



American
Gear Manufacturers
Association

Technical Resources

14FTM21

AGMA Technical Paper

On the Correlation of Specific Film Thickness and Gear Pitting Life

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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 effect of the lubrication regime on gear performance has been recognized, qualitatively, for decades. Often the lubrication regime is characterized by the specific film thickness defined as the ratio of lubricant film thickness to the composite surface roughness. It can be difficult to combine results of studies to create a cohesive and comprehensive dataset. In this work gear surface fatigue lives for a wide range of specific film values were studied using tests done with common rigs, speeds, lubricant temperatures, and test procedures. This study includes previously reported data, results of an additional 50 tests, and detailed information from lab notes and tested gears. The dataset comprised 258 tests covering specific film values [0.47 to 5.2]. The experimentally determined surface fatigue lives, quantified as 10-percent life estimates, ranged from 8.7 to 86.8 million cycles. The trend is one of increasing life for increasing specific film. The trend is nonlinear. The observed trends were found to be in good agreement with data and recommended practice for gears and bearings. The results obtained will perhaps allow for the specific film parameter to be used with more confidence and precision to assess gear surface fatigue for purpose of design, rating, and technology development.

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October 2014

ISBN: 978-1-61481-113-8

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Introduction

The power density of a gearbox is an important consideration for many applications and is especially important for gearboxes used on aircraft. One factor that limits gearbox power density is the ability of the gear teeth to transmit power for the required number of cycles without pitting or spalling. Methods for improving surface fatigue lives of gears are therefore highly desirable.

Gear and bearing performance is strongly influenced by the lubrication condition and the topography of the contacting surfaces. Research to understand and optimize the performance of systems using gears and bearings has a long history, and studies continue today to refine the qualitative understanding and quantitative relationships. The lubrication condition and surface topography have a strong influence on all of friction, scoring and scuffing, wear, micropitting, and surface fatigue of gears and bearings.

The effect of oil viscosity and surface finish on the scoring load capacity of gears was investigated experimentally more than 50 years ago [1]. Patching [2] evaluated the scuffing properties of ground and superfinished surfaces using turbine engine oil as the lubricant. The evaluation was performed using case-carburized steel discs. The discs were finish ground in the axial direction to orient the lay perpendicular to the direction of rolling and sliding, thereby simulating the conditions normally found in gears. Some of the discs were superfinished to provide smoother surfaces. The Ra of the ground discs was about $0.4\text{ }\mu\text{m}$ ($16\text{ }\mu\text{in}$), and the Ra of the superfinished discs was less than $0.1\text{ }\mu\text{m}$ ($4\text{ }\mu\text{in}$). They found that compared with the ground discs, the superfinished discs had a significantly higher scuffing load capacity when lubricated with turbine engine oil and subjected to high rolling and sliding speeds. They also noted that under these operating conditions, the sliding friction of the superfinished surfaces was the order of half that for the ground surfaces. Others have reported similar trends while producing more refined understanding of the relationships of surface texture and operating conditions to gear scoring and scuffing [3-6].

The influences of lubricant viscosity and additives on gear wear were evaluated by Krantz and Kahraman [7]. Gears tested to study surface fatigue were evaluated to quantify gear wear rates as influenced by lubricant viscosity and additives. The gears of that study were case-carburized and ground finished. The wear rates when gears were lubricated by a 9-centistoke oil were about 10 times lower than the wear rates when lubricated by a 3-centistoke oil. The measured gear tooth wear rates strongly correlated to the lubricant viscosity.

Studies of rolling element bearings have shown that the bearing surface fatigue life is influenced by the lubricant viscosity and the surface roughness [8-11]. The influences have been condensed using the concept of specific film thickness, also often termed the "lambda ratio". The specific film thickness is a ratio of the lubricating oil film thickness to the composite surface roughness of the two contacting surfaces. When the specific film thickness is less than unity, the service life of the bearing is considerably reduced. The Society of Tribologists and Lubrication Engineers (STLE) has published a recommended life factor for bearings that is a function of specific film thickness [12]. Some investigators have speculated that the effect of specific film thickness on gear life could be even more pronounced than is the effect on bearing life [13]. To improve the surface fatigue lives of gears, the film thickness may be increased, the composite surface roughness reduced, or both approaches may be adopted. These two effects have been studied separately for gears.

Townsend and Shimski [14] studied the influence of viscosity on gear fatigue lives using seven different lubricants of varying viscosity. Tests were conducted on a set of case-carburized and ground gears, all manufactured from the same melt of consumable-electrode vacuum-melted (CVM) AISI 9310 steel. At least 17 gears were tested with each lubricant. They noted a strong positive correlation of the gear surface fatigue lives with the calculated film thickness and demonstrated that increasing the film thickness does indeed improve gear surface fatigue life.

Several investigations have been carried out to demonstrate the relation between gear surface fatigue and surface roughness. One investigation by Tanka [15] involved a series of tests conducted on steels of various chemistry, hardness, and states of surface finish. Some gears were provided with a near-mirror finish by using a special grinding wheel and machine [16]. The grinding procedure was a generating process that provided teeth with surface roughness quantified as R_{max} of about $0.1\text{ }\mu\text{m}$ ($4\text{ }\mu\text{in}$). A series of

pitting durability tests were conducted and included tests of case-carburized pinions mating with both plain carbon steel gears and through-hardened steel gears. They concluded that the gear surface durability was improved in all cases because of the near-mirror finish. They noted that when a case-hardened, mirror-finish pinion was mated with a relatively soft gear, the gear became polished with running. They concluded that this polishing during running improved the surface durability of the gear.

Nakasuji [17, 18] studied the possibility of improving gear fatigue lives by electrolytic polishing. They conducted their tests using medium carbon steel gears and noted that the electro polishing process altered the gear profile and the surface hardness as well as the surface roughness. The polishing reduced the surface hardness and changed the tooth profiles to the extent that the measured dynamic tooth stresses were significantly larger relative to the ground gears. Even though the loss of hardness and increased dynamic stresses would tend to reduce stress limits for pitting durability, the electrolytic polishing was shown to improve the stress limit for which the gears were free of pitting by about 50 percent.

Hoyashita [19, 20] completed a third investigation of the relation between surface durability and roughness. They conducted a set of tests to investigate the effects of shot peening and polishing on the fatigue strength of case-hardened rollers. Some of the shot-peened rollers were reground and some were polished by a process called barreling. The reground rollers had a roughness average (Ra) of 0.78 μm (31 μin). The polished rollers had a Ra of 0.05 μm (2.0 μin). Pitting tests were conducted using a slide-roll ratio of -20 percent on the follower with mineral oil as the lubricant. The lubricant film thickness was estimated to be 0.15 ~ 0.25 μm (5.9 ~ 9.8 μin). The surface durability of the rollers that had been shot peened and polished by barreling was significantly improved compared with rollers that were shot peened only or that were shot peened and reground. They found that the pitting limits (maximum Hertz stress with no pitting after 107 cycles) of the shot-peened/reground rollers and the shot-peened/polished rollers were 2.15 GPa (312 ksi) and 2.45 GPa (355 ksi), respectively.

Krantz [21, 22] studied the surface fatigue of gears with an improved surface finish using case-carburized gears made from AISI 9310 steel. Testing was done on the same high-speed power recirculating gear tester used by Townsend and Shimski [14]. The AISI 9310 gears with improved surface finish had longer lives as compared to standard ground gears by a factor of about four times. Motivated by these results, similar testing was later done using the same test rigs and test methods using gears made from aerospace quality, case carburized AMS 6308B alloy steel [23], and the relative life improvement was a factor of about three.

All of these previous works [1-23] provide strong evidence that the specific film thickness parameter is an effective engineering concept for assessing the surface fatigue lives of gears. The review of previous works just presented is not exhaustive. Other work has been published offering results that, from a qualitative view, are consistent with the preceding discussion. However, it has been difficult to combine the results of these studies of the surface fatigue lives of gears to provide a comprehensive quantitative correlation of the lubrication conditions and surface fatigue lives. Because of differing test rigs, specimen geometry, gear alloys and processing, and ranges of operating conditions such as speed and load, it is challenging to combine results. The present study was therefore carried out to quantify the correlation of the surface fatigue lives of gears to specific film thickness. In this work, experimental data from four studies are combined into one dataset. All experiments were conducted on the NASA Spur Gear Test Rigs using consistent test procedures and test conditions (identical speed, torque, temperature, oil jetting and filtration, test gear geometry, and test gear manufacturing quality). This study comprises 258 gear surface fatigue tests. The fatigue data for the majority of the dataset have been published previously [14, 21, 23]. Townsend and Shimski [14] reported results of gear tests using seven lubricants. Later, using gears made from the same melt of steel as used in [14], Townsend completed an additional 50 tests using three more lubricants, but he did not openly publish the data. Those 50 fatigue tests are included into the dataset for this study. Along with previously reported information in [12, 21, 23], many of the tested gears and laboratory records were still available, and access to this information provided a unique opportunity to compile sufficient detail of information to correlate the experimentally measured gear surface fatigue lives to a wide range of specific film thickness.

Test facility and testing procedure

The gear fatigue tests were performed in the NASA Glenn Research Center's gear test apparatus. The test rig is shown in Figure 1(a) and described in [24]. The rig uses the four-square principle of applying

test loads, and thus the input drive only needs to overcome the frictional losses in the system. The test rig is belt driven and operated at a fixed speed for the duration of a particular test.

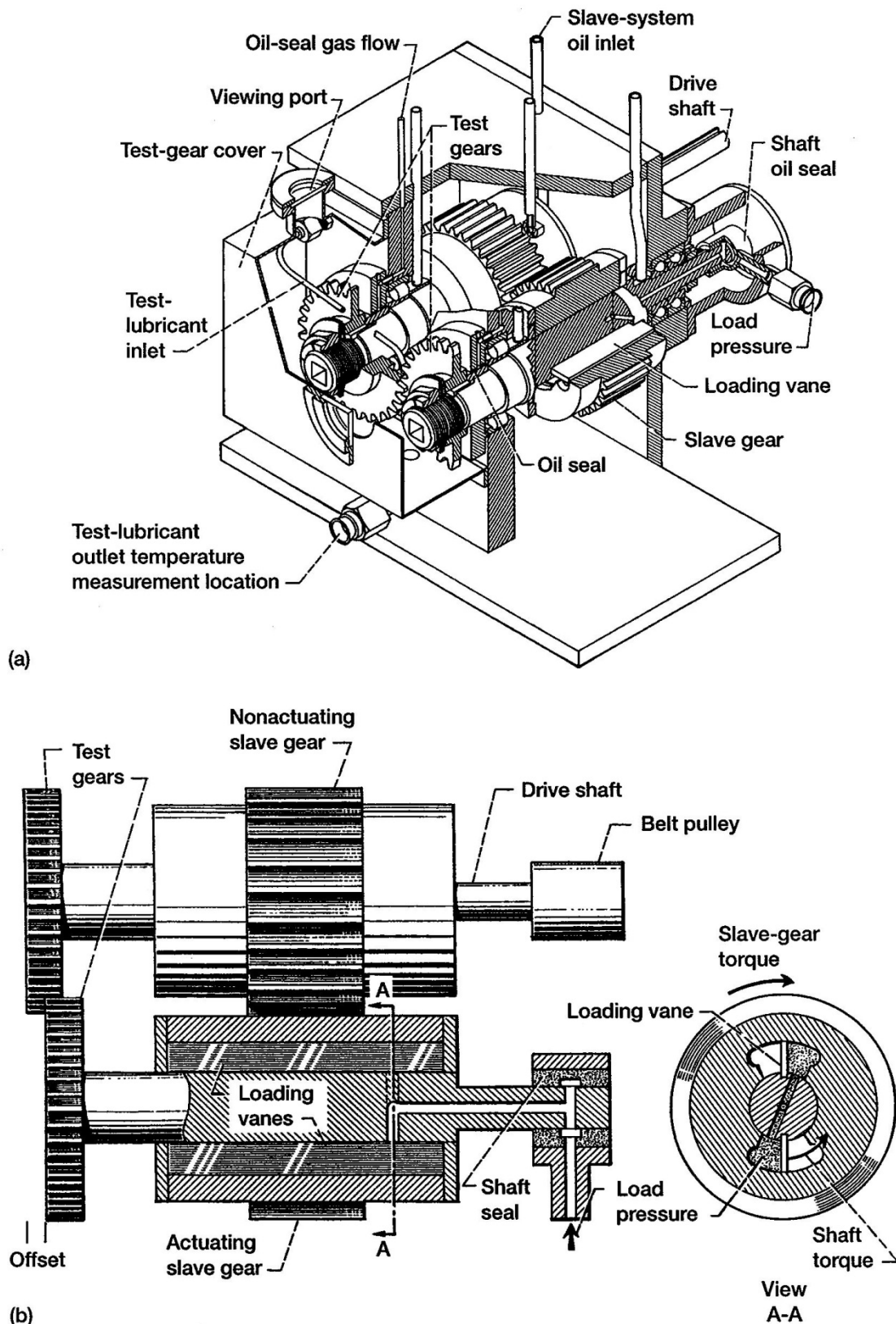


Figure 1. NASA Glenn Research Center gear fatigue test apparatus: (a) cutaway view; (b) schematic view

A schematic of the apparatus is shown in Figure 1(b). Oil pressure and leakage replacement flow is supplied to the load vanes through a shaft seal. As the oil pressure is increased on the load vanes located inside one of the slave gears, torque is applied to its shaft. This torque is transmitted through the test gears and back to the slave gears. In this way power is circulated, and the desired load and corresponding stress level on the test gear teeth may be obtained by adjusting the hydraulic pressure. The two identical test gears may be started under no load, and the load can then be applied gradually. To enable testing at the desired contact stress, the gears are tested with the faces offset as shown in Figure 1. By utilizing the offset arrangement for both faces of the gear teeth, a total of four surface fatigue tests can be run for each pair of gears. The test gears were run with the tooth faces offset by a nominal 3.3 mm (0.130 in) to give a nominal surface load width on the gear face of 3.0 mm (0.120 in). The precise width of the running track will be influenced by gear tooth facewidth tolerances and by the shape and radius of the edge breaks. In this work, post-test inspections were used to determine the running track widths, as will be discussed later in this report.

All tests were run-in at a torque load of 14 Nm (130 in-lb) for at least 1 hour. The torque was then increased to the test torque of 72 Nm (640 in-lb). For this test torque, the peak of the Hertz pressure distribution for line contact condition at the pitch-line and static torque equilibrium is 1.7 GPa (250-ksi). Typical dynamic tooth forces have been measured using strain gages located in tooth fillets. Using calibration coefficients determined by specialized calibration experiments [25] typical gear tooth forces were calculated from measured tooth fillet strains (Figure 2). The resulting peak dynamic tooth force is about 1.3 times greater than the force for static equilibrium, and the resulting peak of the Hertz pressure distribution for this peak dynamic force is 1.9 GPa (285 ksi). The Hertz pressure values stated herein are idealized stress indices assuming perfectly smooth surfaces and an even pressure distribution across a 2.79 mm (0.110 in) line contact (the line length is less than the face width allowing for the face offset and the edge break radius).

The gears were tested at 10 000 rpm, which gave a pitch-line velocity of 46.5 m/s (9154 ft/min). Inlet and outlet oil temperatures were continuously monitored. Cooled lubricant was supplied to the inlet of the gear mesh at 0.8 liter/min (49 in³/min) and 320 ± 7 K (116 ± 13 °F). The lubricant outlet temperature was recorded and observed to have been maintained at 348 ± 4.5 K (166 ± 8 °F). The lubricant was circulated through a 5 µm (200 µin) rated fiberglass filter to remove wear particles. For each test, 3.8 liter (1 gal) of lubricant was used.

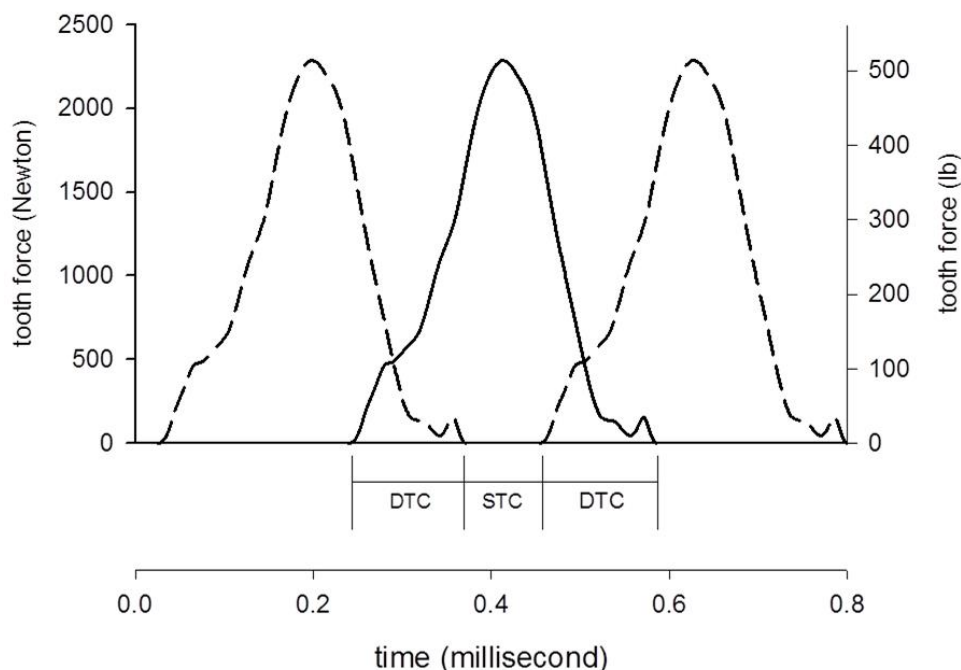


Figure 2. Measured dynamic tooth force at nominal test conditions [22]. The solid line is the measured data, and the dashed lines are replicates of the measured data spaced along the ordinate at the equivalent of one tooth pitch. The zones of double tooth contact (DTC) and single tooth contact (STC) are illustrated.