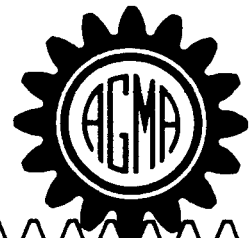


Modeling and Measurement of Sliding Friction for Gear Analysis

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American Gear Manufacturers Association



TECHNICAL PAPER

Modeling and Measurement of Sliding Friction for Gear Analysis

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[The statements and opinions contained herein are those of the author and should not be construed as an official action or opinion of the American Gear Manufacturers Association.]

Abstract

This paper discusses background studies needed for the prediction of sliding resistance on gear teeth. Various elastohydrodynamic and mixed lubrication theories for coefficient of traction are examined. These theories are evaluated with respect to experimental results from two-disk tests for several parameters that simulate the continuously varying properties during gear engagement. For one mesh cycle, coefficient of friction for each tooth in contact is predicted as a function of roll angle. Normal load is estimated by the Load Distribution Program and is combined with friction coefficient to compute the total friction force. Dynamics tests are carried out on a pair of gears to measure shaft displacement in the off-line of action direction. Using the bearing stiffness and shaft motions, measured friction force is calculated and compared with analytical results predicted from lubrication principles. Frequency response of the test rig is used to eliminate the influence of cross-coupling terms with line-of-action forces and filters are applied to the measured data to curtail the dynamic effects of the test rig. Based on the comparison of theoretical predictions and test results on a spur gear pair, the lubrication models are evaluated, with special emphasis being placed on the dynamic modeling of friction.

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MODELING AND MEASUREMENT OF SLIDING FRICTION FOR GEAR ANALYSIS

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1. Introduction

Sliding resistance in gear meshing has a considerable effect on off-line-of-action dynamics. Most existing studies [1] have assumed simplified Coulomb friction to model this phenomenon. In reality, the lubrication regime includes elastohydrodynamic and boundary lubrication. The coefficient of friction used in lubrication calculations depends upon parameters like radii of curvature, sliding speed and Hertzian pressure and varies significantly during a mesh cycle [2]. Furthermore, a unified theory for fluid rheological properties is not available and they all depend upon the particular application.

This article presents a critical analysis of various tribological principles [3, 4, 5, 6, 7, 8, 9] in the context of gear dynamic behavior. Experiments are specially designed to tackle the issue of sensitivity, arising due to low magnitude of friction force as compared to the torque transmitting loads. With this information, a comparison could be made between the disk tests and friction measured in a running gear. The validity and limitations of this comparison are discussed.

A preliminary evaluation of lubrication models is done using two-disk tests. Results of these tribology experiments are applied for modeling sliding forces in a gear mesh. Friction forces are computed from load distribution on the gear teeth and instantaneous coefficients of friction. To validate this model, dynamics tests are carried out on a gear pair. Shaft displacement is measured in the off-line of action *direction and combined with bearing stiffness* to get the friction force. Analytical methods are applied to minimize the dynamic and cross-coupling effects of the test rig.

2. Methodology

Two lubricants with distinct properties are tested on a two-disk friction testing machine. The coefficient

of traction is studied for four different parameters, namely load, rolling speed, slide-to-roll ratio and temperature. The parameters are selected such that they simulate the mechanism of gear meshing under typical conditions. From the comparison of experimental and predicted coefficients of friction, certain formulations are selected on the basis of their applicability over this range of parameters.

Subsequently, a spur gear pair is modeled as a quasi-static system. A variable center distance test rig is used to measure shaft deflection under different torque conditions. For quasi-static assumption to hold, the dynamic effects need to be minimized. This is achieved by restricting the tests to low rotational speeds. Frequency response of the test rig is used to attenuate the signal near resonant conditions. From transmissibility studies, it is shown that the coupling between line of action (LOA) forces, resulting from transmission error or mesh stiffness variation, and off-line of action (OLOA) displacement is negligible. Thus the shaft displacement in OLOA direction shall directly provide the information on sliding forces.

Finally, different lubrication models are applied to the gear mesh, using the time varying parameters like Hertzian pressure and sliding velocity in the mesh cycle. The meshing load, as predicted by the Load Distribution Program [10], is multiplied with the traction coefficient to get the combined friction force on all teeth in contact. This force is compared with the experimental values to evaluate different lubrication models. This modeling technique is expected to provide the basis of a full multi-degree of freedom dynamic model of gear meshing, including off-line of action forces.

The overall methodology that is applied in this paper is shown as a flowchart (Fig. 1).

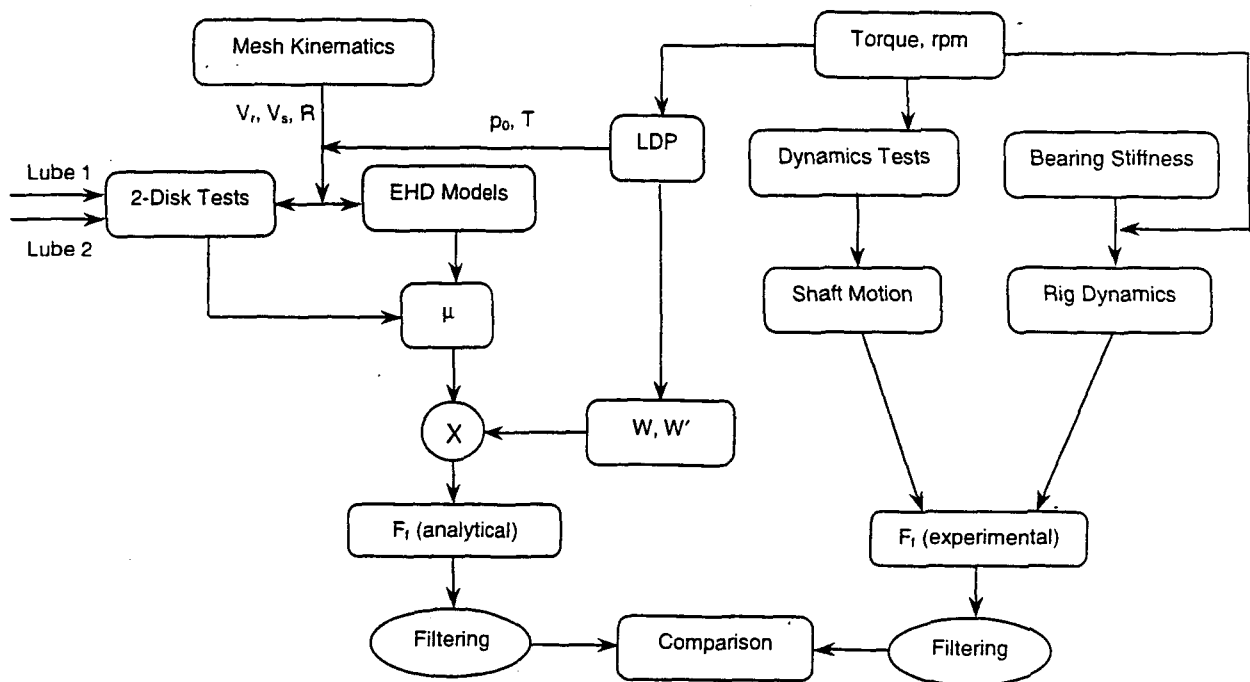


Figure 1. Flowchart showing the analysis method

3. Two-Disk Friction Tests

For testing rheological properties of fluids, 2-disk experiments are usually applied to emulate the varying gear mesh conditions. Here, the two-disk friction test rig (Fig. 2) located in the Tribology Lab at the Ohio State University has been used to test two different lubricants, Automatic Transmission Fluid (fluid A) and high viscosity gear oil (fluid B). The selection of lubricants was based on their widely distinct properties (Table 1), so the friction effects measured during gear tests should have pronounced differences.

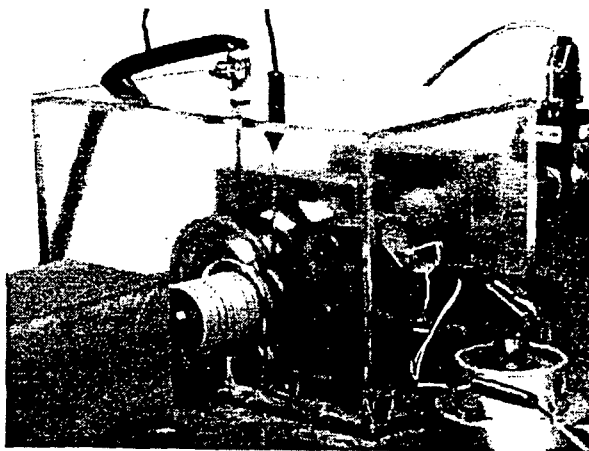


Figure 2. Two-disk lubricant test rig

The rig consists of two disks of radii ¼ inch and 1.5 inches, each crowned at a radius of 3 inches across the face. The disks have a hardness of 50 HRC and 20 μ-in Ra surface roughness value. The friction disks are propelled by two motors whose individual speeds can be varied using a controller. Normal load between the disks is applied pneumatically and is measured using a load cell. The other parameters that are recorded are the inlet temperature and output torque. The coefficient of friction is calculated from the output torque and the normal load.

Table 1. Properties of test lubricants

		Fluid A	Fluid B
Kinematic viscosity (40C)	cSt	36	460
Viscosity index	/°C	184	95
Pressure-viscosity coeff	/Pa	2.2e-8	2.2e-8
AGMA number		-	7 EP

Since most lubrication models are relevant only over a limited range of parameter values, the required range for a gear application is established. In this study, a unity ratio spur gear pair with 28 teeth, quarter inch face width and diametral pitch of 8, was analyzed. These gears have a profile contact ratio of 1.638. For any given running condition, say 1200 rpm and 900 in-lb, the various meshing parameters can be estimated (Table 2).

Table 2. Gear mesh parameters

(a) At beginning of contact

		Pinion	Gear
Radius of curvature	in	0.296	0.901
Effective Radius	in	0.223	
Surface speed	in/s	37.2	113.2
Mean rolling speed	in/s	75.2	
Sliding speed	in/s	76.0	
Hertz contact stress	psi	-	

(b) At the pitch point

		Pinion	Gear
Radius of curvature	in	0.599	0.599
Effective Radius	in	0.299	
Surface speed	in/s	75.2	75.2
Mean rolling speed	in/s	75.2	
Sliding speed	in/s	0.0	
Hertz contact stress	psi	200,000	

From these representative values, parameters for any running condition can be estimated, since all velocities are proportional to the rpm and the contact stresses are proportional to (Torque)^{1/3}. To simulate a wide range of speeds and torque, within the constraints of the test machine, the following conditions were selected (Table 3). Only one controlled parameter was varied at a time and others were approximately held at baseline values.

Table 3. Parameters for 2-disk tests

		Range	Baseline value
Mean rolling speed	in/s	12 – 150	97.5
Slide-to-roll ratio		0 – 1.2	1.08
Load (Hertz stress)	lb (ksi)	11 – 135	67 (257)
Inlet temperature	°C	33 – 73	33
Lubricant		A, B	

Figure 3 depicts the behavior of the coefficient of friction (μ) for the above conditions for both the lubricants. The four parameters that were independently varied are the rolling velocity, slide-to-roll ratio, normal load and temperature. In a gear mesh, these would correspond to the rpm, position within a mesh cycle, torque and gearbox design respectively. μ has been calculated from the average of 100 points, sampled at 4 Hz.

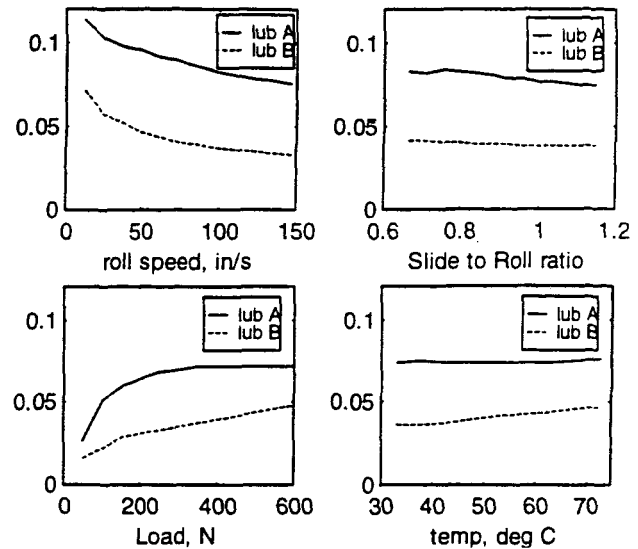


Figure 3. Variation in coefficient of friction for lubricants A and B

It can be seen from Figure 3 that for all the conditions, ATF has a value of μ nearly one-half of that of fluid B. Hence, it can be deduced that viscosity of the lubricant is one of the primary determinants of its traction properties. These graphs also show a strong dependence on rolling velocity and contact stress. On the other hand, μ increases very gradually as the inlet temperature is raised, and drops with increasing specific sliding velocity. All four graphs show a asymptotic behavior at extreme conditions. In the following section, these results will be evaluated against the available theoretical predictions.

4. Lubricant Formulations

Gear tribology depends upon various parameters, including the surface roughness, lubricant rheology and meshing conditions. During one tooth roll cycle, sliding friction can vary significantly due to changes in speed, load distribution and surface profile. As a result, the lubrication regime may oscillate between elastohydrodynamic to thin film boundary lubrication. Miscellaneous theories exist under such varying conditions, which may depend upon totally different sets of parameters. In particular, equations for gear friction have long been debated because of the lack of reliable friction measurement. This problem arose due to the small magnitude of friction forces and the dynamic effects during a gear mesh [11].

The following table (see Table 4) gives a summary of some common formulations to calculate the coefficient of friction, along with the factors that contribute to its calculation. First, the minimum fluid film thickness is calculated, which determines the applicable lubrication regime. Two alternative formulations are presented here (list of symbols is provided at the end of the paper):