



AGMA Technical Paper

Impact of Surface Condition and Lubricant on Effective Gear Tooth Friction Coefficient

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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

The actual power loss due to friction in a gear pair is of significant interest to the transmission community. This loss impacts fuel efficiency on almost every form of transportation. In aircraft applications, this loss converts to heat, which must be removed using cooling equipment, inherently detracting from the payload of the aircraft. Consequently, the impact of gear tooth surface characteristics and lubricant on friction and frictional losses at the gear mesh is of significant interest to designers of high performance gear boxes. While a prior paper [1] compared the change in losses due to these factors, this effort is intended to actually evaluate the effective coefficient of friction at the gear tooth flank under the conditions described below.

By instrumenting the input and output ends of the test box of a four-square, power re-circulating gear test rig with high-accuracy torque transducers, losses at the meshing gear pair in the test box are measured. The difference between the readings of the two torque transducers are the losses in the test gear box. By subtracting bearing and seal losses and neglecting windage losses, the friction losses at the test gear mesh are obtained. The nominal torque in the four-square loop is known, therefore the “effective coefficient of friction” at the tooth flank is calculated.

The gears evaluated in this study are 6.25-mm face width, 21-tooth, 4.233-module, 20-degree pressure angle spur gears. Gears were fabricated from AMS 6308 steel, gas carburized, and oil quenched, resulting in a surface hardness of 60–64 on the Rockwell C scale. This experimental procedure was carried out for three operating speeds (5000, 8100 and 10 000 rpm) and three different gear tooth flank conditions: ground, Isotropic Superfinished (ISF®), and Tungsten Diamond Like Carbon (W-DLC) coated. Surface roughness (R_a) on the tooth flank ranged from 0.06 to 0.29 μm . Two different gear lubricants (Mobil Jet Oil II/MIL-PRF-23699/ISO VG22 and Mobil SHC 626/ISO 12925-1 CKD/ISO VG68) and two different torque levels (96 and 192 N-m) are also considered in this study. This paper summarizes the results obtained and the effective coefficient of friction at the gear tooth flank under the conditions described above.

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INTRODUCTION

The impact of gear tooth surface roughness and advanced coatings on frictional losses in a gear mesh is of significant interest to the transmission community, as these losses reduce the efficiency of the drive system. Maximizing efficiency to save fuel has long been an objective in vehicle industries. In recent years, the power generation sector has realized that improved efficiency while maintaining robust, long-life gearboxes can improve profit margins. Further, in aircraft applications, these frictional losses are converted to heat that has to be removed from the system. This is done using large heat exchangers and by carrying larger-than-ideal quantities of lubricant onboard the aircraft. Both solutions subtract from the available payload of the aircraft. The impact of lubricant properties on frictional tooth mesh losses is also of interest. More importantly, the gearbox designer needs a simple method to estimate the losses due to friction in a gear box using known parameters, such as speeds, torques, and lubricant properties. The concept of experimentally computing an “effective coefficient of friction”, in a gear mesh that could be utilized in estimating total friction losses in a gear box, was considered a worthwhile effort.

A prior study [1] presented a comparison of frictional losses at the gear tooth flank for three different surface conditions and two lubricants, with the gears operating at three speeds and two different load conditions. The most exhaustive experimental study quantifying gear tooth friction is by Yoshizaki, et al. [2], in which spur gears with various geometries were operated in a power re-circulating test rig and frictional losses were measured. Various lubricants and additives were also evaluated, and tooth surface finishes ($R_{\max}$) ranging from 0.5 to 4 μm were considered. Britton, et al. [3] describes another experimental study that specifically evaluated the effect of superfinishing on gear tooth friction on a power re-circulating gear test rig. A 30% reduction in frictional losses is measured and documented. In another experimental study on gear tooth friction, Petry-Johnson, et al. [4] measure frictional losses in a power re-circulating test rig operating ground and chemically polished gears with two different tooth sizes in three different lubricants, which were further utilized to define guidelines for the design of gear meshes and transmissions. Martins, et al. [5] experimentally measured the friction coefficient in FZG (ground) gears utilizing two lubricants. Several attempts to model and predict the friction losses [6, 7, 8] are also evident in literature, where the experimental effort is utilized to correlate to analytical results.

Based on the available literature, a comprehensive experimental study to evaluate the effective coefficient of friction at the meshing tooth surfaces, resulting in a parameter that could be utilized in estimating frictional losses in a gear box—especially as it relates to aerospace gearing—was considered useful. In this study, the variables being evaluated include superfinishing and coated gear teeth against a ground base line. Two lubricants are also evaluated.

EXPERIMENTAL SETUP

The gears evaluated in this study are 6.25-mm face width, 21-tooth, 4.233-module, 20-degree pressure angle, 0.005-mm crown spur gears. These gears were fabricated from AMS 6308 steel, gas carburized, and oil quenched, resulting in a surface hardness of 60–64 on the Rockwell C scale. This experimental procedure was carried out for three operating speeds (5000, 8100 and 10 000 rpm), which were determined using a vibration survey.

Three different gear tooth flank conditions were used. The first group of test gears utilized an as-ground gear for both the driver and driven gear. The second group utilized a Tungsten Diamond Like Carbon

(W-DLC) coated driving gear that was previously Isotropic Superfinished (ISF®¹) against an ISF-driven gear. The third test group utilized an ISF gear for both the driver and driven gear. Two different gear lubricants (Mobil Jet Oil II/MIL-PRF-23699/ ISO VG22 and Mobil SHC 626/ISO 12925-1 CKD/ ISO VG68) and two different torque levels (96 and 192 N-m) are also considered in this study. All tests were conducted with a nominal oil temperature of 46 °C. Oil temperature was controlled by a closed-loop heater / chiller via feedback from a thermocouple on the bearing and gear mesh oil supply line. Test conditions are summarized in Table 1.

Table 1: Test Details

Oils	Gear Tooth Flank Conditions		Test Torques	Test Speeds	Oil Temperature
Mobil Jet Oil II (MIL-PRF-23699/ ISO VG22) Mobil SHC 626 (ISO 12925-1 CKD/ ISO VG68)	Specimen (Driver)	Mate (Driven)	96 N-m 192 N-m	5000 RPM (23.27 m/s) 8100 RPM (37.70 m/s) 10000 RPM (46.55 m/s)	46 °C
	As Ground	As Ground			
	ISF with W-DLC Coating	ISF			
	ISF	ISF			

After evaluating several alternate experimental configurations to establish the losses (T_{F1}) in the test gear box, the configuration, illustrated in Figure 1, was selected as the best alternative. A four-square, power-recirculating gear test rig was modified with additional instrumentation for the purpose of this study. The additional instrumentation consisted of incorporating two precision torque transducers (T_1 and T_2) to measure the input and output to the test gear box. The third torque transducer (T_3) to measure the total losses in the four-square loop had been incorporated earlier. Specifications for the torque transducers used for acquiring measurements T_1 and T_2 are given in Table 2.

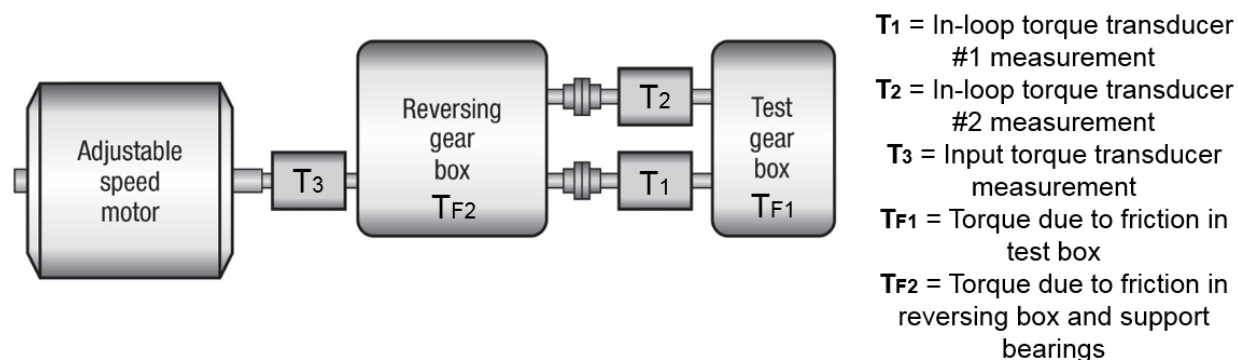


Figure 1: Schematic of the Experimental Setup

Table 2: Torque Transducer Specifications

Manufacturer	HBM
Model	T12
Maximum Test Torque	500 N-m
Measurement Resolution	18 bit resolution = .004 N-m
Measurement non-Linearity and Hysteresis	0.007% of range = 0.035 N-m
Max. Speed	12,000 RPM

¹ ISF® is a registered trademark of REM Chemicals Inc.