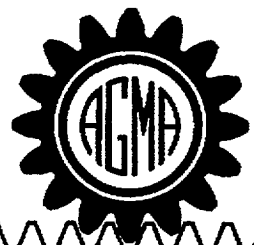


Dynamic Measurements of Gear Tooth Friction and Load

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American Gear Manufacturers Association



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ABSTRACT:

A program to experimentally and theoretically study fundamental mechanisms of gear dynamic behavior is being conducted at the NASA Lewis Research Center in support of a joint research program between NASA and the U. S. Army. This paper presents the results of dynamic tooth-fillet strain gage measurements from the NASA gear-noise rig, and it introduces a technique for using these measurements to separate the normal and tangential (friction) components of the load at the tooth contact.

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American Gear Manufacturers Association
1500 King Street, Suite 201
Alexandria, Virginia, 22314

October, 1991

ISBN: 1-55589-607-3

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1. INTRODUCTION

The dynamic forces at the point of tooth contact are of considerable interest to the designers of high-speed, light-weight gearing. Accurate prediction of the dynamic loads can assist in minimizing the size and weight of a transmission. In a helicopter application, where the transmission is a significant fraction of vehicle weight, such a reduction would be an important factor in overall vehicle performance.

A program to experimentally and theoretically study fundamental mechanisms of gear dynamic behavior is being undertaken at the NASA Lewis Research Center in support of a joint research program between NASA and the U.S. Army. This paper presents the results of dynamic tooth-fillet strain gage measurements from the NASA gear-noise rig, and it introduces a technique for using these measurements to separate the normal and tangential (friction) components of the load at the tooth contact. Resolution of the contact force is desirable for several reasons. Two of these reasons are the following:

(1) A primary output of analytical models of gear dynamic behavior is typically the normal force at the point of contact (e.g., [1] and [2]).

(2) The measurement of dynamic friction of meshing gears does not appear to have yet been carried out successfully.

An interesting trial was carried out by Benedict and Kelly [3], but it was discontinued because of dynamic response

problems. Anderson and Lowenthal [4] computed overall losses due to friction and found good agreement between theoretical predictions and experimental data. Krantz and Handschuh [5] applied a similar technique to an epicyclic gear rig, obtaining good correlation at low oil temperatures, but poorer correlation at higher oil temperatures. However, this technique cannot detect the variation in friction during the tooth engagement cycle. There is also the problem of separating the power loss due to gear tooth friction from power losses due to other sources such as bearings, windage, and so forth.

Extensive measurements of lubrication conditions at a sliding-rolling contact have been carried out on disk machines [6]. These experiments are of considerable value in confirming the existence of elastohydrodynamic lubrication and in identifying the separate regimes of lubrication that prevail under the various slide-to-roll ratios. However, the usefulness of the modes of behavior and friction coefficients in predicting lubrication conditions at an actual tooth contact, where the degree of sliding changes throughout the tooth engagement cycle (typical duration, 250 μ sec), needs to be verified. In this short period of time, large changes occur in the lubricant temperature, shear, and viscosity at pressures up to 1.4 GPa (200 000 lbf-in.²). Dyson [7] reported temperatures up to 400 °C and oscillatory shear rates up to 10⁷ sec⁻¹. These conditions cannot readily be produced outside of an actual tooth mesh.

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Friction at the tooth contact is important for determining not only power loss and efficiency, but also for understanding gear-tooth scoring and wear. An important parameter in scoring is the friction coefficient [3]. Friction greatly affects the heat input to the lubricant when sliding velocities are high.

This report presents dynamic, gear-tooth strain measurements from low-contact-ratio spur gears tested in the NASA gear-noise rig. The technique used to convert these strain measurements into normal and tangential (friction) tooth loads is described. Plots of normal and tangential forces, for both static and dynamic conditions, are presented for a representative range of loads and speeds. The normal force and dynamic strain data have been used to verify a gear dynamics code in another related report [8].

2. APPARATUS

2.1 Test Facility

These tests were conducted in the NASA Lewis gear-noise rig (Fig. 1). This rig comprises a simple gearbox powered by a 150-kW (200-hp) variable speed electric motor, with an eddy-current dynamometer that loads the output shaft. The gearbox can be operated at speeds up to 6000 rpm. The rig was built to carry out fundamental studies of gear noise and of dynamic behavior of gear systems. It was designed to allow testing of various configurations of gears, bearings, dampers, and supports.

A poly-V belt drive served as a speed increaser between the motor and input shaft. A soft coupling was installed on the input shaft to reduce input torque fluctuations caused by a nonuniformity of the belt at the splice.

Test gear parameters are shown in Table 1, test rig parameters in Table 2, and gear tooth profile traces in Fig. 2. The tooth surface roughness was measured by using an involute-gear-checking machine with a diamond stylus of approximately 10- μ m (0.0003-in.) radius. The surface roughness varied along the length of the tooth, with the region near the root appearing to be lightly polished. The maximum surface roughness was estimated to be 1.34 μ m (34 μ in.) peak-to-peak, or an average of 0.43 μ m (11 μ in.) (Fig. 3). The gear rig was operated at an oil fling-off temperature of 54 $\pm$ 2 $^{\circ}$ C (130 $\pm$ 5 $^{\circ}$ F). At the mean temperature of 54 $^{\circ}$ C, the viscosity of the synthetic oil (Table 2) used in the tests was 14 cSt (11.6 cP). Natural frequencies from a four degrees-of-freedom eigensolution [8] are also shown in Table 2.

2.2 Instrumentation

General-purpose, constantan foil, resistance strain gages (gage length, 0.38 mm (0.015 in.)) were installed in the

tooth-root fillets on both the loaded (tensile) and unloaded (compression) side of two adjacent teeth on the output (driven) gear (Fig. 4). To measure maximum tooth bending stress, the gages were placed at the 30 $^{\circ}$ tangency location [9].

Strain gage signals were conditioned by two methods: for static calibration and measurement, a strain gage (Wheatstone) bridge was used; for dynamic measurements, the strain gages were connected via a slip-ring assembly to a set of constant-current strain gage amplifiers.

A 4-channel, 14-bit digital data acquisition system was used to record the dynamic strain data. Sample rates of 20 to 50 kHz per channel were used, depending on test gear speed.

An optical encoder was mounted on the input shaft to measure roll angle and hence determine load location; the position of the encoder was adjusted so it would produce 1 pulse/revolution at a known roll angle.

3. TEST PROCEDURE

3.1 Calibration

Calibration of the strain gages on the instrumented (driven) gear was conducted to provide a matrix of strain output versus applied load. Before commencing the strain gage calibration, the gears were demagnetized. This demagnetization reduced the apparent strain resulting from the gages moving through the magnetic field of the adjacent gear. At normal gear operating speeds, magnetic effects can induce an error signal in the gage.

For calibration, the instrumented gear was meshed with a special gear whose adjacent teeth had been ground away; this permitted loading of a single tooth only. The calibration was carried out for each of the two instrumented teeth for roll angles ranging from 12 $^{\circ}$ to 30 $^{\circ}$. At each test position (roll angle) the torque was applied at three levels - 45 percent, 88.5 percent, and 132 percent of the nominal value of 71.8 N-m (635 in.-lb). At each of these load levels the sliding direction was reversed (by reversing roll direction), and a linear curve was fit to the data for each sliding direction. By reversing the roll direction, the instrumented gear was effectively tested as both the driven gear (output) and driving gear (input). In each instance the gear was rotated a small angle (approximately 1 $^{\circ}$) in the intended direction of roll until the desired roll angle was reached, so as to definitely establish a sliding direction.

The strain gage calibration apparatus is shown in Fig 5. The results of the calibration for gages 1 to 4 are given in Figs. 6 and 7, for loading on tooth 1. The arrows indicate roll direction. The results for loading on tooth 2 were very similar.