LOADED TRANSMISSION ERROR MEASUREMENT SYSTEM FOR SPUR AND HELICAL GEARS

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* * * * *
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ABSTRACT

The majority of loaded static transmission error test stands developed in the past had little success generating accurate results versus analytical predictions for parallel-axis gearing. Design flaws historically caused issues with speed and torque control, ultimately, leading to erroneous results. Fortunately, some of these issues were corrected through the years, most recently by Schmitkons [1], for loaded transmission error testing of bevel gears sets. The original goal of this thesis was to translate those successes into a test rig for parallel-axis gearing that can measure static transmission error and shaft deflections to take a look at transmission error, shuttling and friction force excitations. However, due to difficulties in achieving a good comparison between experimental results and analytical predictions, the goal was shifted towards simply assessing the performance of the new test stand. By using virtually the same control setup and measurement setup as the loaded bevel gear static transmission error test stand, the new test stand generated static transmission error results for both spur and helical gears at various torque levels. Those results were compared to analytical prediction software codes (WindowsLDP, RomaxDesigner and Helical3D), using optimal and measured micro-geometry topographies. The static transmission error results compared well at low torque values, but deviated from the predicted trends at higher torque values. Ultimately, lessons learned from this test setup will be reflected in future experimental work in order to better assess the accuracy of prediction tools.
Dedicated to My Family
Mom, Dad, Matt and Mike
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CHAPTER 1
INTRODUCTION

1.1 Introduction

The main focus of this thesis is the experimental testing and analytical analysis of loaded static transmission error for parallel-axis gearing. Transmission error is a proven contributor to vibration and noise within gearing applications, yet less is known about the individual contributions of transmission error, shuttling and friction force excitations on overall noise and vibration. Tom Schachinger [2] designed three gear pairs to independently isolate these forces in 2004, and the goal of this thesis is to study those gear pairs and compare experimental results to analytical prediction software. Moreover, transmission error and mesh forces are a function of torque, so extremely precise experiments are needed in order to quantitatively express results. Testing was conducting at The Ohio State University GearLab, using a test stand specifically developed for slow-speed and high-torque measurements simulating static conditions. By using a direct drive system utilizing a DC motor as the power source and an air-brake as the power absorption, a large range of torque values were able to simulate the working range of the gear sets. Shaft rotations were measured using angle encoders and processed using the ROTEC rotational analysis system which calculates transmission error. Ultimately, if the measured transmission error values are similar to the analytical predicted trends, software
packages can be utilized to help alleviate some of the time, and more importantly money, spent prototyping and testing gear sets prior to mass production.

1.2  Research Background

1.2.1  Helical Gears

Used throughout the gear industry, in general applications such as automotive transmissions to more advanced aerospace topics, helical gears serve as one of the major contributors to an overall reduction in powertrain noise. By machining the gear teeth at an angle, illustrated in Figure 1.2.1.1, helical gears tend to run smoother and quieter than spur gears due to an increase in contact ratio. Additionally, helical gears create an axial force that needs to be taken out through a trust bearing, increasing the importance of bearing selection during the transmission design process. Figure 1.2.1.1 also shows a simplified figure of a manual transmission, illustrating a traditional use of helical gearing. The power comes through the lay shaft and is then transferred to the output shaft via whichever gear set is engaged by the driver. Ultimately, a complete transmission, including bearings, shafts, gears, housings, etc., is shown subsequently in Figure 1.2.1.2.

Figure 1.2.1.1: Helical gears and a simplified manual transmission setup
1.2.2 Introduction to Transmission Error

Theoretically, if two mating gears are geometrically perfect with infinite stiffness, when one is rotated from a reference angle the other should rotate exactly the same angle multiplied simply by the gear ratio, but because of manufacturing imperfections and material deflections an error in motion transfer, or ‘transmission error,’ occurs.

Simply stated, transmission error is

\[ TE = \theta_2 - \theta_1 \left( \frac{N_1}{N_2} \right) \]  

(1)

where \( \theta_1 \) = the rotation angle of the pinion, \( \theta_2 \) = the rotation angle of the gear, \( N_1 \) = the number of teeth on the pinion, and \( N_2 \) = the number of teeth on the gear. Figure 1.2.2.1
illustrates the difference between the position of the driven gear and where it should be theoretically, which ultimately leads to a dynamic excitation within the transmission system as the gear set rotates through multiple cycles.

![Diagram](image)

Figure 1.2.2.1: Definition of transmission error [11]

History has shown that there is typically a direct correlation between transmission error amplitudes and sound pressure levels radiating from a transmission housing [3]. So from an engineering design standpoint, minimizing transmission error throughout the working torque range of whatever machinery they are designing is extremely important. For example, a perfect involute spur gear set has theoretically zero transmission error at zero torque, but once load is introduced to the system, deflections and corner contact cause a linear increase in transmission error; simply the more load, the greater the transmission error. So as an engineering solution, micro-geometry modifications are intentionally included on the surface of the gear tooth, potentially causing non-zero
transmission error at zero load. Transmission error then decreases towards a minimum value at the design load, and then increases once the relief is no longer affective.

Figure 1.2.2.2 illustrates a typical transmission error result, where the fluctuation in motion transfer is averaged to the pinion shaft. The major sine wave, or low frequency contribution, is due to the gear eccentricity, or runout, and the super-imposed high frequency contribution is due to tooth-to-tooth errors. By taking the angular units (µ-rad) and multiplying them by the base radius of the pinion, transmission error can then be expressed in linear units (µ-m).

![Figure 1.2.2.2: Average total transmission error](image)

Moreover, Figure 1.2.2.3 shows the Fast Fourier Transform (FFT) of the total transmission error from Figure 1.2.2.2, illustrating specific orders that contribute to the spectral content of the signal. The amplitude of runout is calculated in linear units and shown in the figure at the first order. Tooth-to-tooth transmission error harmonic
amplitudes are located at integer multiples of the pinion tooth count, which for this specific gear set are orders 50, 100, 150, 200 and 250. Once this analysis is performed for multiple torque values, a comparison can show how the mesh harmonic amplitudes change as a function of torque.

Figure 1.2.2.3: Transmission error spectrum of total transmission error

1.3 Objectives

The main objective was to perform measurements on a series of helical gear designs that allowed gear excitations (TE, shuttling, and friction) to be isolated. The gears were to be inspected to verify that their manufacturer met the design goals and then measurements and predictions of gear excitations were to be compared. In the course of this study, an existing OSU test rig was chosen for this experimental analysis and a donated fixture was to be incorporated into the existing rig. The rig will include all of the
hardware necessary to mount the test gears, apply power and torque to the system, measure relative shaft motions, and calculate transmission error.

Once the test stand is built and the speed and torque control are working properly, a series of tests will be completed to validate the new setup. First, un-loaded transmission error tests will be completed using both the Gleason/Goulder Single Flank Tester and the new test stand to see if the two measurement systems produce similar results. Then, repeatability studies will be completed in order to determine the percent fluctuation in measured transmission error for tests started at different locations during the hunting ratio. Thirdly, loaded transmission error measurements will be performed throughout a range of torque values to see how mesh harmonic amplitudes change as a function in torque.

Unfortunately, some of the early transmission error measurements did not make sense so additional very high accuracy spur gears were also to be tested on the rig. TE had been measured previously on these gears so it was expected that the current study would provide similar results. Because of the sidetracking of the study to obtain better TE data and TE correlations, the objectives related to the shuttling and friction forces were cut back and the primary focus of the study became to assess the TE measurement performance of the donated fixtures when mounted in the OSU rig.

1.4 Thesis Overview

Chapter 2 includes the development of a new loaded static transmission error test stand for spur and helical gears. In addition to outlining the hardware included in the test
stand, it describes the control setup, as well as the measurements setup for transmission error measurements.

Chapter 3 discusses preliminary test results, comparing transmission error data between the new test stand and the Gleason/Goulder Single Flank Tester. Also, it includes the repeatability studies to validate the accuracy of the new test stand, and illustrate the experimental spread in data for tests started at different locations throughout the hunting ratio of a gear set. Ultimately, Chapter 3 includes the loaded static transmission error results for several gear pairs throughout a reasonable range of torque values.

Chapter 4 compares the experimental results from the new loaded static transmission error test stand to analytical predictions generated using commercial software packages.

In conclusion, Chapter 5 will serve as a summary of lessons learned, and also include future recommendations for whoever takes over the responsibilities related to loaded transmission error measurements in the future.
CHAPTER 2
LOADED STATIC TRANSMISSION ERROR TEST STAND DEVELOPMENT

2.1 Introduction

This chapter includes the history of the loaded static transmission error test stand at OSU, as well as all of the current components utilized for the testing of parallel-axis gearing. First, all of the donated and existing hardware is discussed, as well as the additional hardware needed to assemble the test stand from input to output. Next, the control and measurement hardware is discussed to show how the test stand is controlled and how transmission error results are acquired from the setup utilizing over-the-shaft angle encoders and a ROTEC data acquisition and analysis system. Once the test setup is established, the next step is to perform actual transmission error measurements, which are reported subsequently in Chapter 3.

2.2 Background

Transmission error measurements historically are completed in one of two ways; first, by using accelerometers mounted close to the gears to measure changes in rotational acceleration along with numerical integration, and secondly, by using angle encoders to measure the relative motion of the gear shafts with an analog multiply/divide technique.

Blankenship and Kahraman [4] performed accelerometer type transmission error
measurements the dynamics rig with successful results for both pseudo-static and
dynamic transmission error. These results are used as a reference for analytical analysis
done in Chapter 4.

The first person at OSU to design a test stand for loaded gear transmission error
measurements utilizing angle encoders was Bassett [5] in 1985. He designed the test
stand with two DC torque motors running against each other to supply power and torque
to the system. The maximum torque for that setup was about 1400 ft-lbs with an
estimated running speed of 2-5 rpm. The signal processing was done by utilizing the
optical encoders previously used by the Gleason/Goulder Single Flank Tester located at
the ends gear shafts. These extremely precise optical encoders, with 18000 lines of
resolution per revolution, generated the signals needed for calculating the motion error
present using the multiply/divide technique. Torque was to be controlled by a DC
servo/amplifier controller utilizing an analog computer for proportional and derivative
speed control and proportional torque control. Considering the optical encoders were
extremely sensitive to misalignments and deflections, he recommended that flexible
couplings and/or extremely accurate alignment be used to help alleviate distortion in the
encoder signals.

Next, Schutt [6] took over the loaded transmission error test stand and made it
operational using a Falk double reduction gearbox with only the first reduction utilized.
He was the first to produce loaded transmission error results, but unfortunately had
trouble with the speed and torque control. Fluctuations in speed and torque caused a
discrepancy between the transmission error measurements and prediction software. He
recommended implementing a digital control system, a digital transmission error setup, and some sort of brake torque control.

Foster [7] took Schutt’s recommendations and designed a digital control system for the speed of the two DC torque motors used in the Loaded Single Flank Transmission Error Test Stand, which was used until Schmitkons [1] implemented DASYLab along with a National Instruments data acquisition system for the Loaded Bevel Gear Static Transmission Error Test Stand.

D. Hochmann [8] was the first to use a power absorbing configuration, by replacing one of the DC torque motors with a pneumatic brake. Ultimately, he ran into the same issue as all of the students before him, and had trouble controlling the torque and acquired erroneous results when compared to analytical predictions.

So after Dziech [9] changed the setup to test non-parallel axis gearing and Poling [10] implemented a PID pneumatic controller for brake torque, Schmitkons [1] was the first to generate viable transmission error results. He implemented new digital control systems for the speed and torque control, used the ROTEC transmission error analysis package for data acquisition and post-processing, and ultimately made the assembly process much more easy and repeatable, something that has always been an issue in the past. Along with his entire control and measurement setup and future recommendations the physical setup of the New Loaded Static Transmission Error Test Stand for Parallel-Axis Gearing is described in the following section.
2.3 Test Stand Development: Physical Setup

2.3.1 Donated Components from Ford Motor Company

Initially, the plan was to use the variable center distance gearbox for the loaded static transmission error tests, but Ford Motor Company was gracious enough to donate a rig already set up for transmission error measurements. It was designed in the 1990’s by Clapper; a past graduate of OSU GearLab, and includes: gear pedestals with shafts and bearings, Heidenhain angle encoders, arbors to hold helical gears, as well as all of the wiring up to, but not including the ROTEC rotational analysis system. Figure 2.3.1.1 shows the section donated by Ford Motor Company. The input and output gear pedestals are adjustable to facilitate changes in center distance and facewidth so a wide range of

Figure 2.3.1.1: Donated section of loaded static transmission error test stand from Ford
gear designs can be tested. Additionally, Figure 2.3.1.2 shows the arbors designed to hold helical gears used in a Ford transaxle and Figure 2.3.1.3 shows the Heidenhain angle encoders utilized for the shaft motion measurement necessary for transmission error calculations. These components were matched with a DC motor for power supply and an air-brake for load application discussed in Section 2.3.2, as well as additional components needed for the overall assembly discussed in Section 2.3.3.

Figure 2.3.1.2: Input and output arbors for Ford transaxle gears

Figure 2.3.1.3: Heidenhain rotary encoder and a side view of gears in mesh
2.3.2 Existing Components from GearLab

To go along with the donated section of the test stand, it was necessary to include some of the test hardware the GearLab owns to control the speed and torque of the test stand, as well as PC based control software and digital processing. The existing components already owned by the GearLab include: a DC torque motor, a LeBow torquemeter, an Eaton Airflex brake, a Fairchild pneumatic controller, a PC with DASYLab already installed, and the ROTEC hardware/software package. All of these components are illustrated in Figures 2.3.2.1 and 2.3.2.2.

The motor is a Sierracin/Magnedyne DC torque motor with peak torque of 12,000 in-lbs and a no-load speed of 15 rpm. The internal workings of the motor include seven pairs of brushes, 28 poles and 253 commutator bars. Along with the Glentek amplifier, this is how the test stand gets its power, and it was originally the second motor in the Loaded Single Flank Test Stand previously discussed in Section 2.2.

The LeBow 1228 slip ring torquemeter has a maximum working torque range of 10,000 in-lbs and is placed on the output shaft of the new test stand. Along with a National Instruments 2310 signal conditioner amplifier, it supplies the feedback signal necessary for the torque control in the system.

The Eaton AirFlex 206 WCB pneumatic brake was added to the test stand instead of the 214 WCB previously used by Schmitkons [1], because the torque range for the gear sets studied have a much lower operating torque than the rear differential previously used. It includes two friction discs, which rotate with the shaft coupled to the output side of the test stand, and piston sections. Once air pressure is applied to the stationary
section of the brake, the friction surfaces slip against one another and create the torque necessary for loaded transmission error measurements.

The Fairchild pneumatic controller is used in addition to an air pressure regulator to control the brake torque. By utilizing the torque meter as feedback, the PID controller allows the right amount of air pressure to stabilize the brake torque. Unfortunately, there is a limit to the torque applied to the system because of an instability occurring above 150 N-m, so the working range of loaded transmission error will only be zero to 100 N-m.

The PC with DASYLab, along with signal conditioning boxes, power supplies and a National instruments data acquisition board, controls the speed of the test stand and the set value for the torque set to the pneumatic controller used to apply air pressure to the brake.

The ROTEC transmission error hardware/software pc is a system that takes the encoder signals and calculates transmission error. How it does the transmission error analysis is discussed later in Section 2.4.

Figure 2.3.2.1: Sierracin/Magntyne DC Torque Motor, Eaton AirFlex 206WB Brake
2.3.3 Additional Hardware Components for Assembly

Ultimately, not all of the donated and existing components directly assembled with each other, so additional hardware components were designed in order to connect the donated section with the input motor and output load, along with mounting the entire setup to the bedplate located in room W066 of Scott Laboratory. When the donated section from Ford first arrived at OSU, it was assembled on a cart so as to roll the test
stand in and out of a test dynamometer for easier assembly and disassembly during their noise and vibration testing process. Since the centerlines of the input motor and output load were much lower than the centerline of the donated section, a new baseplate along with risers needed to be designed. Figure 2.3.3.1 shows the new baseplate and risers machined to facilitate the change of centerlines. Additionally, couplings and a spacer, shown in Figure 2.3.3.2, were designed and machined in order to connect the shafts of the input motor and output brake to the donated section of the test stand. Once all of these components were assembled and aligned, the final hardware setup of the test stand, shown in Figure 2.3.3.3, was complete.

Figure 2.3.3.1: New baseplate and risers for loaded static transmission error test stand

Figure 2.3.3.2: Coupling and spacer needed for assembly of test stand
2.4 Test Stand Development: Measurement and Control Setup

2.4.1 DASYLab – Speed Control and Torque Set Value

In order to control the hardware described in Section 2.3, DASYLab was chosen by previous students as the software package. The majority of the control theory and setup was completed and validated by A. Schmitkons [1] in 2005, so minimal changes were needed in order to operate the new loaded static transmission error test stand. By adding an extra block to take into consideration the discrepancy between the previous and current optical encoder resolution (18000 lines of resolution previously and 9000 lines of resolution currently), and modifying slightly the PID control set values, the speed control theory was complete. Figure 2.4.1.1 illustrates the flowchart of the control theory.
and Figure 2.4.1.2 shows an actual snapshot of the DASYLab module during test stand operation.

Figure 2.4.1.1: DASYLab flowchart setup for speed control and torque set value

Figure 2.4.1.2: DASYLab illustration of speed control and torque control
2.4.2 Fairchild Pneumatic Controller

In order to control the air pressure applied to the brake, a Fairchild T7950 pneumatic PID controller is used, along with an industrial quality air pressure regulator to help minimize fluctuations in building air supply lines. This controller uses both the torque set value from DASYLab (discussed in Section 2.4.1), and the torque signal to determine how much air pressure is applied to the brake. By implementing this system, Schmitkons [1] was able to decrease the torque fluctuations previously experienced.

2.4.3 Heidenhain Encoders

Two Heidenhain ERA type angle encoders record the angular motion of the input and output shafts using the imaging scanning principle. Two graduations, with equal grating periods are moved relative to each other. The scale is the section of the encoder that is attached to the shaft, and the scanning reticle is stationary. “When parallel light passes through a grating, light and dark surfaces are projected at a certain distance. An index grating with the same grating period is located here. When the two gratings move relative to each other, the incident light is modulated. If the gaps in the gratings are aligned, light passes through. If the lines of one grating coincide with the gaps of the other, no light passes. Photovoltaic cells convert these variations in light intensity into electrical signals [15].” This sinusoidal signal is then digitized to create a square wave, which is later interpreted by the ROTEC transmission error system to calculate the error in relative motion between the two shafts or transmission error. Figure 2.4.1.1 shows the photoelectric scanning technique used by the current optical encoders located on the New Loaded Static Transmission Error Test Stand.
2.4.4 ROTEC System

The data acquisition and analysis system used for the new loaded static transmission error test stand is the ROTEC Rotary Analysis System (RAS). It is a hardware/software packaged tailored towards geartrain analysis. Unlike the traditional transmission error analysis techniques, such as divide/divide or multiple/divide for analog signals [14], ROTEC uses a time-stamp technique. By using the internal quartz oscillator, it stamps the incoming signals to eliminate typical phasing issues. A detailed description of how these type of encoders work can be found in Chapter to of Schmitkons [1].
2.5 **Summary**

So Chapter 2 outlined all of the donated, existing and additional components needed in order to control the test stand and measure data necessary for transmission error measurements. Figure 2.5.1.1 is a flowchart illustrating the data flow for speed and torque control as well as the measurement of transmission error. With the whole test stand assembled and ready for transmission error measurements, the next thing step is to complete actual results and compare them to analytical predictions. Chapter 3 discusses a series of static transmission error test and Chapter 4 compares those results to analytical models.

![Flowchart for speed and torque control as well as transmission error measurement](image)

Figure 2.5.1.1: Flowchart for speed and torque control as well as transmission error measurement
CHAPTER 3

STATIC TRANSMISSION ERROR TEST RESULTS

3.1 Introduction

This chapter includes test results from the new loaded static transmission error test stand for spur and helical gears. First, a series of preliminary tests were completed to determine the accuracy and repeatability of the new test setup. These tests include an unloaded comparison between the new test stand and the Gleason/Goulder Single Flank Tester, as well as repeated start tests to determine experimental spread. Next, extremely precise dynamics rig spur gears, previously used by Blankenship and Kahraman [4], were tested at various torque values to determine how transmission error changes with increased torque. Thirdly, loaded transmission error tests were completed for the three Tom Schachinger (TS) designs. Ultimately, these results are compared to the predicted trends from WindowsLDP, RomaxDesigner and Helical3D subsequently in Chapter 4.

3.1.1 Test Specimen

The test specimen for the initial studies with the new loaded static transmission error test stand are the dynamics rig spur gears and the TS helical gear designs. Figure 3.1.1.1 shows the macro-geometry of the two sets of gears. The dynamics rig gears are a unity ratio gear set, with 50 teeth on both the pinion and gear, and TS helical gears have a
ratio of 0.9661, with 59 teeth on the pinion and 57 teeth on the gear. Since the dynamics rig gears have the same macro-geometry, i.e. number of teeth, module, pressure angle, center distance, etc, they only differ due to micro-geometry modifications on the surface of the teeth. The 10V1 gear set is a perfect involute, 9KB1 has 5 μ-m of tip relief on both the pinion and gear starting at 20.9º of roll angle, 9KB2 has 5 μ-m of tip relief beginning closer to the tip at 22.2º of roll angle, and 9KB3 has 5 μ-m of tip relief very close to the tip starting at 23.6º of roll angle. Additionally, all of the dynamics gears have 5 μ-m of circular lead crown. Schachinger [1], in his attempt to separate out transmission error, shuttling and friction forces, designed his gears with different macro-geometry and micro-geometry. Design #1 has a 2.10898 mm module, 17º pressure angle, 33º helix angle, with 4 μm of tip relief starting at 24.144º of roll angle of the pinion and 24.404º of roll angle for the gear, and 3 μ-m and 4 μ-m of circular lead crown on both the pinion and gear, respectively. Design #2 has a 2.24046 mm module, 18º pressure angle, 27º helix angle, and the exact same modifications as Design #1. Finally, Design #4 has a 2.01725 mm module, 15º pressure angle, 35º helix angle, with 4 μ-m of circular profile crown of the pinion and gear, and 3 μ-m and 4 μ-m of circular lead crown on both the pinion and gear, respectively. Ultimately, Table 3.1.1.1 summarizes the difference between the each of the four dynamics rig spur gears, as well as the differences between the three TS helical gears. Figures 3.1.1.2 and 3.1.1.3 illustrate the 3D micro-geometry modifications for the dynamics and TS gear, respectively. (Note: the detailed gear information for all of the gear sets analyzed in this section is included in Appendix C.1).
Figure 3.1.1.1: Dynamics rig spur gears (left) and TS helical gears (right)

Table 3.1.1.1: Gear information for dynamics rig spur gears and TS helical gears

<table>
<thead>
<tr>
<th>Macro-Geometry</th>
<th>Units</th>
<th>10V1</th>
<th>9KB1</th>
<th>9KB2</th>
<th>9KB3</th>
<th>Design #1</th>
<th>Design #2</th>
<th>Design #4</th>
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<tbody>
<tr>
<td>Number of Teeth Pinion, N1</td>
<td></td>
<td>50</td>
<td>50</td>
<td>50</td>
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<td>59</td>
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<tr>
<td>Module</td>
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<td>3.0</td>
<td>3.0</td>
<td>3.0</td>
<td>2.10898</td>
<td>2.24046</td>
<td>2.01725</td>
</tr>
<tr>
<td>Pressure Angle, φ</td>
<td>°</td>
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<td>20</td>
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<td>15</td>
</tr>
<tr>
<td>Helix Angle, ψ</td>
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<td>0</td>
<td>0</td>
<td>33</td>
<td>27</td>
<td>35</td>
</tr>
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<td>150.0</td>
<td>150.0</td>
<td>150.0</td>
<td>146.0</td>
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<td>146.0</td>
</tr>
<tr>
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<td>20.0</td>
<td>20.0</td>
<td>18.0</td>
<td>18.0</td>
<td>18.0</td>
</tr>
</tbody>
</table>

Profile Modifications

| Start of Modification Pinion | °     | N/A | 20.900 | 22.200 | 23.600 | 24.144    | 24.144    | N/A       |
| Linear Modification Pinion  | u-m   | N/A | 5.0 | 5.0 | 5.0 | 4.0 | 4.0 | N/A |
| Start of Modification Gear  | °     | N/A | 20.900 | 22.200 | 23.600 | 24.404 | 24.144 | N/A |
| Linear Modification Gear    | u-m   | N/A | 5.0 | 5.0 | 5.0 | 4.0 | 4.0 | N/A |
| Circular Profile Crown Pinion| u-m    | N/A | N/A | N/A | N/A | N/A | N/A | 4.00 |
| Circular Profile Crown Gear | u-m   | N/A | N/A | N/A | N/A | N/A | N/A | 4.00 |

Lead Modifications

| Lead Crown Pinion           | u-m   | 5.0 | 5.0 | 5.0 | 5.0 | 3.0 | 3.0 | 3.0 |
| Lead Crown Gear             | u-m   | 5.0 | 5.0 | 5.0 | 5.0 | 4.0 | 4.0 | 4.0 |
Figure 3.1.1.2: Total micro-geometry modifications for the dynamics gears
Figure 3.1.1.3: Total micro-geometry modifications for the TS gears
3.2 Preliminary Test Results

3.2.1 Un-Loaded Transmission Error Measurements – Comparison between New Test Stand and the Gleason/Goulder Single Flank Tester

As in most new test stand designs, an initial comparison to existing hardware is extremely important to validate accuracy and repeatability. In the case of the new loaded static transmission error test stand, un-loaded transmission error tests were completed and compared to the Gleason/Goulder single flank tester for both the spur and helical gears previously described in Section 3.1.1. If the results at no load compare within acceptable limits, then it is reasonable to assume that the measurement setup, i.e. the angle encoders and the ROTEC system, is working properly and will produce believable results throughout the test matrix of loaded experiments.

A comparison of the average total transmission error curves and transmission error spectrums for the dynamics gear 10V1, measured by both the Gleason/Goulder and new test stand, is shown in Figures 3.2.1.1 and 3.2.1.2. Note that for the total transmission error figures, the runout peak-to-peak values are much different, 60 µ-m for the Gleason/Goulder and 400 µ-m for the new test stand. This is due to the difference in eccentricity of the arbors from one test stand to the other. The main focus of this study is on the mesh frequency component of transmission error, so this can be disregarded at this time. Since there are also knicks present in the average total transmission error figures for dynamics gear 10V1, the first harmonic of transmission error is not visible above the noise floor. Knowing that the first harmonic of transmission error for a perfect involute gear set should be small this is alright for the time being. Once load is introduced to the gear set in Section 3.3, the harmonic of transmission will peak out above the noise floor.
Also, note that there is a peak at the 253 order in the new test stand transmission error spectrum, which is due to the motor windings and is present throughout all testing.

Figures 3.2.1.3 and 3.2.1.4 show similar comparisons for dynamics gear 9KB1, which should have un-loaded transmission error due to the modifications outline in Section 3.1.1. In Figure 3.2.1.3 there is a definite high frequency component superimposed on the runout, and the first harmonic of transmission error is visible above the noise floor in Figure 3.2.1.4. Since the peak value of the first harmonic of transmission error from the Gleason/Goulder and new test stand are 2.805 µ-m and 2.997 µ-m, respectively, there is only a 10% percent difference from the 0.192 µ-m variation.

Additionally, Figures 3.2.1.5 and 3.2.1.6 show the same comparisons for dynamics gear 9KB2, with a percent difference of 16 % for the first harmonic of transmission error from a 0.255 µm variation. Dynamics gear 9KB3 shows a 25% difference from a 0.260 µm variation illustrated in Figures 3.2.1.7 and 3.2.1.8. Ultimately, Figures 3.2.1.9 through 3.2.14 show similar comparisons for the TS designs. Design #1 has a percent difference of 20% due to a 0.106 µ-m variation, Design #2 is harder to compare due the noise floor, and Design #4 has a percent difference of 20% due to 0.484 µ-m variation.
Figure 3.2.1.1: Gleason/Goulder (top) to New Test Stand (bottom) Comparison Dynamic 10V1 Comparison of Total TE
Figure 3.2.1.2: Gleason/Goulder (top) to New Test Stand (bottom) Comparison Dynamic 10V1 Comparison of TE Spectrum
Figure 3.2.1.3: Gleason/Goulder (top) to New Test Stand (bottom) Comparison
Dynamic 9KB1 Comparison of Total TE
Working Spectrum (Revolutions) Channel (Ch2 - Ch1) (i=50:50) ref. Channel Ch1

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<th>Orders of Ch1</th>
<th>0.03125</th>
<th>0.0625</th>
<th>0.125</th>
<th>0.25</th>
<th>0.5</th>
<th>1</th>
<th>2</th>
<th>4</th>
<th>8</th>
<th>16</th>
<th>32</th>
<th>64</th>
<th>128</th>
<th>256</th>
<th>512</th>
<th>1024</th>
</tr>
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<tbody>
<tr>
<td>Transmission Error, peak [um]</td>
<td>15.84</td>
<td>0.805</td>
<td>0.1184</td>
<td>0.1696</td>
<td>0.1423</td>
<td>0</td>
<td>50</td>
<td>100</td>
<td>150</td>
<td>200</td>
<td>250</td>
<td>300</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Orders Input Shaft

<table>
<thead>
<tr>
<th>Orders Input Shaft</th>
<th>0.03125</th>
<th>0.0625</th>
<th>0.125</th>
<th>0.25</th>
<th>0.5</th>
<th>1</th>
<th>2</th>
<th>4</th>
<th>8</th>
<th>16</th>
<th>32</th>
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<th>128</th>
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<tr>
<td>Transmission Error, peak [um]</td>
<td>197.2</td>
<td>1.997</td>
<td>0.2479</td>
<td>0.2081</td>
<td>0.1696</td>
<td>0.1423</td>
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<td>50</td>
<td>100</td>
<td>150</td>
<td>200</td>
<td>250</td>
<td>300</td>
<td></td>
<td></td>
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</tr>
</tbody>
</table>

Figure 3.2.1.4: Gleason/Goulder (top) to New Test Stand (bottom) Comparison
Dynamic 9KB1 Comparison of TE Spectrum
Figure 3.2.1.5: Gleason/Goulder (top) to New Test Stand (bottom) Comparison Dynamic 9KB2 Comparison of TE Spectrum
Figure 3.2.1.6: Gleason/Goulder (top) to New Test Stand (bottom) Comparison
Dynamic 9KB2 Comparison of TE Spectrum
Figure 3.2.1.7: Gleason/Goulder (top) to New Test Stand (bottom) Comparison
Dynamic 9KB3 Comparison of Total TE
Figure 3.2.1.8: Gleason/Goulder (top) to New Test Stand (bottom) Comparison Dynamic 9KB3 Comparison of TE Spectrum
Figure 3.2.1.9: Gleason/Goulder (top) to New Test Stand (bottom) Comparison
TS Design #1 Module 2.10898 Comparison of Total
Figure 3.2.1.10: Gleason/Goulder (top) to New Test Stand (bottom) Comparison
TS Design #1 Module 2.10898 Comparison of TE Spectrum
Figure 3.2.1.11: Gleason/Goulder (top) to New Test Stand (bottom) Comparison
TS Design #2 Module 2.24046 Comparison of Total TE
Figure 3.2.1.12: Gleason/Goulder (top) to New Test Stand (bottom) Comparison
TS Design #2 Module 2.24046 Comparison of TE Spectrum
Figure 3.2.1.13: Gleason/Goulder (top) to New Test Stand (bottom) Comparison
TS Design #4 Module 2.01725 Comparison of Total TE
Figure 3.2.1.14: Gleason/Goulder (top) to New Test Stand (bottom) Comparison
TS Design #4 Module 2.01725 Comparison of TE Spectrum
3.2.2 Repeatability Tests

When the TS helical gears were tested, it was noticed that the mesh frequency transmission error time traces changed significantly depending on when the test was started during the hunting ratio (Note: There are over 3000 combination of tooth meshes for the 59 to 57 tooth gear set). Having said that, there is a limitation to the OSU-ROTEC analysis system, because it can only store about one-sixth the data necessary to capture the whole hunting ratio (somewhere around 500 tooth meshes). Therefore, in an attempt to determine the repeatability of the test stand two tests were conducted; one which attempted to store exactly the same tooth mesh data by starting the test when tooth 1 of the pinion meshed with tooth 1 of the gear and time-averaging over the subsequent 10 revolutions, and the other attempted to determine the spread of the first harmonic of transmission error when the test was started randomly during the hunting ratio.

Figure 3.2.2.1 shows the total transmission error time traces for three repeated runs of the first test. It is interesting to note that the three time traces are nearly identical in shape throughout all ten revolutions. It is also interesting to note that the mesh frequency component is very apparent in the last four to five revolutions. This shows the sensitivity of the test to the starting tooth pair. If the test was started five revolutions earlier, there might not be any noticeable high frequency content at all. Conversely, if the test was started five revolutions later there might be a significant amount of high frequency content throughout all ten revolutions. Ultimately, this can dramatically change the average, showing much less or much more mesh frequency transmission error than what would be calculated if all of the tooth mesh combinations were recorded. In addition to the total transmission error comparison, the transmission error spectrum of all
Figure 3.2.2.1: Repeatability Study – Same Start Time Total TE for 10 Revolutions (x3)
Figure 3.2.2.2: Repeatability Study – Same Start Time TE Spectrum (x3)
three runs is shown in Figure 3.2.2.2. When compared to each other, the three runs had a percent different of about 10% for the first harmonic of transmission error.

Furthermore, to see how much experimental spread is created when the test is started at random positions throughout the hunting ratio, the second test was run. Figure 3.2.2.3 shows the first harmonic of transmission error versus torque for which the test was started randomly during the hunting ratio five different times. Now the variation between these tests is more than the first test, showing a percent different from the mean of about 20-30%. But having said that, the difference in absolute dimensions is only 0.2-0.3 µ-m (around 10 µ-in), which is quite small.

A more complete repeatability analysis would also include assembling and disassembling the gears in the test rig and restarting the test. This was qualitatively done and found to give repeatable results on the same order of magnitude as the first two tests.

Figure 3.2.2.3: Experimental Spread of the 1st Harmonic of Transmission Error Random Start Time During Hunting Ratio for TS Design #1 (Module 2.10898)
3.3 Primary Test Results

3.3.1 Dynamics Rig Spur Gears Loaded Transmission Error Results

The next step in the test stand validation is to perform loaded static transmission error tests for the dynamics gears to serve as a baseline when studying the less precise TS helical gears. Since the micro-geometry modifications are very close to the design modifications, the transmission error harmonics versus torque should be quite accurate when compared to predicted trends.

Figures 3.3.1.1 through 3.3.1.3 include both the average total transmission error curve and the transmission error spectrum for dynamics gear 10V1 at 10, 50 and 90 N-m. When compared to each other, there is virtually zero change in the runout amplitude, but the transmission error harmonic values change. For the 10 N-m transmission error spectrum it is difficult to determine if the calculated first harmonic value is above the noise floor or not, but once 50 N-m is applied to the gear set the first harmonic of transmission error is quite visible above the noise floor at 1.425 µ-m. A summary for the first, second and third harmonic of transmission error versus torque from zero to 90 N-m is shown in Figure 3.3.1.4 (Note: the first couple data points at zero, 5 and 10 N-m may be inaccurate because there is no way to determine if the peak is above or below the noise floor).

Similar results for 9KB1, 9KB2 and 9KB3 are shown in Figures 3.3.1.5 through 3.3.1.16. Since these gears include modifications, the first harmonic of transmission error is visible at low torque values, so the entire summary of the first, second and third harmonics versus torque can be believable.
Figure 3.3.1.1: Dynamics gear 10 V1
Total TE, TE Spectrum and Tooth mesh – 010 N-m
Transmission Error, Averaged Revolution (n=10) (Output Shaft - Input Shaft) (i=50:50)

Revolutions Input Shaft

Transmission Error [um]

Orders Input Shaft

Transmission Error [um]

Figure 3.3.1.2: Dynamics gear 10 V1
Total TE, TE Spectrum and Tooth mesh – 050 N-m
Figure 3.3.1.3: Dynamics gear 10 V1
Total TE, TE Spectrum and Tooth mesh – 090 N-m
Figure 3.3.1.4: Dynamics gear 10V1
1st, 2nd, and 3rd Harmonic of Transmission Error vs. Torque Summary
Figure 3.3.1.5: Dynamics gear 9KB1
Total TE, TE Spectrum and Tooth mesh – 010 N-m
Figure 3.3.1.6: Dynamics gear 9KB1
Total TE, TE Spectrum and Tooth mesh – 050 N-m
Figure 3.3.1.7: Dynamics gear 9KB1
Total TE, TE Spectrum and Tooth mesh – 090 N-m
Figure 3.3.1.8: Dynamics gear 9KB1
1\textsuperscript{st}, 2\textsuperscript{nd}, and 3\textsuperscript{rd} Harmonic of Transmission Error vs. Torque Summary
Figure 3.3.1.9: Dynamics gear 9KB2
Total TE, TE Spectrum and Tooth mesh – 010 N-m
Figure 3.3.1.10: Dynamics gear 9KB2
Total TE, TE Spectrum and Tooth mesh – 050 N-m
Figure 3.3.1.11: Dynamics gear 9KB2
Total TE, TE Spectrum and Tooth mesh – 090 N-m
Figure 3.3.1.12: Dynamics gear 9KB2

1\textsuperscript{st}, 2\textsuperscript{nd}, and 3\textsuperscript{rd} Harmonic of Transmission Error vs. Torque Summary
Figure 3.3.1.13: Dynamics gear 9KB3
Total TE, TE Spectrum and Tooth mesh – 010 N-m
Figure 3.3.1.14: Dynamics gear 9KB3
Total TE, TE Spectrum and Tooth mesh – 050 N-m
Figure 3.3.1.15: Dynamics gear 9KB3
Total TE, TE Spectrum and Tooth mesh – 090 N-m
Figure 3.3.1.16: Dynamics gear 9KB3
1st, 2nd, and 3rd Harmonic of Transmission Error vs. Torque Summary
3.3.2 TS Helical Gear Loaded Transmission Error Results

Having already analyzed extremely precise gears, the next step in the test matrix is to test less accurate gears, i.e. the TS helical gears supplied to GearLab by the Ford Motor Company, in order to see if any noticeable changes occur to the transmission error with increase in load.

Figures 3.3.2.1 through 3.3.2.3 illustrate similar trends to those in Section 3.3.1, where we see the average total transmission error, along with the FFT of that signal and the average tooth mesh for the Ford Design #1 gears with module 2.10898 mm. Figure 3.3.2.4 summaries the mesh harmonic of transmission error versus torque for completeness.

Figures 3.3.2.5 through 3.3.2.7 illustrate similar trends for the Ford Design #2 gears with module 2.24046, along with the mesh harmonic summary in Figure 3.3.2.8.

Figures 3.3.2.9 through 3.3.2.11 illustrate the same trends for the Ford Design #4 gears with module 2.01725, along with the mesh harmonic summary in Figure 3.3.2.12.
Figure 3.3.2.1: TS design #1 Module 2.10898
Total TE, TE Spectrum and Tooth mesh – 010 N-m
Figure 3.3.2.2: TS design #1 Module 2.10898
Total TE, TE Spectrum and Tooth mesh – 050 N-m
Figure 3.3.2.3: TS design #1 Module 2.10898
Total TE, TE Spectrum and Tooth mesh – 090 N-m
Figure 3.3.2.4: TS design #1 Module 2.10898
1\textsuperscript{st}, 2\textsuperscript{nd}, and 3\textsuperscript{rd} Harmonic of Transmission Error vs. Torque Summary
Figure 3.3.2.5: TS design #2 Module 2.24046
Total TE, TE Spectrum and Tooth mesh – 010 N-m
Transmission Error, Averaged Revolution (n=10) (Output Shaft - Input Shaft) (i=57:59)

Revolutions Input Shaft

Transmission Error [um]

Total Signal

Working Spectrum (Revolutions) (Output Shaft - Input Shaft) (i=57:59) ref. Input Shaft

Orders Input Shaft

Transmission Error [um]

Total Signal

Figure 3.3.2.6: TS design #2 Module 2.24046
Total TE, TE Spectrum and Tooth mesh – 050 N-m
Figure 3.3.2.7: TS design #2 Module 2.24046
Total TE, TE Spectrum and Tooth mesh – 090 N-m
Figure 3.3.2.8: TS design #2 Module 2.24046

1st, 2nd, and 3rd Harmonic of Transmission Error vs. Torque Summary
Figure 3.3.2.9: TS design #4 Module 2.01725
Total TE, TE Spectrum and Tooth mesh – 010 N-m
Figure 3.3.2.10: TS design #4 Module 2.01725
Total TE, TE Spectrum and Tooth mesh – 050 N-m
Figure 3.3.2.11: TS design #4 Module 2.01725
Total TE, TE Spectrum and Tooth mesh – 090 N-m
Figure 3.3.2.12: TS design #4 Module 2.01725
1st, 2nd, and 3rd Harmonic of Transmission Error vs. Torque Summary
3.4 Summary

All of the loaded static transmission error measurement for the four dynamics gear designs and the three TS designs are completed. The average total transmission error curves along with the transmission error spectrum results resemble standard trends. Very few nicks are present once torque is introduced into the system, so the majority of transmission error harmonics are visible above the noise floor in transmission error spectrum. The transmission error harmonics versus torque seem to create a smooth curve for all of the gear sets. It is interesting that the peak values of the first harmonic of transmission error occur at a reasonably high torques for most of the gears (even the pure involute). This peak value ranges from about 1.5 µ-m for the perfect involute to around 3 µ-m for each of the other gears except the TS helical design #4, which has a peak value of under 1 µ-m. However, when looking at the spectrum for the #4 helical pair one sees heavy sidebands around mesh frequency and its harmonics. These sidebands are far less significant for the spectra of all of the other gears. Since the plotted first harmonic of transmission error is taken from the spectra at the mesh order only, if one included sideband energy, one would expect that the values would be closer to 2 or 3 µ-m for the #4 gear pair. Further tests and analysis should be performed on gear pair #4 to ascertain why these sidebands exist on this gear set and not the others. Ultimately, these experimental results need to be compared to analytical predictions, which are included in Chapter 4. Here we will see if transmission error harmonics versus torque following similar trends, and if they do the new loaded transmission error test stand can be used for additional testing in the future.
CHAPTER 4

COMPARISON BETWEEN EXPERIMENTAL RESULTS AND ANALYTICAL MODELS FOR LOADED STATIC TRANSMISSION ERROR

4.1. Introduction

In this chapter the static transmission error results from Chapter 3 are compared to the transmission error results versus torque from analytical predictions. Transmission error trends from WindowsLDP, RomaxDesigner, and Helical3D are compared to the experimental loaded transmission error results from Section 3.3. First, descriptions of the individual software packages are covered to illustrate their differences and/or similarities, including simulated figures of the gears tested. Actual tooth topography is used in the analytical prediction stage for more realistic description of the gear set analyzed. And finally, there is a summary of transmission error predictions versus torque for the individual software packages and the experimental results to evaluate the accuracy of the new loaded static transmission error test stand.

4.2. Description of Analytical Models

4.2.1. WindowsLDP

Developed by The Ohio State University GearLab, WindowsLDP is a contact based gear analysis software package for spur and helical gears. It is used to calculate transmission error, load distribution, root and contract stresses, tooth forces, mesh stiffness, etc. Because it is an analytically based program it can compute information quickly in comparison to finite element software packages. Additionally, it can perform
multi-torque simulations and robustness studies to determine how micro-geometry modifications affect the overall performance of a gear pair. Shafting and bearing information can be included in WindowsLDP by using the ComplexShaft program, to see how gear mesh forces cause shaft deflections and misalignments across a gear facewidth, leading to changes in load distribution, transmission error, root stress, etc.

When the WindowsLDP prediction for the involute spur gear pair was compared with some very precise loaded TE measurements that were made previously for that gear pair, we found that the LDP prediction for the peak to peak TE was about 30% lower than the measurement. Based on this comparison, using the exaggeration factor, a 30% increase in the LDP compliance was used for all subsequent simulations that are presented in this chapter.

4.2.2. RomaxDesigner

Used throughout the gearing community RomaxDesigner is an advanced software tool for conceptual design and sizing for gearing and transmission systems. Within the design modules, shafting, bearings, gears and housing can be included in order to determine the global deflections and misalignments for multiple gear meshes. The software provides modeling, sizing and flexibility analysis so engineers can design and analyze a transmission system quickly. Figure 4.2.2.1 illustrates the RomaxDesigner models used for the analytical comparison to experimental results from Chapter 3.
4.2.3. Helical3D

For a complete finite element analysis, Helical3D developed by ANSol Inc., was included in the analytical comparison. This software package can be used for 3D analysis of internal and external helical gears. Similar to WindowsLDP and RomaxDesigner, Helical3D can include micro-geometry modification into its analysis routines. The user inputs the macro-geometry and the finite element mesh in generated automatically, with additional refinements available. Transmission error can be calculated quite simply with Helical3D, and the program uses iGlass as a post-processing tool to see 3D representations of stress within the gear teeth and blank, as well as the contact on the surface of the teeth. Figure 4.2.3.1 shows the finite element meshes for the spur and helical gears previously tested in Chapter 3.

Figure 4.2.2.1: RomaxDesigner models for the spur and helical gears and An example output of transmission error
4.3 Experimental Results Compared with WindowsLDP, RomaxDesigner and Helical3D

4.3.1 Comparison Figures

Figure 4.3.1.1 through 4.3.1.4 show the comparison for the four dynamics rig spur gear sets. Here we notice that the analytical prediction are extremely close to one another, yet the loaded test results from the new test stand deviate with increases in torque. Unfortunately, the seemingly close correlation at no load is not reflected in the test results as more and more load is introduced to the system. Figure 4.3.1.5 through Figure 4.3.1.7 show a similar trend for the three helical gear sets donated by Ford Motor Company.
Figure 4.3.1.1: Dynamics gear 10V1
Comparison of 1\textsuperscript{st} Harmonic of Transmission Error vs. Torque

Figure 4.3.1.2: Dynamics gear 9KB1
Comparison of 1\textsuperscript{st} Harmonic of Transmission Error vs. Torque
Figure 4.3.1.3: Dynamics gear 9KB2
Comparison of 1\textsuperscript{st} Harmonic of Transmission Error vs. Torque

Figure 4.3.1.4: Dynamics gear 9KB3
Comparison of 1\textsuperscript{st} Harmonic of Transmission Error vs. Torque
Figure 4.3.1.5: TS design #1 Module 2.10898
Comparison of 1st Harmonic of Transmission Error vs. Torque

Figure 4.3.1.6: TS design #2 Module 2.24046
Comparison of 1st Harmonic of Transmission Error vs. Torque
Figure 4.3.1.7: TS design #4 Module 2.01725
Comparison of 1st Harmonic of Transmission Error vs. Torque
4.4 Summary

It is obvious that the measurements at higher torques are always greater than the predictions, no matter which prediction is used. The lone exception is helical gear pair 4, but if sideband energy were added to the plots, the same could be said for this gear pair.

Several things were considered as causes of these differences:

1) Shaft deflections of the overhung gears could cause changes in tooth contact. Simulations showed that the shafts of the spur gears deflect the same amount but in opposite directions so that contact would not shift. This was verified with contact pattern shifts. In the helical gears the shafting of the two pairs is not symmetric, so it is possible that some contact shift could occur. However, contact patterns did not indicate significant contact shift.

2) Modeling errors. Since three models that have been previously been shown to be reliable all predict similar results that differ from the measurements, this certainly points to the measurement being excessively high.

3) Encoder measurement difficulties. The encoders are mounted on the deflecting shafts and as such, there could be some internal deflections of the encoders that cause both data offset and changes in the rotary motion between the measurement parts. Offsets could result in errors called interference errors and the motion irregularities could cause changes in sensitivity called modification errors [16]. It is the author’s conclusion that this is the most likely cause and the best cure would be to mount the encoders between bearings so the internal deflections are minimized. If the donated fixture is to be used in its current configuration, tangential accelerometers used in an angular acceleration configuration should be used for transmission error measurement.
CHAPTER 5
CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

A new loaded transmission error test stand was created by merging an existing motor and brake with a donated gear mounting fixture that had encoders built in. After some redesign, the fixture was assembled and seemed to function well. The speed control for the new STE test stand works well, with the average torque fluctuations being ± 5 N-m from the set value. The repeatability of the measurement seems to be good. By starting the test at the same time, when tooth 1 from the pinion is meshing with tooth 1 of the gear, the first harmonic of transmission error is repeatable to within 10%. The experimental spread when the test is started randomly during the hunting ration of the gear set is somewhere between ± 0.2-0.3 microns.

Gear changeovers are quite simple with no change in center distance. On average it may take 10-15 to change from one gear set to another as long as they have the same macro-geometry. Changes in center distance and/or facewidth are more difficult, but they can be done in a half of a day.

All of the analytical predictions are complete for the four GM and three Ford gear sets tested in this thesis. Since the experimental results versus analytical predictions
deviate at higher torque values, and there are significant deflections happening due to the overhung arrangement, one can conclude something is happening to the encoder measurement of the shaft angular motion. Even just the smallest deflection, or slope change, to the measured medium in the angle encoder can cause distortion to the analog signal according to the technical support from Heidenhain.

5.2 Recommendations

1. Encoder location needs to be changed. Currently overhung orientation does not seem to be working right. Shaft deflections may cause a distortion in the signal created by the optical rotary encoder. Deflection in the measuring standard may lead to inaccurate transmission error measurements.

2. Increase data storage for ROTEC, because currently the memory limitations do not allow the whole hunting ratio to be recorded.

3. Accurate alignment needs to be studied. Commercial tools for pulley alignment might be helpful to aligning gears.

4. The effect of runout should be studied using analytical techniques to see how much the eccentricity of the individual gears affects STE.

5. If the donated fixture is to be used in the future, tangential accelerometers in a rotary acceleration configuration should be used for transmission error measurements.

6. One of the original goals of this thesis was to measure the effects of shuttling and friction excitations. This did not occur in this work so it is recommended that the
current fixtures be mounted in a running rig and rotating tri-axial accelerometers be mounted close to the gears in order to measure transmission error, shaft radial motion, shaft axial motion and shaft rocking. Non contact displacement probes could also be used for some of these measurements.

7. Additional comparisons of the different transmission error prediction software should be made. This should include the FE and shaft deflection modules within LDP.
REFERENCES


APPENDIX A

EQUIPMENT SETUP AND OPERATION INSTRUCTIONS
A.1 Changeover, and Equipment Adjustments

A.1.1 Gear Specimen Changeover

General Motors Spur Gears

1. Loosen lock nut on the gear by using the three-prong wrench.
2. Unscrew/Remove the lock nut from the arbor.
3. Remove the gear from the shaft.

*Note:* Be sure to use a rubber hammer to remove the gear.

4. Repeat for the pinion.

Ford Helical Gears

1. Loosen bolt holding gear cap on the output arbor.
2. Remove the cap and the parking gear from the output arbor.
3. Remove the gear from the shaft.

*Note:* If the gear does not slide off easily, try rotating the output shaft.

4. Repeat for the pinion on the input arbor.

A.1.2 Change of Center distance

1. Unbolt the input coupling.
2. Loosen the floor bolts on the DC torque motor.
3. Loosen the set screw on the input gear pedestal.
4. Crank the ball screw until the appropriate center distance is achieved.

5. Lock the set screw on the input gear pedestal.

6. Move the motor on the bedplate.

7. Align the motor using the laser alignment tools.

8. Tighten the floor bolts on the DC torque motor.

9. Reconnect input coupling and tighten the bolts.

A.1.3 Change in Facewidth

1. Loosen the set screw on the hub closest to the pneumatic brake.

2. Loosen the bolts on the output coupling and slide the section towards the pneumatic brake to create a gap.

3. Loosen the floor bolts on the brake pedestal.

4. Slide brake away from the output gear pedestal.

5. Loosen set screw on the output gear pedestal.

6. Crank the ball screw until the appropriate facewidth is achieved.

7. Tighten the set screw on the output gear pedestal.

8. Slide the brake close to the output gear pedestal, leaving slight gap.

9. Align the output shaft to the shaft connected to the brake using alignment tools.

10. Bolt brake pedestal to the bedplate.
11. Slide output coupling section back into place.

12. Tighten the coupling bolts.

13. Finally, tighten the set screw on the output hub closest to the brake.

A.2 Loaded STE Test Stand for Parallel-Axis Gearing Operation

(Reference from A.W. Schmitkons [8] page 133.)

Motor Amplifier

1. Make sure the fuse box switch is ON.

2. Switch ON “Main Power” switch.


4. Press the GREEN “Run” button.

Torque Sensor Amplifier

1. Press the RED “Power” button to turn the amplifier ON.

2. Make sure the “Excitation” is ON and set to 10 volts.

3. Make sure the “Gain” is set to 1000X.

4. Make sure the “Filter” is set to 100 Hz.

5. Adjust “Trim” knob to zero out the bridge balance.

A.3 DASYLab Program (Reference from pg. 135 of [8].)

1. Start DASYLab on the PC.

2. Open Drive Motor Controller Program “Loaded Static TE Tester in-lbs and Nm.dsb”
3. Press Play on the Function Bar to Start the Program.

4. Adjust the Slider bars to the Desired Speed and Torque values.

5. Press Stop on the Function Bar to Stop the Program

Note: Speed may runaway if adjusted to quickly under low loads.

A.4 ROTEC System (Referenced from pg. 136 of [8].)

1. Turn ON ROTEC Computer.

2. Open Rotec-RAS Program.

3. Select appropriate “Username” and “Password.”

4. Under the “Setup” menu, configure the “Measurement” and “Evaluation.”

5. Once the test stand is running, select the “Measure” menu. Measurement may be initiated automatically or manually depending on the parameters set in the previous step. If the measurement is set to trigger manually, click the “Start” button to begin the measurement.

6. Once the measurement is complete, provide a detailed description on the acquired data file.

7. Analysis of the data is done by selecting the “Evaluate” menu.

8. Previous data files can be selected and analyzed under the “File/Measurements” menu.

For additional guidelines on ROTEC operation, see pg. 136-139 of [1].
APPENDIX B

DRAWINGS OF TEST HARDWARE
Figure B.1:  Input shaft of new test stand

Figure B.2:  Output shaft of new test stand
Figure B3: Callout Drawing of donated components from Ford Motor Company
Figure B.4: Baseplate for loaded STE test stand

Figure B.5: Riser for STE test stand
Figure B.6: Input coupling hub for STE test stand

Figure B.7: Spacer for output shaft of STE test stand
Figure B.8: Arbor section 1 for dynamics gears

Figure B.9: Arbor section 2 for dynamics gears
## C.1 Gear Data Sheets

### C.1.1 Dynamic spur gears

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### C.1.3 TS helical gear – Design #2

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<td>GEAR1 Offset</td>
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<th>Trans : PCD US CURVATURE (mm)</th>
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<td>Theoretical R1ch</td>
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<thead>
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<th>Trans : TOOTH THICKNESS (mm)</th>
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<td>Root</td>
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<thead>
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<th>Normal Tooth Thickness (mm)</th>
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<td>Normal Backlash (mm)</td>
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<td>Percent of Backlash (%)</td>
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<td>Root Clearance (mm)</td>
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### C.1.4 TS helical gear – Design #4

<table>
<thead>
<tr>
<th>Number of Teeth</th>
<th>C1</th>
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<tr>
<td>59</td>
<td>0.986</td>
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#### CENTER DISTANCE (mm)

- **Oper ating**
  - Standard: 142,831
  - Ratio: 1.022

#### CONTACT RATIO

- Profile: 1.63
- Face: 1.629
- Total: 3.269

#### MODULE (mm)

- Normal Theoretical: 2.01725
- Normal Operating: 2.04655
- Transverse Theoretical: 2.46261
- Transverse Operating: 2.51724

#### PRESSURE ANGLE (degree)

- Normal Theoretical: 15°
- Normal Operating: 17.841°
- Transverse Theoretical: 18.113°
- Transverse Operating: 21.584°

#### HELIX ANGLE (degree)

- Theoretical: 35° - 35°
- Operating: 35.593° - 35.593°
- Base: 33.644° - 33.644°

#### R.T.O (mm)

- Base: 7.35312
- Circular: 7.90815
- Act: 11.04888

#### LENGTH OF CONTACT (mm %)

- Approach: 5.967 (49.94%)
- Recess: 6.001 (50.06%)
- Total: 11.968

#### ROLL ANGLES (degree)

- SAP (Starting Active Profile): 17.71° - 17.523°
- Operating Pitch: 22.678° - 22.745°
- Transverse: 18.742° - 18.742°
- Transverse Pitch: 27.658° - 27.658°
- Pitch: 21.556° - 21.556°
- Effective Outside (Tip): 23.812°

#### DIAMETERS (mm)

- Root: 142.141
- Base: 138.093
- SAP (Starting Active Profile): 144.539
- Theoretical Pitch: 145.293
- Operating Pitch: 148.517
- Effective Outside (Tip): 153.341
- Pitch: 147.544
- Effective Outside (Tip): 149.544

#### FACEWIDTH (mm)

- Actual: 18
- Active: 18

#### TRANS.: RND US CURVATURE (mm)

- SAP (Starting Active Profile): 21.34204
- Operating Pitch: 27.32665
- Transverse Pitch: 33.30216
- Effective Outside (Tip): 5.77531

#### TRANS.: TOOTH TH ONESS (mm)

- Root: 5.77531
- SAP (Starting Active Profile): 5.20128
- Theoretical Pitch: 4.96791
- Operating Pitch: 3.91797
- Effective Outside (Tip): 1.88746

#### NORM. TOOTH TH ONESS (mm)

- Root: 4.76462
- SAP (Starting Active Profile): 4.26791
- Theoretical Pitch: 4.06666
- Operating Pitch: 3.18653
- Effective Outside (Tip): 1.51795

#### BACKLASH AT OPERATING (mm)

- Transverse: 0.0682
- Normal: 0.0559
- Percent of Backlash (%): 53.03

#### Root Clearance (mm)

- 0.7755