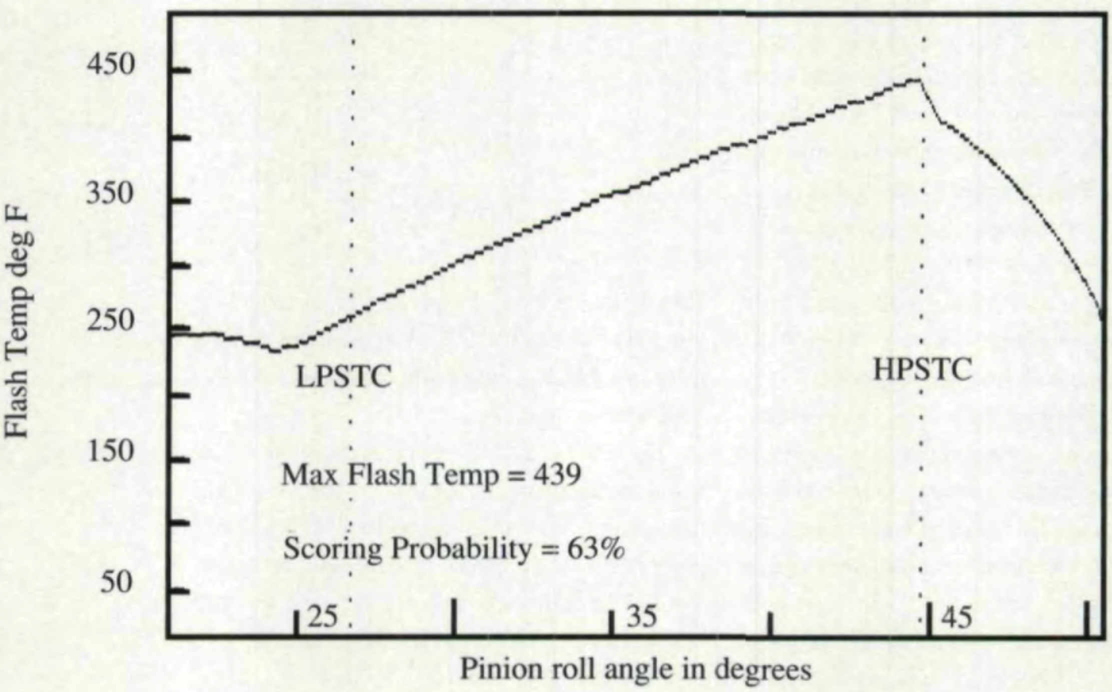

Fig. 5 - Flash temperature versus pinion roll angle for gear tooth geometry of scuffed gearset.

tooth near the lowest point of single tooth contact (LPSTC) where h_{min} = 2.1 micro inches. The specific film thickness, based on 20 μin rms surface roughness for both profiles, is $\lambda = 0.073$. Fig. 1 shows that the gears operate in the boundary lubrication regime. The program predicts that the probability of wear is greater than 95%.

The contact temperature was also calculated with the program. The scuffing temperature for the ISO BG 100 lubricant was calculated with the equation for non-anti-scuff mineral oils:

$$Ts = 146 + 59\ln(100) = 418°F$$

Fig. 5 shows a plot of the contact temperature versus position on the pinion tooth. The maximum contact temperature occurs high on the pinion tooth near the highest point of single tooth contact (HPSTC) where Tc = 439°F. The program predicts that the probability of scuffing is 63%. This is considered to be a high risk of scuffing. The relatively high temperature peak near the tip of the pinion tooth was caused by the geometry of the gears. The designer selected a long addendum tooth for the pinion. Long addendum pinions perform well in speed reducers, where they increase the amount of recess action and decrease the amount of approach action of the gear mesh. Since recess action is much smoother than approach action, long addendum pinions give speed reducers smooth meshing characteristics. When operated as a speed increaser, however, the approach and recess portions of the gear mesh reverse, making a long addendum pinion rough running and vulnerable to scuffing.

To explore the possibilities for reducing the scuffing risk, new gear tooth geometry was proposed with the pinion and gear addenda designed to minimize the flash temperature rise. The new gearset, analyzed with the program, assumed the lubricant was a mineral oil with anti-scuff additives, with a viscosity of ISO 220, and with the following properties:

$$\mu 0 = 10 \text{ cP}(1.45 \times 10^{-6} \text{ Reyns})$$

$$\alpha = 1.09 \times 10^{-4} \text{ in}^2/\text{lb}$$

$$Ts = 245 + 59\ln(220) = 563°F$$

Fig. 6 shows that the film thickness increases to h_{min} = 2.7 μin, and the specific film thickness increases to $\lambda = 0.097$. Fig. 1 shows that the gears still operate in the boundary lubrication regime, however, the probability of wear is reduced to 94%. Fig. 7 shows that the optimized gear geometry reduced the maximum contact temperature to Tc = 302°F. The combination of reduced contact

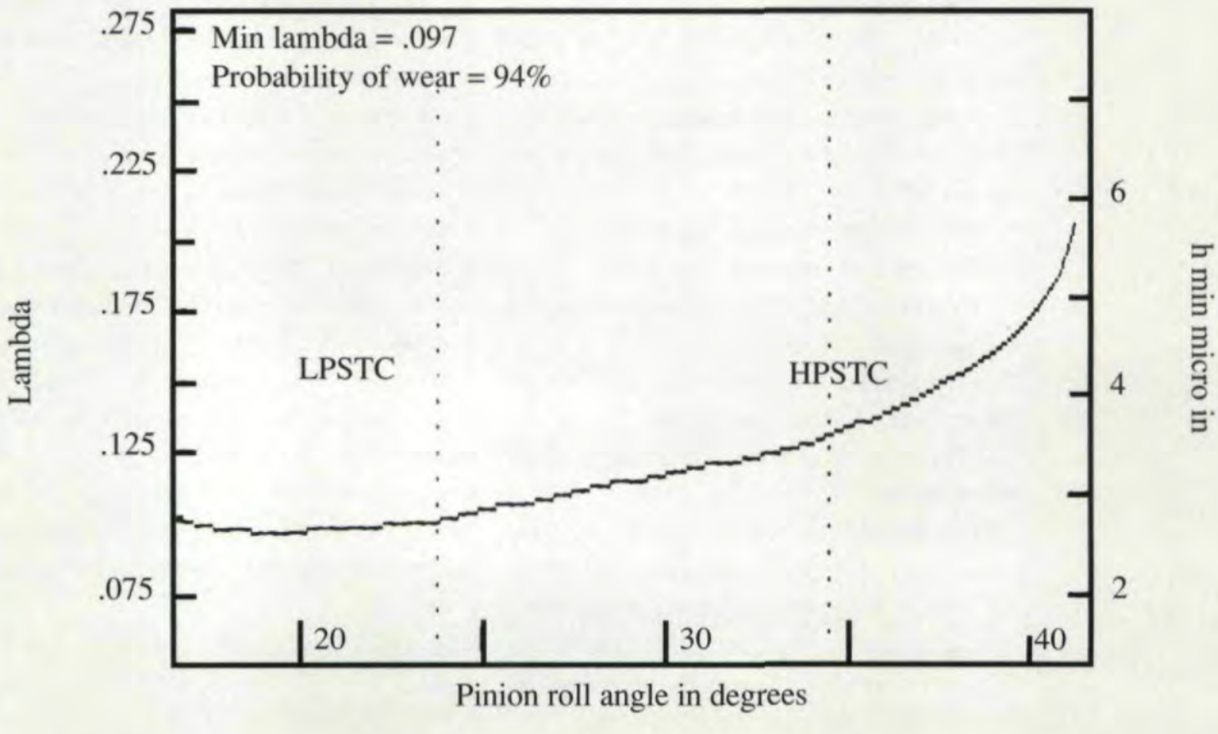

Fig. 6 - Film thickness versus pinion roll angle for gear tooth geometry optimized for maximum scuffing resistance.

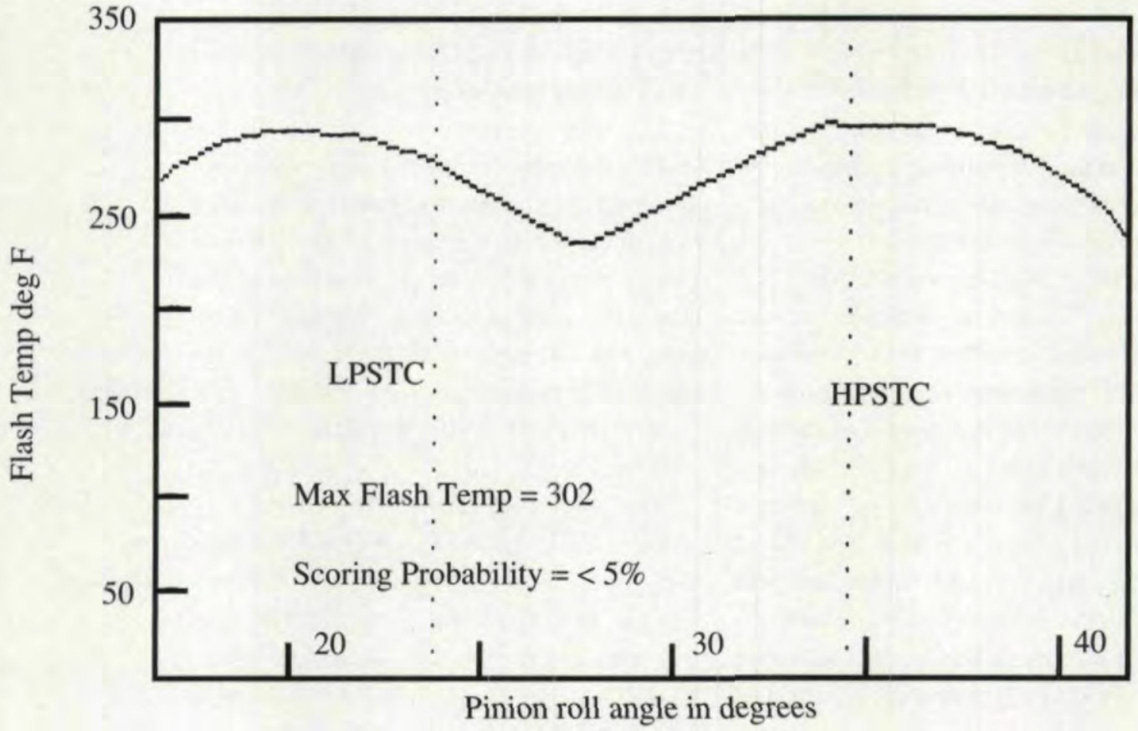

Fig. 7 - Flash temperature versus pinion roll angle for gear tooth geometry optimized for maximum scuffing resistance.

temperature and the increased scuffing resistance provided by the higher viscosity mineral oil with anti-scuff additives reduces the scuffing probability to < 5%.

Typical of many gear failures, this case history shows that several factors contributed to the failures:

•The lubricant viscosity was too low.

•No anti-scuff additives were used.

•A gearbox designed as a speed reducer was used as a speed increaser.

•The gear teeth were provided with a coating or plating to ease running-in.

•The gears were not run-in properly under reduced loads.

Gear failures, as exemplified by the case history, can be avoided if designers and operators recognize that the lubricant is an important component of a gearbox, and appreciate that the tribology of gearing requires the consideration and control of many interrelated factors.

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