LUBRICATION IN HELICAL GEARS

G. MORDUKHOVICH
General Motors Powertrain, Advanced Power Transfer, 30240 Oak Creek Drive, Wixom, Michigan, 48393, USA;
e-mail: gregory.mordukhovich@gm.com

N. ANDERSON
General Motors Powertrain, Advanced Power Transfer, 30240 Oak Creek Drive, Wixom, Michigan, 48393, USA;
e-mail: neil.e.anderson@gm.com

SUMMARY
A method to determine elastohydrodynamic film thickness in helical gears is developed by combining the Dowson-Higginson EHD formulation with helical gear geometry and kinematics. Comparisons are made to traditional gear film thickness models. This analysis is then used to characterize film thickness and lambda ratio in automotive planetary gearset. Methods to measure surface roughness in fine pitch gears are described.

Keywords: Gear, Lubrication, Rough Surfaces, Pitting

NOTATION
\( n_p \) Input pinion speed, rpm
\( d \) Pinion operational pitch diameter, m.
\( \Phi_t \) Operating transverse pressure angle, degrees
\( \omega_p \) Pinion angular velocity rad/sec
\( r \) Pinion operating pitch radius, m
\( \Psi \) Operating helix angle, degrees
\( \Phi_n \) Operating normal pressure angle (NPA)
\( V_{fp} \) Lengthwise pinion velocity at pitch line, m/sec
\( V_{rp} \) Profile pinion velocity at pitch line, m/sec
\( V_{fg} \) Lengthwise gear velocity at pitch line, m/sec
\( V_{pg} \) Profile pinion velocity at pitch line, m/sec
\( R_e \) Effective Radius, m
\( \alpha \) Angle between the total velocity vector and profile velocity in the tangent plane, degrees
\( C \) Operating center distance, m
\( z \) Gear reduction ratio
\( \alpha_0 \) Pressure/viscosity coefficient, m\(^2\)/N
\( \eta_0 \) Viscosity, Pa sec

1 INTRODUCTION
Elastohydrodynamic (EHD) lubrication analyses have been used to determine bearing operating film thickness for many years [1]. These analyses have also been applied to aerospace gears with some success [2]. There is evidence of successful operation, i.e., little or no metal-to-metal contact when the calculated film thickness is well below composite roughness [3]. Such evidence suggests the need to review film thickness calculations and lubrication regime definitions. In this work these methods are revised to properly account for helical gear geometry and kinematics.

Similarly, surface roughness measurements are routinely made on bearing components and coarse pitch gears. Measurements on fine pitch gears, typically used in automotive transmissions, however, present a challenge due to limited space available to make these measurements. A non-destructive method to obtain these measurements was developed and is described herein.

Finally, with these tools in hand, the lambda ratio (ratio of film thickness to composite surface roughness) was determined to characterize the lubrication conditions in fine pitch automotive gears.

2 FILM THICKNESS CALCULATION

2.1 Contact in spur and helical gears
Contact line motion is one of the most important differences between spur and helical gears. The line of contact in spur gears travels from the start to the end of contact across the tooth in the profile direction only. The line of contact in helical gears travels in the profile and facewidth direction simultaneously. Rolling motion occurs only in the profile direction in spur gears. In helical gears, rolling motion exists both in the profile and facewidth directions [2]. This difference tends to increase film thickness in helical gears as will be shown below.

2.2 Film thickness equation
The Dowson-Higginson film thickness equation [4] was selected for this study:

\[
h = 1.6 \alpha_0^{0.6} (\eta_0 \mu)^{0.7} (E')^{0.03} R_e^{0.43} w^{-0.13}
\]

The Dowson-Higginson method was selected because of wide acceptance in the bearing and gearing industries, and it is one of the most conservative estimates.

This equation can be used directly except for the velocity and equivalent radius of curvature terms, which are developed below.

2.3 Helical gear velocities
Coleman determined velocities in helical gears as part of a sliding velocity study in [5]. Velocities were given as vector projections in the tangent plane.

The total velocity projection in the tangent plane can be split into facewidth and profile velocities, \( V_f \) and \( V_p \). These projections were used for sliding velocity calculations in [5] and can also be used for determination of rolling velocity (Fig.1). The total rolling velocity can be found using equations 2 through 6 from [5] as follows:

\[
V_r = \alpha_p r \cos \Psi
\]
Tangential velocity vector in the tangent plane at pitch line.

\[ V_{tp} = V_{tg} = V_n \tan \Psi \]  \hspace{1cm} (3)  
\[ V_{pp} = V_{pg} = V_n \sin \Phi \]  \hspace{1cm} (4)  
\[ V_{tp} = \omega_p r \cos \Psi \tan \Psi = \omega_p r \sin \Psi \]  \hspace{1cm} (5)  
\[ V_{pp} = V_{pg} = \omega_p r \cos \Psi \sin \Phi \]  \hspace{1cm} (6)  
\[ V_{FR} = 2 \frac{V_{tp}}{2} = \omega_p r \sin \Psi \]  \hspace{1cm} (7)  
\[ V_{PR} = 2 \frac{V_{pp}}{2} = \omega_p r \cos \Psi \sin \Phi \]  \hspace{1cm} (8)  
\[ V_{Rolling} = \left( V_{FR}^2 + V_{PR}^2 \right)^{0.5} \]  \hspace{1cm} (9)  
\[ V = \pi n_p d \left( 1 - \cos^2 \Phi \cos^2 \Psi \right)^{0.5} / 60 \]  \hspace{1cm} (10)  

**Fig. 1** Contact line velocities in helical gears.

### 2.4 Effective radius of curvature

Let us consider a plane perpendicular to the tangent plane containing the projection of the total velocity vector. Radii of curvature in this plane are the correct values for helical gears. By using these values an additional increase in film thickness is seen.

\[ R_e = (C \sin \Phi_n / \cos^2 \alpha) (z/z+1)^2 \]  \hspace{1cm} (12)  
\[ \alpha = \tan \left( \tan \Psi / \sin \Phi_n \right) \]  \hspace{1cm} (13)  

### 2.5 Modified vs. standard film thickness

Rolling velocity and effective radius of curvature as defined in [6] were used to compare the proposed changes in minimum film thickness in a helical gear mesh. The values from [6] are labeled “standard”.

\[ V = \pi n_p d \sin \Phi_n / 60 \]  \hspace{1cm} (14)  
\[ R_e = (C \sin \Phi_n / \cos^2 \Psi) (z/z+1)^2 \]  \hspace{1cm} (15)  

All other parameters in the Dowson-Higginson equation were kept constant. The difference in film thickness between the standard and the modified methods was up to 4.7 times higher for the modified method for a given range (Fig. 2). The difference between the two approaches decreases when the helical angle approaches zero. For spur gears, both equations give the same value of film thickness. The difference is maximum for the lowest pressure angle and the highest helical angle gears. High-pressure angle gears have smaller differences. This can be explained by the more dominant role of the profile rolling velocity in high-pressure gears. The difference between profile velocities in the normal and transverse planes is quite small.

Low-pressure angle gears show greater differences in the two approaches. Here, the face-rolling velocity is dominant and this adds to the curvature effect.

The difference between helical angle and \( \alpha \) angle increases more quickly in high-pressure angle gears as shown in Figure 2.

The standard film thickness expression is primarily driven by pressure angle with some impact from helical angle (Figure 3).

The modified film thickness is primarily driven by helical angle with some impact from pressure angle (Figure 4).
3 SURFACE ROUGHNESS INSPECTION

3.1 Inspection method comparison

A number of surface inspection methods were considered for this study but the optical and stylus methods were determined to be the most suitable for a production environment.

The stylus method is most commonly used but has the following problems:

1. Radius of the stylus tip acts as a short cut off which affects sensitivity. When a small tip radius is chosen, contact stresses in contact increase. These increases in contact stress lead to increased plastic deformation in contact.

2. Statistically stable values are reached by dividing the measurement length into small traces equal to the selected cut-off length. Homogeneity of the texture through the whole length of the trace is the underlying assumption. This assumption is not valid for shaved or honed gears, where the operating pitch radius of the cutter is located along the flank of the gear. See figure 5.

3. Multiple inspection traces are necessary to get enough data for statistical significance.

The optical interferometric method has a 1 nm vertical resolution, a 2 \( \mu \)m lateral resolution, and area measurement capability [6]. By taking measurements over a small area the texture is relatively homogeneous. The optical method was selected for this study for these reasons.

3.2 Replicating techniques

Direct optical surface finish measurement of fine pitch gears is difficult because of line of sight problems. Space limitations between teeth prevent positioning the surface perpendicular to the light beam. Angular positioning leads up to 20% loss of the reflected light. It became necessary to use replicas to get surface finish measurements in fine pitch gears or determination of accumulative damage without cutting out gear teeth. The following replicating techniques were considered: Plastic Tape and Acetone.

1. Aluminum Foil
2. Silicone Rubber - SILFLO
3. Tempanol - Temporary Filling Material
4. Thin Set - Temporary Cap and Crown Cement

5. Timken
6. GM Method (modified Timken method).

Repeatability problems were found with the first five methods and they could not be used. The Timken Method is very reliable and accurate but certain materials required for proper replication and reflection were very difficult to prepare properly. Replicas produced with the GM method were much easier to work with. This methodology was developed by GM and was used to monitor gear surface finish changes through testing.

3.3 Surface Finish Inspection Results

Five zones along the path of contact were monitored during accelerated bench testing (Fig.6 and 7).

Fig. 6 Pinion Ra in the profile direction.

Fig. 7 Sun gear Ra in the profile direction.

Results were obtained before the test, after 1.5 hrs. run-in at 50% load, after 3, 40 and 60 hrs. The test criteria, 1 mm² pit was reached at 60 hrs. The Ohio State University “Load Distribution Program” (LDP) was used to determine the long cut-off values in the profile direction. A 0.162mm long cut-off was chosen. This value is equal to the Hertzian contact width at the pitch line at the test load. The original surface roughness changes from the SAP to the tip and is a function of relative position of the operating pitch line during shaving in the gear mesh (Fig. 6 and 7).

Fig. 8 Gear tooth flank condition after run-in.
Surface roughness changes drastically after run-in and then stabilizes in each zone (Fig. 6, 7, 8, and 9). It appears that heavy metal-to-metal contact exists only during run-in and good fluid film separation occurs through the balance of the test with only mild wear.

![Fig. 9 Gear tooth flank condition after 60 Hrs.]

4 LAMBDA RATIO

Lambda ratio ($\lambda$) or specific film thickness is a ratio between the minimum film thickness and the composite roughness of two mating surfaces. $R_{a g}$ and $R_{ap}$ are the average roughness values for the gear and pinion respectively

$$\lambda = h_{min} / S ; \quad S = (R_{ag}^2 + R_{ap}^2)^{0.5}$$

Lambda is based on the non-deformed, “free state” roughness. In an actual gear mesh the roughness asperities might elastically deform thus reducing the effective surface roughness. This might explain why metal-to-metal contact is sometimes not observed when the film thickness is comparable with the composite surface roughness of the gear pair [3]. This may also explain the increase in relative pitting life as $\lambda$ increases to 3 [2].

The application of the proposed film thickness equations to a typical automotive gearset is shown in Figure 10. Lambda was evaluated for both the as-machined condition and after run-in since there was a significant difference in these roughness values. The as-machined value moved from the boundary to the mixed lubrication regime and the run-in value moved from the lower to the upper end of mixed lubrication. This better reflects the condition of the test gears and suggests that smooth contact EHL equations are more appropriate for this case. Also, different life improvement methods must be targeted for each lubrication regime. These methods range from extreme pressure to mild anti-wear additives [6]. The proposed methodology will allow selection of a better strategy for contact fatigue life improvement in helical gears.

5 CONCLUSIONS

Rolling velocity and equivalent radius of curvature expressions used in standard helical gear film thickness prediction methods were re-evaluated. To properly account for rolling velocity, the axial component found in helical gears must be included. Also, the radius of curvature must be evaluated in the plane perpendicular to the actual contact. These effects significantly increase the predicted film thickness (up to 4.7 times) under certain conditions.

Surface roughness in fine-pitch gears can be determined with optical methods. An accurate surface replicating technique was developed to check surface finish of a running gear at various intervals of testing. In an automatic transmission gears test, surface finish changed after break-in and stabilized to very low values in the critical pitting area below the pitch point. These data indicate that smooth surface EHL equations can be used after run-in.

Based on the observed condition of the test gear surfaces after running, the proposed helical gear EHL equation has better correlation with the test results than the standard equation.

![Specific film thickness, \(\bullet\)]

1. Standard equation as machined: 0.25
2. Modified equation as machined: 0.40
3. Standard equation after run in: 0.46
4. Modified equation after run in: 0.75

Fig. 10 Lambda ratio vs. Relative life.

The proposed EHL equation may allow selection of more appropriate methods to improve contact fatigue life. The initial surface finish effect on changes in roughness after run-in and pitting fatigue life requires additional study.

6 ACKNOWLEDGEMENTS

The authors wish to acknowledge the support provided by GM Powertrain Advanced Power Transfer, GM R&D and GMPT Materials Engineering personnel. Thank you all for your support in this project. The special thanks to Dr. A. Singh, GMPT for his constant support and comments on this work, Dr. H. Nixon and Dr. D. Cogdell of Timken Corp. for their collaboration in replicating techniques research.

7 REFERENCES