EXPERIMENTAL ASSESSMENT OF THE DYNAMIC FACTORS OF FZG GEAR TESTS AS WELL AS AN EVALUATION OF A MICROPITTING LOAD CAPACITY FORMULA

THESIS

Presented in Partial Fulfillment of the Requirements for the Degree Master of Science in the Graduate School of The Ohio State University

By

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Abstract

This research investigates the effect of dynamics on the performance of an FZG test rig as well as develops further understanding and analysis of micropitting by the development of a MATLAB based computer program and an experimental study of this phenomenon. Lubrication, speed, torque, and surface finish were the parameters studied in the dynamics portion of the research to understand their influence on the dynamics of the system. Post-processing the data involved reproducing the plots that were acquired during the experiment and calculating a dynamic factor, which is the peak root stress divided by the static root stress. A statistical analysis was done to get a minimum, maximum, and average dynamic factor and observe how it changes with respect to torque and speed. Computer programs were utilized to supplement the dynamics investigation in an attempt to interpret the experimental data. The ISO/TR: 15144 is a procedure that outlines a detailed review of determining the micropitting load capacity and the computer program that was developed in this research computes the equations of this procedure and determines the safety factor that is used to establish the onset of micropitting. The main detection of micropitting was by observing the profile changes of the pinion and gear after each time varying test. Two lubricants were used in the experiment which had different viscosity grades. Only ground gears were tested for the micropitting testing.
Dedication

This document is dedicated to my Savior Jesus Christ, my dad, mom, Isaiah, Darius, and my future wife.
Acknowledgments

First I would like to thank Dr. Ahmet Kahraman for giving me this opportunity to join the Gearlab for my research endeavors. I am deeply grateful for Dr. Donald Houser bringing me under his research project and offering his rich breadth of engineering knowledge to fully equip me for my engineering career. He is a true engineer and the knowledge and skills I learned under him are irreplaceable and invaluable. I would like to thank Sam Shon for his help in providing me the skills and knowledge for my experiment. His patience and support throughout this process cannot go unnoticed. I am also deeply grateful for Chad Bivens and his help with getting my machine together. I bothered him a lot throughout this process and his patience and kindness deserves praise. He played a major role in the success of this research. Joe West was an amazing help in aiding me in the electronics portion of my experiment. Without these three guys I would not have completed my project so I cannot express how thankful I am for them. I am grateful for Nick Breckenridge and his mention of Dr. Kahraman and also helping get the fellowship I received. Anindo thanks for the many conversations and good times we had. I cannot reiterate enough that above all I would like to thank God for giving me life and opening up this door.
Vita

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Fields of Study

Major Field: Mechanical Engineering
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CHAPTER 1

INTRODUCTION

1.1 Background and Motivation

The ISO/TR 15144-1:2010E [1] is a fairly recent document that outlines the procedure to create a safety factor that identifies when micropitting will occur. This is a purely analytical procedure that includes numerous equations to ultimately pinpoint the minimum specific lubricant film thickness across the entire meshing surface. A key factor in determining the safety factor is the permissible specific film thickness. The procedure gives a plot of how to retrieve this value and it is supposedly based on experimental results. However, there is no experimental data available for public use to validate this procedure which is wearisome considering the fact this it is an internationally approved report. Essentially all scientists and engineers who use this procedure are following this method with the hope that it is accurate. Part of the motivation in conducting research
with respect to this procedure is to provide the scientific community with experimental
data that would either validate or disprove this ISO document.

Micropitting has attracted significant interest over the recent years, in part because of
its subtle impact on the tooth flanks that occurs during mesh. There are numerous factors
that can account for micropitting such as the sliding interactions that happen between
teeth, flash temperature, load distribution, lubrication, additives in the lubricant, and
contact stresses to name a few. The hope in performing actual fatigue tests is to analyze
such causes of micropitting and observe how the wear forms in the post processing
analysis. There are also a lot of assumptions made in the lubricant film thickness
calculation in the ISO document that may not reflect what actually happens when the gear
teeth are in mesh.

For the most part, the calculation follows a modified version of the Dowson
elastohydrodynamic (EHD) equation. This equation assumes steady state rolling, a
constant radius of curvature, and no sliding interaction. This limitation led to a
micropitting model developed by Li [2], a senior researcher at The Ohio State University,
which accounts for transient interactions during mesh of rough surfaces as well as sliding
velocities.

The AGMA supplied tribology gears will be used in this study and the experimental
results will be analyzed in this micropitting model to compare with the ISO procedure for
determining the micropitting load capacity. Part of this study was the assembly of a
refurbished FZG tester to not only test these gears for micropitting but also to reenact a test conducted by Houser, et al [3] in 2006.

In 2006, an evaluation of the FVA Information Sheet No. 54/I-IV [4] was conducted using the AGMA tribology gears that were of the same kind that is expressed in the technical paper [5] and load stage testing was done to determine the micropitting load capacity according to the information sheet. The technical paper done by Houser, et al [3] prompted future investigation of the dynamic performance of the FZG test rig. A peculiar T-shaped contact pattern was found on the flanks of both the pinion and gear. The deduction following the publication of the results was that dynamics was the cause of the T-shaped contact pattern. Figures 1.1 and 1.2 show the contact pattern of the pinion and gear, respectively, with the dynamics likely causing the broad contact region. Since the dynamics increases the amount of load in the system as indicated by the wider contact across the face width, it is important to retrieve the appropriate factor to account for the increased load in load distribution analysis and in calculating the proper contact stress.

Dynamic loading is becoming more of an interest in the gear research community because of its impact on the performance of any gearbox system. There is a corresponding factor, $K_v$, that is applied in calculating the Hertzian contact pressure distribution. This factor is vitally important because of the potential increase of the local pressure on the tooth flank. The dynamics can be such a problem that the actual torque in the system is close to double of the nominal torque. Determining an appropriate dynamic loading factor is vital in correctly assessing not only the contact stress in the tooth mesh
cycle but also the performance and consistency of the test rig. The results that come from the dynamic testing were to hopefully bring confidence in accounting for the load distribution for predictive analyses such as LDP.

Figure 1.1 Pinion contact pattern of 2006 test

Figure 1.2 Gear contact pattern of 2006 test
1.2 Dynamic Loading Literature Review

There have been previous studies done on the dynamic performance of certain gearbox systems. Dynamic analyses, whether theoretical or experimental, have been investigated in a manner similar to the research that is discussed in Chapter 2 of this thesis in the “Experimental Methodology” section. It is vitally important to understand specifically the parameter that is being identified in the experiment. In this particular case, dynamic loading is a broad area of research that can be studied in many different ways. These have dynamic factors that were studied in great detail by Harianto [6] more than a decade ago. In his thesis, he documented the fact that there are many different dynamic factors that should be considered by the engineer when studying the effect of this phenomenon in gear system analyses. He noted that there are six different definitions of dynamic factors that include Dynamic Load Factor, Dynamic Tooth Force Factor, Dynamic Bending Moment Factor, Dynamic Contact Stress Factor, and Dynamic Root Stress Factor. The sixth definition comes from AGMA and their Dynamic Load Factor, $K_p$, is determined by very different means than the first five as will be shown in the table below. The AGMA factor is highly dependent on gear manufacturing quality and pitch line velocity and is determined by empirical means. Table 1.1 shows the six factors with their corresponding definitions.
Table 1.1 Dynamic Factor Definition

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<td>Dynamic Load Factor</td>
<td>Constrained to only single tooth pair contact where the static load of a tooth pair is divided by the dynamic load of the same tooth pair.</td>
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<tr>
<td>Dynamic Tooth Force Factor</td>
<td>Considers tooth pair in contact and computes the static force of the particular tooth in contact and compares it with the dynamic tooth force of that same tooth (very similar to dynamic load factor).</td>
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<tr>
<td>Dynamic Bending Moment Factor</td>
<td>Utilizes dynamic tooth force and multiplies it by moment arm at the location of critical root stress found by geometry factor calculation and is compared with the static moment.</td>
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<tr>
<td>Dynamic Contact Stress Factor</td>
<td>Used mostly in surface durability analysis, it is computed by dividing the static contact stress on a tooth pair to the peak contact stress and is distinguished from the Dynamic Tooth Force Factor due to the location of the highest dynamic load.</td>
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<tr>
<td>Dynamic Root Stress Factor</td>
<td>This factor divides the static root stress by the highest dynamic root stress. The difference between this and the Dynamic Bending Moment Factor is the location at which the highest root stress occurs. This is more accurate since the critical stress occurs mostly at the root.</td>
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</table>
Harianto narrowed his research to comparing only the Dynamic Load Factor, Dynamic Tooth Force Factor, and Dynamic Bending Moment Factor with the experimental results of NASA (National Aeronautic and Space Administration) [7]. He used four different computer program simulations to compare the natural frequencies obtained by NASA with the results of the program. One particular program developed by Ozguven [8], DYTEM, was utilized to compare dynamic results with that of NASA. With the specifications of the gears noted in the literature [6], the computer results showed that the dynamic factors ranged from 2.1 to 2.8. Even though these dynamic factors are of little consequence in this study since they used a different test machine and a different gear pair, a similar path was followed in that computer programs were used to understand the experimental results.

Also important in understanding the full range of dynamic activity that takes place during a gear mesh cycle is determining the natural frequency of the system. Harianto [6] compared the natural frequencies obtained by the computer programs to that of NASA. Rebbechi, et.al. [7] investigated NASA’s gearbox system of the gear specifications expressed in Harianto [6]. The natural frequency that was determined was 3400 Hz. Rebbechi, et.al. [7] have an experimental setup that is similar to the one expressed within the contents of this thesis. Mounting strain gages to the roots of the gear of interest and a data acquisition system were the instruments used to observe the dynamics and retrieve strain signals. The main difference in their experiment is the specific gear data that NASA [7] used which was a unity gear pair ratio of 28 teeth. Thus it is difficult to
compare exact results of their experiment with that of the experiment within this thesis. The gain from observing NASA’s experiment is comparing the speeds that they used (800, 2000, 4000, 6000 rpm) with the sample rates that they took (20000 Hz and 50000 Hz). Figure 1.3 shows one of the strain signals that were obtained in their experiment. Even though their gear geometry and parameters are different than the gears used in the dynamic portion of this thesis, this plot reveals the same signal that was acquired from the tests that were done that is expressed in later chapters of this thesis. Essentially for Rebbechi’s experiment, strain gages were placed on adjacent teeth to observe the load sharing and effective contact ratio. Also in this figure are the speeds that were mentioned above and its corresponding dynamic strain signal. The strain is plotted against the roll angles of the two strain gaged teeth. The oscillations are another important feature in this plot which are essential in determining the natural frequency. It was beneficial to understand the pertinent correlation between sampling rates and gear speed to get an uncompromised signal.
Dynamic loads are also of interest because of their impact on the noise that may result. This noise can generally be broken down into two parts, whine and rattle. The desire to minimize noise in the gear housing when a gearbox is running led to many research endeavors to understand this phenomenon. One in particular, by Tamminana et al [9], studied the relationship between the dynamic transmission error (DTE) and different components of the dynamic factor. Experimental and analytical methods were employed to determine this relationship and propose a formula to calculate such values.
Their experiment involved utilizing gears of modified and unmodified profiles, gears of different contact ratios, and testing at different speeds (different mesh frequencies). In this analysis, their focus was only on the dynamic mesh force factor, dynamic tooth force factor, and the dynamic bending stress factor. Their data found that the dynamic factor based on tooth mesh forces was consistently higher than the other two. Even though the gear geometry was different than what was used in the experiment expressed in this thesis, it is good to get an idea of the range of the dynamic bending stress factors retrieved. Since Harianto [6] only conducted a predictive analysis, this paper [9] is helpful due to its experimental verification.

Wang [10] also did a study of dynamics of a spur gear system but his focus was on the inclusion of backlash in the analysis which presents a nonlinear parameter in a linear system. Since backlash causes tooth separation to occur during mesh, Wang used analytical methods to account for this. He also took into account the static transmission error and the time-varying mesh stiffness. His results mostly focus on the amount of power in the system, comparing 12 kW and 15 kW at a speed of 2000 rpm and documents the results. Within backlash, he analyzed a no impact scenario, single-sided impact, and double-sided impact. He concluded that the external load had an immense effect on the dynamic response and the larger the load, the more teeth separation occurred.

Since there are many parameters that affect the dynamics of a gear system, there have been many studies on focusing on one or two of those parameters and their influence.
Sliding, for example, has tremendous potential to provide an external excitation as a frictional force and amplify the dynamics considerably. This was studied by Vaishya and Houser [11] when they compared an analytical and experimental gear dynamic system against the different modes of excitation. They also looked at the different calculations of the lubricant film thickness and different formulations of the coefficient of friction. Their predictions and results showed that sliding friction had a large impact on the dynamic behavior of the gear system.

Influencing factors, including dynamics, are documented in the calculation of the load capacity of spur and helical gears of the ISO 6336-1:2006 [12]. The goal of the procedure is to establish formulae that calculate pitting resistance and the bending strength capacity. Most of these calculations are directly related to assessing the dynamics of the system. Tooth mesh stiffness, equivalent misalignment, and shaft bending deflections are just a few of the calculations made that influence the overall dynamics of the system. There are however, strict restrictions with applying this method. For example, the contact ratio has to be greater than 1 and less than 2.5. Overall this procedure gives an adequate estimation of the optimum tip relief for gears with modification which is directly connected to the determination of the micropitting load capacity [1].

In Dudley’s [13] *Handbook of Practical Gear Design*, he discusses dynamic load theory and the different ways they arise, which may be similar, if not the same reason to why the FZG experiences dynamic loads. One reason is due to the variation in torque between the input and output stages. He says the load may change because of
perturbations with the motor/engine making the power and the machine absorbing the power. A second reason could be from torsional vibration of the apparatus making the power, in this case the motor, which could have a natural frequency that is equal to that of another inertia rotating in the gear system. The third theory that Dudley postulates, which may be the most applicable, is the fact that the torque may vary as gear teeth roll through mesh. This could be a result of changes in tooth mesh stiffness, tooth alignment, and the effect of transmission error.

1.3 Micropitting Literature Review

Since this research is comprised of both dynamics and micropitting, it is appropriate to document literature that is related to the experiment and investigation discussed in this thesis. Since there are a plethora of micropitting papers, a literature search of micropitting was done and is included in the appendix section of this thesis. The most pertinent document is the ISO/TR 15144-1:2010 (E) procedure [1] that outlines the standardized way to determine the micropitting load capacity. It details specific equations that all affect the overall determining factor in micropitting detection, the micropitting safety factor. According to the procedure, the safety factor is the parameter that predicts if micropitting occurs. The safety factor is the minimum specific film thickness divided by the permissible film thickness that is shown in equation 1.1:

\[ S_\lambda = \frac{\lambda_{GF, min}}{\lambda_{GFP}} \geq S_{\lambda, min} \]  

(1.1)
where $\lambda_{GF,\text{min}}$ is the minimum specific film thickness which is the lubricant film thickness divided by the mean roughness. According to Kissling [14], who performed a calculation of the procedure [1], the risk of micropitting increases with decreasing values of the safety factor. Table 1.2 summarizes the scheme of how micropitting is determined.

<table>
<thead>
<tr>
<th>Micropitting Risk</th>
<th>$S_\lambda$</th>
</tr>
</thead>
<tbody>
<tr>
<td>No risk</td>
<td>$&gt; 2$</td>
</tr>
<tr>
<td>Medium risk</td>
<td>$1 \leq S_\lambda \leq 2$</td>
</tr>
<tr>
<td>High risk</td>
<td>$&lt; 1$</td>
</tr>
</tbody>
</table>

This procedure, however, is very empirical and the actual method by which these equations in the procedure were procured is unknown to the scientific community. It is important to note that there are two different methods (A and B) for determining both the local specific film thickness and permissible film thickness expressed in the procedure.

One way of calculating the local specific film thickness is by using a load distribution program that calculates the contact stresses on the whole tooth topography. This is described as Method A and LDP (Load Distribution Program) [15] is an applicable program that will give the contact stress at any location across the face width, which will be discussed in greater detail in chapter 2. Method B is a simplified approach to
determine the minimum local film thickness taken at seven locations along the line of action. These seven points (A, AB, B, C, D, DE, E) correspond to the start and end of the active profile, the pitch point, and two points between each of the start and end of the active profile and the pitch point. The most challenging part of the entire procedure is obtaining the permissible specific film thickness. There are also two different methods in determining the permissible specific film thickness, a Method A and B (different from the earlier methods A and B). Method A emphasizes experimental investigations in running actual gears, which is dependent upon experience by the engineer in micropitting. Method B utilizes the critical specific film thickness, which is found by any standard micropitting test such as the FVA-Information Sheet 54/7 [4]. The permissible specific film thickness for non-ISO viscosity grades is found by equation 1.2:

\[ \lambda_{GFP} = 1.4W_w \lambda_{GFT} \]  

(1.2)

where \( W_w \) is the material factor and \( \lambda_{GFT} \) is the specific film thickness retrieved by tests.

Kissling [14] has documented calculations using the ISO/TR 15144-1:2010 (E) procedure [1] where he used both Method A and Method B to determine the micropitting load capacity. He also analyzed the incorporation of profile modification and how that changed the results of Method B. The ISO procedure [1] describes a way to determine the optimum profile modification, in which the tip relief of the pinion, \( C_a \), must equal the effective tip relief \( C_{eff} \). In Kissling’s study, he actually found that the highest safety factor (lowest risk of micropitting) had a profile modification lower than the optimum tip relief, \( C_{eff} \).
In this thesis, a Matlab program has been written that, much like Kissling, applies both methods A and B, but for all examples, only method A is used, since it properly deals with the heavy shifts in load distribution that are due to the large profile and lead crowning of the test gears. With regard to the permissible lube film thickness, this thesis essentially follows the standard testing numbers of Kissling, but no tests were run to actually experimentally obtain the permissible values. In theory, lubricant manufacturers should provide these numbers for their lubricants, but little data of this nature is available in their literature.

Errichello [16] wrote a technical paper on the morphology of gears and roller bearings with respect to micropitting. He discussed the numerous factors that can affect where and how gears micropit. He emphasized the importance of sliding velocities and the higher probability that micropitting will occur in the region where sliding is the highest, which is typically on the dedendum of the tooth flank. He also noted how the micropitting area may not always be in the center of the root of the flank, but could change in location due to the geometric stress concentration. According to Errichello [28], this could occur at the edges of teeth, corners of tip relief, boundaries of debris or wear indentations, and where the elastohydrodynamic lubrication (EHL) layer is disrupted. With this understanding, it will be easier to document the formation of micropits with the experimental tests conducted in this thesis.
1.3 Scope and Objective

Since the research in this thesis comprises of three distinct features, it is important to articulate the range of parameters that were being studied by each main feature. As stated earlier, the components of this thesis are an experimental investigation of the dynamics of the FZG machine, an observation of the formation of micropitting with respect to load and lubrication, and the creation of a MATLAB created program to accommodate the features of Method A and Method B of ISO/TR 15144 procedure [1]. Dynamics is an in-depth field, as well as micropitting, and there are a number of ways to approach both topics. The scope and objectives are as follows:

- Focus on the Dynamic Root Stress Factor since strain gages are mounted to the roots of the gear. The strain signal, multiplied by Young’s Modulus, most closely resembles the root stress, thus, the peak stress divided by the static stress would yield this particular factor.

- The external factors in the dynamic experiment were limited to varying torque, lubrication, and speed while the internal factor was the finish type of the gear. Both superfinished and ground gears, with the same amount of teeth, were compared against each other.

- While dynamics can vary greatly between different families of gears, (helical, spur, bevel, worm) only spur gears were tested.

- When testing for micropitting, only two lubrications were used, Mobil Jet Oil II MIL-PRF-23699 synthetic oil and Castrol Optigear Synthetic x320 oil.
While there are many factors that initiate micropitting, the main goal was to
determine and observe the effect of load and lubrication on micropit
formation and propagation.

- A computer program was created in accordance with ISO/TR 15144 [1] with
  an analysis done for both helical and spur gears to compare Method A and
  Method B for the determination of the micropitting load capacity. The
  objective is to validate this particular procedure to ensure that it does in fact
  predict when micropitting will occur.

1.4 Thesis Outline

Chapter 2 discusses the test machine with all of the details of how it was
assembled and the complementary features, such as the measurement devices. The post-
processing machines for profile, lead, and surface roughness measurements are also be
discussed. Specific information regarding the gears and lubrication is included. Lastly,
the experimental test plan of both the dynamics testing and micropitting testing will be
examined to provide a thorough understanding of the experimental methodology.

Chapter 3 focuses on the experimental results of the dynamic experiments.
Figures and tables are included to enhance results as well as a detailed overview of the
pertinent tests that aid in developing a better understanding of effect of dynamics on the
gearbox system. The computer models are discussed to complement the experimental
study.
Chapter 4 focuses on fatigue testing and a thorough overview of the micropitting investigation that took place in the research. Experiments are discussed in detail and supplementary profile inspection charts and photos are shown to elaborate on this surface phenomenon. Also the MATLAB program, created to have a visual and easy to understand way of interpreting the micropitting load capacity as expressed by the ISO document, is utilized and discussed in conjunction with the experimental tests to compare an analytical approach in determining micropitting with an experimental approach.

Chapter 5 recapitulates the objectives and results to remind the reader the point of the research and the necessary steps to take place in the future for further research. This chapter highlights the main findings in the experiment, of both dynamics and micropitting, and brings out its importance. This chapter also brings to light the tests that were done in 2006 and compares its results with the results of the tests conducted in this thesis. Pictures from the 2006 test are included to emphasis the difference in results with the two different FZG machines. This is a relatively short chapter but its significance is explained in detail to connect the 2006 data with the current research. Recommendations are documented to give direction for the next researcher for further investigation of stated problem.
Chapter 2

Experimental Methodology

2.1 Test Machine Description

Both the dynamic and pitting testing was conducted on a German made FZG gearbox system. This particular gearbox system is also referred to as a back-to-back tester because of the reaction gear pair transmitting the angular velocity and torque to a test gear pair in a power recirculating mode. Since the current research was initiated and motivated in part by the 2006 pitting test [3] and the strange contact pattern that appeared on the gear tooth came about by a particular FZG test machine, a refurbished FZG machine was assembled to confirm if indeed the same results occur in the new FZG test machine.
The beginning stages of research involved this author’s assembling from scratch an entire FZG machine (having previously had little hands-on experience, this in itself was and daunting experience that was extremely educational). A heavy duty machine table was purchased and set up to prepare for the installation of the base of the FZG. Holes were drilled in the table in such a way that it was intentionally off-centered to accommodate the torque moment arm. The reaction gearbox was pre-installed on the FZG base and the test gearbox was aligned to match up accordingly with its parallel counterpart. As will be reiterated later in the test machine description, alignment is critical in obtaining proper operation. The bearings, shafts, keys, coupling flanges, coupling between the motor and the reaction shaft, and the necessary bolts were assembled accordingly. Since there is approximately 9 inches of separation between the shaft of the reaction driver and the motor shaft, a motor mount was designed to provide the necessary vertical difference so both shafts will be on the same plane. The motor stand was bolted to the table in such a way that the shaft of the motor and the shaft of the reaction driver had coincident axes of symmetry for optimal performance. An induction heater was used to heat the reaction gear pair, test gear pair, and the inner races. This was done to heat the parts to a high enough temperature to expand the gears so they can easily slip onto the shaft. It is important that the gears were not heated beyond its tempering point, which was about 220°F, so that the fatigue tests would not be compromised. The schematic (top view) of the assembled FZG machine is shown in Figure 2.1 while an actual picture of the assembled FZG machine is shown in Figure 2.2.
Figure 2.1 Top view of FZG with description of machine elements [17]

Figure 2.2 Picture of the assembled FZG machine with 20/30 test gear pair
Measurement devices, such as shaft motor shaft revolution tracking, input speed, and temperature of lubrication in the test gearbox, were purchased and installed to assist in the experimental analysis. A Panasonic LC4H Counter, Monarch ACT-1B Series Panel Tachometer with a 0-5Vdc analog output, and a West 6100 Temperature Controller were all purchased and installed in an electrically compliant safety box on the side of the experimental table. Initially, a magnetic pick-up sensor was designated as the sensor that will give the signal to both the tachometer and digital counter. It works by pointing the face of the sensor on a metallic object, such as a small screw, and the rotation of the motor coupling would produce a change in the magnetic field which produces a current in the wires and gives an input feed to the desired devices, in this case the tachometer and counter. An L-shaped bracket was designed to place the sensor on the same plane as the set screw, which was aligned with the axis of symmetry of the coupling. After the magnetic sensor broke due to too close of proximity between the sensor and the screw, an infrared optical sensor was installed to replace it. The infrared optical sensor works by having an infrared light source and picking up a signal by the slight changes in the light of the detecting object, which is again a set screw, thus, creating a current. A solenoid valve was bought to provide the electromechanical operation needed to receive a current from the temperature controller which energizes its coils and provides an electromotive force that causes its valve to open and allow water to flow through and effectively cool the lubricant to a preset temperature. Pipes, hoses, bushings, and pipe nipples were assembled to provide a cooling system for the FZG machine. A heat exchanger was
installed in both the reaction gearbox and test gearbox to provide a channel for the cooling water to flow through. A type K temperature probe is connected to the inputs of the temperature controller which detects the temperature of the oil in the test gearbox.

The reaction gear pair is of little concern when performing post-processing analysis of any test, whether in a pitting test or a dynamics test. The reason is because of the large face width of both the pinion and gear facilitates a high longevity and prevents major surface wear. Also, the mesh frequency of the reaction set is different from the test set, making it easy to isolate the dynamic analysis of the test gear pair. Milliren [17], who conducted pitting tests on a similarly designed FZG machine, discussed some of the FZG design parameters in his thesis. The torsion shaft, which transmits the torque from the reaction side to the test side, has a diameter of 23 mm and is 315.8 mm in length. Due to the dimensions of the shaft, a torsional stiffness of 6,960 N-m/rad is used in the dynamic analysis. On the torsional shaft that was used in the FZG machine discussed in this particular thesis, a strain gage bridge was mounted in the middle of the shaft to allow for a strain gage channel connector to be attached. This will be discussed in greater detail in section 2.3 when detailing the test setup and procedure.

A Baldor AC variable speed motor that produces 20 horsepower and has a maximum speed of 5000 rpm is the prime mover. Initially, an aluminum flexible coupling was used as the coupler between the motor shaft and driver shaft of the reaction gearbox. This particular coupling is designed to slip fit on both shafts with no use of keys.
2.2 Torque Calibration

Before dynamic or fatigue testing could be accomplished, it was necessary to be confident in the amount of torque that was applied to the load clutch and transmitted throughout the back-to-back tester. This concern was addressed in part, in the 2006 micropitting technical paper [3] in which a calibration test was done to establish the torque that was actually being applied. This concern was brought up for two main reasons. The first reason is the fact that a different moment arm was being used for the testing, which is heavier and longer than the FVA [4] moment arm. Since the weights that were used were according to the FVA procedure, there would be some discrepancy between the torques because of the moment arm. The second reason is related to the way the loading was done. Typically the weights are placed on the weight hanger in a fashion that can cause hysteresis in the torque. A different torque value is found if one would lightly place the weights on the hanger, as opposed to dropping them on it. Thus, a more definitive way of determining the torque was sought to reduce any inconsistencies in the loading. Table 2.1 shows the FVA weights and the corresponding torques according to the new moment arm shown in Figure 2.3.

Figure 2.3 Moment arm used in dynamic and fatigue testing
<table>
<thead>
<tr>
<th>Weight Number</th>
<th>Weight (lbs)</th>
<th>Weight (N)</th>
<th>Torque(Nm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>W1</td>
<td>11.46</td>
<td>50.97</td>
<td>38</td>
</tr>
<tr>
<td>W2</td>
<td>14.94</td>
<td>66.45</td>
<td>50</td>
</tr>
<tr>
<td>W3</td>
<td>18.53</td>
<td>82.42</td>
<td>62</td>
</tr>
<tr>
<td>W4</td>
<td>21.60</td>
<td>96.08</td>
<td>72</td>
</tr>
<tr>
<td>W5</td>
<td>25.13</td>
<td>111.78</td>
<td>84</td>
</tr>
<tr>
<td>W6</td>
<td>28.08</td>
<td>124.90</td>
<td>93</td>
</tr>
<tr>
<td>W7</td>
<td>31.69</td>
<td>140.96</td>
<td>105</td>
</tr>
<tr>
<td>W8</td>
<td>34.84</td>
<td>154.97</td>
<td>116</td>
</tr>
<tr>
<td>W9</td>
<td>37.84</td>
<td>168.31</td>
<td>126</td>
</tr>
<tr>
<td>W10</td>
<td>41.40</td>
<td>184.15</td>
<td>138</td>
</tr>
</tbody>
</table>

The goal of calibrating the torque was to have a clear readout, via a voltmeter, of the torque applied to the test rig. The torsion shaft was strain gaged with a full Wheatstone bridge configuration along with a female end of the strain gage connector. A signal conditioner and a digital voltmeter were the instruments used to measure the shaft torque. As noticed in Table 2.1, each weight contributes to an added torque, as well as the weight hanger, and the moment arm. As stated earlier, since the FVA moment arm is
different than the one used in the experiment, it was necessary to figure out the individual
torque of moment arm to properly calibrate the overall torque. Initially, the center of
mass of the moment arm was found to multiply the distance from the centroid to the load
clutch by the weight of the moment arm. After realizing the moment arm had distributed
loading, which meant the center of mass would have been difficult to find, a new
methodology was enacted to remove the unknown of the center of mass. The moment
arm was disregarded entirely by applying the moment arm to the load clutch and using
the signal conditioner to zero the voltmeter so it will assume the load clutch is
“weightless”. Three tests were conducted to get a full range of hysteresis values. Since
there are two dowel pin holes, the tests that were done were done at zero degrees, 180
degrees, and zero degrees again. Figures 2.4, 2.5, and 2.6 show the hysteresis results,
respectively. Since the signal conditioner has a calibration setting, it was simple to apply
a linear relationship between the calibration voltage and torque. The slope was found
from the equation of the interpolated line between the points. The fundamental
relationship is as follows:

\[ V = mT \]  \hspace{2cm} (2.1)

where V is the calibration voltage of the signal conditioner, m is the slope of the points
taken from the calibration test, and T is voltage needed to read the voltmeter in terms of
Nm.
Figure 2.4 Zero degree calibration test

Figure 2.5 180 degree calibration test
The average of the three tests was taken and a slope of 0.0119 was used to calculate the needed calibration setting. As noticed in the above figures, the hysteresis was not uniform and the points were somewhat scattered. This posed problems for the readout of the voltmeter which still varied with any slight movement of the moment arm. Two more tests were done over the next few months, with the last one yielding the most consistent results. It was found that the test rig side plates and alignment were a cause of the large hysteresis bands. Due to the inconsistency of the torque readings of the first two tests, the final test was done after the FZG was dismantled and reassembled. Since dynamic testing was conducted under one particular calibration setting, a correction factor was used to establish the true torque that was used. The calibration voltage of the first test was found to give a torque reading that produced a root stress that was well off from what LDP [15] predicted in its boundary element and finite element root.
compressive stress prediction at the strain gage location. This was discovered during dynamic testing when the root stress given by the experiment was much higher than the root stresses of LDP. Thus, the torque that was noted under dynamic testing was divided by the final calibration value that was used for fatigue testing.

As with the first test, the procedure for getting data points at each particular weight was done in three motions: lifting the moment arm and setting it at its equilibrium point, applying a downward force on the moment arm and setting it at its equilibrium point, and bobbling the moment arm. Table 2.2 shows the voltage readings when the test was done in these three motions during the final calibration test. Figure 2.7 shows the final hysteresis plot after the FZG was put back together. The figure shows a much smoother and linear plot with the more accurate calibration voltage values.

<table>
<thead>
<tr>
<th></th>
<th>Down Force (V)</th>
<th>Lift Force (V)</th>
<th>Bobble (V)</th>
<th>Torque (Nm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A+H</td>
<td>0.53</td>
<td>0.01</td>
<td>0.07</td>
<td>0</td>
</tr>
<tr>
<td>W1</td>
<td>1.12</td>
<td>0.579</td>
<td>0.628</td>
<td>38</td>
</tr>
<tr>
<td>W2</td>
<td>1.3</td>
<td>0.808</td>
<td>0.924</td>
<td>49</td>
</tr>
<tr>
<td>W3</td>
<td>1.5</td>
<td>0.961</td>
<td>0.94</td>
<td>61</td>
</tr>
<tr>
<td>W4</td>
<td>1.55</td>
<td>1.01</td>
<td>1</td>
<td>71</td>
</tr>
<tr>
<td>W5</td>
<td>1.73</td>
<td>1.25</td>
<td>1.2</td>
<td>83</td>
</tr>
<tr>
<td>W6</td>
<td>1.9</td>
<td>1.365</td>
<td>1.31</td>
<td>93</td>
</tr>
<tr>
<td>W7</td>
<td>2.01</td>
<td>1.53</td>
<td>1.54</td>
<td>105</td>
</tr>
<tr>
<td>W8</td>
<td>2.24</td>
<td>1.59</td>
<td>1.64</td>
<td>115</td>
</tr>
<tr>
<td>W9</td>
<td>2.44</td>
<td>1.7</td>
<td>1.74</td>
<td>125</td>
</tr>
<tr>
<td>W10</td>
<td>2.59</td>
<td>1.93</td>
<td>1.9</td>
<td>137</td>
</tr>
</tbody>
</table>
2.3 Test Procedure and Conditions for Dynamic Testing

In analyzing the dynamics of the FZG machine, it is important to first establish the variables that are in effect in this study. Speed, torque, lubrication, and the finish type were the four factors in determining how the dynamics would be affected by controlling one or two of them and varying another. It was unknown as to which variable would contribute the most to the dynamics of the system, so it was necessary to investigate all probable causes to gain understanding on each variable’s impact.

Strain gages were mounted in the roots of three test gears. Two AGMA gears were used, with the finish type of normal and superfinished, and one FVA type C [4]
Three strain gages were mounted in the root of one tooth with another strain gage mounted in an adjacent tooth, also in the root. Single gage bridges were the means by which the strain gages were mounted. In order for the strain gaged gear to connect to a data acquisition system (DAQ), a hollow shaft was installed that could funnel the wire through it and connect to the DAQ. A slip ring assembly was bolted to an end piece for the shaft. Wires from the strain gage were soldered to the slip ring which was connected to the appropriate DAQ chassis channel. As far as the data acquisition is concerned, a National Instrument chassis was used as the intermediary between the strain gages of the gear and observing the signal on the monitor. Dummy resistors of 350 ohms resistance were connected in the channel board to correspond to the resistance of the strain gages. A bridge excitation voltage of 5 volts was used. A Labview [18] program was created by a previous graduate student that included a block diagram with the necessary blocks to acquire a strain signal. In the block diagram there was a mathematical block to multiply the strain response by Young’s modulus to observe the root stresses.

Figure 2.8 Picture of strain gaged gear in test gear box
As noted earlier, there are numerous dynamic factors that can be investigated. The root stress dynamic factor was the one that was emphasized in the experiment, since the strain gages are mounted in the roots of the gear teeth on the compressive side. Equation 2.2 shows the fundamental equation for determining the dynamic factor.

\[
DF = \frac{\text{Peak Dynamic Stress}}{\text{Static Stress}}
\]

The first objective in obtaining the dynamic factor is to establish a static strain (stress) trace. Initially, manual rotation of the load clutch was done to attain a “static speed”. Even though this produced a signal, the inconsistency of rotating the clutch by hand proved to yield unreliable results. Thus, a low speed of 30 rpm was selected as the static speed which was the reference speed for all of the dynamic testing. Figure 2.9 shows a static plot for a torque of 224 Nm. The plot resembles very much the strain signals shown by Rebbechi and the team at NASA [7] who conducted similar experiments as shown in Figure 1.3. The figure below is for a superfinished set and has much of the same features as Figure 1.3. For one, the noise is minimized due to the higher loading. Noise can be an issue at lower loads because the lower torques where the noise generating by the slip rings and data acquisition system are much larger relative to the strain signal. The two signals correspond to two adjacent teeth that the strain gages are placed on. The difference in the peak values of the signals is a result of each gage having a slightly different strain gage diameter location along the tooth root. However, since dynamic factor is the ratio of the dynamic stress to the dynamic stress, this does not
affect dynamic factor results. The negative sign for the stress means that the strain gages are located on the compressive side of the tooth flank. Another important feature of these types of plots is the load sharing. At higher loads, more load sharing takes place due to the higher load distribution and greater double tooth pair contact time. The effective contact ratio can easily be calculated to determine the amount of double tooth pair contact occurs during one mesh cycle which is equal to one base pitch. The equation is as follows:

$$ECR = \frac{(B - C)}{(A - C)}$$  

(2.3)

where C is the point at the beginning of the mesh cycle, B is the final point when the pair of teeth that contained C leaves contact, and A is the first point when the second pair of teeth comes into contact. The resulting contact ratio was calculated to be 1.38 which means there is double tooth contact 38% of the time during the mesh cycle. This contact ratio that will be tracked in later tests, changes with torque level due to the tip relief that is applied to the gears.
Figure 2.9 Static strain signal at 224 Nm

The torques that were chosen were based upon the modified loading procedure that were shown in the 2006 technical paper [3]. The FVA procedure [4] describes a procedure for perfect involute gears which would necessitate a different loading scheme. Since a modified gear pair (gears with tip relief and lead crown) intentionally increases the peak contact stress of the gear pair, the new loading procedure was necessary to match up the FVA [4] Hertzian contact pressures with ones modeled in LDP [15] that included the lead crown and tip relief of the AGMA gears. Since the load is mostly absorbed by the lead crown which is centralized in the middle of the tooth, very low torques were needed to achieve the same contact stress.

For the first half of the dynamic experimentation, steady state speeds were selected to compare its dynamic response with the static speed. In order to get a proper
and uncompromised signal, the sampling rate and low pass filter settings were critical in their selection. For post processing purposes, there had to be enough data points to determine speed (which will be of more concern in the speed run up test) and effective contact ratio. The selection of the frequency of the low pass filter was chosen based on a basic understanding of system dynamics. Since the low pass filter attenuates any signal above the cut-off frequency and retains all of the frequencies below the cut-off frequency, a frequency had to be chosen that would not remove the dynamic oscillations while simultaneously removing overbearing noise in the signal. Since the goal of the experiment was to observe the dynamics, the mesh natural frequency played an important role in seeing oscillations in the system. The low pass filter is dependent on the speed at which the test rig is running. Thus, the observance of oscillations in the signal is dependent on the setting of the low pass filter. If the cut-off frequency is too high, the signal may get too much noise. If the cut-off frequency is too low, then the DAQ will attenuate the mesh natural frequency and give a perfectly smooth sinusoidal signal. This is confirmed by the calculation of the experimental mesh natural frequency as expressed in equation 2.4

\[
MF = \left( \frac{\sum_{i} Z_i}{60} \right)N \quad \text{(Hz)}
\]  

(2.4)
where $Z_t$ is the total number of teeth and $N$ is the speed (in rpm) at which the pinion/gear is rotating. Another way to calculate the mesh natural frequency is to invert the amount of single tooth contact time. In essence, this would be subtracting “C” from “A” in Figure 2.8. This value will correspond with the mesh time period and gives succinctly the amount of time only one pair of teeth will be in contact. However, this approach may give a rather crude mesh natural frequency which may not produce a mesh natural frequency coincident with Equation 2.4. Figure 2.10 gives an example of how the mesh time period is calculated which can easily be inverted to yield the mesh natural frequency. The time of 0.3259 seconds is taken from 0.3273 seconds which gives a single tooth contact time of roughly 0.0013 seconds. For the trace of Figure 2.10 the mesh natural frequency is calculated to be 770 Hz, however, according to Equation 2.4 and with 30 gear teeth and a speed of 1500 rpm, the analytical mesh natural frequency is 750 Hz.
There are two main data processing options in the Labview simulation window. One being the “Rate As” option which allows the user to select the frequency at which the data is taken. This is taken as samples/second, and as stated before, a large enough frequency needed to be selected for best post-processing analysis. The challenge came in at the memory buffer. This buffer can only hold so much data for a period of time and if an amount exceeds its capacity, then no data can be taken. The maximum frequency was noted at 60,000 Hz where the buffer was able to sustain for a couple of seconds. This leads to the second important parameter that the user selects, which is the “Samples/Channel” option that accommodates the use of four channels if needed. These two options are interrelated. The amount of time the DAQ records is dependent on these
two parameters. For example, if one wants to have 4 seconds worth of data recorded, the frequency could be 1000 Hz and the samples/channel could be 4000. This is shown in the below calculation.

\[
\frac{4000 \ samples}{\text{channel}} \times 1 \ \text{channel} \times \frac{\sec}{1000 \ samples} = 4 \ \text{seconds}
\]

In order to get good repeatability, there needs to be enough peaks generated to have some statistical idea of how the dynamics vary as the teeth roll through mesh. Thus, having four or five strain cycles will not suffice to have a good understanding of the dynamics of the system. The number of cycles generated is directly related to the amount of time the DAQ records for the test which is inevitably related to the two important parameters discussed above. To continue with the example from above, if the test rig is run at 240 rpm then 16 distinct signals would be produced. This is shown in the below calculation.

\[
240 \ \frac{\text{rev}}{\text{minute}} \times \frac{1 \ \text{minute}}{60 \ sec} \times 4 \ sec = 16 \ \text{rev} \Rightarrow 16 \ \text{cycles}
\]

With this in mind, the buffer rate and sampling rate were chosen specifically to meet the purposes of not only acquiring a sufficient number of cycles, but ensuring the signal can be analyzed appropriately without any compromise. Since there was little need
to reproduce many static strain plots, there was a 1 to 1 ratio of samples per second. The sampling rate was 40000 Hz, the samples per channel were 40000, and the cut-off frequency for the low pass filter was 500 Hz. These parameters allow for one second worth of data to be taken which produces one full strain signal for a static speed of 30 rpm. For the steady state speeds, 60 cycles, give or take, are desired in each test. Intuitively, the faster the test rig runs, the shorter the amount of time that is needed to capture a plethora of peaks. For example, one of the steady state speeds that was run was 1410 rpm. Thus, 23.5 peaks were generated per second so the DAQ only needed to record data for nearly 3 seconds to get 60 cycles.

The same parameters and methodology were applied during the second phase of the dynamic testing with the speed run-up testing. In this experiment, a BNC cable was connected from two analog ports in the vector drive to an I/O channel board on a separate computer. A jumper cable was attached from the “Enable” port on the vector drive to the “Forward” port. This had to be run in a standard two wire configuration that necessitated the remote control to be switched to the remote option instead of local. Another Labview file was modified to apply a linear speed run up from rest to a maximum speed of 1750 rpm for the gear shaft. Within the block diagram of the second Labview file, it was possible to adjust the time that it takes the rig to go from rest to maximum speed. If the time it takes the rig to get to maximum speed is too low then the DAQ may not capture all of the peaks in between the speeds, which is why intervals of 10 seconds were chosen initially to get a medium run up. Figure 2.11 shows an example of a superfinsihed gear
pair speed run-up at 224 Nm. As can be seen in the figure, at the slower speeds, there are larger spaces in between the spikes at low speed. As the speed increases, the time interval between the spikes gets smaller and thus the space between the cycles get smaller culminating in the appearance of multiple cycles overlapping each other as the test rig reaches its maximum speed. As stated earlier, the combined torque of the moment arm and hanger were computed iteratively and during the speed run up experiment the second iteration was employed. The torques that were run in the speed run up test can be easily corrected by dividing the assumed torque by that factor. Run ups were done for each of the torques shown in Table 2.3.
<table>
<thead>
<tr>
<th>Torque (Nm)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
</tr>
<tr>
<td>17</td>
</tr>
<tr>
<td>28</td>
</tr>
<tr>
<td>39</td>
</tr>
<tr>
<td>56</td>
</tr>
<tr>
<td>64</td>
</tr>
<tr>
<td>73</td>
</tr>
<tr>
<td>82</td>
</tr>
<tr>
<td>88</td>
</tr>
<tr>
<td>103</td>
</tr>
<tr>
<td>111</td>
</tr>
<tr>
<td>224</td>
</tr>
</tbody>
</table>

Table 2.3 Adjusted Torque
2.4 Test Methodology for Micropitting Testing

All of the gears tested for fatigue testing were donated by the AGMA foundation. Table 2.4 shows the design features of the pinion and gear. Even though the FVA micropitting procedure [4] provides a detailed review and procedure for observing micropitting failure, the experimental testing that was done deviated in part with respect to the load staging. The weights that were used were the same as the ones described in the FVA procedure [4], however, the incremental loading was altered and implemented as a
result of observed fatigue testing results. For example, if the test rig was run at 65 Nm and a 48 hour test was done, after the inspection of the test gears, the decision was made to either run the 65 Nm longer or increase the load. The main variable was the duration of the test which varied from 4 hours to 48 hours. Since the 2006 technical paper [3] was presented at a time when the authors’ understanding of micropitting was still relatively new and due to the lack of the proper microscope for observing micropits, what was once concluded to be micropitting then may not be considered micropitting now. Thus, the “observation” methodology of the fatigue tests provided step by step evaluation to determine what the next load or testing interval should be. Table 2.5 shows the test matrix of the micropitting tests.
Table 2.4 AGMA Gear Design Parameters

<table>
<thead>
<tr>
<th>Dimension</th>
<th>Unit</th>
<th>Pinion</th>
<th>Gear</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of Teeth</td>
<td>-</td>
<td>20</td>
<td>30</td>
</tr>
<tr>
<td>Normal Module</td>
<td>mm</td>
<td>3.63</td>
<td></td>
</tr>
<tr>
<td>Normal Pressure Angle</td>
<td>deg</td>
<td>20</td>
<td></td>
</tr>
<tr>
<td>Normal Helix Angle</td>
<td>deg</td>
<td>0</td>
<td></td>
</tr>
<tr>
<td>Center Distance</td>
<td>mm</td>
<td>91.5</td>
<td></td>
</tr>
<tr>
<td>Total Contact Ratio</td>
<td>-</td>
<td>1.58</td>
<td></td>
</tr>
<tr>
<td>Transverse Backlash</td>
<td>mm</td>
<td>0.38</td>
<td></td>
</tr>
<tr>
<td>Young’s Modulus</td>
<td>GPa</td>
<td>206</td>
<td></td>
</tr>
<tr>
<td>Poisson’s Ratio</td>
<td>-</td>
<td>0.3</td>
<td></td>
</tr>
<tr>
<td>Face width</td>
<td>mm</td>
<td>14</td>
<td>20</td>
</tr>
<tr>
<td>Tip Outside Diameter</td>
<td>mm</td>
<td>81.6</td>
<td>116.4</td>
</tr>
<tr>
<td>Root Diameter</td>
<td>mm</td>
<td>63.2</td>
<td>98</td>
</tr>
<tr>
<td>Start of Active Profile</td>
<td>mm</td>
<td>69.07</td>
<td>104.57</td>
</tr>
</tbody>
</table>

Table 2.5 Micropitting Testing Matrix

<table>
<thead>
<tr>
<th>Torque (Nm)</th>
<th>Lubrication</th>
</tr>
</thead>
<tbody>
<tr>
<td>65</td>
<td>Mobil Jet Oil II (MIL L 23699)</td>
</tr>
<tr>
<td>90</td>
<td>Castrol Optigear Synthetic x320</td>
</tr>
<tr>
<td>108</td>
<td></td>
</tr>
<tr>
<td>165</td>
<td></td>
</tr>
<tr>
<td>262</td>
<td></td>
</tr>
</tbody>
</table>
Though the test plan depended on the previous test, certain things remained constant in all the testing. For one, the signal conditioner and voltmeter were used every time torque was added in the system. This was done to ensure repeatability in torque loadings. Also the lubrication system was subjected to dip lubrication where the oil was poured into the gear box. The laboratory supervisor advised that the oil limit should be halfway up the test shaft which corresponded to halfway up the gear pair.

With fatigue testing, it is necessary to have appropriate post-analysis machines to understand the necessary parameters that affect micropitting. One of the key machines is the M&M coordinate measuring machine that that measures the profile, lead, tooth thickness, spacing, and topography of the desired gears. A profile inspection is typically done to see if any wear has initiated after a load stage run. Lead crown and profile measurements are the most common in assessing the condition of the gear, before and after the test. Essentially, one would insert the gear into an arbor which is then attached vertically to the connecting apparatus of the M&M machine. A probe is then used to take all of the necessary measurements. There is an I-beam that is the reference block to zero the probe whenever a new gear is being measured. A power screw type device is attached to the arbor so there will not be any lag in the rotational motion when the probe is traversing the teeth of the pinion or the gear. The CMM (Coordinate Measuring Machine) is an excellent way to observe if there are manufacturing errors in the profile.
modifications of the gear or pinion. A picture of a test pinion getting measured is shown in Figure 2.12.

In order to complement the profile measurements to determine if indeed micropitting has occurred, it is necessary to take actual photos of the flanks of the pinion and gear. Digital imaging is an important asset to acquire photos that represent the worn tooth as accurate and clear as possible. The OSU GearLab has two microscopes that are available for imaging. The first one is an Olympus DP25 that has three focusing lenses which are 10x, 20x, and 50x, respectively. This microscope has great focusing and magnification ability which amplifies the smallest of micropits. If the user wants to look at a very detailed view of the pitting area, this would be a good microscope to use, however, if one wants to observe the contact pattern then the second microscope would be ideal. The second microscope is an Olympus DFPLFL 0.5xPF and has sufficient magnification power to photograph the entire contact area which is ideal when attempting to observe dynamic contact patterns as well as show the micropitted area. Figures 2.13 and 2.14 show the comparison of the two microscopes for the same tooth flank.

Figures 2.15 and 2.16 shows sample profile traces of an AGMA pinion and gear, respectively. These two figures show a profile plot of each tooth in the pinion and gear respectively. The probe was aligned directly in the middle of the face width where the wear is typically the largest. Figures such as these and plots in which 16 profile traces at equidistant points along the face width are the ones that give the best observance of wear. The 16 profile trace plot shows the development of the wear from the end of the tooth.
flank being measured to the opposite end of the same flank with the traces towards the ends showing a straight measurement and the traces in the middle showing the hollow measurements. Figures such as these will be explained in greater detail in Chapter 4 when micropitting is discussed. For now it is sufficient to point out the wear generated in the pinion at 14 degrees of roll. The hollow region signifies that some material has been removed as the probe moves along the profile direction of the tooth flank. This particular test was run at 265 Nm for 16 hours and Figure 2.16 shows little to no wear forming in the gear.

Roughness is also an important parameter in assessing any tooth surface change. The Taylor Hobson Precision surface finish measurement system was used to extract roughness profiles and measurements. Typically, tooth surface finishes of around 0.1 $\mu m$ are characterized by superfinished gears and ground gears usually have an average roughness of around 0.45 $\mu m$. A gear fixture is used to slide the pinion or gear on the shaft to get it in close contact with the probe. The main difficulty in attaining correct data is ensuring the tooth that is being measured is as straight as possible. Also, since the roughness profile is measured on a micron level, any perturbation of the experimental table could alter the results. It is also difficult to quantify the diameter at which the very short profile trace is made on the gear. Figure 2.17 shows a picture of the Taylor Hobson roughness device with Figure 2.18 showing a sample of a roughness profile of a ground gear that was actually measured after a test run.
Figure 2.12 Profile measurement of test pinion using M&M machine

Figure 2.13 Olympus DP25 photo of pinion at 90 Nm
Figure 2.14 Olympus DFPLFL 0.5xPF photo of pinion at 90 Nm

Figure 2.15 CMM Pinion Profile trace for AGMA gear

Local micropitting
dedendum of test pinion

Wear region at 14°
Pinion roll angle
Figure 2.16 CMM Gear Profile trace for AGMA gear

Figure 2.17 Talysurf roughness profile measurement with probe engaging pinion
A MATLAB program was created to analyze spur or helical gear pairs according to the ISO procedure [1]. As stated earlier, the purpose of the procedure is to determine the micropitting loading capacity and produce a safety factor that yields a numerical value that corresponds to the risk of micropitting. This program is user friendly and allows one to choose whether they want to analyze a gear pair using Method A or Method B. Each method is equipped with the pertinent parameters needed to properly analyze the state of the gear at given load. Method A features plots of contact stress, contact temperature, specific film thickness (lambda ratio) before and after mesh, lubricant film thickness before and after mesh, and safety factors with tables of each as well at the beginning of the facewidth, 25% of the facewidth, 50% of the facewidth, 75% of the
facewidth and 100% of the facewidth. Since Method B is a simplified version of Method A with only 7 points calculated along the line of action, only two dimensional plots such as contact stress, lubricant film thickness, and safety factors vs. pinion diameter as well as multiple tables with all of the important parameters such as sliding velocity parameter, load parameter, flash temperature, mean friction, etc. tabulated. The user simply has to input the normal gear information and the program will evaluate it. Figures 2.19 and 2.20 show the menu of Method A and Method B, respectively. The AGMA test gears, among several of the ISO example gear pairs, were analyzed using the program to predict if micropitting was apt to occur and to verify that the computer program was working properly.

![Figure 2.19 “Method A” of MATLAB micropitting program](image-url)
Another note about the ISO procedure is the permissible specific film thickness that is used for the analysis. There are two methods expressed in the ISO procedure to retrieve this important parameter. One way is experimentally in which gears are run under specific operating conditions. Experience is required for this method because the design gears have to match up exactly with the actual gear pair that was pre-analyzed in a load distribution program. The second method is to use the results that have been published by engineers and have used a standardized method in determining the load stage failure. Typically, the FVA procedure [4] is the standard by which the failure load stage is determined and a critical specific film thickness is found. Then equation A1 in the ISO document is used to calculate the permissible specific film thickness. Equation 1.2 poses problems when one wants to the safety factor with different torques. The issue with this equation is that it uses the minimum specific film thickness to calculate the
permissible specific film thickness and this presents counterintuitive results when going from a lower torque to a higher torque. One of the limitations in using this value is the inability to ascertain the actual value of the permissible specific film thickness. Figure 2.21 is another way to determine this parameter. All is needed is the load stage number and the viscosity grade. One of the problems with this approach is the too simplified and unverifiable means by which these graphs are produced. This procedure does not take into account any additives or anti-wear chemicals that have an immense impact on the formation of micropitting or any wear for that matter. The permissible specific film thickness that was chosen was 0.08 which is based on Figure 2.21. Even though the Mobil Jet Oil II is a synthetic oil, an assumption had to be made that the oil fell under the category of mineral oil to be able to use this parameter. Castrol Optigear x320 was the other oil used in the fatigue test and this is also a synthetic gear that has a much higher viscosity than the Mobil oil. According to the information sheet of the Castrol oil [10], the failure load stage was reported to be greater than load stage 12. Since Figure 2.21 only goes up to load stage 10, that was the one selected to retrieve the permissible specific film thickness value.
2.5 Summary

The first course of action was to get an accurate reading of the torque that was applied to the system. Without confidence in the loading, it would be hard to have confidence in the results. Calibrating the torque to get a digital reading, was necessary to ensure integrity in the results. The dynamic testing will be a two stage process, one prior to fatigue testing and one after fatigue testing. The overall goal of these tests is to understand and identify the dynamic factor that should be used in load distribution analysis. Seeing how the dynamic factor is affected by torque and speed is essential in the
research. This will give an overview of the variability of the FZG tester and its influences. Also understanding and calculating the natural frequencies of the system offers insight into the resonance that takes place during a given test. Concerning micropitting, an analytical and experimental approach went into the research. Experimentally, proper profile inspection for both the pinion and the gear was essential in observing the progression of the wear from test to test at a given load. Analytically, the ISO procedure [3] was utilized and programmed into MATLAB to predict the micropitting load capacity and compare analytical results with experimental results.
Chapter 3

Dynamic Experimental Results

3.1 Introduction

Chapter 2 discussed the methodology of both the dynamic and micropitting experiments. The order that each phase of research was discussed earlier corresponded with the chronology of the testing. Thus, the dynamic results will be presented first and then micropitting. As far as the dynamic experimentation is concerned, as stated earlier, the data procured during the testing corresponded to a slightly lower torque than what was initially expected. Therefore, two sets of results will be presented in this thesis; one that reflects the initial dynamic testing done in the first phase of research and the second one reflecting the final dynamic experimentation done after the micropitting tests in the final phase of research. The point of returning to dynamic experimentation was to increase the time length of the testing at a given torque during a speed run up. Also it is a great way to observe the repeatability of the initial testing at the correct calibration.
voltage which would yield the actual torque. As observing the dynamic overshoots in the plots are important to assess the dynamic factor, determining the natural frequencies of the FZG are equally important to observe if any resonance is taking place which could be the cause of a high dynamic factor. Since numerous speeds and torques were tested, only select plots that give a definitive observation of dynamics and are unique in nature will be included, otherwise more than 100 potential plots could be included. Since there are a number of changing variables, in order to coherently and lucidly explain how a particular variable varies with another, the following structure that will be imposed will reflect it. Thus, controlling torque and varying speed will, in the ground gear and superfinished case, is explained in one section and varying torque while fixing the speed will be explained in another section. Comparing the different lubricants while holding the torque and speed constant will also be explained in another section.

3.2 Speed Variation with Torque Controlled

The first part in testing included fixing the torque while the speed was adjusted to see how the dynamics changed in that scenario. Within this case, the results of a low speed, medium speed, and high speed at a certain torque level would give an understanding of the influence of speed on the system. Varying the speed may have the most effect on the dynamics of the system due to the fact the variable speed drive may trigger an excitation frequency that coincides with the natural frequency of the system.
The AGMA ground gears were the first ones tested. The following figure shows the results of a steady state strain plot at 40 Nm in which the moment arm and weight hanger were the only loads that were applied. Unless otherwise stated, all of the plots have a value of the torque according to the correcting factor discussed in Chapter 2.

Figure 3.1 Static strain signal at 40 Nm
Figure 3.2 Strain signal at 510 rpm at 40 Nm

Figure 3.3 Strain signal at 900 rpm at 40 Nm
Figures 3.1, 3.2, and 3.3 illustrate the variation in the signal at three different speeds. The two signals that are shown in each plot represent a strain gage on two consecutive teeth adjacent to each other. This was done to observe the load sharing characteristics of the gear pair during mesh at a particular load. It is apparent that the load sharing is constant throughout the changing of the speeds because it is mainly a function of the load distribution and since the torque is controlled, aside from spacing errors, there should be no change in the amount of single tooth pair contact.

Figure 3.4 shows the overshoot that took place during the high speed test. A black line is drawn across one of the strain plots to illustrate the profile of the static strain signal. The hypothesis was that the higher speeds would have oscillation peaks about the
static line with nearly symmetric oscillations about that line. In most of the tests, this type of result was found. The higher speeds, especially Figure 3.4, had oscillations at a much higher point than where the static line indicates. Also, a point to notice is the amount of overshoot at the higher speed. This shows a great example of the logarithmic decrement in which the first peak is typically the largest overshoot and the following peaks show to decrease exponentially. The first peak (since it was the largest) was the one chosen for the dynamic factor calculations in which the stress at that level was compared with the static stress.

To get an idea of whether the torque truly has much effect on the dynamics of the system an observation of strain signals is necessary at a different load. Since 40 Nm is a relatively low torque, the next torque that will be analyzed is at 103 Nm. A 63 Nm torque increase should show enough difference to observe the impending influence. Figure 3.5 shows a static strain signal at the 103 Nm load and one of the parameters that changes is the calculated root stress which reaches 30 ksi which corresponds to 207 MPa. The other parameter is the increase in the load sharing activity that results from the double tooth pair contact.
Figure 3.5 Static strain signal at 103 Nm

Figure 3.6 Strain signal at 720 rpm at 103 Nm
For observation purposes, it will be sufficient to show different signals at two different speeds, which consist of a medium and high speed. Very little can be gathered from Figure 3.6 which suggests that the lower and medium speeds do not show much activity that can give evidence of a strange dynamic behavior occurring. The high speeds however, consistently show the overshoot that takes place. A line similar to the one in Figure 3.4 is drawn in Figure 3.7 where the static signal is superimposed. Interestingly, the static signal at the higher torque (103Nm) coincides, as discussed earlier, with the mean of the oscillations. It appears that at the higher torque, the expectation of the strain signal to oscillate about the static line is true for this case. By utilizing equation 2.2, a
dynamic factor based on the plots was found to be 1.1 which suggests that a 10% increase in load occurred during mesh at this speed and torque.

For analysis purposes, it is helpful to get a gauge of the calculated root stress which is an output feature of LDP. There are two different output tabs that is capable of performing stress calculations, whether tensile or compressive, and give some idea of what the load distribution predicts. The “BEM Stress” tab calculates the stress based on the idea of the tooth under consideration modeled as 2-D boundary elements that are extended to the face width direction using Jaramillo’s method [19]. The dynamic factor that most closely resembles the calculation would be the dynamic root stress factor as explained in Table 1.1. Another reason why this particular stress calculation is not as helpful is because of its calculation method. From the start of active profile to the end of active profile the algorithm searches for the largest stress, which may not correspond to the strain gage location. Figure 3.8 shows the stress peaks of the compressive side of the gear. As one moves further in the profile direction, the peak stress location actually changes. Figure 3.9 shows the corresponding stress at 50% of the facewidth. The shape is different in Figure 3.9 than the experimental data because it is the peak stress whose location moves along the root, thus, it is not of the strain gage location.
Figure 3.8 BEM stress solver at 103 Nm

Figure 3.9 Peak stress calculation at 103 Nm using boundary element method
The most relevant plot in Figure 3.9 is the stress calculation taken at 50% of the facewidth since it corresponds to the location of the strain gages. This plot closely resembles the static strain signal taken from the experiments and interestingly enough, the stresses are nearly identical. The peak stress was found to be 28.3 ksi which is very close to what the strain signal indicates in Figure 3.5. The “FEA Stress” output tab also calculates the stress but the user is able to identify the strain gage location so a finite element analysis can be performed at the diameter that coincides with the strain gage. Figure 3.10 shows the gage stress plot at the three different locations. The actual strain gage face width locations were at 4 mm, 7 mm, and 10 mm from one extreme side of the gear tooth flank. For the FEA analysis, only the 10 mm gage was selected to observe the resulting root stress directly in the middle of the face width of the gear flank since the total face width is 20 mm. This is because the second strain gage is directly in the middle of the facewidth (50%) which is where the peak stress in Figure 3.9 indicates. These figures show how close the results of two different analytical methods (BEM and FEA) accurately reflect and predict the actual experimental results. The gage stress can be approximated at 175 MPa which corresponds to about 25 ksi which is slightly lower than the BEM stress, but more accurate since the radius at which the gages are located can be inserted.
3.3 Torque Variation with Speed Controlled

Even though one can get an idea of how the dynamics are influenced by changing the torque and holding the speed constant from the previous section, it would be beneficial to analyze the system at the speed that the micropitting tests were ran in 2006 [3] which was a gear speed of 1500 rpm (2250 rpm pinion speed). Nine torques were ran at this speed but only a couple will be shown to get an assessment of the how the dynamics change with low, medium, and high torques. Figures 3.11, 3.12, and 3.13 show strain signals at 17 Nm, 88 Nm, and 224 Nm respectively.
Figure 3.11 Strain signal of 17 Nm at 1500 rpm

Figure 3.12 Strain signal of 88 Nm at 1500 rpm
For each figure, the static strain signal is again modeled by the black line that is sketched along the red strain signal. At the low torque, the static signal is directly in line with the oscillations observed at 1500 rpm gear speed. The medium torque shows the activity occurring above the static signal while apparently the activity of the high torque occurs below the strain signal. Not only does this show the inconsistency in the dynamic gearing system as loads are changed, it reveals the arduous nature in determining a specific dynamic factor that can be applied to account for the increase in load.

Figure 3.14 shows a plot of the dynamic factors versus torque. The peak dynamic factor is at 28 Nm while factors less than 1 are found at the higher torques. Even though this seems intuitive in the fact one expects to see most of the dynamic activity occurring
at the lower torque and less at the higher torque, the peaks and valleys that occur in
between show that there is be a linear increasing or decreasing pattern of the dynamic
factors. The strain signal for each set of data was chosen somewhat arbitrarily. For each
dynamic test, whether at a particular speed or torque, a range of peaks were produced to
observe any patterns or changes at steady state. Typically, sixty strain signal peaks were
chosen as a sufficient number to notice any significant changes. The dynamic factors
shown in Figure 3.14 came from strain signals in a set of sixty signals with the average
strain signal being chosen. In the short time interval that the data was taken and the set of
sixty peaks was taken there was a range of stress values discovered.

As stated before, during a particular steady state test run at a given load, there
were typically approximately 60 cycles recorded for data analysis. Within the given time
period of recording the different cycles, there were obvious strain fluctuations that, in
some cases, showed great differences. Figure 3.14 reveals exactly how much fluctuation
occurs at a given load. The speed chosen to be held constant was 1500 rpm which again
was selected to match the speed at which the fatigue tests were being conducted.
Figure 3.14 Hysteresis plot of torques at 1500 rpm

Figure 3.15 Average dynamic factors plot of torques at 1500 rpm
Figure 3.14 shows the range at which the dynamic root stress factor can be applied for a given torque. The one with the largest difference was at 28 Nm. It appears that there is just as much difference between the minimum and average root stress as it is between the maximum and average which is why the dynamic factor can be between 1.22 and 1.57. It is important to note that as the torque load increases, the difference between the minimum and maximum gets smaller, specifically at around 60 Nm where the average dynamic factor is very close to the minimum dynamic factor.

Figure 3.15 shows the average dynamic factors calculated at various torques at 1500 rpm. This data was taken from Figure 3.14 where the average value coincides with the data point that is in between the minimum and maximum value. Since there was such a wide range of strains detected at particular torques, taking the average value is the best and most approximate dynamic factor with the amount of fluctuation that was observed during testing.

3.4 Speed Run-Up with Torque Variation

As expressed in Chapter 2, conducting a speed run-up test offered an interesting viewpoint of the effect of speed on the dynamic factor. The superfinished set was the first pair tested for the speed run up. Figure 3.16 shows a speed run-up test at 17 Nm. Even though the original data included time on the x-axis, it was converted to speed to get a
sense of the change of the strain signals as the gears go through mesh cycles. The test was
done over a time period of 10 seconds.

Figure 3.16 Speed run-up at 17 Nm
For the above figure, a clearer picture of the variation of root stress levels at a certain torque level is made evident. Since the speed acts as a forcing frequency or an input to the system, one would expect that higher forcing frequencies would yield higher strain levels only at the natural frequencies. It is interesting that the gear flank experiences a compressive stress of 10 ksi at a speed of about 625 rpm which is the highest stress level observed in that 10 second time cycle. Figure 3.17 shows a speed run-up test at a higher load of 55 Nm. As it can be seen, the large strain fluctuations decreased due to the increased load. The large dynamic strain is still evident, regardless of the torque change, around the 650 rpm gear speed marker. The static strain signal for
the 55 Nm load which is outlined in Figure 3.17, at first inspection, appears to produce smaller dynamic factors. As the speed increases and moves towards the maximum speed, it seems the strain signals are showing less fluctuations and are more consistent than the 17 Nm load. The static root stress values in Figures 3.16 and 3.17 are 5720 psi and 18000 psi, respectively. Figure 3.18 shows computed dynamic factors at constant speeds. With the superimposed values from the speed ramp, it can be seen that the values are somewhat similar but certainly more tests need to be made in order to better assess the differences.

![DF vs. Speed @17 Nm](image)

Figure 3.18 Dynamic factor vs. speed plot for speed run-up at 17 Nm
As stated earlier, there is different behavior observed in lower loads than higher loads. If the noise level from the DAQ reduces as the load increases, then it is possible that when the gears experience the same transient speed test run at a higher load, a smaller range of compressive root stresses will be produced. Figures 3.21 and 3.22 show the speed ramp test at 113 Nm and 224 Nm, respectively, along with the static strain signal superimposed on the test run. The most noticeable difference with this figure and Figure 3.16 is clearly the less fluctuating stresses. It appears that at the higher load, the compressive root stresses seem more consistent and aggregated. The maximum compressive stress is found at close to 4 seconds of the test which corresponds to a speed approximately 670 rpm which is relatively close to the speed at which the largest strain signal was found at 17 Nm. This will be analyzed in greater detail later. For now, the

![DF vs. Speed @ 55 Nm](image_url)
important fact to observe is that the fluctuation of root stress (which is an indication of a fluctuation of load) is not nearly present in higher loads than it is in lower loads. This is most likely a result of the dynamic increments in load remain relatively constant while the denominator torque increases by a factor of about 13. This would reduce the dynamic factor from 1.5 to 1.03. The understanding of this is rooted in the dynamic increment equation which is a derivative of Equation 2.2. Knowing that the transmission error is the excitation that produces the dynamics, the dynamic increment comes from the difference between the peak strain (root stress) value and the static strain (root stress) value. Equation 3.1 emphasizes the importance of the dynamic increment and the role that the increased load plays in reducing the dynamic factor.

$$\text{DF} = \frac{S + DI}{S}$$  \hspace{1cm} (3.1)

In an ideal dynamic system, the dynamic increment would be the same regardless of an increase in torque and as the torque was increased, the dynamic factor would decrease. If this is the case, one can look at the 17 Nm load and observe the reduction in the dynamic factor as the load was increased. If the static load was 17 Nm and the strain signal showed a 3 Nm increase then based on equation 3.1 the computed dynamic factor would be 1.18. Now if the load was quadrupled to 68 Nm then the dynamic factor would reduce to 1.04. In the actual signals of Figures 3.16 and 3.17, there dynamic increment actually reduced from 4300 psi to 3130 psi at the largest strain signal around the 630 rpm
mark. The peak values at that largest strain signal for the 55 Nm and 17 Nm was 21,130 psi and 10,110 psi, respectively.

Figure 3.20 Speed run-up at 113 Nm
As stated earlier, the maximum compressive stress seemed to consistently appear at around 630 rpm. It is also important to note that at that particular speed, there was a very loud whine noise that was heard. Figure 3.22 shows the plot at the steady state speed. The figure shows an unusual strain shape that is atypical compared with all other observed strain signals. It was at no other speed that that noise was heard which caused incentive for further analysis. Figure 3.23 shows the strain signal at that particular speed during the speed run-up test. Figure 3.24 shows the same strain signal at 224 Nm. The strain signal is very distinct from Figure 3.23 where the same strain signal was captured at the same speed at the load that is nearly 13 times higher. The difference is apparent.

Figure 3.21 Speed run-up at 224 Nm
with the 224 Nm strain producing a nearly static signal which does not capture the natural frequencies as Figure 3.22 does.

Figure 3.22 Steady state zoomed at 625 rpm at 17 Nm

Figure 3.23 Speed run-up zoomed at 625 rpm at 17 Nm
A lot can be said about the above plot. The fact that there appears to be two natural frequencies occurring at this speed is the main observation from 625 rpm gear speed. To test to see if this signal only shows when the test rig undergoes a speed run-up, a steady state test was employed at the speed of 625 rpm, which was around the speed at which the strange strain signal was discovered in the speed run-up. Utilizing Equation 2.4 gives greater understanding in the determination of the actual natural frequency that is observed in Figure 3.23. A sine wave was superimposed on Figure 3.23 to illustrate the two natural frequencies present in the system at that speed. The mesh natural frequency calculated from that particular equation is 312 Hz which will be confirmed later by the dynamic program used to determine the natural frequencies of the system. The tooth
mesh frequency excites the second natural frequency which apparently occurs at a greater frequency. This can be calculated by counting the number of peaks in between the time period and multiplying it by the tooth mesh natural frequency. There are approximately 12 oscillation peaks which if multiplied by the mesh natural frequency would result in a second natural frequency of 3744 Hz. The steady state strain signal of Figure 3.22 is identical with the speed ramp. What is projected to be happening at this speed is that there are combinations of two torsional natural frequencies, one at the mesh frequency of 312 Hz and the second one at the mesh resonant frequency of 3744 Hz. Again, these natural frequencies will be identified later in the dynamic simulations of the FZG tester and their mode shapes will be discussed.

Figures 3.25 and 3.26 show that at higher speeds, the lower natural frequency is not excited, but in each instance, the higher mesh natural frequency appears. Also these figures show that as the load is increased, the dynamic factor also decreases which was discussed earlier. The natural frequency can easily be determined in these figures. An inch ruler of a 1/32 scale was utilized to determine the natural frequency. The width of the strain signal is approximately 1” and the width of the oscillations is 20/32 inches. At the speed of 1604 rpm, the tooth mesh natural frequency is equal to 802 Hz based on Equation 2.4. The natural frequency is thus calculated by multiplying the mesh frequency by the total width of the strain signal and dividing it by the width of the total oscillations of the signal and dividing that number by the total number of time periods which in this case is only two. This is expressed in the below calculation.
\[ f_n = \frac{802 \times 2 \times 1}{.625} = 2566 \text{Hz} \]

Figure 3.25 Steady state test run at 1604 rpm at 28 Nm
The same process was utilized to determine the natural frequency of Figure 3.26 which revealed a natural frequency higher than that of the one discovered in Figure 3.25. Even though the speed is 100 rpm lower at 1505 rpm, the applied torque just about doubled which increases the load sharing which essentially increases width of the strain signal. Based on the same methodology that was explained earlier in determining the natural frequency, the natural frequency for Figure 3.26 is 4146 Hz which is much higher than the 2566 Hz natural frequency calculated in Figure 3.25.

Since there were smooth and distinct oscillations at other low torques, specifically 40 Nm, various speeds were tested at that load to see if any differences arose. Theoretically, the natural frequency should be the same at every speed since the mass-
spring-damper system is dependent on the transmission error as the excitation and natural frequency is proportional to the mesh stiffness. Figures 3.26a and 3.26b show the slight changes in oscillation at that load. The figures show the results of a standard finished AGMA gear pair.

Figure 3.27 Steady state test run at 1500 rpm
a) 510 rpm b) 1590 rpm c) 28 Nm d) 224 Nm
The four above figures give great insight into how apparent and consistent the oscillations are with respect to a controlled variable. Focusing on Figures 3.27 (a) and (b), it is verified that speed actually does play a role in the assessing the dynamics of the gearbox. This fact should be obvious but nevertheless, as the speed increases for a given torque, it appears that it goes closer towards a natural frequency. Also, as observed earlier, the torque plays perhaps an even greater role in producing a resonant-like signal. This is confirmed in (c) and (d) of Figure 3.27 where the speed is held constant at 1500 rpm. This speed was chosen because all of the micropitting tests were conducted at this particular speed. There is nearly a 200 Nm difference between (c) and (d) and it is apparent that at the higher load, the dynamics are somewhat suppressed and very minute oscillations occur.

3.5 Dynamic Factor Simulation

One of the benefits of doing experimental work is to verify the predicted the results of analytical simulations. The one that will be discussed in detail is a frequency response of a single degree of freedom system with an input trapezoidal ramp load profile. The ramp load acts as a reference point for the frequency response where its peak point is unity, making it easy to observe the dynamic factor. This simulation was initially created by Dr. Donald Houser and was enhanced by the author. There are a few parameters that have to be determined and included in order to produce the transient
response. With the enhanced version of the model, speed can be varied to see what the potential dynamic factor would be. LDP had to be incorporated to figure out the mesh cycle of the tooth forces. Figure 3.28 shows a plot of the tooth forces of a single tooth pair along the mesh cycle.

![Tooth Forces for Each Tooth](image)

**Figure 3.28 Tooth forces along mesh cycle at T=40Nm**

This plot shows the load sharing that takes place at each successive mesh position. For this particular case, the point where both pairs of teeth carry the same load is at two thirds along the mesh cycle. Figure 3.29 shows the pertinent data for incorporation into the single degree of freedom trapezoidal load profile simulation. This figure analyzes just one pair of teeth and actually shows the ramp up to maximum load.
In the above figure, the trapezoidal load profile is seen clearly. This is the same ramp profile that was shown in Figure 3.29 that represents the computer simulation of the transient response. In order to get a response based on the speed, the mesh position at
which the ramp reaches a “steady state” load is the mesh position that is recorded for the simulation. This position coincides with the ramp time that is used as the calculation for the transient response. Thus in this case, the mesh position at which the ramp is excited to maximum load is 0.0999. This poscon value is multiplied by the time period of the mesh natural frequency of the actual gear pair and is simply the square root of the mesh stiffness divided by the effective mass of the gear pair. Since the tooth mesh frequency is directly related to the speed, the ramp is excited at this particular frequency which is why the ramp time is inextricably connected to this mesh frequency. The natural frequency of the second order system had to be chosen. The estimated natural frequency of the system was 3800 Hz which was observed in the experiment as expressed in the second natural frequency in Figure 3.23. A damping ratio of 0.05 was chosen to simulate the response as reasonably as possible. With all of parameters either chosen or calculated, the response shows the corresponding overshoot from which a dynamic load factor may be calculated.

A few of the torques that were tested in the previous sections were used as inputs in the ramp model. For example, the largest observed average dynamic factor was at 28 Nm at 1.42 at the speed of 1500 rpm. Table 3.1 shows the parameters used in the single degree of freedom analysis for a torque of 28 Nm and a speed of 1500 rpm. The main input parameters are the natural frequency and the damping ratio. The resulting dynamic factor was found to be 1.54 which is a little off from the average experimental value of 1.42. However, if one wanted to observe the maximum experimental strain which would
correspond to the maximum dynamic factor, the results would be very close since the maximum experimental dynamic factor is 1.57.

Table 3.1 Speed Ramp Model Parameters

<table>
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<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$f_m$ (Hz)</td>
<td>750</td>
</tr>
<tr>
<td>$f_n$ (Hz)</td>
<td>3800</td>
</tr>
<tr>
<td>$\zeta$</td>
<td>0.05</td>
</tr>
<tr>
<td>$\phi$ (rad)</td>
<td>3.04</td>
</tr>
<tr>
<td>$\omega_n$ (rad/sec)</td>
<td>23876</td>
</tr>
</tbody>
</table>

Figure 3.31 Dynamic response of model at 55 Nm at 1505 rpm
Figure 3.32 Dynamic factor vs. speed of model at 55 Nm

Figure 3.33 Dynamic factor vs. torque of model at 1500 rpm
The single degree of freedom ramp model seems to predict the dynamic factor for the largest experimental strain pretty well. Again, analyzing Figure 3.21, it can be seen that at the largest tested torque of 224 Nm, the calculated dynamic factor was 1.05. The LDP poscon value is changed to 0.3996, which simply alludes to the fact that it takes longer for the ramp to reach its steady state value. The resulting dynamic factor is 1.03 which is very close to the experimental value for the maximum dynamic factor. Figure 3.32 shows the simulation predicts the dynamic factors with respect to gear speed. This was done at 55 Nm and if compared with Figure 3.19, it can be seen that there is less fluctuation of dynamic factors as the gear speed changes. The 625 rpm does not produce the largest factor as the experimental results show and also at 1500 rpm Figure 3.32 shows a minimum dynamic factor value which is very different than the high dynamic factor value given by Figure 3.19. Figure 3.33 shows the dynamic factor simulation with respect to torque held constant at the gear speed of 1500 rpm and it follows Figure 3.14 fairly closely. Even though there is much fluctuation of the strain signal in that figure, there also appears fluctuation towards the higher torques with the maximum dynamic factor being found in the lower torques as expected.

Another dynamic modeling program that was mentioned before was utilized to get a more complex analysis of the gear dynamic system. In reality, there are seven degrees of freedom for the system as a whole which is shown in Table 3.2. The rotation of the prime mover, the rotation of the gear and pinion on the reaction side, the rotation of the gear and pinion on the test side, and the length of the line of action (LOA)
translation of the gear and pinion on the test side are all important parameters in performing a complete dynamic analysis. The “Gearvib9BB” program [20] is a MATLAB created program written by the Gearlab and was modified to model back to back testers such as the FZG machine. It has since then been updated to include more input options for a broadened investigation. To compare with the previous analysis of the single degree of freedom model, the seven degree of freedom program was analyzed first at 28 Nm. The program was analyzed in English units so the 28 Nm corresponds to 248 lb-in. The transmission error was used as the input for the analysis in which five transmission error harmonics are extracted from LDP and incorporated in the MATLAB program. When the system as a whole is analyzed, the closest natural frequency to the experimental is at 2744 Hz. Figure 3.34 shows the mode shapes of the test gear and pinion at each of the seven natural frequencies. Since typically one natural frequency (sometimes two) can be captured by the strain signal, the figure below gives insightful developments into the displacement of the test gear and pinion when a lower or higher natural frequency is detected. At the lower natural frequencies, there appears to be the close to the same rotational displacement of the pinion and the gear. When it comes to the natural frequency in question, the one at the fourth mode, the pinion appears to rotate one way, while having some recoil in the clockwise direction. The gear has a very small displacement with respect to the pinion.
Table 3.2 Seven Degree of Freedom Gear Dynamic System

<table>
<thead>
<tr>
<th>Mode Number</th>
<th>Degree of Freedom</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Motor Rotation</td>
</tr>
<tr>
<td>2</td>
<td>Reaction Gear Rotation</td>
</tr>
<tr>
<td>3</td>
<td>Test Pinion Rotation</td>
</tr>
<tr>
<td>4</td>
<td>Test Gear Rotation</td>
</tr>
<tr>
<td>5</td>
<td>Reaction Pinion Rotation</td>
</tr>
<tr>
<td>6</td>
<td>Test Pinion LOA Translation</td>
</tr>
<tr>
<td>7</td>
<td>Test Gear LOA Translation</td>
</tr>
</tbody>
</table>


Figure 3.34 Test pinion and gear mode shapes of 7 natural frequencies at 28 Nm
The system was also analyzed to try to predict and understand what is happening at the strange speed of 625 rpm which is expressed in Figure 3.22. The transmission error input was changed accordingly to correspond with the 17 Nm load. The mode shapes are shown in Figure 3.34. The final mode at 5096 Hz is very close to the determined natural frequency value of 5000 Hz that was found in the unusual signal. The results confirm the original induction that there was a combination of bending and torsion that was detected at that particular speed. Focusing on Figure 3.35, one can observe exactly what takes place in the system at that natural frequency. At 5096 Hz, there appears rotational and translational displacement in the test pinion and gear. Even though there is considerably more displacement in the pinion, the gear also experiences displacement in two different ways. These results compare very closely with what was found in the experimental results. Even though a natural frequency around the value of 3744 Hz (which was the other natural frequency found in Figure 3.22) did not show up in the analysis, much can be gained by observing the other natural frequencies predicted. The second natural frequency determined in Figure 3.34 of 371 Hz is very close to the mesh frequency that was determined in Figure 3.23 which was 312 Hz. Also the fifth natural frequency of the system is very close to the natural frequency of 4146 Hz in Figure 3.26. If anything, the seven degree of freedom system revealed the nature of Figure 3.22 which is also related with the loud whine noise observed at that speed. Figure 3.36 shows the force analysis done in the seven degree of freedom system. The bearing forces, dynamic transmission error and more importantly, the mesh force, show particularly the frequencies that excite
the largest force. The mesh force shows that at the last two natural frequencies, the mesh force reaches 600 lbs which is a lot of extra load considering this is more of an internal load than an external one. This gives some idea of what happens when a natural frequency is excited and the resulted added load during mesh.

Figure 3.35 Mode shapes of the seven natural frequencies
3.6 Summary

Understanding the dynamics of a gear system is essential in modeling accurately the load distribution of a particular gear pair. The strain gaged results shed light into how the torque, speed, lubrication, and surface finish played a major role in what type of strain signal the gear would see. During the course of one steady state test, the fluctuation of the strain (stresses) provided insight into how much the load varies during a test cycle. If the dynamic factor is to be included in the input for an LDP analysis, it would be best to take the average dynamic factor due to the variance in the signal. Based on the results, a different dynamic factor has to be applied for each load. It is still inconclusive as to
providing one dynamic factor that can be applied at any load. The natural frequency was calculated experimentally based on the time period of the oscillations. At least one consistent natural frequency of 3744 Hz was found with a natural frequency of 2500 Hz and 5000 Hz showed up in some strain signals. The speed of 625 rpm reveals the resonance occurring which as stated earlier could be a result of the mesh natural frequency exciting a second natural frequency. While the tests were initially being performed at steady state speeds, conducting tests in a transient environment with the speed of the test rig slowly increasing to maximum, proved to give a more comprehensive assessment how the strain changes over time with respect to speed. Even though the speed does not change the natural frequency, the observance of the oscillations is greatly affected by the torque and speed. The implications of the earlier tests reveal that the dynamic factors do decrease with increasing load.
Chapter 4

Micropitting Experimental Results

4.1 Introduction

As mentioned earlier, part of the goal in conducting fatigue tests was to repeat the tests that were done in 2006 [3] and to get a better understanding of the micropitting that supposedly occurred during those tests. The main purpose in performing each load test was to visibly see effects of micropitting, by profile inspection after the test and microscopic photos. Since the refurbished FZG has never run for long hours, there was a preliminary test pair that “wore in” the machine to prepare it for the actual tests. After gaining confidence in the performance of FZG, the micropitting test commenced. In certain micropitting studies, the engineer documents the number of pinion cycles to classify their micropitting results. In this particular test, the amount of time that each test...
was conducted was the main consideration along with the amount of torque that the test was run. A comparison between two different lubricants with different viscosities will be shown as well. Also a run through of the MATLAB based micropitting program with graphs and tables to compare its results with the actual fatigue testing are shown in this section. As noted in the scope of the thesis, the determination of the micropitting load capacity according to the ISO procedure [3] was the purpose of the creation of the program. In many micropitting studies, one either strictly analyzes this phenomenon by experimental means or by using the ISO procedure [3]. Rarely has there been a study comparing both methods to understand the feasibility and accuracy of the standardized procedure. Thus, the comparison of an analytical approach to an experimental approach in determining micropitting will be shown in this section.

Since the pinion is likely to see more wear than the gear, only pinion profile traces will be shown unless the gear shows signs of wear so this section will not be inundated with Figures. One of the reasons the pinion is more apt to see wear is because its teeth will see many more cycles than the gear due to the less number of teeth. It is the same reason why most micropitting Figures are in pinion cycles and not gear cycles. Also pitting seems to occur more in approach action which is the region where friction is of greater concern. Since the pinion is the driver, the first point of contact on the gear tooth is its tip and it traverses to the root at the end of the length of action. Conversely, the first point of contact on the pinion begins in the root and ends at its tip. Thus the driven gear constantly digs into the root of the pinion teeth which culminates in a heavily worn
region. Since the AGMA gears used in this experiment had tip relief and crowning, the digging action is of little consequence since material is removed to eliminate corner contact. Also, micropitting often occurs in the region where sliding and rolling are in opposite directions. Dudley [13] expounds on this when he discusses that sliding is away from the pitch line on the pinion.

In order to properly prepare for fatigue testing with a refurbished FZG machine, it was necessary to see how high the temperature gets at a certain load. The FVA procedure [4] identifies the inlet (sump) temperature as 90°C. Since our motivation was to follow this procedure closely apart from the torque, it was in our best interest to observe whether or not the FZG can get up to 90°C without auxiliary heating. The first temperature test was run at a gear speed of 1500 rpm, which corresponds to a pinion speed of 2250 rpm, and at a torque of 33 Nm. Every 10 minutes the temperature was recorded. This went on for 100 minutes for a total of 1 hour and 40 minutes. The second temperature test was run at 1505 rpm at a torque of 265 Nm. Time intervals of 30 minutes were used for this particular test. Figures 4.1(a) and (b) show the respective temperature test results.
It is important to note that the FVA procedure [4] outlines the micropitting procedure as having specific operating conditions. While the pitch-line velocity, which is 8.3 m/s, is the same in the procedure as it is in the actual test gear pair, the operating
lubricant temperature (sump temperature) is mandated to be 90°C. As it can be seen from the Figures above, the re-furbished test rig never reached the intended operating temperature. Naturally, at the higher load, the temperature would reach a higher sump temperature more quickly, however, the maximum lubricant temperature during any of the tests that will be discussed below was at 83°C. Even though the previous test that was conducted in 2006 [3] was said to have the same operating conditions as the FVA procedure [4], the discrepancy of the sump temperature was not too much of a determent expecting similar results. Since the experimental testing was done in conjunction with the analytical prediction of micropitting as expressed in the ISO procedure [1], the input parameters had to be updated to match the same operating conditions as the actual experiment. Thus, the temperature at which the test was conducted, even though it was under 90°C, was used in the analysis.

Table 4.1 shows a table of the testing that was conducted. The test number is associated with a new gear pair while the torque in the next column is the load at which the gear pair was tested. In some instances, one test gear pair underwent multiple loads such as test number 5 indicates. The lubrication column shows the type of lubricant the test was conducted under, with ‘M” representing the MIL L 23699 oil and “C” representing the Castrol synthetic oil. The total amount of operating hours for each test is listed in the next column. Since the first test that was conducted under 65 Nm was the first fatigue test of the re-furbished FZG test rig, the exact number of total hours was not documented. However, since the goal of the testing rests in the next two columns, the
consequence of knowing the number of hours for that test was nonexistent. Thus, the discussion of the test results begins at the 90 Nm load and not at 65 Nm. Area micropitting is considered “classical” micropitting where the micropits occupy are congregated in a certain area on the tooth flank where from a distance, the micropitted area may look like a small patch where the roughness is obvious incongruous with the rest of the tooth flank. Local micropitting is the opposite in which micropits may generate in a line perpendicular to the profile direction, typically in the dedendum region of the pinion tooth flank and the region above the pitch-line for the gear tooth flank. However, local micropitting also may be scattered in an uncharacteristic manner in the contact pattern. It is also important to note that the loading scheme was different than the one in the 2006 test [3]. This will be elaborated more on in Chapter 5 of this thesis.

<table>
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<th>Test #</th>
<th>Torque (Nm)</th>
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<th>$\lambda_{GFP}$</th>
<th>Test Duration (hrs)</th>
<th>Local Micropitting</th>
<th>Area Micropitting</th>
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<td>3</td>
<td>108</td>
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<td></td>
<td>80</td>
<td>Yes</td>
<td>Yes</td>
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<td>4</td>
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<td>No</td>
</tr>
<tr>
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<td>108, 165, 265</td>
<td>C</td>
<td>.18</td>
<td>48, 48, 32</td>
<td>Yes</td>
<td>No</td>
</tr>
</tbody>
</table>
4.2 Fatigue Test at 90 Nm

When the FZG ran for approximately 48 hours at 90Nm, as will be the case for all of the mentioned tests, there was virtually no sign of micropitting that would classify as “micropitting failure”. There was however, some wear occurring on both the pinion and gear that resulted in some material being removed that showed up on the profile inspection sheet but was almost nonexistent in the photographs at this torque. The FVA procedure [4] classifies failure due to micropitting occurring when the amount of material removed is greater than 7.5 micron in their staged tests. That was never seen at this load and for this amount of time. Figure 2.13 shows a magnified view of the contact pattern after the 48 hour test for the flanks of the pinion. With the 20x magnified lens, there are visible micropits shown. The phenomenon of micropitting is known to form in areas as opposed to scattered sections. It is interesting to note that that figure shows scattered micropits in the middle of the pinion tooth flank while the gear shows a micropitting area. However, along the machining marks of the tooth flank, the micropitted area does not grow.

Even though there appears area that may have micropit, the CMM machine gives a true assessment of whether or not and how much material has been removed. The diameter of the stylus on the probe can play an important role, especially when trying to determine whether a small amount of material has been removed. If the diameter is quite larger, then it may not detect these scattered micropits that are shown in Figure 2.14.
However, the stylus that was used is sufficient enough to traverse the tooth flank and measure profile deviation to detect the smallest amount of material that was removed. Figure 4.3 shows a plot of the profile deviation with respect to roll angle for the pinion. The start of active profile roll angle for the pinion was noted at 8 degrees by the CMM inspection while the end of active profile was recorded at 38 degrees. There is a possibility for the probe to slightly roll off of the tooth flank and the program will still record the data. In this case the profile deviation is recorded at -9.99 micron which is obviously unreasonable. In this case, data that revealed that number was ignored and the cutoff point for the number of data points ended at that point.

There appears to be a slight removal of material in the middle trace which is plotted alone in Figure 4.2. This is representative of tooth 1. A calculated 0.7 micron of material is removed after 48 hours which is considerably less than the FVA failure value of 7.5 micron. Thus, another test done for repeatability at the same load was initiated in an attempt to see more wear and potentially micropitting.

More wear was observed when the gear pair was inspected after the same gear pair underwent 32 more hours. Figure 4.3 shows twenty profile traces along the middle of each tooth on the pinion. As stated before, since the middle of the face width coincides with the middle of the contact pattern, this is the ideal trace for wear inspection. Approximately 1.6 micron of material has been removed for tooth 1 which means after 32 added hours, only 0.9 micron of wear has been added.
Figure 4.2 Pinion profile trace after 80 hours at 90 Nm

Figure 4.3 Pinion profile deviation of 20 teeth after 80 hours at 90 Nm
After 80 hours of testing, another 32 hour test was administered to enhance the wear on the gears. Figure 4.4 shows the profile deviations of four teeth that are exactly 90 degrees apart. It can be seen in the figure that not all four teeth show the same profile deviation. The calculated values of the wear depth are 3.2 micron for “Tooth 1”, 2 micron for “Tooth 6”, 2.3 micron for “Tooth 11”, and 2.6 micron for “Tooth 16”. As before, the profile trace reflects the probe measuring directly in the middle of the face width. With the added 32 hours, the total amount of time the gears were tested for amounted to 112 hours. Figure 4.5 shows a zoomed view of tooth 11 with the wear region clearly identified. Figure 4.6 shows how the wear progressed over time and the amount of material removed for tooth 1 on the pinion. It is evident that the hollow region deepens as the amount of test time increases. Figure 4.7 shows the exact amount of material removed with respect to time.

Figure 4.4 Profile traces of 4 equidistant teeth after 80 hours at 90 Nm
Figure 4.5 Fine view of wear region of pinion tooth 11 at 90 Nm

Figure 4.6 Time effect of progressive wear of pinion tooth 1 at 90 Nm
4.3 ISO Analysis at 90Nm

One crucial point in performing an analytical analysis such as the one given in the ISO procedure [3] is the fact that there are limitations when comparing such a method with an actual experiment and in this case a different set of gears. One of the biggest limitations in this procedure, as Kissling [26] confirmed in his independent study, is the inability of the ISO procedure to properly account for any tooth modifications. Even though there are a set of equations to apply a certain amount of linear tip relief, the approach of this standard is faulty only for Method B. Methods A and B were both described in Chapter 1 as well as the fact that the ISO procedure decides that the
optimum tip relief is when the amounts of tip relief of the pinion and gear are equal to the effective tip relief. This has been proven untrue by Kissling. Essentially, to accurately perform a post-processing analysis using the standard, a load modification factor should be implemented if the gear pair has modifications and it should only be used for Method B. Since Method A utilizes the contact stresses of LDP and tip relief and lead crown can be accounted for, there is no need to apply this load distribution factor for this particular method. In this case, the AGMA test gear pair is heavily crowned and does have tip relief. LDP [15] does a zone of contact feature that calculates a load distribution factor based on the load sharing that takes place. This factor would typically be multiplied by the torque in the ISO procedure to get a true estimate of the actual load distribution for the modified gear pair. However, since Method A produces the most pertinent results, only tables and Figures are shown to convey how the integrated contact stresses of LDP predicts the safety factor according to the ISO procedure. Thus, to simulate a real micropitting prediction, results from Method B are not included in this thesis.

Method A gives a greater breadth of information regarding contact stress, film thickness, specific film thickness, and safety factor because this method utilizes a load distribution program to calculate the load distribution at every point on the face width. Besides the geometry and basic input parameters of the gear pair, the main inputs of this program are the lubrication type, value of the permissible specific film thickness, and the sump temperature. Indeed the K factor product plays a role in the increased contact stresses but as far as the determination of the safety factor according to the ISO
procedure, these are the three necessary inputs. For all of the simulations conducted, the K product value (which includes the application factor, dynamic factor, and mounting factor) was chosen to be 1.1 and the gear quality was chosen to be 5. For each simulation done that tried to predict the experimental results, the permissible specific film thickness that was used was .08 as explained in Chapter 2. As shown in Table 4.1, the lubricant for this test was the Mobil Jet oil while the sump temperature was recorded at 53°C.

Hence, the MATLAB program is programmed to use the contact stresses from LDP [8] and calculate the flash temperature, lubricant film thickness, specific film thickness, and safety factor from it. Figure 4.8 shows a plot of the contact stress over the facewidth of the pinion with respect to the pinion roll angle. Table 4.2 shows the specific film thickness after mesh, respectively, at the beginning of the face width, 25% of the face width, 75% of the face width, and at the end of the face width. All of these values are with respect to the face width of the pinion. These values depend on the multiplier value that is selected in LDP when one wants a certain number of points in the face width direction. Thus, the algorithm created in MATLAB selects the value closest to the 25% and 75% mark. The program included specific film calculations before mesh to observe the impact of the bulk temperature alone. However, since the specific film thickness after the mesh was calculated using the flash temperature which is a calculation that includes the contact stresses, friction, and sliding velocity. This parameter is best when trying to evaluate the lubrication properties. The “NaN” in the tables represent the “Not a Number” value which essentially alludes to the fact that there is no contact at those
points. Table 4.3 shows the safety factor table given along the seven points of the length of the line of action. Thus, at a torque level of 90Nm, the analytical method appears to match up with the experimental results. It is unknown as to whether or not the ISO procedure had in mind classical or local micropitting when the safety factor is calculated. According to table 4.3 it appears that there would be a very low risk of micropitting and according to Table 4.1 the experimental results showed that only local micropitting was discovered for this particular test and area micropitting was nonexistent. Figure 4.9 shows a 2-D graphic of the safety factor. This helps visualize the region where micropitting is apt to occur. Table 4.3 shows that lowest safety factor is 2.2 which says that there is marginal chance of micropitting which corresponds to the experimental observations where no area micropitting was observed.
Figure 4.8 Predicted contact stress on pinion face width at 90 Nm

Table 4.2 Specific film thickness after mesh at 90 Nm

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Table 4.3 Safety factor against micropitting at 90 Nm

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Figure 4.9 Safety factor against micropitting plot at 90 Nm

4.4 Fatigue Test at 108 Nm

When there was no micropitting detected at 90Nm, the load was increased by 20%. A new gear pair was tested and was first run for 32 hours. After the inspection, there appeared some wear on both the pinion and gear. The Olympus DFPLFL 0.5xPF
microscope was used for this particular photo inspection to get a fuller range of the contact area and wear region. Figure 4.10 shows the gear contact area with a clear sign of wear in the dedendum region of the tooth flank at the very bottom of the contact pattern. There also appears scattered micropits above the pitch-line which is a region where high sliding occurs. Figure 4.11 shows a magnified view of the bottom wear area. This fits more with the understanding of classical micropitting where it generates in a continuous area as opposed to random patches over the contact area. What is interesting about the results of this gear pair is the fact that the gear appears to be seeing more wear than the pinion. This can be observed in Figure 4.12 where the pinion contact area is seen to have the scattered micropits and not an area where micropits are concentrated. Dykem bluing was applied to three consecutive pinion teeth prior to the test to observe if any dynamics were taking place during the test. Figure 4.13(a) shows the profile traces of four teeth on the pinion. The fact that more wear appears on the gear than the pinion is corroborated by Figure 4.13(b) with the gear profile trace because of the deeper hollow region. The maximum amount of material removed on the pinion as assessed by the four teeth can be approximated at 1.6 micron and is found on tooth 16, whereas the maximum amount of material removed on the gear can be approximated at 3.2 micron.
Figure 4.10 Gear contact area at 108 Nm after 32 hours

Figure 4.11 Magnified view of contact region of dedenda of gear at 108 Nm

Figure 4.12 Pinion contact area at 108Nm after 32 hours
Figure 4.13 Profile trace at 108 Nm after 32 hours
a) Four pinion teeth b) Gear tooth
With the same gear pair, an extension of the test was implemented to accrue more wear at the same load. The next test was run for 48 hours and as expected, the micropitting was enhanced when the pinion and gear were inspected. The micropitting on the flank of the gear still appeared to be greater than that of the pinion and Figure 4.18 reveals the increased micropitting in the addendum and dedendum region of the contact pattern. Comparing Figure 4.10 with Figure 4.14, it can be seen that the micropitting area increased in length and width. Figures 4.15, 4.16, and 4.17 reveal the difference in the micropitting areas between the four teeth 90 degrees apart that were inspected under the microscope. There is not a consistent micropitted area for the four teeth in the dedenda of the contact pattern which shows there is tooth to tooth variability of load distribution. There is however, a consistent micropitted pattern in the addendum region of the four teeth.

Figure 4.14 Gear tooth 1 contact area after 80 hours at 108 Nm
Figure 4.15 Gear tooth 9 contact area after 80 hours at 108 Nm

Figure 4.16 Gear tooth 16 contact area after 80 hours at 108 Nm

Figure 4.17 Gear tooth 24 contact area after 80 hours at 108 Nm
Figure 4.18 Pinion and Gear profile trace at 108Nm after 80 hours

Figure 4.19 Pinion and Gear wear region at 108Nm after 80 hours
Figure 4.18 shows the pinion and gear profile measurement after 80 hours. These traces were taken in the middle of the face width and it confirmed the previous results of the 32 hour extension in which the flank of the gear had significantly more wear on it than the pinion. The total amount of material removed for the pinion after 80 hours of testing was approximated at 2.4 micron, whereas 5 micron of material was removed for the gear which is more than double than the pinion. This is confirmed in Figure 4.19 where a zoomed view of Figure 4.18 is exposed and analyzed. The data points indicate the profile measurement point and its corresponding roll angle. There was so much variability between the propagation of the micropitting on adjacent teeth of the gear that a diagnosis was made for each tooth in regards to how serious the wear was on the tooth with respect to how it was on other teeth. A low, medium, and high assessment was made for each tooth by simple observation of the flank under a flashlight. Even though the observation was subjective, because of the large discrepancy in micropitting from the different teeth, a reasonable determination of the severity was able to be made. Even Figures 4.14, 4.15, 4.16, and 4.17 can give some insight into how micropitting was assessed. Table 4.4 gives a survey of the gear teeth and their resulting wear.
Table 4.4 Gear Tooth Micropitting Severity

<table>
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Figure 4.20 shows a picture of one of the pinion flanks that was tested. As with the gears, four different teeth were photographed and there is a consistent contact pattern for all of the teeth. Even though the micropitting area was not as large in the dedendum
region of the pinion as the gear, a different wear pattern showed in the region above the pitch-line. As it can be seen by the figure, there appears a different roughness as characterized by a brownish staining that begins on the top of the contact pattern and expands outward. This particular area is emphasized by the red arrow in the Figure. The uniqueness of this wear led to an investigation to see if the roughness area was actually different in the distinct wear regions of the contact pattern. Roughness profile measurements were taken at four different sections of the tooth flank. Since a datum was necessary to see the progression of the roughness of the wear, one section was a non-contacting section of the flank where there was no sign of wear. The next three sections were in the darker region in the micropitting area, the middle of the contact pattern in between the bottom section, and the top section where the red arrow is located in Figure 4.20. These three sections are broken down in Figure 4.21. Figure 4.22 show the roughness profiles of the section that contains no contact, top portion of the contact pattern (section 1), middle portion of the contact pattern (section 2), and bottom portion of the contact pattern (section 3) and their roughness values are 0.4541, 0.6476, 0.2912, and 0.4687 micron, respectively.
Figure 4.20 Pinion flank contact pattern after total 80 hours at 108 Nm

Figure 4.21 Three partitions for roughness profile measurements
Figure 4.22b Roughness profile after 80 hours at 108 Nm
a) No contact section b) Section 1 c) Section 2 d) Section 3
These roughness profiles are significant in showing the degree of variability in not just the roughness as the teeth interact under load during mesh, but also the different types of wear that results. It is important to note that the charts have much variation in the horizontal axis, thus the variety in the number of peaks on each plot. In each case there are roughly 40-50 peaks per millimeter. In section 1 where no contact occurred, there are sharp peaks on both the top and the hollow of the surface. In each of the other plots the tops of the roughness peaks have obviously been worn off to some degree. It is also important to note that the roughness of section 2 has a much lower value than the other sections, yet the peaks are not flattened. Since this location is near the pitch-point where there is no sliding, one would not expect much smoothening. In section 3, which is the main region of interest since that is where the micropitting is located, there is indeed flattening. The flattening is also obvious in Figure 4.22(d), however, the trace is dominated by one section with a pronounced cavity that is greater than any of the others. This observation is likely the cause of the large roughness value. As shown by Figure 4.21, the middle section, section 2, has a polishing type wear. This is confirmed by Figure 4.22(b) with a roughness value of 0.2912 micron. This is significantly lower than the roughness values at the other sections. The strange brownish area which is characterized by section 1 and a non-micropitting area, has a very high roughness value. In this region, the roughness traces show some smoothening along the top peaks. This is likely to be polishing wear since micropitting was not observed in the microscope shots. Polishing wear is observed and explained in an AGMA tribology test report, compiled by Bradley
This experiment confirms that the roughness across the tooth flank is not continuous. In fact, it gives credence to some of the interactions that may be unaccounted for in post-processing analysis and a detailed look into the different kinds of wear that occurs during testing.

### 4.5 ISO Analysis at 108Nm

When going from 90 Nm to 108 Nm, it is naturally understood that increasing the torque would increase the contact pressure which would decrease the safety factor. The load distribution factor acts inversely with that of the torque. So increasing the torque decreases the load distribution factor. This can be visualized by Figures 4.23 and 4.24 where the load distribution can be seen to spread across the face width. To emphasize the point, a picture at 90 Nm and 108 Nm is given below. This makes sense since as the load increases, a larger area across the face width would receive more pressure so in essence a lower distribution factor is needed to account for any area on the flank that has low or zero Hertzian contact stress. The predicted contact stress and pattern is pivotal in assessing the nature of the impending results of the contact area. Comparisons between the predicted and experimental contact patterns at 90 Nm and 108 Nm are shown in Figures 4.25 and 4.26.
Figure 4.23 Load distribution across face width at 90 Nm

Figure 4.24 Load distribution across face width at 108 Nm

Figure 4.25 Pinion at 90 Nm
a) Experimental b) LDP prediction
As stated earlier in the experimental findings, the reason the flank of the gear appears to have greater wear could be a result of the sliding that takes place during mesh. The ISO procedure does account for rolling and sliding velocities and Figure 4.27 gives a plot of the sliding that takes place at 108 Nm. By examining the figure, it can be argued that there is high sliding in the region below the pitch-line of the gear which agrees with the figure. Even though the plot is with respect to the pinion roll angle, there is high sliding, both above and below the pitch-line for both the pinion and gear flank. All of the same inputs that were used for the 90 Nm analysis were used, even the sump temperature which remained at 53°C. Figure 4.28 shows the LDP surface temperature distribution of the pinion. This gives an idea where LDP predicts where the peak surface temperature will occur on the contact pattern.
Figure 4.27 Sliding velocities at 108 Nm

Figure 4.28 Local contact temperature at 108 Nm
To understand this more, it would be necessary to have a plot of the local contact temperature to pinpoint the high likelihood of wear. It has been proposed that where the flash temperature is the highest, that is the most probable area where wear and micropitting will first occur. This could be due to the elevated temperature, but is just as likely because the arrangement of variables in the flash temperature equation is coincidentally similar to the resulting parameters that affect wear. Figure 4.28 shows the contact temperature, taken from LDP, at 108 Nm. The predictions of LDP are remarkably similar to the experimental results. It confirms that the region where the contact/flash temperature is highest coincides with the wear shown in Figures 4.14, 4.15, 4.16, and 4.17. It also shows that there will be wear in the region close to the one that is shown in those figures above the pitch-line. Table 4.5 shows the safety factor table of the seven points of contact across the face width. By examining the figure, it is seen that the point that has the lowest safety factor (highest micropitting risk) is at AB directly in the middle of the face width. This point corresponds to the dedendum region of the pinion which is congruent with the experimental results. Even though all of the safety factors in the figure are above one, the degree of discrepancy could still be explained by the micro-geometries of the actual gear pair. To reiterate, Table 4.5 is indicative of Method A of the ISO procedure which has greater benefit in analyzing the load distribution.
Table 4.5 Safety factor distribution at 108Nm for Method A

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4.6 Added Testing Time for Test at 108Nm

A new gear pair was used to generate repeated data and to better understand the propagation of wear with respect to time. Since it was difficult to observe the inception of micropitting, observing the progression of it would enhance our understanding of where and even how micropitting evolves on the tooth flank. Thus, the next course of action was to detect the propagation of wear by increasing the amount of time the test was run. The first time set was 4 hours which was followed by 12 hours, 28 hours, and finally a 48 hour test. All of the time extensions were run at the same load of 108 Nm. Figures 4.29, 4.30, and 4.31 all show pinion profiles after 12, 28, and 48 hour extensions, respectively.
Figure 4.29 Tooth 16 Pinion profile deviation after 16 hours at 108Nm

Figure 4.30 Tooth 16 Pinion profile deviation after 44 hours at 108Nm
Figure 4.31 Tooth 16 Pinion profile deviation after 112 hours at 108Nm

Figure 4.32 Wear progression of the pinion dedendum at 108Nm
For this progression test, a new pair of AGMA gears was tested to get a fresh observation in the development of wear at the same load and to gain some information on repeatability. As it is shown in the above figures, there is natural wear progression as the amount of time for the test increases. After 16 hours there is also hardly any indication that material has been removed and the amount is so small that it is hardly quantifiable. After 44 hours, about 2 microns can be approximated as the amount of material that has been removed. Finally, after 92 hours there is more wear and the amount of material removed is about 3.75 micron. Figure 4.33 supplements the profile inspection charts by visually showing the progression of wear from 4 hours to 48 hours. The conclusion of the matter is that no micropitting is observed, while the earlier tests showed micropitting. Since wear occurs but there is no micropitting polishing wear is most likely occurring.

Four teeth were again captured under the microscope to see any major similarities or differences. For the most part, the same polishing pattern was observed and there was little variability between the teeth. After the 4 hour extension, there was a similar local micropitting line near the bottom edge of the contact pattern of both the pinion and gear flank. However, it was observed after 16 hours, the flank of the gear again appeared to resemble area micropitting. This can be shown in Figure 4.34 where a photo of the contact area of the first tooth of the gear and pinion flank are shown after 16 hours. This is observed even moreso in Figure 4.35 where photos were taken at the end of the test after 92 hours. The same thing was observed for the previous tests at 90 Nm and 108 Nm,
thus prociding some repeatability. This is because the polishing wear occurred on the pinion and micropitting was found on the gear.

Figure 4.33 Pinion Tooth 16
a) After 4 hours b) After 16 hours c) After 44 hours d) After 92 hours
Figure 4.34 Micropitting observation after 16 hours
   a) Gear tooth 24 b) Pinion tooth 1

Figure 4.35 Micropitting observation after 92 hours
   a) Gear tooth 24 b) Pinion tooth 1
4.7 Fatigue Test at 108 Nm Using Castrol Oil

As stated in the opening section, another goal in conducting micropitting tests was to compare lubricants with different viscosities to notice the changes that would occur as a result of the different lubricants. The oil used in the earlier tests was Mobil Jet Oil II synthetic oil which is mostly used for aerospace applications. The Castrol optigear synthetic oil has applications in the additive manipulation to improve surfacing that utilizes hydrocarbons and plastic deformation of a unique kind for optimum performance [21]. The properties of interest in the post-processing analysis for gear are the kinematic viscosity at 40°C, 100°C, and the density of the lubricant at 15°C. These properties of the two test lubricants are summarized in Table 4.6.

Table 4.6 Lubricant Properties of Test Gears

<table>
<thead>
<tr>
<th>Property</th>
<th>Mobil Jet Oil II Synthetic</th>
<th>Castrol Optigear x320 Synthetic</th>
</tr>
</thead>
<tbody>
<tr>
<td>Kinematic viscosity @40°C (mm²/s)</td>
<td>27.6</td>
<td>325</td>
</tr>
<tr>
<td>Kinematic viscosity @100°C (mm²/s)</td>
<td>5.1</td>
<td>34.9</td>
</tr>
<tr>
<td>Density 15°C (kg/L)</td>
<td>1.003</td>
<td>.854</td>
</tr>
</tbody>
</table>

As shown in the table, it is expected that using the wind turbine oil would minimize the probability of having wear early on in the testing process. The wind turbine
oil also has additives to reduce wear and since it has a kinematic viscosity of more than 11 times that of the aircraft oil at 40°C which encouraged the first test of a new gear pair to be run at 24 hours while all the while expecting little wear. The profile inspection charts proved this to be true and thus an added 24 hours of testing was instituted for a total of 48 hours. Figure 4.36 shows the profile traces after 48 hours. From the below figure, 2.6 micron of material was removed as the testing time of the gear pair increased by 24 hours.

Figure 4.36 Profile deviation after 48 hours at 108Nm using wind turbine oil
4.8 ISO Analysis at 108Nm Using Castrol Oil

To understand the effect of the lubrication and the additives, the post-processing analysis will show how much some of the key parameters are changed by the different oil. Since the torque is kept constant, it will be easy to see how the safety factor is changed by the change in lubrication. As stated in Chapter 2, the permissible specific film thickness for the aircraft oil was chosen to be 0.08 and for the wind turbine oil was 0.18. The flash temperature is inevitably affected by the change in the viscosities since those properties play a key role in the pressure-viscosity coefficient for the bulk temperature. The flash temperature is a function of the pressure, friction, and sliding velocities. The mean friction is a function of the dynamic viscosity which is heavily influenced by the density and kinematic viscosity of the lubricant. Thus, the higher viscosity fluid would result in a lower flash temperature while the lighter viscosity lubricant will yield higher surface temperatures. The nuances in Figures 4.37 allude to the effect the viscosity has on the development of wear. There is a wider degree distribution in the aircraft oil with the peak flash temperature coinciding in the same location as wind turbine oil. The lubricant film thickness is another parameter that is influenced by the viscosity and density properties. Naturally, the higher the kinematic viscosity the higher the lubricant film thickness. This is expressed in Figures 4.38 with a lower film thickness shown in the aircraft oil. The minimum film thickness, which is the location where the safety factor is determined, is 0.2 micron lower in the wind turbine oil. There is also a wider distribution of film thickness levels in wind turbine oil. This is verified in Tables 4.7 and 4.8 with the
lower velocity and sliding parameter in the wind turbine oil. Table 4.9 shows the resulting safety factors which are nearly 3 times higher than the aircraft oil shown in Table 4.6. The peak points in which coincide with AB and DE of the points on the line of action according to the ISO procedure are represented by A, B, C, and D which result in 139°C, 120°C, 145°C, and 128°C, respectively. The lubricant film thickness of the extreme points in the same locations are represented by E, F, G, and H which reveal a film thickness of 0.081 micron, 0.078 micron, 0.262 micron, and 0.249 micron, respectively. The comparison of the safety factors of the two different lubricants can be made since the safety factor table for the aircraft oil is shown in Table 4.5.

![Surface Temperature Distribution](image)

Figure 4.37 Surface temperature at 108 Nm
a) Mobil Jet oil b) Castrol oil
Figure 4.38 Lubricant film thickness at 108 Nm
a) Mobil Jet oil II b) Castrol oil

Table 4.7 Parameters for Mobil Jet Oil II at 108Nm

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<th>Velocity Parameter</th>
<th>Load Parameter</th>
<th>Sliding Parameter</th>
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### Table 4.8 Parameters for Castrol Optigear oil at 108Nm

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### Table 4.9 Safety Factor for Castrol Optigear oil at 108Nm

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<th>50% of FW</th>
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<th>100% of FW</th>
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</table>

### 4.9 Fatigue Test at 165Nm using Castrol Oil

After performing a fatigue test at 108Nm, the next step was to decide how much the load should be increased. The MATLAB program was used to do an analysis through ISO procedure and vary the torques to determine which one would yield a safety factor of one. A load of 165 Nm was decided upon since it gave a relatively lower safety factor than the other extensions but would still have a safety factor higher than 1.0. Similar to the 108 Nm load test and using the same gear pair, two 24 hour time extensions were conducted using the wind turbine oil at a load of 165 Nm. It was interesting to see no
increase in wear was measured after 24 hours and 48 hours. Both time extensions observed approximately 2.5 micron of material. Figure 4.39 shows 4 profile traces directly in the middle of the face width of four teeth of the pinion while Figure 4.40 shows 16 profile measurements in the middle of the gear face width. The figure shows that virtually no wear was accrued over the 48 hour testing period. Figure 4.41 shows both tests of the pinion traced on top of each other and it can be seen that there is virtually no difference between the two tests.

![Graph showing profile traces](image)

Figure 4.39 Four profile pinion traces at 165 Nm after 48 hours
Figure 4.40 Sixteen profile gear traces at 165 Nm after 48 hours

Figure 4.41 Profile traces of pinion tooth number 6 after 24 and 48 hours
Figures 4.42(a) and (b) corroborate the CMM inspection of the profile charts expressed in Figure 4.39. There appears no significant change in micropitting over a span of 24 hours. These photos were true for all the teeth on the pinion. This is likely due to the higher viscosity and powerful additives and anti-wear chemicals that are in the wind turbine oil. There is only local micropitting observed for both of these runs located along the machining marks. These results are even more interesting since the previous test at
108 Nm had the same time increment for testing. Yet, there was an increase of wear going from 24 hours to 48 hours at that particular load. Since it is obvious there is no significant micropitting and the fact there was no change in wear for the different time sets, it is not necessary to discuss the safety factor results. Based on the ISO results for the last test of 108 Nm, it is apparent the safety factor would still be larger than one at 165 Nm. It is interesting, however, to see the predicted contact pattern of LDP at 165 Nm and comparing it with the experimental results. Figure 4.43 shows the contour plot of the contact stresses. The outline of the contact pattern matches very closely to Figures 4.42a and 4.42b.

Figure 4.43 LDP prediction of contact stresses at 165Nm
4.10 Fatigue Test at 265 Nm using Castrol Oil

The final torque extension was done at 265Nm, the largest load applied to these gears and the same load used in the AGMA tribology test [5]. It is important to note that at this load, AGMA experienced all three failure modes namely, micropitting, polishing wear, and macropitting using several lubricants. The first run was for 16 hours and inspected for wear. Increasing the load by approximately 100 Nm was motivated to push the gear pair to a very high loading capacity so that more wear would be induced. The same gear pair was used since the beginning of testing the gears with the wind turbine lubrication. There still was not much wear after the first run and the slope of the profile measurement makes it difficult to ascertain the approximate amount of wear that was on the pinion. After the second run, which ran for another 16 hours, the profile measurement remarkably appeared to show reduced wear after 32 hours than after only 16 hours. Two distinct inspection tests were conducted to see if the researcher did something wrong for the first test and perhaps tested the wrong side of the tooth flank. There the second inspection test was meticulously done to ensure that a proper trace was conducted. However, both tests proved consistent and the same traces with a sloped beginning at the start of active profile roll angle. Figures 4.44 and 4.45 shows profile traces of 16 teeth taken directly in the middle of the face width of the pinion and gear, respectively. Figure 4.46 shows the progression of wear from 16 hours to 32 total hours of the sixth tooth of the pinion. Figures 4.47(a) and 4.47(b) will shed more light into the wear progression or
lack thereof. The pictures show nearly exact results of micropitting so it is a quandary as to how it appears more material is removed for a less amount of time.

Figure 4.44 Sixteen pinion profile traces at 265 Nm after 32 hours
Figure 4.45 Sixteen gear profile traces at 265 Nm after 32 hours

Figure 4.46 Profile traces of pinion tooth number 6 after 16 and 32 hours
The figures above show nearly identical contact patterns with local micropits in generally the same location at the bottom of the contact pattern. One conspicuous difference is the entry of what appears to be a large pit near the tip of the contact pattern in Figure 4.47b. This pit is slightly to the left of the middle of the face width where the probe measured the profile. Another rather subtle difference between the two figures is the contact pattern shape. The contact pattern after 16 hours has a more rounded edge.
across the pitch-line and a more tapered incline to the tip of the contact pattern. After 48 hours, the once rounded edge across the pitch-line appears to be sharper and the top of the contact pattern has more of a point to it.

4.11 ISO Analysis at 265 Nm Using Castrol Oil

The final comparison between the experimental results and the prediction by the ISO procedure gives results that are similar to the previous loads. Figure 4.48 shows a 3 dimensional view of the contact stresses that are the result of a 265 Nm load. This figure shows that the maximum stress could reach 2.2 GPa. The prediction shown in Figure 4.49 in Method A of the ISO procedure, which is produced by LDP, shows a more curved area near the start of active profile than what the experimental results gave. The safety factor is plotted against the roll angle and face width to get a 3 dimensional visual of the high spots with respect to micropitting.
Figure 4.48 Three dimensional contact stress distribution at 265Nm

Figure 4.49 Two dimensional contact stress distribution at 265Nm
Figure 4.50 Three dimensional safety factor distribution at 265Nm

Table 4.10 Safety factors at 265Nm

<table>
<thead>
<tr>
<th></th>
<th>0% of FW</th>
<th>25% of FW</th>
<th>50% of FW</th>
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It is important to reiterate that the ISO procedure documents the risk of micropitting by the safety factor. It is not absolutely certain that micropitting will occur but it gives a reference frame to observe whether it is probable that micropitting will occur. Method A shows that there is little risk of micropitting occurring at this load which is exactly what the experiment showed. Apparently, at this particular load micropitting is close to happening. The safety factors are fairly close to 1.0 even though it is slightly above it. As stated earlier, 265 Nm was the highest load applied to the system. It would be interesting to observe the comparison of the safety factors of the different lubricants at this particular load. Figures 4.51 and 4.52 show the roughness profile measurements of the pinion and gear, respectively. These figures show that polishing wear was evident due to the lack of flattening of the peaks and the many peaks and valleys. Both measurements were taken directly in the middle of the face width along the contact pattern.

Figure 4.51 Roughness profile measurement of pinion
4.12 Li’s Thermal EHL Model Comparison

As stated in Chapter 1, another motivation for creating a MATLAB program for the ISO procedure was to provide gear micropitting data for the evaluation of a micropitting and wear modeling approach developed by Sheng Li [2] of the OSU Gearlab. Li’s model uses the roughness profile as an input to determine the lubricant and surface temperature. He conducts the analysis on the whole flank and divides it between 9 and 11 sections, however, usually omits the two end sections since there is no contact stress in those regions. In Li’s model, the inlet temperature is synonymous with the bulk temperature whereas in the ISO document, the inlet or sump temperature is considered when the lubricant temperature reaches steady state and a separate bulk temperature calculation includes the sump temperature. Thus, if the two models were used in their
purest form, the ISO model would naturally be higher by a difference of the bulk temperature and inlet temperature. This is expressed in Figure 4.53 where the ISO surface temperature of the pinion is approximately 20 degrees higher than Li’s model. The inlet temperatures for both models were 66°C but as stated above, Li does not perform a bulk temperature calculation. Figures 4.54 and 4.55 show the comparison between the surface temperature of Li’s model and the surface temperature of the ISO model at 108 Nm and 165 Nm, respectively. For both figures, the data was collected directly in the middle of the face width along the profile direction where the temperature is most likely the highest. Since Li’s thermal elastohydrodynamic model includes many discrete points, in order to overlay the results of the ISO procedure with it, the number of equally spaced positions to analyze had to be adjusted in LDP. The desired number of partitions is simply divided by the contact ratio to determine the correct number of positions needed to simulate the same model. It is important to note the difference in inlet parameters for the determination of the surface temperature for both the ISO and Li’s model. The bulk temperature for the ISO model was 86°C which explains the difference in the results of the two models. However, if the ISO model was considered as Li’s model and the sump temperature was the same as the bulk temperature, there the results would be a lot closer, as it is shown in Figure 4.54. Figure 4.55 shows the pinion surface temperature with the bulk equal to the inlet at 165 Nm.
Figure 4.53 Pinion Surface Temperature of Li’s and ISO’s model at 108 Nm

Figure 4.54 Model comparison at 108 Nm with bulk temp equal to inlet temp
It is apparent in both of the above figures that the ISO predicted surface temperature (using the LDP contact stresses) is higher than Li’s surface temperature. The profiles of both plots reflect the inherent nature of the input parameters. As stated above, Li’s model uses five roughness tests and averages them as the input whereas the ISO procedure mainly depends on the permissible specific film thickness and lubricant properties as the inputs. Li’s profile even reflects to some degree how a roughness profile looks with the many peaks and valleys.

4.13 Summary

There were four different torque levels discussed in this section that were tested and inspected for wear and micropitting. Two different lubricants were used to compare
results. The difference in viscosity levels was great enough to yield a very high safety factor for the high viscosity lubricant and also little to no wear on certain tests. The ISO procedure attempts to determine the micropitting load capacity analytically and the goal of the research presented in this thesis was to validate or point to errors in their analysis. The two methods expressed in the procedure were both utilized to compare the results. It is apparent that there may be some discrepancy between experimental results and what is predicted analytically. Experimentally, there were both scattered micropits over the contact area and also deep micropitting areas that showed up on certain teeth. Getting consistent wear results was also a challenge. There was virtually no micropitting observed using the Castrol synthetic oil. Even after increasing the load by a factor of four there was still no sign of micropitting. In the lighter viscosity lubricant, increasing the load and the time intervals after each previous test was designed to promote micropitting and this proved to be successful.
Chapter 5

Discussion and Conclusions

5.1 Introduction

As stated in Chapter 1, part of the motivation for conducting recent research in micropitting and dynamics on the FZG test rig was to enhance understanding of the tests that were done years ago [3]. The hope was to observe some of the same contact patterns, which were believed to be due to dynamics and also to come up with some of the same wear trends. The assembly of a refurbished FZG machine and the testing thereafter was constantly compared with the test results of the previous test and observations [3]. Repeatability is always a desired method when conducting experiments and the lack
thereof brings about questions and further investigations as to how and why the data diverges. As discussed in the previous chapters, much effort went into matching all of the parameters that were used in the earlier tests. The testing did not “match” the staged testing of the FVA tests, but it did in most other features. As stated above, the two main goals in observance for the new tests was the T-shaped contact pattern and also the micropitted areas when the fatigue tests were done. Chapters 3 and 4 provided the experimental results and analytical analyses and this chapter discusses whether or not that data is similar to the data and results from the 2006 tests.

5.2 Dynamics Observation Assessment

It was alluded to in Chapter 3 but there was not a repeat in the contact pattern that was observed in 2006. Dykem bluing was applied to the flanks of three adjacent teeth so that the contact pattern can be clearly seen gears tested at each of the loads that were mentioned in Chapter 3 had the bluing applied to them. The following pictures were taken for AGMA ground gears and at speed of 1410 rpm. The speed of 1410 rpm was chosen because that was the speed that appeared to have the most dynamic activity. The below pictures give an idea of the dynamics that took place at that particular load. For example, Figure 5.1 shows the contact pattern at 39 Nm and even though the picture may be slightly blurred, the contact pattern can still be clearly seen.
Figure 5.1 a) Gear contact pattern at 39 Nm  

Figure 5.2 a) Gear contact pattern at 56 Nm  

Figure 5.3 a) Gear contact pattern at 71 Nm  

Figure 5.4 a) Gear contact pattern at 82 Nm
As it can be seen, none of the above figures reflect the dynamic patterns that were observed in 2006, which are also shown in Figures 1.1 and 1.2. All of the pictures on the left are the static experimental dynamic contact patterns and the figures on the right are the LDP predicted contact pattern at the same torque load. The predicted contact stress pattern and the dynamically measured contact pattern match quite well indicating very little dynamics. Figure 5.7 shows a photo from the 2006 test [3] at a load of 70 Nm. A comparison of the contact patterns of the previous test [3] and the current contact pattern with the load of 71 Nm can easily be made. The T-shaped contact pattern blamed on dynamics is clearly evident in the 2006 tests and not evident in these later tests. This would indicate that the new test rig obviously performs differently than the earlier tester.
5.3 Micropitting Observation Assessment

Even though the dynamics and micropitting phases of research were presented as separate chapters, making one believe that they are completely unrelated, there is actually a correlation between the two that coincide with the motivation of the research in its
totality. Accounting for the dynamic load for load distribution modeling is essential to have an accurate assessment of not just the contact stress but the impending wear, thus, a desire to observe the same wear patterns as of the 2006 test [3] was also the goal. An important factor that plays a role in the comparison of the fatigue experiments expressed in this thesis and the previous test [3] is the loading procedure. When the test was conducted in 2006, the test gear pair underwent a staged loading scheme where it modeled the FVA procedure [4]. Since the gear teeth were heavily crowned, the procedure done in 2006 was modified to account for the contact stresses. LDP was used to predict the same contact stress that the FVA procedure used and correspond those stresses with the new torque load. As expected with crowned teeth, the torque would inevitably be lower than an unmodified gear pair.

Even though the same gear pair was used in the recent tests, the loading was based more off of observation than a particular staged loading scheme. After each test

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was conducted and the post-processing analysis was done, the results of the profile inspection and photos dictated the next course of action. Typically, if there was no significant micropitting then the load was increased by a decent amount to initiate the surface degradation. Thus, only single loading was used for each test which could have played a major role in not seeing the same results in the previous test.

In discussing the comparison of the results of the two testing periods, probably the most significant and even alarming results is the fact that micropitting was not detected as early or as severe as what discovered in the 2006 test. What brings this comparison to an even more interesting conclusion is the fact that much higher load was used than the ones used in the previous test [3]. Even though the highest load of 265 Nm was applied to the test gear pair under the Castrol (much higher viscosity) lubricant, initially the test gears were tested under the same lubricant as the 2006 test. Figures 5.9 and 5.10 compares the pinion and gear of the 2006 test with the current test while figure 5.11 shows the LDP prediction of the pinion and gear at 265 Nm, which was the same load that the comparison is being made against with the 2006 test. Even though the load and lubrication at which the pictures of the 2006 test were taken is unknown, these pictures will be compared with the highest load of the recent tests which were at 265 Nm to convey the disparity.
It is evident that even macropits developed in the pinion of the 2006 test, whereas there was no such indication of that occurring in any of the current tests. The gear of the previous test also has a much larger micropitting area than the current test. Even if the maximum load that was used in the 2006 test, which was 65 Nm, was compared with the photos of the current test, there is nearly 70% more load in the current test than in the 2006 test.
previous test. This leads to another important point of the manifestation of micropitting. As alluded to in Chapter 4, in 2006 what was thought to be a failure mode of micropitting was actually something that is now known as “polishing wear”. This seems to be when there is apparent wear but it cannot be classified as micropitting. This can be taken from the Tribology Report compiled by Bradley [5] for the Helical Gear Rating Committee.

5.4 Redesign of Test Pinions

The testing of the current AGMA gear pair provided insight into some modifications that can be made for better future testing. As stated earlier, the lead crown of the gears is great for correcting for misalignment and accounting for the location where the contact stresses are the highest if there is also tip relief. Also, as expressed in the 2006 technical paper [3], the modified loading procedure that was used was done to correspond the contact stresses predicted by LDP which accounted for tooth modification with that of the FVA procedure [4] that had no tooth modification. Thus, high contact stresses were yielded for a low torque load for the modified gear pair. As a result of wanting to increase the torque capacity and get higher loads that are less vulnerable to rig dynamics, six AGMA pinions were re-modified to remove the lead crown. Also, the removal of the lead crown allows for applying higher torques when getting high contact stresses. Based on earlier results this not only spreads out the stress pattern but also further reduces the chance for high dynamics due to reduced profile contact ratios. The original plan was to use the Kapp form grinding wheels so that the gear and pinion would
experience the same tip relief but due to the fact that manufacturers were under heavy work loads and since they had these grind wheels, Ford’s gear prototyping lab in Livonia, Michigan volunteered to use their Maag generation grinder to trim the lead crown on the pinion. As a result, there is less tip relief for the pinion.

Since LDP was an integral part in assessing the predicted contact stresses associated with a pinion that does not have any lead crown, it would be beneficial to observe the results of the contact stress model. Figure 5.12 shows the profile of the contact stresses that LDP predicts for a pinion with profile modification. It shows that the peak contact stress will reach 1478 MPa. Comparing that with Figure 5.13 where the contact stress reaches only 1003 MPa, it can be seen more clearly the effect of the removal of the lead crown. In order to get a peak contact stress of 1478 MPa, the new gears would require a torque of 148 Nm. This is confirmed in Figure 5.14. An indicator of the likely reduced dynamics of this gear is that the transmission error of the new gear at 1478 MPa is reduced by 17% from the transmission error of the original gear. Table 5.2 summarizes the peak contact stress at the load that was tested in the experiment discussed in the previous sections. Based on the above figures and the above table, it is apparent that the re-modification of the test pinions to have no lead crown, will result in an overall ability to equip the pinion with uniform load distribution and load carrying capability. Figures 5.15 and 5.16 show the lead charts of two teeth at four equidistant locations along the face width.
Figure 5.12 Contact stress distribution with lead crown at 65 Nm

Figure 5.13 Contact stress distribution without lead crown at 65 Nm
Figure 5.14 Contact stress distribution without lead crown at 148 Nm

Figure 5.15 Pinion with lead crown
Table 5.2 Peak contact stress of pinion with and without lead crown

<table>
<thead>
<tr>
<th>Torque (Nm)</th>
<th>With no Lead (MPa)</th>
<th>With lead crown (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>90</td>
<td>1169</td>
<td>1639</td>
</tr>
<tr>
<td>108</td>
<td>1285</td>
<td>1723</td>
</tr>
<tr>
<td>165</td>
<td>1563</td>
<td>1950</td>
</tr>
<tr>
<td>265</td>
<td>1957</td>
<td>2237</td>
</tr>
</tbody>
</table>

Figure 5.16 Pinion without lead crown
5.5 Conclusions

The contents of this thesis provided an extensive overview of both dynamics and micropitting. These experiments offered a better understanding into exactly what took place during the 2006 tests [3]. A lot was unknown about the development of micropitting at that time and the experiments conducted in this thesis shed more light into not only the strange “T-shaped” contact pattern that arose but also the repeatability of the micropitting tests that were done. Also a thorough analysis of the ISO procedure [1] was performed in conjunction with the experiments to test its validity. As stated in Chapter 2, the main parameter of uncertainty was the permissible specific film thickness and the ability to ascertain its value. Reasonable assumptions were made in determining this value, however, the fact that Figure A.1 in the ISO document is only for mineral oils puts limitations and a lack of confidence in the validity of the whole procedure. However, even with the uncertainty of the permissible specific film thickness, the results expressed in Chapter 4 revealed decent prediction accuracy with respect to the experimental results, specifically when the analysis using the wind turbine oil was done. Thus, the following conclusions can be made through the research that was conducted:

- What was thought to be the observation of micropitting in the 2006 test [3] was actually “polishing wear” which was something that was not recognized at the time of the writing of that paper.
• The “T-shaped” contact pattern that was observed in 2006 was a result of the rig dynamics. This was confirmed by LDP when it predicted that the true loading at that time was 50% above static loading. The recent experiments attempted to reproduce that contact pattern but were unsuccessful which led to the conclusion that even though dynamics were the cause of that pattern, it was perhaps specific to the test rig that was used in 2006.

• Due to the fluctuation of load during a particular test run, it is difficult to determine a specific and “universal” dynamic factor that can be applied for a given test. As a result, dynamics will not only have to be adjusted with respect to load but also the change of load during mesh should be taken into account and the ideal dynamic factor would be the average dynamic factor.

• At the speed of around 625 rpm, there is definitely a resonance occurring with the likelihood of two natural frequencies being excited. A whine type noise was heard and the strain gage showed a strange strain signal that showed two distinct natural frequencies.

• Though it has been observed that at lower loads, more dynamics were observed, there was still a sharp overshoot for even the highest load at 224 Nm. There also
seems to be little relationship between the type of finish and lubrication on the dynamics.

- Though the permissible specific film thickness is difficult to procure, the value that was used by deductive means proved to provide fairly decent results with regards to the prediction of micropitting. Since there is no experimental data provided by the researchers of the ISO document for the public, the results of this thesis presents experimental comparisons to assess the validity of the procedure.

- A second ISO [22] analysis was created by Houser which uses the load distribution contact stress for evaluation. In this spreadsheet, the flash temperature was determined by LDP and the diameter at which it occurred (which corresponded to points AB and DE of the ISO) was used in the calculation. The results showed that the safety factors were similar to the ones in MATLAB created program.

- For the experimental tests that were done, local and area micropitting were observed with no consistent pattern. However, the loads tests that were done with the aircraft oil consistently showed the gear flank have the classical area micropitting while the pinion flank had mostly local micropitting. However, most of the observed wear was not due to micropitting but was due to polishing wear.
5.6 Recommendations for Future Work

- With the redesigned pinion, the next course of action is to actually test a gear pair without the lead crown to observe the results. As stated above, the testing of the gear pair should have a positive effect on the dynamics of the system and since the strain gages are mounted on the uncrowned test gears, it will be possible to repeat the strain gage tests.

- Revisit dynamic testing to get a more enhanced database of the effect of lubrication, finish type, and speed run-up time on the dynamics of the system.

- Apply strain gages in the location where the contact stresses are maximum to have a more comprehensive understanding of the different dynamic factors that would result, mainly the dynamic contact stress factor as explained in Table 1.1.

- Conduct a fatigue test on the same FZG test rig that was used in 2006 to see if indeed it was the actual machine that produced the “T-shaped” contact pattern since that pattern was not observed on the refurbished test machine.

- Perform an actual staged loading test that is outlined by the FVA procedure [4] to have similar results to compare with the 2006 test. The staged loading provides a
run-in at lighter loads and steadily increases in torque which culminates in an endurance test ran for 80 hours.

- Ascertain the correct value of the permissible specific film thickness by contacting lubrication companies. Even though this was done in the past, perhaps they conducted recent tests that would be of some assistance.

- Assess the aerospace lubricants and reference specifically micropitting using the FVA procedure outlined in Method A of the ISO procedure.
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