EXPERIMENTS ON THE HIGH-POWER AND HIGH-TEMPERATURE PERFORMANCE OF GEAR CONTACTS

Thesis

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ABSTRACT

In this study, gear contact tests were performed using a recently developed test methodology capable of both high-power (pitch-line velocities up to 50 m/s and pinion torques up to 450 N-m) and high-temperature (oil inlet temperatures up to 150°C) operating conditions. Test specimens and operating conditions were chosen in order to simulate high-power automotive and aerospace applications. Automotive test specimens were made from a typical automotive transmission gear steel, SAE 4118M, at surface roughnesses typical of hard ground gears. Aerospace test specimens were made out of a high performance (high-temperature) proprietary gear steel. These aerospace specimens were either chemically polished or super-finished following grinding to achieve roughness amplitudes more than 10 times smoother than typical ground surfaces. Throughout each test interim inspections were used to identify and monitor failure modes. Experimental testing for automotive applications is shown to consistently produce contact fatigue failures in the form of micro-pitting and macro-pitting. Tests were suspended when macro-pits exceeded the test methodologies pre-determined failure criteria. Experimental testing for aerospace applications is shown to be absent of any contact fatigue failures due to the extremely smooth contact surfaces. The primary mode of contact failure in aerospace tests is observed to be scuffing.
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CHAPTER 1

INTRODUCTION

1.1 Background and Motivation

Gears fail in various ways as a result of the interaction between many parameters including the operating conditions (input torque, speed, and temperature), surface roughness, micro-geometry modifications, manufacturing error and misalignment, material properties (fatigue strength, residual stress, hardness, etc.), and lubricant properties (viscosity-pressure-temperature relationships and additives). Pitting (macro-pitting) and micro-pitting are two common contact failure modes associated with fatigue of contact surface materials due to cyclic compressive loading while scuffing (also known as scoring) represents a common surface failure associated with the surface temperatures. Regardless of failure mode, increases in vibration are often the only indication of gear failure until total failure occurs. Detailed examinations and failure analysis are often necessary in order to identify the actual mode of failure [1].

During operation, gear teeth are subjected to cyclic loading as they repeatedly enter and leave gear mesh with the mating teeth. For the driving gear (pinion), the
contact initiates at the start-of-active-profile (SAP), sweeps along the dedendum region (negative sliding) up to the pitch point where only rolling exists (sliding velocity is zero), and continues along the addendum (positive sliding) to the tip. The opposite is true for the driven gear (gear). This meshing action causes various time-varying contact parameters including the normal tooth force, radii of curvature, surface velocities, and slide-to-roll ratio (ratio of the sliding velocity to the rolling (mean) velocity). As described in detail by Li and Kahraman [2], these time-varying parameters can be used to model both the elastohydrodynamic lubrication (EHL) film between the mating surfaces and the potential asperity contacts. In general, the resulting normal and shear stress distributions from this complex contact result in multi-axial stress states on or below the tooth surface.

The cyclic nature of these localized stresses initiates the formation of fatigue cracks on or below the tooth surface. These cracks propagate along a shallow angle (typically 30°) into the material, turn parallel to the surface at a certain depth, and branch back to the tooth surface to form a surface initiated pit. Normally pitting occurs in the dedendum region of the tooth surface where both the negative sliding magnitude and the contact pressure are high. Pits with the diameters smaller than one millimeter are typically referred to as non-progressive pitting failure. Such pits remove high asperities until contact pressure and load are more evenly distributed. Contrarily, progressive pits are those with diameters larger than one millimeter. Such pits usually continue growing until a significant portion of the tooth surface has pitted and spalling, the coalescing of
multiple pits, has occurred. At this point the bending strength of the tooth may have decreased enough for bending failure to occur as well [3].

In certain gearing applications, micro-pitting often precedes the formation of macro-pits. In contrast to the sudden removal of a large amount of material, micro-pitting is a progressive wear phenomenon. By continuously removing material of micron size, micro-pitting can result in substantial profile changes after a large number of contact cycles. Micro-pitting is closely related to the local contact stresses induced by surface asperities and is a function of the normalized film thickness parameter \( \lambda \) (also referred to as specific film thickness or lambda ratio). The lambda ratio is defined as the ratio of the minimum film thickness under ideally smooth conditions to the root-mean-square (RMS) amplitude of the surface roughness. The AGMA [4] uses \( \lambda \) as a measure for the prediction of micro-pitting occurrence. Generally, \( \lambda < 1 \) corresponds to the occurrence of asperity contacts, allowing micro-pitting to ensue. When micro-pitting occurs, small pits, only a few microns in size, form on the tooth surface. Often early stage micro-pitting is not apparent through visual examination and additional inspection with a microscope is required to determine its presence. As the cyclic contact continues, the micron sized pits increase in number and conglomerate to form well-defined grey stained areas. Under certain operating conditions micro-pitting can stabilize after the surface wear redistributes and relieves the contact pressure, however, micro-pitting can also lead to pitting failure fairly quickly [5].
Scuffing is frequently observed in high speed contact applications. Unlike pitting and micro-pitting, which occur after a period of running time, scuffing has the possibility to occur immediately. When scuffing occurs material is transferred from one gear to the other due to welding and tearing of the mating tooth surfaces [1]. Scuffing, which is also referred to as scoring, is often considered to result from a lubrication failure that leads to an increase in friction and instantaneous temperature in the gear mesh [6]. Typically scuffing damage occurs in the addendum and dedendum where high sliding velocities exist, in bands that are oriented in the direction of sliding. Scuffing can vary in severity, however, when significant portions of the addendum and dedendum become scuffed, it often becomes a progressive failure [3].

While the extensive amount of research completed on pitting failure has provided engineers with an improved ability to design against such failures, additional experimental research is still required to quantify how improved materials, manufacturing methods, surface finishes, etc., perform against pitting. Additionally, experimental data is always critical for validation of the advanced modeling efforts continuously taking place. Since contact fatigue failures are common in automotive transmissions and aerospace applications, this experimental study aims at providing much needed information for the further advancement of both applications. Furthermore, the lack of pitting data at high-power and high-temperature conditions makes such research even more essential and timely.
1.2 Literature Survey

Substantial efforts have been made over the last several decades in order to research the various modes of gear failure, including both surface fatigue and scuffing. Unfortunately, the cost of such research often keeps the results of gear failure studies out of the public domain for competitive purposes, especially for experimental work. While this severely reduces the amount of published results, numerous published studies do exist.

1.2.1 Influence of Lambda Ratio on Surface Fatigue

Townsend, et al. [7] studied the influence of $\lambda$, the lambda ratio, on surface fatigue lives using four-square test machines. While keeping the surface roughness amplitudes constant, different $\lambda$ values were achieved by using synthetic polyol-ester lubricants with different viscosities to vary the elastohydrodynamic lubrication (EHL) film thickness. These fatigue tests utilized consumable electrode vacuum melted (CVM) AISI 9310 spur gears that were case carburized, hardened, and ground. The gears were operated under 10,000 rpm rotational speed, pitch line contact stress of 248 ksi, and 49°C oil inlet temperatures. The results of these fatigue tests showed that gears experienced surface fatigue lives more than 8 times longer when lubricants that yielded lambda ratios of $\lambda > 1$ were used in comparison to lubricants that yielded lambda ratios of $\lambda < 1$. 
Furthermore, lubricants with equivalent viscosities and similar additive packages produced very similar gear surface fatigue lives.

Using the same test rig and operating conditions as Townsend, et al. [7], Krantz et al. [8] increased the lambda ratio by reducing the surface roughness amplitudes of the test samples. Test gears made from CVM AISI 9310 steel were case carburized, ground, and super-finished to achieve an arithmetic roughness amplitude of $R_a = 0.07 \mu m$. Compared to baseline ground gears with $R_a = 0.4 \mu m$, the super-finished gears showed an improvement of nearly four times the surface fatigue life. Tanaka, et al. [9] also examined the effect of reducing surface roughness on the pitting life of gears. Test specimens made from AISI 1045, 4118, and 5135 steels were finished with various processes. One such process created a near mirror-like finish with surface roughness values near $0.02 \mu m$ $R_a$ that largely improved surface fatigue lives.

Bluestein [10] investigated the influences of different surface treatments on the pitting life of gears manufactured out of AISI 8620 steel. Experiments were conducted on standard FZG type gear test machines utilizing a back-to-back power circulation loop arrangement. Tests were carried out at 1,440 rpm, an oil inlet temperature of $90^\circ C$, and several stress levels. The pitting lives of shot-peened and plastic honed surfaces, chemically polished surfaces, and chemically polished and CrN coated surfaces were compared to those of the baseline hobbed-shaved surfaces. Statistical analysis of the results showed that the pitting life of the chemically polished gears was nearly three times that of the baseline hobbed-shaved gears. Furthermore, the pitting life of the chemically
polished gears was slightly improved by the addition of the CrN coating. For the shot-peened and plastic honed process, no noticeable difference was observed compared to the baseline due to limited data points.

Using the same methodology and test machines, Klein [11] expanded the pitting investigation initiated by Bluestein [10]. In addition to the previously studied surface finishes, this study included ground AISI 8620 and ground AISI 4620M gears while dropping the chemically polished and CrN coated surface finish. Results confirmed the increased pitting life of the chemically polished gears and showed life improvements for the shot-peened and plastic honed gears over the baseline hobbed-shaved gears. Finally, limited testing suggested that both 8620 and 4620M ground gears had longer contact fatigue lives than the baseline.

In their modeling studies, Li and Kahraman [12] and Li, et al. [13] developed physics-based pitting models of roller and gear contacts, respectively. Both models relied on new mixed EHL models including asperity contacts to predict the surface pressure and shear stress distributions of rough surfaces in combined rolling and sliding. The stress states into the surfaces due to these tractions were computed to obtain mean and alternating amplitudes of stress components. A multi-axial fatigue criterion was used to predict crack initiation pitting lives. The rolling fatigue model [12] was validated by using the roller contact fatigue experiments of Li [14] while the gear fatigue model [13] was validated by using the experiments of Klein [11]. Both sets of experiments as well
as the models showed that macro-pitting fatigue lives of roller contacts and gears can be increased significantly by reducing the surface roughness amplitudes.

In a recent study, Leque [15] developed a new test methodology for performing gear contact fatigue testing at much higher speeds and temperatures than were achievable using the FZG type machines previously described [10-11]. In comparison to the FZG speed limit of 1,440 rpm and oil temperature limit of 90°C, the methodology developed by Leque [15] has the capability of input speeds up to 13,500 rpm, input torques up to 450 N-m and oil inlet temperatures up to 150°C. Along with developing the methodology, the study also presented preliminary results with aerospace quality gears. The aerospace quality gears were made out of a high performance (high temperature) aerospace gear steel and were either hard ground, hard ground and chemically polished, or hard ground and super-finished. The preliminary results showed severe micro-pitting failures for the hard ground samples due to the large surface roughness amplitudes of about 0.5 μm $R_a$. Additionally, both chemical polishing and super-finishing were shown to successfully eliminate the occurrence of micro-pitting through the reduction of the surface roughness amplitudes to $R_a < 0.1 \, \mu m$.

1.2.2 Influence of Gear Material on Surface Fatigue

Kaneko [16] performed gear pitting experiments to study both the influence of gear materials and surface finishes on pitting life. Fatigue test machines similar to FZG
machines were used in these experiments. Speed, temperature, and lubricant type were held constant across all tests while the test specimens and load were changed. Spur gears made out of either NiCrMo or NiCr steel were heat treated through case carburization. Helical gears made of either CrMo or a carbon steel were heat treated using quenching and annealing, induction hardening, or flame hardening. The results from the study showed that NiCrMo steel, CrMo steel, and induction hardened samples outperformed NiCr steel, standard carbon steel, and flame hardened samples respectively.

Townsend [17] investigated the manufacturing process of powder metallurgy by comparing fatigue lives of gears made with such a process with machined and ground AISI 9310 gears. Tests were performed at pitch line contact stresses of 222 ksi and 248 ksi, rotational speed of 10,000 rpm, and jet lubrication conditions with an oil inlet temperature of 77°C. The results of the study indicated that powdered metal gears cannot provide the same surface fatigue life achieved with the machined and ground AISI 9310 gears. In a later study, Townsend [18] compared three new high temperature gear materials to the standard AISI 9310 aerospace steel. The three new materials included CVM VASCO max 350, CVM VASCO matrix II, and nitralloy N. Operating conditions of 10,000 rpm and 248 ksi contact stress were maintained while the oil temperature was reduced to 49°C. Results showed that VASCO matrix II performed the best out of the three new materials with fatigue lives several times longer when compared to AISI 9310. The nitralloy N showed no improvements compared to AISI 9310 and VASCO max 350 performed below AISI 9310.
In a recent study, Milliren [19] expanded on Klein’s [11] database of ground AISI 8620 and 4620M gears. Using the same test equipment and methodology as Klein [11], Milliren [19] compared ground gears made out of AISI 8620, 4620M, and 5120M to their shaved counterparts. The study showed that in comparison to the shaved gears, the ground gears had substantially longer pitting lives. It was also observed that gears made out of AISI 4620M steel performed slightly better than both AISI 8620 and AISI 5120M.

In the previously described study by Leque [15] the preliminary results also compared different gear materials in terms of automotive and aerospace quality steels. In this comparison the automotive quality gears were made from hard ground SAE 4118M steel and the aerospace quality gears were made with a high performance gear steel. The results showed that the SAE 4118M samples experienced micro-pitting that propagated into pitting, while the aerospace samples achieved much longer surface fatigue lives that only experienced micro-pitting.

1.2.3 Influence of Operating Conditions on Scuffing

Standardized scuffing procedures are typically completed with either roller specimens or gears themselves to reduce variability between studies. One such scuffing study was performed by Castro and Seabra [20] to investigate the influence of lubricant viscosity on scuffing failure using three separate lubricants. In the order of increasing viscosity, the lubricants tested were ISO VG 68, VG 220, and VG 680. Utilizing the
standard FZG-A/8.6/90 test the scuffing performance of these lubricants were compared to show that the highest viscosity lubricant, VG 680, prevented scuffing at much higher loads and temperatures than the other two lubricants. Since lubricant film thickness decreases as temperature increases due to reduced viscosity, the high-viscosity lubricant was able to maintain a sufficient film thickness to prevent scuffing.

In a study by Snidle, et al [21] scuffing experiments were performed at various sliding speeds while the applied force between steel disks was increased stepwise. During testing both bulk temperatures of disk surfaces and the frictional force between the disks were recorded until scuffing occurred and the test was suspended. The results of the study illustrated that the frictional force between disks decreased before scuffing initiation and then experienced a large increase upon initiation. A similar two-disk scuffing experimental study was performed by Liou [6] who showed the scuffing load (normal force at which scuffing occurs) increased with reduced surface roughness. His experiments suggested that there is a maximum temperature limit (surface bulk temperature plus instantaneous contact temperature rise), which defines the onset of scuffing.

The previously described study by Klein [11] also included scuffing experiments in addition to the surface fatigue research that was conducted. For the scuffing experiments spur gears made from AISI 8620 steel were testing in accordance with the standard ISO 14635-1 FZG Scuffing Test. The spur gears were either uncoated or coated with an experimental PVD coating. Although the experimental data was limited, the
results showed that while the uncoated gears scuffed during Stages 11 or 12 of the standardized test, the PVD coated gears completed State 12, the last stage of testing, without any scuffing failure. Additionally the PVD coated gears correlated with the lowest recorded temperatures during testing. Finally, the study also illustrated that when scuffing failures initiated, the amperage being delivered to the drive motor increased in order to overcome the increased frictional forces caused by scuffing.

In addition to monitoring surface fatigue failures, Leque [15] also illustrated that avoiding scuffing failures was a major task of his study. To avoid scuffing failures, the study utilized a gear design that would minimize the maximum sliding velocities that occur near the root and tip of the gear teeth. In addition to the gear design, high pressure jet lubrication was used to remove as much heat from the tooth surfaces as possible. The combination of these two design choices were reported to effectively eliminate scuffing failures during preliminary tests.

While the above scuffing studies were solely experimental, a recent study by Li, et al. [22] proposed a physics-based model of scuffing. The point contact EHL model of Li and Kahraman [23] was modified to include thermal effects. The surface traction distributions were predicted and converted to friction heat flux. A heat balance model of the contacting surfaces was then constructed to predict the bulk surface temperatures as a function of time. The results of this model were validated through direct comparisons to the friction coefficient and bulk temperature measurements from a set of ball-on-disk scuffing experiments.
1.3 Scope and Objectives

While the existing studies have generated an extensive database for surface contact fatigue, results at high-speed and high-temperature conditions are limited. Additionally, there is still a need for additional studies on the impact of surface roughness on contact fatigue life. While Leque [15] successfully developed a test methodology capable of performing contact fatigue tests at high-speed and high-temperature conditions, the preliminary testing results presented in the study are only the beginning of the formation of a valuable database such a methodology can generate. Accordingly, the primary goal of this study is to utilize the test methodology developed by Leque [15] in order to expand upon the preliminary results generated in his study. By expanding the high-speed and high-temperature testing results for both automotive and aerospace quality gear samples, commercial applications of gear technology at those operating conditions will benefit greatly. Additionally, such results will also provide significant value in validation for analytical modeling efforts.

The specific objectives of this study are as follows:

- Apply the previously developed test methodology of Leque [15] to expand on the contact fatigue pitting results of SAE 4118M automotive steel under high-speed and high-temperature conditions. Construct preliminary statistical confidence intervals for the pitting life of SAE 4118M steel under the torque values and contact stresses tested.
• Apply the same test methodology to evaluate the experimental contact performance of a proprietary high performance (high temperature) aerospace gear steel under high-speed and high-temperature conditions.

• Evaluate the scuffing performance of gear specimens during high-speed and high-temperature testing. Investigate the impact of surface roughness, defined by the chemically polished and super-finished surface finishes, on the occurrence of scuffing.

1.4 Thesis Outline

Chapter 2 will describe the test equipment, test methodology, and test specimens used in this study. The two identical four-square test machines will be detailed along with their operation and the routine maintenance established to maintain machine integrity. The gear inspection process and measurement devices will also be presented.

Chapter 3 will present the contact fatigue results of SAE 4118M steel at the various load levels tested. Fatigue failures will be complied and presented in graphical form as torque-life and stress-life curves. Gear profile traces and surface roughness measurements taken at periodic inspection points will be documented to illustrate trends related to the eventual failure mode. Digital pictures will also be presented to document the onset and propagation of pitting. Finally, a preliminary statistical analysis of the pitting life of SAE 4118M specimens will be presented.
Chapter 4 will present the experimental results of the high performance aerospace steel at the various testing conditions performed. The lack of contact fatigue failure will be illustrated along with scuffing failures in its place. Gear profile traces, surface roughness measurements, and digital images taken at periodic interim inspections will be documented to illustrate trends related to the eventual failure modes.

Chapter 5 will summarize the thesis research described in the previous chapters. The main conclusions generated by the research results will be identified. Finally, recommendations for future work to further expand the high-speed and high-temperature results will be stated.
CHAPTER 2

EXPERIMENTAL METHODOLOGY

2.1 Test Machine

Gear durability tests were performed on two identical high-speed, four-square test machines previously described in great detail by Leque [15]. Figure 2.1 shows both machines in The Gleason Gear and Power Transmission Research Laboratory of The Ohio State University. In order to simulate high-power (high speed and high load) and high-temperature operating conditions common to aerospace applications the machines were designed to be capable of:

- Pinion (smaller gear of the gear pair) rotational speeds up to 13,500 rpm, corresponding to a maximum pitch-line velocity of 50 m/s.

- Pinion torque up to 450 N-m applied in either direction.

- Duty cycle simulation through the ability to induce user defined changes to both speed and load during operation.
Figure 2.1 High-speed gear test machines used for this study.
Jet lubrication with oil pressures up to 150 psi and temperatures up to 150°C.

Consequently, these machines represent significant upgrades from the standard FZG type gear test machines used in earlier studies by Bluestein [10], Klein [11], and Milliren [19]. Such machines typically operate under dip lubrication, at constant torque, and constant rotational speeds below 1,500 rpm, corresponding to a pitch line velocities lower than 8.6 m/s.

The high-speed test machines shown in Figure 2.1 achieve high power conditions (up to 632 kW) by creating a power circulation loop in a back-to-back (four-square) arrangement. Figure 2.2 illustrates the top view schematic lay-out of one of the test machines, while both machines were identical in every aspect. Compliant shafts and flexible couplings are utilized in order to connect the test gearbox to the larger reaction gearbox. These compliant shafts and flexible couplings are essential in order to isolate the two gearboxes from each other dynamically. In order to maintain machine integrity, this setup allows vibrations generated in the test gearbox to be effectively damped before entering and affecting the reaction gearbox. Similarly, it also restricts vibrations in the reaction gearbox from subjecting the test samples to undesirable disturbances.

Both the test and reaction gearboxes have matching speed ratios, but different gear arrangement and geometry. The reaction gearbox was designed to be very robust and heavy duty. The gearbox utilizes helical gears with large face widths to reduce contact pressure and tooth bending stress. Also, a chemical polishing process was used on these helical gears to decrease the surface roughness amplitudes to prevent micro-
Figure 2.2 Machine layout schematic [15].
pitting. Overall, the oversized nature of the reaction gearbox design increases the service life of this gearbox greatly by decreasing the likelihood of gear failure. The test gearbox was designed to accommodate a 17 tooth gear, referred to as the “test pinion,” paired with a 26 tooth gear, known as the “test gear,” operating at a center distance of 91.5 mm. This gearbox features relatively rigid shafts and high-capacity cylindrical roller bearings. Each gear is held on its shaft using precision spacers, lock nuts, and two symmetric keys to avoid unbalanced forces at high rotational speeds. Finally, the front and top covers of the test gearbox were designed to be easily removed for the periodic gear inspections that were performed.

As previously shown in Figure 2.2, a 40 HP AC drive motor connected to the reaction gearbox via a belt drive system provides the rotational input to the power circulation loop. This AC drive operates at moderate speeds since the reaction gearbox acts as a speed increaser. Speed is measured using a magnetic proximity sensor mounted near the flexible coupling nearest the test pinion. Variable pinion torque values of up to 450 N-m are obtained with a torsional actuator mounted on the end of the coaxial input shaft of the reaction gearbox. This torsional actuator is capable of dynamically adjusting torque between the inner and outer layers of the input shaft using proportional valve control. The applied torque is continuously monitored using a non-contact torque sensor within an accuracy of 0.05%. The sensor consists of a rotating sensor attached to the input shaft and a stationary receiver base mounted on the test bed. A strain gauge bridge housed inside the rotating sensor communicates with the stationary receiver base using infrared diodes.
The high-speed and high-power test conditions performed by these test machines require jet lubrication systems to be used for both the reaction and the test gearboxes. Such a system is required to obtain sufficient oil penetration into the root of the gear teeth in order to effectively lubricate and cool the gear teeth. Each gearbox is equipped with its own independent lubrication system, while the lubrication system of the reaction gearbox also accommodates the middle bearing support box. Each system contains a 5-gallon reservoir, a 3 GPM pump, a lubricant temperature control facility via a heat exchanger, and a pressure relief valve for the manual adjustment of the system pressure. In order to remove debris from each system a 74 μm strainer and a 5 μm filter are positioned on either side of each system’s pump.

The reaction gearbox utilizes a high viscosity, Mobile SHC 629 synthetic gear oil. This oil is maintained at 45°C and delivered both into and out-of the various gear meshes using spray bars with sets of 0.5 mm diameter nozzles. This lubrication spans the entire face width of the reaction gears and a portion is also delivered to the gearbox bearings. The test gearbox utilizes the test lubricant of choice. In this study, either a typical automotive automatic transmission fluid (Dexron VI) was used with the automotive test gears or BP Turbo Oil 2380 with the gears representative of aerospace applications. A 6 kW heater in the lube reservoir is employed to reach the desired oil temperature for tests. It is also important to note that a portion of the test gearbox oil is diverted through a separate heat exchanger and delivered to the test gearbox bearings at a lower temperature. This helps ensure prolonged life of the test gearbox bearings. Pressure of the test system is held between 140 and 142.5 psi with a flow rate between 0.7 and 1.0 gallons per
minute to the test nozzles. Using the nozzles shown in Figure 2.3, oil is delivered both into and out-of the gear mesh formed by the test samples. The automotive quality gears were lubricated by two manifolds each with three nozzles 0.020 inches in diameter, as shown in Figure 2.3(a). The aerospace test gears were lubricated by two manifolds each with six nozzles 0.020 in diameter, depicted in Figure 2.3(b).

Extensive health monitoring instrumentation allows the machines to automatically shut down if measurements exceeding user defined limits are detected. This instrumentation allows laboratory safety to be maintained as well as machine integrity. In order to monitor vibrations, two uni-axial PCB accelerometers are stud mounted on the high-speed test bearing housing and the reaction bearing housing. Type T thermocouples are located in various locations around the machines in order to monitor bearing and oil temperatures. These thermocouples trigger both automatic shutdowns and appropriate action from the machine’s heat exchangers and heater. Finally, drive motor amperage and filter clog detectors are also continuously monitored.

In addition to the active health monitoring instrumentation, machine components were routinely inspected at predetermined intervals and gear inspection points in order to maintain machine integrity and test repeatability. Machine components that experienced the most wear and fatigue included the test shafts and test gearbox bearings. Whenever such components were observed to have reached their usable life, they were replaced by new ones. Detailed maintenance records were kept in order to better predict and prepare for maintenance intervals.
Figure 2.3  Test gearbox lubrication nozzles used during (a) SAE 4118M fatigue tests and (b) high performance gear steel fatigue tests.
2.2 Inspection of the Test Specimens

The test pinion and test gear were routinely inspected prior to, at predetermined inspection intervals, and at the conclusion of each test. Such inspections included involute and lead profile traces, surface roughness measurements, and digital imaging of the contact surfaces. These inspections allowed the specimens to be monitored throughout the duration of each test in order to observe the initiation and propagation of the various gear failure modes. By observing changes in roughness, gear tooth geometry, and contact surface condition, trends could be captured. These intermittent inspections helped suspend testing, when failure criteria were met, with better accuracy.

A Gleason M&M 225 gear coordinate measurement machine, shown in Figure 2.4, was used during inspections to measure both lead and involute profile traces of the test pinion and test gear. In total, twelve lead traces and twelve profile traces were recorded for each specimen. These traces were gathered from four teeth of the gear specimen at three different lead and three different profile locations each. Subsequent inspections were performed at the same locations such that any changes in the form of surface wear could be monitored as each fatigue test progressed. Figure 2.5 and Figure 2.6 show example traces from a brand new pinion and brand new gear, respectively.

A Taylor-Hobson Form Talysurf-120 surface profiler was used to perform surface roughness measurements on the specimen contact surfaces. Figure 2.7 shows the surface profiler probe as a test gear specimen is being inspected. As illustrated in this figure, roughness measurements were taken along the transverse mid-plane of the specimens in
Figure 2.4  CMM measurement of a test gear.
Figure 2.5 Example set of (a) lead and (b) profile traces from a pinion inspection.
Figure 2.6 Example set of (a) lead and (b) profile traces from a gear inspection.
Figure 2.7  Surface roughness measurement of a test gear.
the profile direction. For each specimen, this measurement was repeated three times on a single tooth and the results were averaged. Typically, each surface roughness measurement traced either 3.2 mm or 4 mm of the contact surface. To filter out low frequency surface waviness, recommended cut-offs (ISO 4288-1996) were used based on the surface roughness amplitude range. Cut-offs of 0.8 mm and 0.25 mm were used for automotive and aerospace test samples, respectively, due to their relative surface finishes.

Figure 2.8(a) and (b) show typical surface roughness profiles of a hard ground SAE 4118M pinion and a hard ground and chemically polished aerospace gear steel pinion, respectively. After the initial inspection, the same tooth was measured for all subsequent inspections so that any changes could be monitored. The commonly used $R_a$ (the arithmetic mean) and $R_q$ (the root-mean-square) roughness parameters were used to describe the contact surfaces and to illustrate changes in surface roughness over the duration of fatigue tests. Along with recording these two main parameters, numerous other roughness parameters were computed and saved for future reference.

The last inspection process consisted of taking digital images of the contact surfaces. Images were taken using a computer station linked with a high resolution camera mounted to a 1/2X objective lens, shown in Figure 2.9. Test pinion tooth surfaces were photographed with 5X magnification while test gear tooth surfaces were photographed with 3.5X magnification. For each specimen, the same tooth was photographed at each inspection interval while additional teeth were photographed if their contact surfaces showed any signs of pit formation.
Figure 2.8  Example roughness plots from (a) a hard ground SAE 4118M pinion inspection and (b) a hard ground and chemically polished high performance gear steel pinion inspection.
Figure 2.9  Digital imaging of a test gear contact surface.
2.3 Test Procedure and Test Conditions

A consistent test procedure was used throughout the testing of all automotive SAE 4118M test specimens. All tests were performed at constant rotational speeds of either 8,000 rpm or 10,650 rpm of the test pinion. As previously described, jet lubrication was used to provide adequate lubrication of the test gears with three nozzles 0.020 inches in diameter positioned both into and out-of the gear mesh. The oil supply pressure was held between 140 and 142.5 psi with a flow rate of 1.0 gallons per minute to the test nozzles. Dexron VI, a typical automatic transmission fluid, was maintained at an oil inlet temperature of 90°C for all tests. To offset for minor leakage of the test gearbox lubrication system, small amounts of Dexron VI were added to the system when necessary. A complete drain and fresh refill of Dexron VI was performed approximately every 30 million cycles.

After the initial inspection, new gear sets were installed on the test machine as shown in Figure 2.10. After ensuring proper oil levels and pressures the control computer was used to initiate the test cycle. At this point heating of the test gearbox lubricant to 90°C was commenced. Upon coming within ±3°C of the set temperature, the machine began ramping both the speed and load simultaneously. Rotational speed and torque were allowed to ramp at 150 rpm/sec and 15 Nm/sec, respectively, until the desired set points were obtained. After ramping, testing continued at the constant set points until the predetermined number of cycles was completed or until the machine was automatically shut down by health monitoring instrumentation. Upon shutting down,
Figure 2.10 Test gearbox with front cover removed and test gear pair installed.
both speed and load were reduced back to zero while the lubricant systems were kept running in order to cool all components back to room temperature.

It is important to note that no ‘run-in’ procedures were completed on the test specimens. All tests operated at constant torque (at one of four load levels) for their entirety. Starting with the lowest torque level and continuing to the highest, the four load levels were labeled as L1, L2, L3, and L4. These load levels along with their corresponding contact pressures are listed in Table 2.1. Due to confidentiality requests from the project sponsor, these torque and contact pressure values have been normalized.

Prior to each test, both the test pinion and test gear were subjected to full inspections as described in Section 2.2. Inspection intervals were then chosen to represent 10% increments of the total expected life. Initially such intervals were estimated for each load level, however, completed tests allowed for more accurate predictions as the study progressed. At these inspection intervals visual inspections and full inspections were completed in an alternating fashion. This combination of visual and full inspections typically allowed failures to be caught and testing suspended within 10% of the failure life. In preparation for full inspections, the test pinion and gear were removed from the test machine, cleaned using isopropyl alcohol to remove oil from the tooth surfaces, and subjected to the three inspections described in Section 2.2. In contrast, visual inspections were performed by simply removing the front cover of the test gearbox and using a flashlight to inspect each tooth surface for signs of failure. After
Table 2.1  Load levels and corresponding normalized stresses used during SAE 4118M fatigue tests.

<table>
<thead>
<tr>
<th>Load Level</th>
<th>Normalized Pinion Torque</th>
<th>Normalized Pitch-line Contact Stress</th>
</tr>
</thead>
<tbody>
<tr>
<td>L1</td>
<td>2.55</td>
<td>1.24</td>
</tr>
<tr>
<td>L2</td>
<td>3.01</td>
<td>1.33</td>
</tr>
<tr>
<td>L3</td>
<td>3.47</td>
<td>1.41</td>
</tr>
<tr>
<td>L4</td>
<td>3.93</td>
<td>1.52</td>
</tr>
</tbody>
</table>
each inspection the gear set was re-installed on the test machine, oil levels and pressures were checked, and the next test cycle was initiated.

Testing continued to follow this sequence of test cycles and inspections until either the test pinion or test gear displayed a failure. The failure criteria used was a slightly modified form of that used by previous FZG pitting studies [10, 11, 19]. The modified criteria stated that failure had occurred if (i) the total area of pits on a single tooth is greater than 1.0 mm$^2$ (0.00155 in$^2$), (ii) the total area of pits on all teeth is greater than 8.0 mm$^2$ (0.0124 in$^2$), or (iii) the maximum profile wear depth is greater than 12 μm (0.0005 in). Criteria (i) an (ii) corresponded with pitting failures that were included in the pitting database for this study while criteria (iii) corresponded with wear failures that were excluded from the pitting database. This modified failure criteria was intended to better capture pitting at an earlier stage since the high-power testing conditions used in this study lead to faster crack propagation. After final inspections, all test specimens were wrapped in rust-inhibiting paper, labeled, and stored for future reference.

In comparison to the test procedure used for SAE 4118M test specimens, testing of aerospace test specimens utilized a slightly modified procedure. All tests were performed at constant rotational speeds of 12,000 rpm while the test gearbox lubricant was replaced with BP Turbo Oil 2380. This synthetic oil conformed to MIL-L-23699 specifications required by the project sponsor. Jet lubrication between 140 and 142.5 psi was maintained while six nozzles 0.020 inches in diameter were positioned both into and out-of the gear mesh. Oil inlet temperatures were varied between 82°C, 107°C, and
120°C across various tests which corresponded to temperature dependent flow rates between 0.7 and 0.9 gallons per minute. Minor leakage of the test gearbox system was again offset with small amounts of additional oil when necessary and a complete drain and fresh refill of BP Turbo Oil 2380 was performed approximately every 200 million cycles.

Once again, no ‘run-in’ procedures were completed on the test specimens. Starting with the lowest torque level and continuing to the highest, the three load levels tested included L5, L6, and L7. These load levels along with their corresponding contact pressures are listed in Table 2.2. Due to confidentiality requests these torque and contact pressure values have also been normalized.

The same test cycle procedure was continued for the aerospace test specimens while the inspection process was changed. Instead of alternating between both visual and full inspections, only full inspections were completed on the test gears. Furthermore, these full inspections were only completed at 20% intervals of the total expected life due to the prolonged operational life of the high performance gear steel. Since no signs of pitting failure were encountered, a modified failure criterion was used. Tests were suspended if scuffing failures were observed or if the gear set reached 100 million cycles without any signs of failure. Once again after final inspections all test specimens were wrapped in rust-inhibiting paper, labeled, and stored for future reference.
Table 2.2  Load levels and corresponding normalized stresses used during high performance gear steel fatigue tests.

<table>
<thead>
<tr>
<th>Load Level</th>
<th>Normalized Pinion Torque</th>
<th>Normalized Pitch-line Contact Stress</th>
</tr>
</thead>
<tbody>
<tr>
<td>L5</td>
<td>2.16</td>
<td>1.14</td>
</tr>
<tr>
<td>L6</td>
<td>2.81</td>
<td>1.26</td>
</tr>
<tr>
<td>L7</td>
<td>3.92</td>
<td>1.45</td>
</tr>
</tbody>
</table>
2.4 Test Gears and Test Matrix

This fatigue study utilized the same test pinion and test gear parameters as were used in the previous study by Leque [15]. These design parameters, used for both the SAE 4118M and aerospace test specimens, are presented in Table 2.3. The SAE 4118M test specimens used in this study were procured from the same batch as those presented in Leque’s [15] preliminary results. All SAE 4118M test specimens therefore featured the same tooth profile modifications signified as modification “C” in both the study by Leque [15] and Table 2.4. These automotive gears were heat treated to achieve a surface hardness of about 59 Rockwell C with a case depth of around 1 mm. The aerospace test specimens used in this study were procured from a separate batch from those presented in Leque’s [15] preliminary results. These specimens featured a new set of tooth profile modifications signified as modification “D” in Table 2.4. These aerospace gears were carburized and hardened to a case depth of around 1 mm with a surface hardness of 60 HRC. For both the SAE 4118M and aerospace samples, the respective micro-geometry modifications were designed to eliminate excessive contact stresses from the tip and root regions, avoid edge loading, and limit dynamic responses caused by transmission error.

While the SAE 4118M specimens were only hard ground ($R_a \approx 0.4 \mu$m), the aerospace specimens were hard ground and then either chemically polished ($R_a \approx 0.04 \mu$m) or given a high-end super finish ($R_a \approx 0.02 \mu$m) to reduce the surface roughness of the contact surfaces. These three different finishes are shown in Figure 2.11 where 2.11(a) shows a typical ground test gear pair, 2.11(b) shows a typical chemically polished
Table 2.3  Basic design parameters of the baseline spur gear pair used in this study [15]

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Pinion</th>
<th>Gear</th>
</tr>
</thead>
<tbody>
<tr>
<td>Module (mm)</td>
<td>4.23</td>
<td></td>
</tr>
<tr>
<td>Center distance (mm)</td>
<td></td>
<td>91.5</td>
</tr>
<tr>
<td>Number of Teeth</td>
<td>17</td>
<td>26</td>
</tr>
<tr>
<td>Pressure Angle (deg)</td>
<td></td>
<td>22.50</td>
</tr>
<tr>
<td>Face Width (mm)</td>
<td>14.00</td>
<td>20.29</td>
</tr>
<tr>
<td>Root Diameter (mm)</td>
<td>62.87</td>
<td>99.95</td>
</tr>
<tr>
<td>Base Diameter (mm)</td>
<td>66.49</td>
<td>101.69</td>
</tr>
<tr>
<td>Outside Diameter (mm)</td>
<td>80.02</td>
<td>117.11</td>
</tr>
<tr>
<td>Circular Tooth Thickness (mm)</td>
<td>7.81</td>
<td>5.65</td>
</tr>
</tbody>
</table>
Table 2.4  Two different tooth profile modifications considered in this study.

<table>
<thead>
<tr>
<th>Type</th>
<th>Parameter</th>
<th>Modifications</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>C Pinion</td>
</tr>
<tr>
<td>Tip relief</td>
<td>Magnitude (μm)</td>
<td>36</td>
</tr>
<tr>
<td></td>
<td>Start roll angle (deg)</td>
<td>34</td>
</tr>
<tr>
<td>Root relief</td>
<td>Magnitude (μm)</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td>Start roll angle (deg)</td>
<td>12</td>
</tr>
<tr>
<td></td>
<td>End roll angle (deg)</td>
<td>34</td>
</tr>
<tr>
<td>Lead crown</td>
<td>Magnitude (μm)</td>
<td>5</td>
</tr>
</tbody>
</table>
Figure 2.11 (a) Ground, (b) ground and chemically polished, and (c) ground and super-finished test gear pairs [15].
test gear pair, and 2.11(c) shows a typical super-finished test gear pair. In addition to these three finishes, an additional batch of aerospace gears was procured that featured a restrained version of the super-finishing process. This restrained (slightly rougher) version of the super-finishing process was aimed at generating roughnesses of \( R_a > 0.0254 \mu m \) (\( R_a > 1 \mu \text{in} \)). Initial inspections of the test gears were used to verify their manufacturing specifications and to provide a baseline for comparison with subsequent inspections. The average initial roughness parameters (\( R_a \) and \( R_q \)) from these initial inspections for each of the different sets of gears are shown in Table 2.5. These average roughness values correlated to a wide range of lambda ratios (defined as the ratio of the minimum film thickness under ideally smooth conditions to the root-mean-square (RMS) amplitude of the surface roughness) between the automotive and aerospace tests. For the automotive test conditions the lambda ratio was calculated to range from 0.406 to 0.511 while the lambda ratio ranged from 2.87 to 7.35 for the aerospace test conditions.

The SAE 4118M test samples were tested at two speeds and each of the four torque levels in order to generate pitting data for a preliminary statistical analysis. The data collected during this study was added to the limited data generated by Leque [15] during his preliminary testing of SAE 4118M. The aerospace test samples were tested at combinations of the three torque levels and the three oil inlet temperatures in order to generate a matrix of pitting results. Lack of pitting failures requires exploration of more extreme testing conditions in future work if pitting failures are to be realized on the aerospace quality gear samples. For the purposes of this study the suspended tests and scuffing failures will be presented and discussed with relation to the test conditions.
Table 2.5  Average initial $R_a$ and $R_q$ roughness values for the various test specimens.

<table>
<thead>
<tr>
<th>Gear Steel</th>
<th>Surface Finish</th>
<th>$R_a$ (μm)</th>
<th>$R_q$ (μm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>SAE 4118M Hard Ground</td>
<td></td>
<td>0.402</td>
<td>0.520</td>
</tr>
<tr>
<td>High performance gear steel</td>
<td>Chemically Polished</td>
<td>0.043</td>
<td>0.055</td>
</tr>
<tr>
<td>High performance gear steel</td>
<td>Super Finished</td>
<td>0.019</td>
<td>0.026</td>
</tr>
<tr>
<td>High performance gear steel</td>
<td>Super Finished (●)</td>
<td>0.029</td>
<td>0.037</td>
</tr>
</tbody>
</table>

(●) Starting $R_a > 0.0254$ μm (1.0 μ-in)
2.5 Summary

This chapter provided relevant details in the description of the test machines, test methodology, and test specimens used in this gear contact fatigue investigation. Details of the test methodology included both the procedures used for cycle accumulation on the test machines and the procedures used to inspect the gear specimens at pre-determined inspection intervals. The failure criteria for each set of gear specimens were listed and the test matrices were described.
CHAPTER 3

CONTACT FATIGUE TEST RESULTS FOR SAE 4118M STEEL

3.1 Introduction

The macro-pitting fatigue performance of gear specimens made out of SAE 4118M gear steel was evaluated using the test methodology described in Chapter 2 and will be presented in this chapter. The test matrix consisted of four loading levels and two rotational speeds intended to represent high-power automotive gearing applications. A total of 15 tests were performed in this study. These tests were combined with the preliminary results generated by Leque [15] in order to form a data set of 18 tests. All SAE 4118M samples used in these studies were procured from the same batch of test gears, manufactured to the same specifications, and tested with the same test methodology. The pitting results will be presented in stress-life (S-N) format with the horizontal axis representing pinion loading cycles and the vertical axis representing pinion torque or the maximum contact stress.

The test methodology described in the previous chapter was systematically followed to accumulate measurements throughout the course of each test with
consistency. These measurements included surface roughness measurements, involute and lead profile traces, and microscope images of the contact surfaces. By taking such measurements prior to, at predetermined inspection intervals, and at the conclusion of each test, surface wear, micro-pitting, and pitting initiation could be monitored. One representative test at each load level will be illustrated to show the various trends that were observed. In addition, images of each pinion tooth that experienced pitting that exceeded the failure criterion will be shown. While the presented data only represents a small portion of the full database of measurements, all other measurements have been recorded and saved for future reference.

3.2 SAE 4118M Gear Pitting Results

Table 3.1 lists the fatigue lives together with the corresponding pinion rotational speeds and load levels for all the eighteen tests. This database includes three tests (Tests 1, 6, and 7) from the preliminary results of Leque [15] which are marked with solid triangles. These three data points were produced at speeds of 8,000 rpm and 10,650 rpm with the lowest load level L1 that requires relatively long testing time. This database also includes one test (Test 5) that experienced a maximum wear depth of over 12 μm. In accordance with the test methodology this test was considered to have failed due to excessive wear, and therefore, was excluded from the pitting database. Aside from this particular wear failure, all other tests resulted in pitting failure on the test pinion.
Table 3.1  Summary of all SAE 4118M tests presented in this study

<table>
<thead>
<tr>
<th>Test Number</th>
<th>Pinion Speed [rpm]</th>
<th>Load Level</th>
<th>Cycles to Failure [millions]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 (▲)</td>
<td>8,000</td>
<td>L1</td>
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</tr>
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<td>L1</td>
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<td>4</td>
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<td>L3</td>
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<td>2.4</td>
</tr>
<tr>
<td>18</td>
<td>10,650</td>
<td>L4</td>
<td>3.6</td>
</tr>
</tbody>
</table>

(▲) Tests performed by Leque [15]
(∆)  Wear failure
Surface roughness measurements were used to track variations in roughness throughout the fatigue tests. Figure 3.1 compares the roughness profiles measured from the initial, two intermediary, and final inspection points for Test 6 operating at 10,650 rpm and load level L1. Figures 3.2 to 3.4 show similar comparisons for Tests 10, 14, and 15, which operated at 10,650 rpm and load levels L2, L3, and L4, respectively. For Test 14, only one intermediate inspection was completed since the resulting fatigue life was much shorter than expected. As seen in these figures, the arithmetic roughness amplitude \( R_a \) is observed to decrease during the first few million cycles and then remain relatively constant. Additionally, roughness peaks are shown to be rounded off during the contact while the deep valleys remain. This correlates with the roughness values decreasing from their initial value of \( R_a = 0.402 \ \mu m \) (\( R_q = 0.520 \ \mu m \)) to \( R_a = 0.307 \ \mu m \) (\( R_q = 0.403 \ \mu m \)) on average. After reaching a minimum, the roughness amplitudes of several tests were observed to slightly increase after a large number of contact cycles, which was due to the formation and progression of micro-pitted regions on the pinion contact surface. For the tests operating at the lower pinion rotational speed of 8,000 rpm, the measurements yield similar results.

The surface wear of each test was monitored through profile traces performed on a coordinate measurement machine (CMM). The same inspection points shown in the previous roughness plots were again used for each test in order to clearly show the wear as it developed during each test. These plots are shown in Figures 3.5 to 3.8 for Tests 6, 10, 14, and 15, respectively. These traces indicate that the majority of wear occurs between the roll angles of 12° and 25° in the dedendum region of the test pinion (the roll
Figure 3.1 Example roughness traces of the SAE 4118M pinion tooth from Test 6 at load level L1.
Figure 3. Example roughness traces of the SAE 4140M pinion tooth from Test 10 at load level L2.
Figure 3.3  Example roughness traces of the SAE 4118M pinion tooth from Test 14 at load level L3.
Figure 3.4  Example roughness traces of the SAE 4118M pinion tooth from Test 15 at load level L4.
Figure 3.5  Variation of measured tooth profiles of the ground SAE 4118M pinion during Test 6 at load level L1.
Figure 3.6 Variation of measured tooth profiles of the ground SAE 4118M pinion during Test 10 at load level L2.
Figure 3.7 Variation of measured tooth profiles of the ground SAE 4118M pinion during Test 14 at load level L3.
Figure 3.8  Variation of measured tooth profiles of the ground SAE 4118M pinion during Test 15 at load level L4.
angle at the pitch line of the test pinion is 24.6˚). This wear zone corresponds to the location where micro-pitting appears and where both the sliding velocity and contact pressure are high simultaneously.

Digital images of the pinion contact surfaces for Tests 6, 10, 14, and 15 are displayed in Figures 3.9 to 3.12, respectively, with the same inspection points shown. As seen in the figures, micro-pitting developed in the dedendum of the pinion teeth where negative sliding exists. These micro-pitted areas take the appearance of dull grey regions (commonly referred as grey staining) and are observed to grow in size as the tests progressed. Unfortunately none of these pre-selected inspection teeth ultimately experienced macro-pitting that met the failure criteria. Figure 3.13 shows the teeth that met the pitting failure criteria for Tests 6 and 7 that operated at 10,650 rpm and load level L1. The remaining teeth that experienced pitting at 10,650 rpm are shown in Figures 3.14, 3.15, and 3.16 for load levels L2, L3, and L4, respectively. The teeth that experienced pitting at the lower rotational speed of 8,000 rpm are shown in Figures 3.17 and 3.18 for load levels L1 and L3, respectively. These images show that while most macro-pitting occurred in the dedendum regions of the pinion teeth, some tests occasionally experienced pitting in the addendum region. This was likely due to macro-pitting that developed in the dedendum region and then quickly spread up the tooth face into the addendum region. Teeth that were photographed prior to macro-pitting failure support this claim since such images show that most macro-pits initiated at the upper boundary of the micro-pitting bands in the shape of an obtuse shell.
Figure 3.9  Digital images of one of the teeth of the ground SAE 4118M pinion during Test 6 at load level L1 at (a) 0 cycles, (b) 5.0M cycles, (c) 15.0M cycles, and (d) 19.0M cycles.
Figure 3.10  Digital images of one of the teeth of the ground SAE 4118M pinion during Test 10 at load level L2 at (a) 0 cycles, (b) 2.4M cycles, (c) 7.2M cycles, and (d) 10.0M cycles.
Figure 3.11  Digital images of one of the teeth of the ground SAE 4140M pinion during Test 14 at load level L3 at (a) 0 cycles, (b) 1.0M cycles, and (c) 3.0M cycles.
Figure 3.12  Digital images of one of the teeth of the ground SAE 4118M pinion during Test 15 at load level L4 at (a) 0 cycles, (b) 1.2M cycles, (c) 2.4M cycles, and (d) 3.0M cycles.
Figure 3.13  Pictures of failed SAE 4118M pinions for tests at 10,650 rpm and load level L1 (a) Test 6, (b) Test 6 enhanced 12.5X magnification, and (c) Test 7.
Figure 3.14  Pictures of failed SAE 4118M pinions for tests at 10,650 rpm and load level L2 (a) Test 8 tooth 4, (b) Test 8 tooth 9, (c) Test 9, (d) Test 10, (e) Test 11, and (f) Test 12.
Figure 3.15  Pictures of failed SAE 4118M pinions for tests at 10,650 rpm and load level L3 (a) Test 13, (b) Test 14 tooth 7, (c) Test 14 tooth 8, and (d) Test 14 tooth 15.
Figure 3.16 Pictures of failed SAE 4118M pinions for tests at 10,650 rpm and load level L4 (a) Test 15, (b) Test 16 tooth 2, (c) Test 16 tooth 9, (d) Test 17, and (e) Test 18.
Figure 3.17  Pictures of failed SAE 4118M pinions for tests at 8,000 rpm and load level L1 (a) Test 1 tooth 9, (b) Test 1 tooth 15, (c) Test 1 tooth 17, and (d) Test 2.
Figure 3.18  Pictures of failed SAE 4118M pinions for tests at 8,000 rpm and load level L3 (a) Test 3 and (b) Test 4.
It can also be seen that the majority of tests exhibited both micro-pitting and macro-pitting that was offset either left or right of the transverse mid-plane of the pinion teeth. This uneven pitting formation to one side of the pinion teeth is likely due to slight misalignment within the test gearbox or lead slope errors present in the gears. While most tests were suspended due to a single pinion tooth meeting the required failure criteria, some tests had two or three pinion teeth meet the failure criteria at the same inspection point.

3.3 Mean Life and Confidence Interval Construction

The S-N curves with the normalized pinion torque versus pitting life and the normalized pitch-line contact stress versus pitting life are presented in Figures 3.19 and 3.20, respectively. The pitch-line contact stresses at each load level were determined using a gear load distribution model, LDP [24]. The two separate pinion speeds, 8,000 and 10,650 rpm, are specified on each graph. Any overlapping data points due to small differences in the measured lives at the same load level are displayed in a slightly offset vertical orientation in order to clearly indicate the multiple failure points. In these figures, the pitting fatigue life is shown to increase as the load is decreased. Although the fatigue lives measured at the lower rotational speed are shown to correspond to larger contact cycles than those of the higher rotational speed, the limited data points cannot be considered conclusive. Definitive conclusions on the effect of rotational speed on pitting life would require additional tests at 8,000 rpm.
Figure 3.19  Normalized pinion torque versus pitting life of ground SAE 4118M pinions.
Figure 3.20  Normalized pitch-line stress versus pitting life of ground SAE 4118M pinions.
Due to the limited fatigue life difference between the two speed levels, all seventeen data points were combined and analyzed statistically at each load level to determine the 50% probability S-N curve and the corresponding 90% confidence interval. This confidence interval represents the expected range in which 90% of the specimens of a population would fail. The formula used to calculate the 90% confidence interval is given as:

$$C = \bar{x} \pm z \left( \frac{p}{\sqrt{n}} \right)$$

(3.1)

where $C$ is the confidence interval, $\bar{x}$ is the 50% probability of failure (mean life), $z$ is a constant equal to 1.645 for a 90% confidence interval, $p$ is the standard deviation of the population (since the population deviation is unknown the sample deviation was used here), and $n$ is the number of data points in the sample (number of failure points at a given torque load).

Figure 3.21 shows the mean life and the 90% confidence interval of the S-N data for the SAE 4118M gears. Although the sample size is relatively small, Figure 3.21 exhibits a mean life curve that is fairly linear and a 90% confidence interval that is reasonably sized. While scatter is typical in any fatigue data set, load L3 displayed the most scatter in this data set and consequently exhibited the largest 90% confidence interval. It can be easily argued that the sample sizes at each load level are not sufficient to apply Equation (3.1) to obtain statistically meaningful confidence limits. The
Figure 3.21  Pitch-line contact stress versus pitting life for SAE 4118M gears with mean lives and 90% confidence intervals.
methodology to establish such limits are provided here for the benefit of future work which might generate additional test points for this particular of SAE 4118M database.

3.4 Summary

Pitting life results from the contact fatigue testing of ground SAE 4118M test samples were presented in this chapter. Each test utilized test specimens manufactured in the same batch to the same specifications. Pitting life results were presented in pinion torque versus pitting life and the pitch-line contact stress versus pitting life graphs. Roughness measurements, tooth profile traces, and digital images were shown at inspection points throughout one representative test at each of the four load levels tested. Roughness measurements illustrated how roughness of the contact surfaces typically decreased and then remained relatively constant during tests. Tooth profile traces in the involute direction demonstrated how the majority of wear was seen in the dedendum region of the pinion teeth. Digital images were also included that illustrated how large micro-pitting regions in the dedendum of the pinion teeth acted as initiation points for macro-pitting. Interim and final inspections of test specimens revealed a typical failure mechanism that follow the sequence of (i) formation of a micro-pitted region in the dedendum of the pinion, (ii) formation of a macro-pit at the upper boundary of the micro-pit zone, (iii) growth of the macro-pit to a size satisfying the failure criterion.
CHAPTER 4

RESULTS OF HIGH-TEMPERATURE AEROSPACE TESTS

4.1 Introduction

This chapter presents the high-temperature and high-speed rolling contact test results performed using gears made out of a high performance (high-temperature) aerospace steel. These specimens operated under three loading levels (L5, L6 and L7 as defined in Chapter 2) and three inlet oil temperatures (82°C, 107°C and 120°C) using BP Turbo Oil 2380. In order to simulate aerospace gearing applications, all tests were run at pinion rotational speeds of 12,000 rpm. The pinion and gear tooth surfaces were smoothened using one of two processes: (i) chemical polishing or (ii) super-finishing, to achieve roughness amplitudes of $R_a < 0.05 \mu m$. A total of 14 tests were performed, which included 6 tests using the chemically polished gear pairs and 8 tests using the super-finished gear pairs. During each test, the test and inspection procedures described in Chapter 2 were strictly followed to ensure consistency.
Due to the very small roughness amplitudes and high rolling velocities, the tooth surfaces in mesh experienced favorable full-film EHL conditions (lambda ratios between 2.87 to 7.35). These optimized lubrication conditions caused none of the 14 tests to experience pitting (neither micro-pitting nor macro-pitting). The large amount of heating (both load dependent gear mesh friction power and load independent windage power) caused by the very high operating speed did however cause scuffing to become a primary failure mode. Representative surface roughness measurements, involute profiles, and digital images will be illustrated for tests that reached 100 million cycles without any failure and for tests that experienced scuffing. Additionally, the digital images for all scuffed tooth surfaces will be shown.

4.2 Test Results for Chemically Polished Gear Specimens

Table 4.1 lists the test results together with the load levels and lubricant inlet temperatures for all six tests performed with chemically polished gear pairs. These tests were performed at load levels L5 and L6 and lubricant temperatures of 82°C, 107°C and 120°C. Out of these six data points, Test 1 and Test 3 were suspended at 1.4 million and 0.14 million cycles, respectively, due to scuffing. The remaining four tests were suspended at 100 million cycles with no observed failure. In terms of test hours, tests that reached 100 million cycles correlate to 139 testing hours on the machine while the two scuffing failures correlate to 2 hours (Test 1) and 12 minutes (Test 3). These extremely short lives for Test 1 and Test 3 are typical of scuffing failures since scuffing has the possibility to occur quickly.
Table 4.1 Summary of all chemically polished aerospace tests presented in this study

<table>
<thead>
<tr>
<th>Test Number</th>
<th>Load Level</th>
<th>Lubricant Temperature [°C]</th>
<th>Cycles to Failure [millions]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 (∆)</td>
<td>L6</td>
<td>120</td>
<td>1.4</td>
</tr>
<tr>
<td>2 (*)</td>
<td>L6</td>
<td>120</td>
<td>100</td>
</tr>
<tr>
<td>3 (∆)</td>
<td>L6</td>
<td>120</td>
<td>0.14</td>
</tr>
<tr>
<td>4 (*)</td>
<td>L6</td>
<td>120</td>
<td>100</td>
</tr>
<tr>
<td>5 (*)</td>
<td>L5</td>
<td>107</td>
<td>100</td>
</tr>
<tr>
<td>6 (*)</td>
<td>L6</td>
<td>82</td>
<td>100</td>
</tr>
</tbody>
</table>

(∆) Scuffing failure
(*) Suspended without failure
Surface roughness measurements were performed at the inspection intervals of each test to track the changes in roughness. Figure 4.1 compares the roughness profiles measured at 0, 20 million, 60 million, and 100 million cycles of Test 5 operating at load level L5 and 107°C. The $R_a$ values are observed to remain relatively constant throughout the test. In contrast to the ground SAE 4118M gears, the roughness peaks do not appear to be removed as the test progresses due to the full lubrication film established between the surfaces. Figure 4.2 shows the same comparison for Test 4, which was operated at load level L6 and 120°C and also reached 100 million cycles. Even under the more extreme operating conditions (higher load and higher temperature), the variation of the roughness profiles is once again found to be very limited. Coordinate measurement machine (CMM) measurements were used to monitor tooth profiles to determine if any surface wear developed throughout the tests. Figures 4.3 and 4.4 illustrate these measurements made at the same inspection points as shown in Figures 4.1 and 4.2 for Tests 5 and 4, respectively. These traces clearly indicate that no detectable wear was accumulated throughout these tests.

Digital images of the pinion contact surfaces from the same inspection points of Tests 5 and 4 are shown in Figures 4.5 and 4.6, respectively. In agreement with both the roughness measurements and profile traces, no significant changes to the contact surface are seen between inspection intervals. Very fine, sub-micron-width scratches can be seen running in the direction of sliding, however, their severity is amplified by the reflectivity of the surface in combination with the microscope’s lighting. These scratches are likely due to debris in the lubricant. Slight bands of discoloration are also found in both the
Figure 4.1  Example roughness traces of the chemically polished aerospace pinion during Test 5 at load level L5 and 107°C.
Figure 4.2  Example roughness traces of the chemically polished aerospace pinion during Test 4 at load level L6 and 120°C.
Figure 4.3 Involute tooth profiles of the chemically polished aerospace pinion during Test 5 at load level L5 and 107°C
Figure 4.4 Involute tooth profiles of the chemically polished aerospace pinion during Test 4 at load level L6 and 120°C
Figure 4.5  Digital images of one of the teeth of the chemically polished aerospace pinion during Test 5 at load level L5 and 107°C at (a) 0 cycles, (b) 20M cycles, (c) 60M cycles, and (d) 100M cycles.
Figure 4.6  Digital images of one of the teeth of the chemically polished aerospace pinion during Test 4 at load level L6 and 120°C at (a) 0 cycles, (b) 20M cycles, (c) 60M cycles, and (d) 100M cycles
addendum and dedendum regions of the teeth where sliding is relatively high. These areas likely represent a thin surface tribo-film formed by the reaction between the surface and the lubricant additives under the high temperature conditions. In all aspects, the other two chemically polished gear pairs that reached 100 million cycles (Tests 2 and 6) yielded similar measurement results as well.

While two tests, Tests 2 and 4, reached 100 million cycles at load level L6 and 120°C, Tests 1 and 3, experienced scuffing under the same operating conditions. Scuffing is often considered to result from a lubrication failure that leads to an increase in friction and instantaneous temperature rise in the gear mesh. While this is typically attributed to the lubricant’s performance under certain operating conditions, such a justification does not explain the two successes and two failures that occurred under the same operation conditions. Furthermore the two successful tests were completed on the opposite flanks of the gear pairs that experienced scuffing (Tests 1 and 2 were both completed on gear pair two while Tests 3 and 4 were both completed on gear pair three). A detailed investigation was carried out to describe this issue. It was found that while the operating conditions and machine performance were consistent among the four tests, the surface hardness of the contact surfaces correlating with the scuffed tests were measured to be less than the required 60 HRC. While this could have been caused by unequal grinding of each flank during manufacturing, it could not be determined whether the reduced hardness was a result of manufacturing error or a result the scuffing event itself. Further investigation confirmed that this hardness difference between flanks was isolated to these two gear pairs.
CMM measurements at the beginning and end of Test 3 are shown in Figure 4.7 for the test pinion and Figure 4.8 for the test gear. These plots illustrate the material removal in the dedendum of the pinion and in the addendum of the gear that resulted from the welding and tearing of the mating surfaces during scuffing. This failure led to increased vibration and triggered the health monitoring equipment to automatically shut down the machine. Digital images of the tooth surfaces before and after scuffing are shown in Figure 4.9 for Test 3 and Figure 4.10 for Test 1. In these images, the scuffing bands are evident at the same locations as those indicated by the CMM measurements for both the pinion and gear contact surfaces.

4.3 Test Results for Super-finished Gear Specimens

Table 4.2 lists the test results along with the corresponding load levels and lubricant inlet temperatures for the super-finished gear pairs. Eight tests in total were performed at load levels L6 and L7 and lubricant temperatures of 107°C and 120°C. Five of these tests used typical super-finished gear pairs while the remaining three tests used gears that featured a restrained version of the super-finishing process aimed at generating roughnesses of $R_a > 0.0254$ μm ($R_a > 1$ μ-in). Out of these eight tests, only two reached 100 million cycles without failure while the remaining six tests experienced scuffing at various points during testing.
Figure 4.7 Involute tooth profiles of the chemically polished aerospace pinion during Test 3 at load level L6 and 120°C.
Figure 4.8 Involute tooth profiles of the chemically polished aerospace gear during Test 3 at load level L6 and 120°C.
Figure 4.9  Digital images the chemically polished aerospace gear pair during Test 3 at load level L6 and 120°C at (a) Pinion tooth at 0 cycles, (b) Pinion tooth at 0.14M cycles, (c) Gear tooth at 0M cycles, and (d) Gear tooth at 0.14M cycles.
Figure 4.10  Digital images the chemically polished aerospace gear pair during Test 1 at load level L6 and 120°C at (a) Pinion tooth at 0 cycles, (b) Pinion tooth at 1.4M cycles, (c) Gear tooth at 0M cycles, and (d) Gear tooth at 1.4M cycles.
Table 4.2  Summary of all super-finished aerospace tests presented in this study.

<table>
<thead>
<tr>
<th>Test Number</th>
<th>Load Level</th>
<th>Lubricant Temperature [°C]</th>
<th>Cycles to Failure [millions]</th>
</tr>
</thead>
<tbody>
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<td>7 (∆)</td>
<td>L7</td>
<td>120</td>
<td>1.2</td>
</tr>
<tr>
<td>8 (∆)</td>
<td>L6</td>
<td>120</td>
<td>23.7</td>
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<td>9 (∆)</td>
<td>L6</td>
<td>120</td>
<td>77.7</td>
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<tr>
<td>10 (∆)</td>
<td>L6</td>
<td>120</td>
<td>17.0</td>
</tr>
<tr>
<td>11 (*,●)</td>
<td>L6</td>
<td>107</td>
<td>100</td>
</tr>
<tr>
<td>12 (*,●)</td>
<td>L6</td>
<td>107</td>
<td>100</td>
</tr>
<tr>
<td>13 (∆)</td>
<td>L6</td>
<td>107</td>
<td>37.0</td>
</tr>
<tr>
<td>14 (∆,●)</td>
<td>L6</td>
<td>120</td>
<td>0.16</td>
</tr>
</tbody>
</table>

(∆) Scuffing failure  
(*) Suspended without failure  
(●) Starting Ra > 0.0254 μm (1.0 µ-in)
Of the six tests that experienced scuffing, only two tests failed due to scuffing before the first inspection point at 20 million cycles. These tests (Tests 7 and 14) failed at 1.2 million and 0.16 million cycles, respectively. Test 7 was run at the most extreme conditions tested, load level L7 and 120°C, and used a gear pair finished with the typical super-finishing process. CMM measurements from Test 7 are shown in Figure 4.11 for the test pinion and Figure 4.12 for the test gear. These plots show substantial material loss in the dedendum of the pinion and both in the dedendum and addendum of the gear. This substantial material loss corresponds to a severe scuffing failure that has spread across the entire contact surface of the gear teeth. This severe scuffing failure can also be seen in Figure 4.13 where it is easily visible that the majority of the contact surfaces have been damaged.

While the scuffing failures seen in Tests 7 and 14 occurred at the beginning of the tests, the remaining four scuffing failures (Tests 8, 9, 10, 13) occurred at 23.7, 77.7, 17.0, and 37.0 million cycles, respectively. It is important to note that, in accordance with the test methodology presented in Chapter 2, the tests were conducted in 20 million cycle increments. Therefore, these four tests (Tests 8, 9, 10, 13) actually scuffed at 3.7, 17.7, 17.0, and 17.0 million cycles into a 20 million cycle test block, respectively. These cycle counts of 3.7 and 17 million cycles correspond with approximately five and twenty-four hours of testing, respectively. Considering all tests are held at constant rotational speeds, loads, and jet lubrication conditions, these delayed scuffing events are uncharacteristic. Since no abnormal events were observed until the scuffing occurred, these delayed failures can hardly be explained since the scuffing phenomenon usually takes place right
Figure 4.11 Involute tooth profiles of the super-finished aerospace pinion during Test 7 at load level L7 and 120°C.
Figure 4.12  Involute tooth profiles of the super-finished aerospace gear during Test 7 at load level L7 and 120°C.
Figure 4.13  Digital images the super-finished aerospace gear pair during Test 7 at load level L7 and 120°C at (a) Pinion tooth at 1.2M cycles, (b) Gear tooth at 1.2M cycles.
after the contact starts due to the extreme local surface temperatures. One possible explanation could be the gradual rise of the gears’ bulk temperatures due to heating by windage and the gear mesh friction power. While this might explain the delayed scuffing events it could not be confirmed with the machines current health monitoring equipment. Similar delayed scuffing events on ball and disk specimens made from the same high performance aerospace steel with the same super-finishing process were observed in a confidential third party study initiated by the project sponsor.

Tests 8, 9, and 10 were run at load level L6 and 120°C while Test 13 was run at load level L6 and 107°C. CMM measurements from Test 13 are shown in Figure 4.14 for the test pinion and Figure 4.15 for the test gear. These plots indicate almost no material loss in the dedendum of the test pinion while the majority of material loss is seen in the addendum of the test gear. These plots also show no tooth profile geometry changes at the 20 million cycle inspection point. Digital images of both the pinion and gear contact surfaces during Test 13 are provided in Figure 4.16 while Figure 4.17 shows images of the remaining four tests that scuffed. Overall these measurements show that aside from differences in cycles to failure, delayed scuffing events failed in the same manner as the immediate scuffing failures.

In order to investigate the increased likelihood of scuffing with the super-finished gear pairs, several gear pairs were given a restrained version of the super-finishing process. These gear sets were procured by the project sponsor in order to investigate the effect of surface roughness on the ability to avoid scuffing failure. Since chemically
Figure 4.14  Involute tooth profiles of the super-finished aerospace pinion during Test 13 at load level L6 and 107°C.
Figure 4.15  Involute tooth profiles of the super-finished aerospace gear during Test 13 at load level L6 and 107°C.
Figure 4.16  Digital images the super-finished aerospace gear pair during Test 13 at load level L6 and 107°C at (a) Pinion tooth at 0M cycles, (b) Pinion tooth at 20M cycles, (c) Pinion tooth at 37M cycles, (d) Gear tooth at 0M cycles, (e) Gear tooth at 20M cycles, (f) Gear tooth at 37M cycles.
Figure 4.17 Digital images the super-finished aerospace gear pairs that experienced scuffing including (a) Test 8 pinion, (b) Test 8 gear, (c) Test 9 pinion, (d) Test 9 gear, (e) Test 10 pinion, (f) Test 10 gear, (g) Test 14 pinion, (h) Test 14 gear.
polished gear sets \((R_a = 0.043 \, \mu m)\) were outperforming super-finished gear sets that are smoother \((R_a = 0.019 \, \mu m)\), these additional gear sets allowed the restrained super-finishing process \((R_a = 0.029 \, \mu m)\) to be added to the comparison. Out of the three tests completed with the restrained super-finishing process, Test 14 resulted in scuffing at load level L6 and 120°C, as previously described, and two tests (Tests 11 and 12) reached 100M cycles without failure at load level L6 and 107°C.

Measurements completed throughout Tests 11 and 12 were very comparable to those seen during the various successful tests completed with chemically polished gear pairs. Roughness measurements taken from Test 11 at 0, 20 million, 60 million, and 100 million cycles are shown in Figure 4.18. Similar to the successful chemically polished tests the arithmetic roughness amplitude is observed to remain relatively constant throughout the test. CMM measurements taken at the same inspection intervals and shown in Figure 4.19 and clearly indicate that no detectable wear occurred during the test. Finally, in agreement with the roughness and CMM measurements, the digital images shown in Figure 4.20 show no significant changes to the contact surface aside from sub-micron level scratches that are amplified by the high reflectivity of the surface. While Tests 11 and 12 were the only super-finished tests to reach 100 million cycles without failure, the limited number of data points with the restrained super-finishing process makes any conclusions about the effect of surface roughness on the ability to avoid scuffing only preliminary until further testing is conducted.
Figure 4.18  Example roughness traces of the super-finished aerospace pinion during Test 11 at load level L6 and 107°C.
Figure 4.19  Involute tooth profiles of the super-finished aerospace pinion during Test 11 at load level L6 and 107°C.
Figure 4.20  Digital images of one of the teeth of the super-finished aerospace pinion during Test 11 at load level L6 and 107°C at (a) 0 cycles, (b) 20M cycles, (c) 60M cycles, and (d) 100M cycles.
4.4 Summary

Contact fatigue testing results for the high performance, aerospace quality gears were presented in this chapter. These tests utilized test specimens that were hard-ground to the same basic specifications and then given one of three different finishes. Gears were either chemically polished ($R_a = 0.043 \, \mu m$), super-finished ($R_a = 0.019 \, \mu m$), or super-finished with a restrained process ($R_a = 0.029 \, \mu m$). Tooth profile traces, roughness measurements, and digital images were shown at various inspection points of a number of tests in order to illustrate the various experimental results. While no evidence of micro-pitting or macro-pitting was encountered, some tests reached 100 million cycles without failure while other tests experienced scuffing. Comparisons between the performances of each finishing process were briefly touched upon in this chapter and will be discussed in further detail in Chapter 5.
CHAPTER 5

SUMMARY AND CONCLUSIONS

5.1 Thesis Summary

In this thesis study, contact surface failures of high-speed gears in the form of fatigue (specifically micro-pitting and macro-pitting) and gear scuffing were investigated experimentally. Specially designed spur gears test specimens were tested in customized test machines operating under high-power and high-temperature conditions. The machines used in this study were designed to be capable of pinion rotational speeds up to 13,500 rpm, pinion torques up to 450 N-m, and jet lubrication operating at oil pressures up to 150 psi and temperatures up to 150°C. Both automotive and aerospace applications were considered in these tests.

Testing for the simulation of high-power automotive applications utilized hard ground gear specimens made out of SAE 4118M material. These tests focused on the macro-pitting fatigue performance of the gear specimens while the progression of micro-pitting was also observed. Tests were conducted at two pinion rotational speeds and four load levels. The lubricant, a typical automatic transmission fluid, was held at an inlet
temperature of 90°C and was delivered to the test gear mesh at high pressure by six nozzles. All tests were shown to produce macro-pits whose lives were presented in a stress-life plot.

Aerospace applications were simulated using hard ground gear specimens made out of a high performance (high-temperature) steel. In addition, the tooth surfaces of the test specimens were finished using one of two processes: (i) chemical polishing or (ii) super-finishing, to achieve the roughness amplitudes of $R_a < 0.05 \mu m$. Tests were conducted at pinion rotational speeds of 12,000 rpm and three load levels. The lubricant, BP Turbo Oil 2380, was delivered to the test gear mesh at high pressure and either 82°C, 107°C, or 120°C by twelve nozzles. Due to the extremely smooth contact surfaces no tests experienced contact fatigue failures (neither micro-pitting nor macro-pitting). Tests were observed to either fail due to scuffing or they were suspended after 100 million cycles if no failure was observed.

5.2 Conclusions

Based on the test results presented in Chapters 3 and 4, the following general observations and conclusions can be made:

- The preliminary statistical analysis of the SAE 4118M pitting results demonstrates that the test methodology previously developed by Leque [15] is
capable of creating a large database of pitting results that resembles typical
gear fatigue testing results.

- The lack of contact failures (specifically micro-pitting and macro-pitting) during aerospace testing confirms the preliminary observation by Leque [15] that the combination of the high performance gear steel and the reduction of surface roughness, using either the chemical polishing or super-finishing processes, significantly improves the fatigue life of spur gears.

- The occurrence of scuffing failures during testing with the high performance aerospace gear specimens illustrates that scuffing is still a major concern and likely failure mode with the current test methodology.

- Gear specimens finished with the chemical polishing process were shown to be more effective at avoiding scuffing failure compared to specimens finished with the super-finishing process.

- It was shown through limited experimentation that surface roughness may be a key factor in the occurrence of scuffing failures. Specifically it might be possible to make surfaces too smooth for reliable performance. Additionally, the unknown manufacturing methods used in the proprietary super-finishing process might adversely affect resistance to scuffing.
5.3 Recommendations for Future Work

While the efforts of this study have made considerable progress since the preliminary results of the test methodology were presented by Leque [15], this methodology can be further employed to generate an even larger database of test results. In order to further understand the behavior observed during this study, the following can be considered areas of future work:

- Further testing of SAE 4118M samples can be performed in order to generate a sufficient sample size for a robust statistical analysis.

- Additional testing with hard ground test specimens made from other gear steels can be completed so comparisons can be made about the pitting performance of SAE 4118M.

- More extreme test conditions, specifically increased contact stresses or an increased cycle limit before test suspension, can be employed in an attempt to produce pitting failure (either micro-pitting or macro-pitting) on the aerospace gear specimens.

- Additional aerospace tests should be performed to define scuffing free regions on a contact stress versus oil temperature plot.

- Modeling of the scuffing and pitting failures observed in these experiments must be done to bring a physical understanding to the behaviors observed.
- The isolation of variables, specifically surface roughness and type of finishing process, can be undertaken in order to identify the root causes of the increased likelihood of scuffing with the super-finished test specimens.

- The analyses of tribo-films formed on gear surfaces and investigations of operating conditions and surface roughnesses that are optimal for such films to form might shed insight into factors leading to scuffing failures.

- Analysis of the lubricant throughout its life span for cleanness and particulates is advisable to determine any effect this might have on scuffing failures.
References


