STEEL GEAR FAULT DIAGNOSTICS USING NON-CONTACT MAGNETIC ROTATIONAL POSITION SENSORS

by

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Abstract

This thesis reports on the investigation of a cost effective method of diagnosing gear faults using non-contact magnetic rotational position sensors. Like all new methods, the proposed sensor needed to be compared directly to an industry standard technique for measuring shaft rotations. The sensor was tested for its ability to record shaft rotational speed under a variety of experimental conditions. The sensor provided optimal measurements when directly compared to an industry standard technique of completing the same task. Using the fully calibrated sensor, experiments were conducted, which involved measuring the presence of faults in a 1:1 gearbox comprised of steel spur gears. Measuring the dynamic transmission error over the course of one full gear rotation allowed for identification of various gear fault types and severity levels. A large dynamic transmission error is the result of the output shaft lagging the input shaft, which ultimately represents the presence of a gear fault. The dynamic transmission error, over the course of one shaft rotation, was also investigated for the presence of a smooth engagement profile. A smooth engagement profile is the result of the sensor accurately detecting each individual tooth entering and exiting the gear mesh. The sensor demonstrated a great deal of promise for the application of a cost effective method for condition monitoring and fault diagnostics in a gearbox containing steel gears and loaded under realistic conditions.
Co-Authorship

This thesis is solely written by the author, David Benjamin Rapos.
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I would like to thank my supervisor, Dr. Chris Mechefske for his direct supervision throughout my thesis. Through his helpful guidance and encouraging leadership I was able to overcome many obstacles that I encountered along the way. It was as a result of his sound knowledge and expertise that ultimately allowed me to reach where I am today.

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Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Unit/Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>lbf</td>
<td>Pound-force</td>
</tr>
<tr>
<td>Nm</td>
<td>Newton meter</td>
</tr>
<tr>
<td>Hz</td>
<td>Hertz</td>
</tr>
<tr>
<td>psi</td>
<td>Pounds per square inch</td>
</tr>
<tr>
<td>RPM</td>
<td>Revolutions per minute</td>
</tr>
<tr>
<td>V</td>
<td>Volts</td>
</tr>
<tr>
<td>Hp</td>
<td>Horsepower</td>
</tr>
<tr>
<td>π</td>
<td>Pi</td>
</tr>
<tr>
<td>σ</td>
<td>Stress</td>
</tr>
<tr>
<td>g</td>
<td>g-force</td>
</tr>
<tr>
<td>s</td>
<td>seconds</td>
</tr>
<tr>
<td>&quot;</td>
<td>Inch</td>
</tr>
<tr>
<td>mm</td>
<td>Millimeter</td>
</tr>
<tr>
<td>1 inch</td>
<td>= 25.4 mm</td>
</tr>
</tbody>
</table>
# List of Abbreviations

The following abbreviations and acronyms are found in this thesis. They are defined here.

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>TE</td>
<td>Transmission Error</td>
</tr>
<tr>
<td>DTE</td>
<td>Dynamic Transmission Error</td>
</tr>
<tr>
<td>DFT</td>
<td>Discrete Fourier Transform</td>
</tr>
<tr>
<td>FFT</td>
<td>Fast Fourier Transform</td>
</tr>
<tr>
<td>VFD</td>
<td>Variable Frequency Drive</td>
</tr>
<tr>
<td>EDM</td>
<td>Electrical Discharge Machining</td>
</tr>
<tr>
<td>AE</td>
<td>Acoustic Emission</td>
</tr>
<tr>
<td>LPF</td>
<td>Low Pass Filter</td>
</tr>
<tr>
<td>FPGA</td>
<td>Field-Programmable Gate Array</td>
</tr>
</tbody>
</table>
Chapter 1

Introduction

1.1 Project Formation

Research into a new method of machine fault detection and diagnostics using a non-contact magnet rotational position sensor was proposed in the research completed in 2010 by Michael Taylor [1]. His thesis examined the application of this non-contact magnet rotational position sensor for detecting faults located in gearboxes. Conclusions from this research showed promising results in regard to accurately detecting gear faults, in comparison to another type of high accuracy shaft rotation measurement device, for a plastic gear set. However, neither a direct comparison between the two sensors used to conduct measurements nor the success of this application for a steel gear set were accomplished. This provided the opportunity for the work that is reported in this thesis.

1.2 Project Background

It has been estimated that approximately half of all operating costs in most industrial facilities are as a result of maintenance [2]. With this in mind, providing appropriate maintenance to a machine, at the correct time, is vital to avoid unnecessary costs. As a result, there are several traditional types of machine fault diagnostics for gearboxes with one of the most common of these being vibration analysis. Gearbox vibration analysis typically requires the use of an accelerometer. Challenges that may be faced using this sensor type include potential mounting difficulty, extremely complex raw vibration signals and interference due to high levels of noise in the measure signal.

The proposed method of fault detection in gearboxes is by means of ‘Dynamic Transmission Error’ (DTE) measurements. Dynamic transmission error is defined as the difference between the
input and output shaft rotational positions as a function of time. With the aid of a non-contact rotational position sensor, an easy to implement, cost effective method of gearbox machine fault diagnostics is introduced and tested. Due to the proposal of a new method for measuring shaft rotational position, a comparison to industry standard techniques of measuring shaft position was examined to allow for a clear justification as to why the adopted technique was explored.

1.3 Scope of the Project
This thesis examines the ability of a non-contact magnet rotational position sensor for the detection and tracking of common steel gear faults. A direct comparison of this sensor to an industry standard method was also completed. Through the comparison of this sensor to another means of measuring shaft rotation, a clear justification as to why the adopted sensor was applied and verification to the accuracy of the data is provided.

1.4 Organization of Thesis
Chapter One outlines a brief introduction to the project. This provides a brief overview of the material without going into too much detail. Chapter Two explores the area of research of machine condition monitoring and fault diagnostics by way of a literature review. Once the previous work conducted in this field of study is stated, Chapter Three moves into the dynamic sensor calibration experiment. The goal here is a direct comparison of the proposed sensor to an industry standard technique. After successful calibration testing occurred, Chapter Four utilizes the proposed method for a steel gear set running with various pre-fault scenarios. Lastly, Chapter Five summarizes the conclusions made from the dynamic sensor calibration and dynamic transmission error test results. The potential areas of improvement for further research are presented to the reader in this chapter as well.
Chapter 2

Literature Review

The proposed method of research utilizes a dynamic transmission error approach to condition monitoring and fault diagnostics. To verify the validity of using the proposed sensor for recording shaft measurements, a comparison was conducted to an industry standard instrument used for measuring shaft rotations at a high degree of accuracy. However, before either of these experiments could be conducted a better understanding to past and present experiments in the field of machine condition monitoring and fault diagnostics, specifically for gearboxes, needed to be explored.

2.1 Machine Condition Monitoring and Fault Diagnostics

The definition of machine condition monitoring and fault diagnostics, according to Mechefske [2], is

“…the field of technical activity in which selected physical parameters, associated with machinery operation, are observed for the purpose of determining machinery integrity.”

Monitoring physical parameters of a select piece of machinery and then using the information gathered to set maintenance priorities is typically the most complex maintenance strategy a company could employ. This strategy is vital for expensive machinery which if run to catastrophic failure could cause significant harm to the surrounding environment as well as the machine itself. An example of a piece of machinery that requires this type of maintenance strategy is a gearbox. A gearbox is often a subpart of a complex piece of machinery. As a result, if failure were to occur to a gear this often leads to destruction of the gearbox as well as damage
to surrounding machinery. Had a successful monitoring model been applied, all of these damage costs could have been reduced to the cost of only the faulted gear.

2.2 Spur Gears

Direct transmission of motion and power from a source to a load is often impossible due to a variety of reasons. These limitations are primarily based around geometry and the need for power/speed increases or decreases. These factors are easily overcome with the aid of a parallel shaft gearbox. The advantages of using a gear set include a high weight to power transmission ratio and a high degree of relative input to output shaft rotational position accuracy. The disadvantages include high levels of noise and vibration and sometimes high purchase cost. The interaction between meshing teeth travels along the pressure line or line of contact. The contact point between any two meshing teeth will travel radially outwards along each tooth until it reaches the tip of the driving gear where it is replaced by the contact of the adjacent approaching tooth against the following driven tooth [3]. However, engagement and disengagement is not perfect from one tooth to another and will result in varying stress and deflections in the teeth.

2.2.1 Transmission Error (TE)

Transmission error is defined by Smith [3] as

“…the difference between the position that the output shaft of a gear drive would be if the gearbox were perfect, without errors or deflections and the actual position of the output shaft. It may be expressed either as an angular displacement from the ‘correct’ position or sometimes more conveniently as a linear displacement along a line of action, i.e., at base circle radius or as a linear displacement at pitch circle radius.”
In an ideal world, under perfect running conditions, there would be zero difference between theoretical and actual shaft position. This however is never the case and a spur gear will generate a TE with a strong regular excitation once per tooth (Figure 2.1).

**Figure 2.1: Typical section of TE for spur gears [3]**

This type of transmission error can be measured using a tangentially mounted accelerometer (Figure 2.2). Accelerometers mounted in this orientation are able to measure torsional accelerations of a shaft. This torsional acceleration can be scaled with respect to gear diameter in order to determine pitch radius movement.

**Figure 2.2: Torsional accelerometer arrangement [3]**
2.2.2 Gear Tooth Bending Stress

During the selection process for gear geometry, consideration should be made towards the stresses the spur gears will encounter during operation. The two main stresses a gear encounters are bending and contact stresses. These stresses can be calculated using stress equations 2.1 and 2.2 for bending and contact stress respectively as defined by the American Gear Manufacturers Association (AGMA). Also, a reference diagram of spur gear terminology is included in Figure 2.3 to help explain some of these terms.

\[
\sigma = W^t K_o K_p K_s \frac{P_d K_m K_B}{F} \quad (U.S.\text{ customary units}) \tag{2.1}
\]

\[
\sigma_c = C_p \sqrt{W^t K_o K_p K_s \frac{K_m C_f}{d_p F I}} \quad (U.S.\text{ customary units}) \tag{2.2}
\]

**Table 2.1: Definition of AGMA bending and contact stress terms**

<table>
<thead>
<tr>
<th>(W^t)</th>
<th>Tangential transmitted load (lbf)</th>
<th>(K_o)</th>
<th>Overload factor (unitless)</th>
<th>(J)</th>
<th>Geometry factor (unitless)</th>
</tr>
</thead>
<tbody>
<tr>
<td>(K_p)</td>
<td>Dynamic Factor (unitless)</td>
<td>(C_p)</td>
<td>Elastic coefficient ((\sqrt{\text{lbf/in}^2}))</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(K_s)</td>
<td>Size factor (unitless)</td>
<td>(C_r)</td>
<td>Surface condition factor (unitless)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(P_d)</td>
<td>Transverse diametral pitch (teeth/in)</td>
<td>(d_p)</td>
<td>Pitch diameter of pinion (inch)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(F)</td>
<td>Face width (inch)</td>
<td>(I)</td>
<td>Geometry factor for pitting resistance (unitless)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(K_m)</td>
<td>Load-distribution factor (unitless)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Figure 2.3: Spur gear terminology [2]

The meaning for each of these variables can be found in textbooks as well as Table 2.1 [4].

Bending stress is the main topic of interest as it is most significantly altered by the fault scenarios in this thesis and in the transmission error measurements conducted by Taylor [5]. Contact stress is included here for completeness but will not be covered in the research described later in this thesis.

Research has been completed on a gear tooth stiffness reduction measurement method using modal analysis [6]. Yesilyurt, Gu, and Ball experimented with stiffness calculations and compared this to tooth wear caused by lack of lubrication in the gearbox. Their results were a linear relationship between surface wear and tooth stiffness. With tooth stiffness reduction leading to increased surface wear, this allows for tooth surface wear to be an explored fault scenario.

2.2.3 Gear Faults and Failure Types

2.2.3.1 Common Gear Faults
Smith outlines the main gear failure categories as pitting, scuffing, root cracking, and wear [7]. These categories are all a result of contact stress except for root cracking which is due to bending stress. Pitting is defined as gear fatigue due to higher than tolerable Hertzian contact stress (stress between curved surfaces). There are two types of pitting. Initial pitting is the removal of high stress spots in order to allow for even loading. If initial pitting does not result in even loading, progressive pitting is the result and is the continuous formation of these high stress spots. Figure 2.4 illustrates a case of progressive pitting where micro pits have formed on the tooth face.

![Figure 2.4: Example of pitting in a steel gear [8]]() 

Scuffing is the result of metal-to-metal contact caused by the breakdown of the oil film layer. Cold scuffing typically occurs at low speeds where insufficient hydrodynamic properties can occur. Hot scuffing results when oil film temperature rises and can no longer keep the gear surfaces apart. Root cracking is the product of small stress raisers in the root, which if allowed to propagate can cause large material losses (for example tooth chunks). If material is removed over the span of the whole tooth face then wear (Figure 2.5) has occurred and is likely the result of oil contamination/impurities.
2.2.3.2 Research Specific Faults

Taylor conducted research on a plastic spur gear set and found promising results for the fault types of root cracks (Figure 2.6) and flat teeth (Figure 2.7) [1]. For purposes of this research, only the root crack test cases will be explored on the steel spur gears. There are two types of root cracks. Tangential root cracks are a fracture across the tooth profile while a radial root crack is defined as a fracture towards the gear centerline.

Figure 2.6: Plastic gear with root cracks [1]
Crack modeling has been conducted in order to determine propagation path [9], [10], [11], [12]. The rim of a gear is the area from the base of the tooth to the keyway. Lewicki and Ballarini used the ratio between the rim thickness and tooth depth/height to express their findings [12]. This ratio is referred to as the ‘back up’ ratio. Their findings proved that for thin-rimmed gears, the propagation path would be towards the centerline and for thick-rimmed gears the path would be across the tooth profile (Figure 2.8).

This result can also be thought of as the root crack preferring to propagate along the path of least resistance and verifies the reasoning for investigating tangential and radial root cracks in the work reported on in this thesis.

### 2.3 Current Fault Diagnostic Techniques in Gearboxes

There are currently three primary methods of diagnosing faults in gearboxes. These three methods are vibrational analysis, acoustic emission (AE) analysis, and oil and wear particle analysis.
2.3.1 Vibrational Analysis

Vibrational analysis is conducted on a gearbox by first collecting a vibration signal using an accelerometer. The most common mounting of an accelerometer is on the outside of the gearbox. This can pose a potential problem since gearboxes can range in size and location within a machine subsystem. If gearbox mounting is available the transmitted vibration signals from all components enclosed inside the gearbox (bearings, gears, etc.), and potentially some sources outside the gearbox, will be found in the measured signal. Often times this can result in extremely complex vibrational signals as well as the presence of high levels of noise in the measured signal. Problems attributed to the gears themselves will typically appear in a properly recorded vibration signal at the gear mesh frequency or by the presence of sidebands at this frequency. This frequency corresponds to the product of the number of teeth and the rotating speed of the gears (RPM). The gear mesh frequency and multiples of this frequency may reveal gear faults.

Dalpiaz, Rivola, and Rubini explore different post processing analysis techniques that can be applied to recorded vibration signals measured from a gear pair affected by a fatigue crack [13]. Here, proven approaches such as cepstrum and time-synchronous averaging analysis are compared to newer approaches that include time-frequency (through wavelet transforms) and cyclostationarity analysis. The comparison explores each of the techniques sensitivity to fault severity and the influence of transducer location determined through signals for two crack depths. It was found that each of these techniques had their own advantages and disadvantages associated with it. For example, cepstrum and cyclostationarity analysis yield information pertaining to spectrum evolution and are therefore required to have more than one vibration measurement. On the other hand, using time synchronous averaging analysis allows for absolute diagnostics of the fault. Furthermore, if wavelet transforms are applied to these signals, the localized damage tooth can be detected. In the end, there is no easy one way to approach signal processing for vibrational analysis, but rather a combination or complex hybrid approach is often employed.
As mentioned earlier, gearbox vibration signals can be extremely difficult to analyze. With this being said, work has been done to look into active vibration control of a gearbox system which experiences vibration excitation due to transmission error [14]. Guan, Lim, and Steve Shepard Jr. proposed that due to the deterministic nature of the discrete narrowband gear tones, an active vibration control system (comprised of a rolling-element bearing to provide forces to the shaft) could be used in order to reduce the vibrational output. Their findings were successful in suppressing housing vibrations at the first and second gear mesh frequency by way of reducing out-of-band overshoot and increasing convergence rate. This control algorithm contributed to a reduction in housing vibrations equivalent to roughly 18 dB at mesh harmonics, however it was not tested and verified at higher speeds or loading scenarios. In the same way, the reduction was seen in lower gear mesh multiples but was not seen in higher GMF multiples.

2.3.2 Acoustic Emission

The technique of measuring acoustic emissions (AE) was originally developed for non-destructive testing of static structures in the early 1960’s. Acoustic emission is the process where transient elastic waves are generated due to a rapid release of strain energy caused by structural alteration [15]. It has only been recently adapted to the field of machine condition monitoring and fault diagnostics.

The primary source of this method in gears and gearboxes is as a result of crack initiation and propagation. Like vibrational analysis, mounting location is critical for AE sensors, because they are required to be as close as possible to the structure of interest to avoid wave dissipation. Work exploring three common gearbox fault diagnostic techniques was completed under a testing rig where natural pitting was allowed to occur [16]. The results found that AE levels showed better sensitivity to pitting rates compared to a vibration analysis but only at high torques. On the
contrary, vibration monitoring had a greater sensitivity to pitting rates at lower torques. Spectrometric oil analysis monitored pit growth better in comparison to a vibration approach at high speeds, however both performed poorly at low speeds. It can be seen from these findings, that by taking an AE approach you may be limited to only higher speed gearboxes.

2.3.3 Oil and Wear Particle Analysis
Oil and wear particle analysis is one of the oldest methods of machine condition monitoring and fault diagnostics. The procedure for conducting this analysis involves looking at the oil of the gearbox for any loss of oil quality or contaminants such as wear particles and foreign substances from outside the gearbox. This method is still commonly used in aircraft and helicopter platforms, however it is typically used in combination with other fault detection methods [16]. It is used in combination with other methods because of its fault prevention limitations (wear has to have already begun) as well as often times the oil is required to be sampled separately and not during machine operation. There is also a time lag between oil sampling and analysis results being available.

2.4 Rotational Position Sensors
Shaft rotational position is a vital part to the gear fault detection method proposed in this thesis. There is a wide variety of common instruments that can measure shaft rotational position. The four main instruments are incremental encoders, interference fringe encoders, absolute position encoders, and Hall Effect sensors. Based on previous research by Taylor, the sensors selected for this work were the incremental encoder and the Hall Effect sensor [1]. The two main differences between these instruments (besides the actual physics of operation) are price and resolution.

2.4.1 Incremental Optical Encoder
An incremental encoder uses photo interrupters to convert rotational motion into electrical impulses. These encoders can be simplified down to three main parts: a stationary disc with ‘n’
number of slits cut in it, a rotating code wheel configured the same way as the stationary disc, and 
a light source. This light source is constantly being directed onto the code wheel. As the shaft 
rotates, the photo detector becomes blocked and unblocked from the light source creating an 
impulse each time the photo detector is unblocked. These impulses are used to determine shaft 
position.

For example, a quadrature incremental encoder with 360 pulses per revolution has a resolution of 
1440 (360*4) pulses per revolution. If this encoder also has three recording channels it has the 
ability to measure absolute shaft position. Two of these channels measure the impulses per 
revolution but are out of phase from each other and the third channel is used to count full shaft 
rotations.

2.4.2 Hall Effect Sensor
Shaft position can be measured in a cost efficient way by using a non-contact magnet rotational 
position sensor. This sensor consists of a magnet (north-south magnetic field axis is transverse to 
the shaft axis) fixed to the end of a shaft used in combination with a Hall Effect sensor. As the 
shaft rotates the magnet will create a changing (rotating) magnet field that can be detected by the 
Hall Effect sensor.

2.4.2.1 Static Sensor Calibration
Taylor conducted a series of experiments on various sensor-to-shaft orientations [1]. His work 
studied the potential use of a solid magnet as well as a ring magnet. The reason for including a 
ring magnet was to provide a viable mounting option if the end of the shaft was inaccessible. For 
each of these magnets he recorded static outputs from the Hall Effect sensor positioned in a 
variety of orientations relative to the shaft. These included axial, transverse and 45° alignment 
with the shaft. His findings proved that for both the conventional magnet (Figure 2.9) as well as 
the ring magnet (Figure 2.10), an axial facing sensor proved to have the best results.
Figure 2.9: Hall Effect sensor (dark grey)/magnet (light grey) configuration for a “puck” shaped magnet [1]

Figure 2.10: Alternative Hall Effect sensor (dark grey)/magnet (light grey) configuration for a “ring” shaped magnet [1]

2.5 Signal Processing Techniques

2.5.1 Data Filtering

Unnecessary high or low frequency noise can be easily removed from a signal using a signal filter. For removing high frequency noise, a low pass filter (LPF) is applied. The tolerance of the filter depends largely on the filter’s order, however the general shape can be seen in Figure 2.11.
Data is characterized as being a part of the pass band (allowed through), transition band (magnitude of response is altered), or rejection band (not allowed through). The main frequency of interest is the cut off frequency. The cut off frequency is chosen as the point at which frequency above or below this selected value is removed or allowed through, respectively. The magnitude response for a Butterworth low pass filter can be seen in Figure 2.12.

The Butterworth filter is often used because of its ability to feature a maximally flat magnitude response over the pass band [17]. This type of filter can also be found preprogrammed into MATLAB software packages.
2.5.2 Shaft Revolution Synchronous Averaging

For repeating data samples (rotating shafts) the removal of any random ambiguities that appear in one revolution but not any other, can be eliminated by averaging one revolution to another. In an ideal world, the results from one revolution point to another will be the exact same if all other variables are held constant. This procedure from a signal processing point of view is limited to integer numbers of samples per revolution. This idea of revolution averaging was adapted from the averaging technique discussed by Smith [7]. An averaging technique is discussed where a once per revolution marker is used to sum successive revolutions in order to eliminate contributions from random vibration levels.

Taylor explored the idea of both a direct approach as well as an averaging approach to his data analysis [5]. For a direct approach the time window is from start to finish. For the averaged approach, a window of one averaged shaft rotation is used. In the end, his findings using an averaged approach were superior to those found using a direct approach, as randomness was eliminated.

2.6 Conclusion

In conclusion, a brief description into the field of machine condition monitoring and fault diagnostics has been presented in order to provide the necessary background and information required for purposes of the research being presented. The presented material highlighted common methods for monitoring and diagnosing faults in gearboxes as well as potential limiting factors that are associated with each method. As a result of these potential limiting factors, the need for an effective method of condition monitoring and fault diagnostics is required for gearbox applications. Also, a look into the measurement and instrumentation used to achieve the proposed method of dynamic transmission error was explored. The main method of measuring shaft rotation is by an incremental optical encoder. Therefore for the proposed Hall Effect sensor used
in this research, the accuracy should be tested in comparison to the industry standard method of measuring shaft position for a dynamic setting.
Chapter 3

Dynamic Sensor Calibration

As described earlier, the proposed method of machine condition and fault monitoring being investigated in this work involves the use of ‘Dynamic Transmission Error’ measurements and analysis. This proposed procedure is achieved by the use of a non-contact magnetic rotational position sensor. These sensors are an alternative option to measuring shaft position with a high-resolution encoder. Although an encoder can accurately track shaft position they are more costly and robust compared to the sensors used here. Since maintenance costs in an average industrial facility can range up to half of all operational costs [2], using a cost efficient sensor would be useful in reducing overall costs. The proposed sensor can also be insulated from high temperatures, shock and vibration and does not require any moving parts for operation.

Previous experimental work has been conducted with different static sensor configurations in order to explore which orientation yielded the optimum static output. However, these sensors have not been fully calibrated in a dynamic setting or in a direct comparison to a high-resolution encoder. The goal of these experiments is to quantify exactly how accurately the proposed method matches other industry standard sensors for measuring and tracking shaft position in dynamic settings.

3.1 Experimental Background

Machinery is often run at a variety of speeds and operating conditions that are specific to the desired task at hand. These operating conditions were subdivided into four common operation conditions that were experimentally simulated in order to determine the accuracy of the proposed sensor. These four operating conditions were: speed dependency, lateral sensor position, magnet...
size, and the effect of environmental factors. A piece of machinery, like a gearbox, can face any or all of these conditions during any given run cycle.

3.1.1 Speed Dependency
Machinery is often operated within a large range of speeds depending on the application. With this in mind, the proposed sensor needs to accurately track shaft rotational speed for each of the required speeds. In order to ensure this universal application, ranges of speeds were sampled and explored. These ranges consisted both of steady state speeds as well as transient running conditions, since both conditions are appropriate for an industrial piece of machinery.

3.1.2 Sensor Position
The sensor configuration consists of a magnet mounted on the end of the shaft. The Hall Effect sensor is then placed co-linearly aligned with the magnet. The sensitivity to the distance between the magnet and sensor is the focus of this part of the study. In order to measure and compare this to the high-resolution encoder, various distances were explored in order to represent different mounting profiles that may be faced due to machinery mounting limitations.

3.1.3 Magnet Size
The complimentary part required for the proposed shaft tracking procedure is the magnet fixed to the end of the shaft. The magnet is a necessary part as it creates the magnet field measured by the Hall Effect sensor. If a stronger field is created (by using a larger magnet), the response of the sensor needs to be understood. In order to explore this, three puck shaped magnets over various sizes were selected in order to compare the results between each other and to the encoder.
3.1.4 Environmental Factors

Machinery often operates in non-ideal running conditions. These non-ideal operating conditions are defined by environmental factors that sensing equipment may experience. The sensor’s ability to measure through these environmental factors is measured by applying different obstructions or contaminants to the sensor head. These obstructions include: metal filings on the magnet, grease, and various different paper films. During the regular running of a machine there is often dirt and debris as well as grease in the surrounding area. This is especially important for gearboxes where the teeth wear during their lifetime. This wear is noticed by the presence of metal filings in the oil bath. Both oil and these filings could obstruct the proposed sensor and alter the readings recorded. Although the paper film does not directly replicate a real working problem, it does simulate how well the sensor works if an obstruction somehow gets in between the magnet and sensor head gap.

3.2 Experimental Apparatus

The experiments were conducted on a SpectraQuest machine fault simulator bench (Figure 3.1 and schematic in Figure 3.2). The machine consisted of an AC motor controlled by an open loop Variable Frequency Drive (VFD) controlling the speed. The motor drove a shaft, mounted on two rolling element bearings, through a flexible coupling. Data was collected using two National Instruments acquisition cards (PCI4472B a 24-bit card with a ±10V range and PXI6221 a 16-bit card with a ±10V range). The encoder used was the HS25 Incremental Optical Encoder by BEI Sensors. This is a three-channel quadrature encoder (dual channel with index) with an output of 360 impulses per revolution. All experimental results were collected using LabVIEW software and were conducted at Laurentian University in Sudbury, Ontario, Canada.
Figure 3.1: Experimental apparatus for dynamic sensor calibration

Figure 3.2: Schematic of experimental apparatus for dynamic sensor calibration

The non-contact magnet rotational position sensor uses the combination of a magnet (attached to the end of a rotating shaft) and a sensor (co-linearly mounted near the magnet, but stationary) that can measure changing magnetic fields (Figure 3.3). As the fixed end of a shaft rotates it creates a rotating magnetic field. This rotating magnetic field is measured using a Hall Effect sensor. The mounting orientation is shown in Figure 3.4. The sensor housing mount had a magnet to sensor distance of 0.035 inches.
Figure 3.3: Hall Effect sensor housing mount

Figure 3.4: Hall Effect sensor mounting CAD drawing
3.3 Experimental Procedure and Data Collection

Table 3.1 shows the testing matrix used to explore the various mounting, environmental and running conditions used. For each test case all other parameters were held constant. The nominal running conditions were: 20 Hz and 30 Hz electric motor open loop input control, mounting space distance of 0.035 inches between the sensor and magnet, clean sensor head, and a 3/8 inches diameter magnet. For all test cases the data was sampled at 85 kHz. For each trial, other than the transient speed tests, the experiment ran for approximately 30 seconds.

Table 3.1: Dynamic calibration experimental test matrix

<table>
<thead>
<tr>
<th>Test Case</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
</tr>
</thead>
<tbody>
<tr>
<td>Speed Dependence</td>
<td>300 RPM or 5 Hz</td>
<td>600 RPM or 10 Hz</td>
<td>1200 RPM or 20 Hz</td>
<td>1800 RPM or 30 Hz</td>
<td>2400 RPM or 40 Hz</td>
<td>3000 RPM or 50 Hz</td>
<td>3600 RPM or 60 Hz</td>
</tr>
<tr>
<td>Transient Speed</td>
<td>Accel to 20 Hz, hold, Deccel to zero</td>
<td>Accel to 30 Hz, hold, Deccel to zero</td>
<td>Manual change: 0-15-30-15-30-0 Hz</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Sensor Positioning</td>
<td>Nominal/Original distance</td>
<td>2 Spacers or 0.008&quot; 20 and 30 Hz</td>
<td></td>
<td>4 Spacers or 0.016&quot; 20 and 30 Hz</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Magnet Diameter</td>
<td>1/4&quot; (Small) 20 and 30 Hz</td>
<td>3/8&quot; (Nominal) 20 and 30 Hz</td>
<td>1/2&quot; (Large) 20 and 30 Hz</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Environmental Factors</td>
<td>Electrical tape 20 and 30 Hz</td>
<td>Masking tape 20 and 30 Hz</td>
<td>Metal Filings 20 and 30 Hz</td>
<td>Grease 20 and 30 Hz</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

3.3.1 Speed Dependency

3.3.1.1 Steady State

Ranges of running speeds were selected using the open loop controller as a reference. The open loop controller provided a set motor frequency. These speeds, as depicted in Table 3.1, are 5, 10, 20, 30, 40, 50, and 60 Hz. For each case, the machine was allowed to reach steady state before sampling commenced.
3.3.1.2 Transient Speed
Two main tests were conducted in order to explore sensor outputs during transient speed conditions. The first of these was a ramp profile. For a ramp profile, testing commenced and then the machine accelerated to a set speed, was held constant, and then decelerated back to zero. The second of these tests involved manual control of the open loop controller, where the user varied speed as follows: 0 Hz – 15 Hz – 30 Hz – 15 Hz – 30 Hz – 0 Hz.

3.3.2 Sensor Position
Sensor position limitations were simulated using washers as spacers. These spacers where 0.004 inches in thickness and were placed between the sensor housing unit and the shaft mount (the right hand face in Figure 3.4). The distances of 0.008 inches and 0.016 inches (two and four spacers) were added to and compared to the no spacer (0.035 inch) data set. The reason why these spacers were used this way was to maintain the original distance of 0.035 inches as a baseline and add to it from there. This distance was previously determined and implemented by the work done on static sensor configuration by Taylor [1].

3.3.3 Magnet Size
Half of the Hall Effect sensor collection apparatus is made up of the oppositely charged bar magnet. This magnet is epoxied to the end of the shaft. Three magnet sizes were tested in order to explore the output deviations. These three magnet sizes were ¼ inch, 3/8 inch, and ½ inch (Figure 3.5).
3.3.4 Environmental Factors
Simulated non-ideal sensor conditions caused by everyday machinery problems were simulated using four test conditions. The first two involved obstructing the sensor head. Using masking tape and electrical tape on the sensor head, obstruction data was obtained. The third test simulated grease from surrounding machinery (or the oil bath for a gearbox). Machinery grease was applied to the sensor head for data collection. Lastly, to simulate contamination from machine wear, metal filings were placed on the magnet during data collection.

3.4 Data Analysis
Data was collected from two sensors that translated to three measured signals. These three signals were the encoder, Hall Effect sine wave, and Hall Effect Cosine wave. For purposes of analysis the two Hall Effect signals will be referred to as Hall 1 and Hall 2. These signals where then passed through the process listed in Figure 3.6. The MATLAB script can be found in Appendix B.
3.4.1 Encoder and Hall Effect Sensor Rotation Speed Counters

The rotation-to-rotation speed was calculated for both the Hall Effect sensor and encoder using the same basic approach. This approach used a built-in MATLAB function to find peaks within a signal. The function works by logging the location of each peak. Then for the case of the encoder the time interval of 360 edges was recorded. This represents one rotation since there are 360 impulses per revolution. For the Hall Effect sensor, since there is only one peak per rotation the time from peak to peak was recorded.

3.4.2 Data Filter

High frequency noise was removed from the Hall Effect sensor using a low-pass Butterworth filter set to 20 Hz above the rotational frequency. This 20 Hz window was determined through trial and error on the data set. Elimination of high frequency noise allowed for the use of the MATLAB peaks function mentioned above.

3.4.3 Normalization

This step takes the Hall Effect sensor signals and transforms them into a signal with amplitude range between 1 and -1. This step is not necessarily needed for the data analysis procedure but was included for completeness and for more easily finding peak values. Since the signal is
between 1 and -1, when multiplied by -1 the peaks become troughs and vice versa. This allowed for the analysis of the troughs if desired.

### 3.4.4 Transient Test Case

For the transient test case the raw data needed to be trimmed to the start of the motor since there was a time before and after the machine ramp commenced where the collection software was logging data but the shaft was not rotating. This only needed to be applied to the Hall Effect sensor because the encoder output is 0 or 0.25 V (true or false). For the Hall Effect signal there was minor noise when the machine was in a stationary position and this affected the peaks function used.

### 3.4.5 Statistical Analysis

The resulting rotation-to-rotation speed was compared as a whole as well as the mean speed. The standard deviation of each data set was calculated and compared illustrating the sensitivity of the sensor. These were the three main variables used to compare the encoder to the Hall Effect sensor.

### 3.5 Results and Discussion

For each of the test scenarios outlined in Table 3.1, there were three comparisons made. The first of these was the rotation-to-rotation speed comparison. For this comparison overall trends were examined between the encoder and Hall Effect sensor signals. Using the mean and standard deviation, each of the test cases were compared on average to one another. The mean speed was also compared to the open loop controller and was the first measurement considered in order to determine that both the sensors were working correctly. For each of the test cases a differences chart was created. These charts outline the magnitude difference between the encoder and Hall Effect sensor. The purpose of this table was to eliminate any test run specific error seen by the deviation from encoder to encoder sample sets sampled at the same operating settings.
It can be noted here that even though all tests were run at 20 and 30 Hz for the non-speed dependent test criteria, only the 20 Hz graphs are included in this section and the remaining graphs can be found in Appendix A. The graphs are included in Appendix A because the same general trends mentioned in the 20 Hz cases can be seen at the rotational speed of 30 Hz.

3.5.1 Speed Dependency

3.5.1.1 Steady State

Rotational frequency as a function of rotation number was graphed for the encoder and two Hall Effect signals (Figure 3.7).

![Graph of Speed Dependency 20 Hz](image)

**Figure 3.7: Speed dependency rotation-to-rotation frequency for an open loop control of 20 Hz**

For the 20 Hz test case listed above as well as all other cases (located in Appendix A), the Hall Effect sensor signals measured a slightly lower rotational frequency in comparison to the encoder. However, other than this slight offset, the overall shapes follow the same general trend. This illustrates that slight changes in speed can be accurately accounted for by way of either sensor. The difference/offset can be quantified by looking at the mean rotational frequency as well as the standard deviation depicted in Table 3.2 as well as Figure 3.8 and Figure 3.9.
Figure 3.8: Speed dependency mean rotational frequency

Figure 3.9: Speed dependency standard deviation of mean rotational frequency

Table 3.2: Speed dependency mean and standard deviation

<table>
<thead>
<tr>
<th>Test Case</th>
<th>Encoder Mean (Hz)</th>
<th>Encoder Standard Deviation (Hz)</th>
<th>Hall 1 Mean (Hz)</th>
<th>Hall 1 Standard Deviation (Hz)</th>
<th>Hall 2 Mean (Hz)</th>
<th>Hall 2 Standard Deviation (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>5 Hz</td>
<td>4.97</td>
<td>0.0056</td>
<td>4.95</td>
<td>0.0092</td>
<td>4.95</td>
<td>0.0054</td>
</tr>
<tr>
<td>10 Hz</td>
<td>9.88</td>
<td>0.0065</td>
<td>9.85</td>
<td>0.0087</td>
<td>9.85</td>
<td>0.0068</td>
</tr>
<tr>
<td>20 Hz</td>
<td>19.81</td>
<td>0.013</td>
<td>19.76</td>
<td>0.013</td>
<td>19.76</td>
<td>0.012</td>
</tr>
<tr>
<td>30 Hz</td>
<td>29.75</td>
<td>0.015</td>
<td>29.67</td>
<td>0.016</td>
<td>29.67</td>
<td>0.015</td>
</tr>
<tr>
<td>40 Hz</td>
<td>39.77</td>
<td>0.014</td>
<td>39.66</td>
<td>0.019</td>
<td>39.66</td>
<td>0.017</td>
</tr>
<tr>
<td>50 Hz</td>
<td>49.80</td>
<td>0.018</td>
<td>49.66</td>
<td>0.024</td>
<td>49.66</td>
<td>0.023</td>
</tr>
<tr>
<td>60 Hz</td>
<td>59.66</td>
<td>0.021</td>
<td>59.50</td>
<td>0.025</td>
<td>59.50</td>
<td>0.026</td>
</tr>
</tbody>
</table>
Examining the results in Figure 3.8 and Table 3.2, there is close agreement between the measurement from the open loop controller and the measured output from the encoder and Hall Effect sensor. The slight difference that appears is as a result of the accuracy of the open loop controller that was used as the reference source. The controller had a course set point knob and had accuracy up to only one decimal place. The deviation from rotation-to-rotation frequency appearing in each sensor in Figure 3.7 can be described using standard deviation. Standard deviation, for the purposes of this calibration work, is used to describe the sensitivity of the sensor and ultimately as a comparison from sensor to sensor (or test case to test case).

One notable phenomenon that occurs in all sensors is the increasing standard deviation value (Figure 3.9). Since, standard deviation is representing the sensitivity of the sensor this value was explored further. The difference between the mean as well as the standard deviation of the encoder and both the Hall Effect sensor signals can be found in Table 3.3.

**Table 3.3: Speed dependency difference of encoder and Hall Effect mean and standard deviation**

<table>
<thead>
<tr>
<th>Test Case</th>
<th>Magnitude of Encoder - Hall 1</th>
<th>Magnitude of Encoder - Hall 2</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Mean (Hz) Standard Deviation (Hz) % Speed</td>
<td>Mean (Hz) Standard Deviation (Hz) % Speed</td>
</tr>
<tr>
<td>5 Hz</td>
<td>0.014 0.0036 0.28</td>
<td>0.014 0.00018 0.28</td>
</tr>
<tr>
<td>10 Hz</td>
<td>0.027 0.0022 0.28</td>
<td>0.027 0.00026 0.28</td>
</tr>
<tr>
<td>20 Hz</td>
<td>0.055 3.4E-05 0.28</td>
<td>0.055 0.00017 0.28</td>
</tr>
<tr>
<td>30 Hz</td>
<td>0.082 0.0012 0.28</td>
<td>0.082 0.00063 0.278</td>
</tr>
<tr>
<td>40 Hz</td>
<td>0.11 0.0048 0.28</td>
<td>0.11 0.0030 0.28</td>
</tr>
<tr>
<td>50 Hz</td>
<td>0.14 0.0068 0.28</td>
<td>0.14 0.0055 0.28</td>
</tr>
<tr>
<td>60 Hz</td>
<td>0.17 0.0046 0.28</td>
<td>0.17 0.0052 0.28</td>
</tr>
</tbody>
</table>

Examining the results from Table 3.3 the mean Hall Effect measured signal is always slightly less than the encoder signal. This was also seen in Figure 3.10 (a zoomed in portion of Figure 3.8) and allows us to quantify the offset from the values seen in Table 3.3. The mean rotational speed difference does increase with rotational frequency; however this variance was consistent and
small as a percentage of the overall measured rotational speed (approximately 0.28% of operating frequency). The differences from the encoder and Hall Effect sensor measured output rotational frequency can be examined in Figure 3.11. The same increasing trend is clearly evident but as mentioned before it is a consistently small fraction of the operating speed. Both of the Hall Effect difference measurements (illustrated in Figure 3.11) are similar in value and when graphed these lines overlap one another. Also, the variance from the linear relationship of the data deviates at the lowest rotational speed. As a result, it is not recommended for measuring at rotational frequencies this slow.

![Speed Dependency 20 Hz](image)

*Figure 3.10: Speed dependency for 20 Hz rotational speed*
In conclusion, as the speed of the shaft increases so too does the standard deviation recorded by both the encoder and Hall Effect sensor. Also, as the rotational frequency increases so too does the mean difference between the signals recorded by the encoder and Hall Effect sensor. This mean difference is consistent, relative to shaft speed. The relative difference had an approximate value of 0.28% of the rotating frequency. Therefore, even though the Hall Effect sensor has a deviation from the encoder this difference is small relative to the overall speed but still should be taken into consideration because it is a fraction of the operating speed (i.e. it is not a consistent numerical value at all operating speeds). The sensitivity also shows this increasing trend and demonstrates that these sensors are affected more by changes of speed even though these values are still small relative to the operating speed.

3.5.1.2 Transient

Transient motor speed is often an important aspect of everyday machine operation. Analyzing an acceleration, constant speed, and deceleration profile was used to test the Hall Effect sensor’s ability to determine operating speed compared to an encoder. Figure 3.12 illustrates that both sensors can accurately track this profile.

Figure 3.11: Speed dependency mean difference between the encoder and Hall Effect sensor

![Mean Difference Encoder and Hall Effect Sensor](image_url)
Figure 3.12: Transient speed rotation-to-rotation frequency for a full open loop control test profile (20 Hz maximum speed)

It should be noted that a similar offset is present in the transient test case as the steady state speed dependency test cases (discussed in the previous section). Using a similar difference comparison between the mean rotational speeds, Figure 3.13 was constructed.

Figure 3.13: Transient speed mean difference comparison for the 20 Hz ramp case

As the shaft ramps up to the operating speed of 20 Hz there appears to be a larger difference between the encoder and the Hall Effect signals (left hand side of Figure 3.13). The same
difference occurs during the deceleration phase (right hand side of Figure 3.13); however this value is now positive. The conclusion that can be drawn here is that the Hall Effect sensor is reading a speed slightly larger than the encoder during shaft acceleration and a speed slightly lower during deceleration. This value appears to peak at a value of 0.4 Hz (2% of the rotational speed at 20 Hz) during the deceleration phase but is constant over the 30 Hz test case as depicted in Appendix A. Therefore, the Hall Effect sensor can accurately measure the rotational speed of a shaft in comparison to an encoder during transient conditions to an accuracy of 0.4 Hz.

3.5.2 Sensor Position
The sensor casing had an original magnet to sensor distance of 0.035 inches and was the baseline distance that was added to for all other measurements. Qualitatively examining the rotation-to-rotation frequency in Figure 3.14 and Figure 3.15, there appear to be minimal differences. The Hall Effect rotation-to-rotation frequency in Figure 3.15 for 4 spacers appears to have a larger range of frequencies. The mean rotational speed and standard deviation can be illustrated in a bar graph having error bars representing the sensitivity/standard deviation (Figure 3.16).

Figure 3.14: Sensor position of 2 spacers (0.008 inches) rotation-to-rotation frequency for an open loop control of 20 Hz
Figure 3.15: Sensor position of 4 spacers (0.016 inches) rotation-to-rotation frequency for an open loop control of 20 Hz

Figure 3.16: Sensor position mean speed with standard deviation error bars
Table 3.4: Sensor position mean and standard deviation

<table>
<thead>
<tr>
<th>Test Case</th>
<th>Encoder Mean (Hz)</th>
<th>Encoder Standard Deviation (Hz)</th>
<th>Hall 1 Mean (Hz)</th>
<th>Hall 1 Standard Deviation (Hz)</th>
<th>Hall 2 Mean (Hz)</th>
<th>Hall 2 Standard Deviation (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>No Spacer 20 Hz</td>
<td>19.81</td>
<td>0.013</td>
<td>19.76</td>
<td>0.013</td>
<td>19.76</td>
<td>0.012</td>
</tr>
<tr>
<td>No Spacer 30 Hz</td>
<td>29.75</td>
<td>0.015</td>
<td>29.67</td>
<td>0.016</td>
<td>29.67</td>
<td>0.015</td>
</tr>
<tr>
<td>2 Spacer (0.008&quot;) 20 Hz</td>
<td>19.81</td>
<td>0.014</td>
<td>19.75</td>
<td>0.015</td>
<td>19.75</td>
<td>0.015</td>
</tr>
<tr>
<td>2 Spacer (0.008&quot;) 30 Hz</td>
<td>29.75</td>
<td>0.014</td>
<td>29.67</td>
<td>0.024</td>
<td>29.67</td>
<td>0.021</td>
</tr>
<tr>
<td>4 Spacer (0.016&quot;) 20 Hz</td>
<td>19.77</td>
<td>0.012</td>
<td>19.72</td>
<td>0.018</td>
<td>19.72</td>
<td>0.018</td>
</tr>
<tr>
<td>4 Spacer (0.016&quot;) 30 Hz</td>
<td>29.67</td>
<td>0.016</td>
<td>29.59</td>
<td>0.036</td>
<td>29.59</td>
<td>0.035</td>
</tr>
</tbody>
</table>

Examining the summarized results in Table 3.4, as the set distance between the sensor and magnet increases so too does the standard deviation. This increased sensor position does not appear to have a significant effect on the mean rotational frequency calculated using the Hall Effect sensor. The differences between the encoder and Hall Effect sensor can be seen in a difference chart Table 3.5.
Table 3.5: Sensor Position difference between encoder and Hall Effect sensor mean and standard deviation

<table>
<thead>
<tr>
<th>Test Case</th>
<th>Magnitude of Encoder - Hall 1</th>
<th>Magnitude of Encoder - Hall 2</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Mean (Hz)</td>
<td>Standard Deviation (Hz)</td>
</tr>
<tr>
<td>No Spacer</td>
<td>0.055</td>
<td>3.4E-05</td>
</tr>
<tr>
<td>20 Hz</td>
<td></td>
<td></td>
</tr>
<tr>
<td>No Spacer</td>
<td>0.082</td>
<td>0.0012</td>
</tr>
<tr>
<td>30 Hz</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2 Spacers</td>
<td>0.055</td>
<td>0.00051</td>
</tr>
<tr>
<td>20 Hz</td>
<td></td>
<td></td>
</tr>
<tr>
<td>30 Hz</td>
<td>0.083</td>
<td>0.0098</td>
</tr>
<tr>
<td>4 Spacers</td>
<td>0.055</td>
<td>0.0064</td>
</tr>
<tr>
<td>20 Hz</td>
<td></td>
<td></td>
</tr>
<tr>
<td>30 Hz</td>
<td>0.082</td>
<td>0.020</td>
</tr>
</tbody>
</table>

The same percent rotation frequency measured in the speed dependency test case re-appears. This proves that changing the sensor position has little effect on the mean rotational speed measured by the Hall Effect sensor. On the other hand, the sensitivity or standard deviation of the sample does change as spacers are added. As the spacing distance between the sensor and magnet head is increased so too does the magnitude of the standard deviation. This proves that the sensor is measuring at an optimal distance of 0.035 inches. For any distance greater than this, as proven by the 0.043 inches (two spacer test case), there is a significant change in sensitivity and samples measured at this distance should factor this into their data analysis and/or avoid measuring at this distance.

3.5.3 Magnet Size

The medium sized or 3/8 inch diameter magnet was used for all other test cases as this was the size used by Taylor [1]. There were also the options of ¼ inch and ½ inch diameter magnets. The
rotation-to-rotation frequency for these two magnet sizes can be found in Figure 3.17 and Figure 3.18.

**Figure 3.17:** 0.25 inches diameter magnet rotation-to-rotation frequency for an open loop control of 20 Hz

**Figure 3.18:** 0.5 inches diameter magnet rotation-to-rotation frequency for an open loop control of 20 Hz
The general shape of the Hall Effect signals matches that of the encoder. For further analysis the signal was broken down into its mean and standard deviation (Figure 3.19 and Table 3.6).

![Magnet Size 20 Hz](image)

**Figure 3.19: Magnet size mean speed with standard deviation error bars**

<table>
<thead>
<tr>
<th>Test Case</th>
<th>Mean (Hz)</th>
<th>Standard Deviation (Hz)</th>
<th>Mean (Hz)</th>
<th>Standard Deviation (Hz)</th>
<th>Mean (Hz)</th>
<th>Standard Deviation (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1/4&quot; 20 Hz</td>
<td>19.72</td>
<td>0.014</td>
<td>19.6</td>
<td>0.014</td>
<td>19.6</td>
<td>0.014</td>
</tr>
<tr>
<td>1/4&quot; 30 Hz</td>
<td>29.72</td>
<td>0.012</td>
<td>29.63</td>
<td>0.012</td>
<td>29.63</td>
<td>0.012</td>
</tr>
<tr>
<td>3/8&quot; 20 Hz</td>
<td>19.81</td>
<td>0.013</td>
<td>19.76</td>
<td>0.013</td>
<td>19.76</td>
<td>0.012</td>
</tr>
<tr>
<td>3/8&quot; 30 Hz</td>
<td>29.75</td>
<td>0.015</td>
<td>29.67</td>
<td>0.016</td>
<td>29.67</td>
<td>0.015</td>
</tr>
<tr>
<td>1/2&quot; 20 Hz</td>
<td>19.76</td>
<td>0.013</td>
<td>19.71</td>
<td>0.013</td>
<td>19.71</td>
<td>0.012</td>
</tr>
<tr>
<td>1/2&quot; 30 Hz</td>
<td>29.74</td>
<td>0.013</td>
<td>29.65</td>
<td>0.012</td>
<td>29.65</td>
<td>0.012</td>
</tr>
</tbody>
</table>

From Figure 3.19 and Table 3.6 it can be seen that the best results occur using the 3/8 inch diameter magnet. These results are the closest to the open loop controller setpoint. The standard deviation is largest for the ¼ inch magnet and smallest for the 3/8 inch magnet. To further examine these results a difference table was constructed for the purpose of comparing how close the Hall Effect measurements are to the encoder measurements.
Table 3.7: Magnet size difference for encoder and Hall Effect mean and standard deviation

<table>
<thead>
<tr>
<th>Test Case</th>
<th>Magnitude of Encoder - Hall 1</th>
<th>Magnitude of Encoder - Hall 2</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Mean (Hz)</td>
<td>Standard Deviation (Hz)</td>
</tr>
<tr>
<td>1/4&quot; 20 Hz</td>
<td>0.055</td>
<td>0.00033</td>
</tr>
<tr>
<td>1/4&quot; 30 Hz</td>
<td>0.083</td>
<td>0.00040</td>
</tr>
<tr>
<td>3/8&quot; 20 Hz</td>
<td>0.055</td>
<td>3.4E-05</td>
</tr>
<tr>
<td>3/8&quot; 30 Hz</td>
<td>0.082</td>
<td>0.0012</td>
</tr>
<tr>
<td>1/2&quot; 20 Hz</td>
<td>0.055</td>
<td>0.00044</td>
</tr>
<tr>
<td>1/2&quot; 30 Hz</td>
<td>0.083</td>
<td>0.00088</td>
</tr>
</tbody>
</table>

Again the same percent rotational frequency difference from the encoder to Hall Effect sensor is present. This has a value of approximately 0.28% of the rotational speed. The mean rotational frequency difference with increasing magnet size stays the same. This small deviation illustrates and describes the variability in the shaft output from test case to test case which is also depicted by the varying encoder results. With this in mind the standard deviation difference is smallest when the 3/8 inch magnet is used for collection. However, these values are all small (fourth decimal place in deviation from the encoder) and prove that magnet size does not have a significant impact on the Hall Effect sensor output. If a magnet were to be recommended, the 3/8 inch magnet would be the best option due to its peak mean performance and small deviation shown in Table 3.6. This peak performance is suspected to be a result of an optimized magnetic field size for the sensor to magnet measuring distance.

3.5.4 Environmental Factors

Sensors used to measure signals from gearboxes often times face harsh operating conditions. These environmental conditions can have an impact on the sensors overall measuring ability. This sub-section is meant to explore these potential factors. The first two simulated conditions include physical obstructions to the sensor using something placed on the sensor head. This physical obstruction is created using electrical or masking tape. Figure 3.20 and Figure 3.21 show the rotation-to-rotation frequency over the measurement period. The Hall Effect sensor output for the
electrical tape condition follows the trends of the encoder output more accurately in comparison to the masking tape condition.

Figure 3.20: Electrical tape rotation-to-rotation frequency for an open loop control of 20 Hz

Figure 3.21: Masking tape rotation-to-rotation frequency for an open loop control of 20 Hz

The two other test cases, which explore factors that affect a gearbox, are shown in Figure 3.22 and Figure 3.23. These tests explored the effect of the oil bath on the sensor. The oil bath can often times contain worn or chipped off pieces of steel from the gears. All gearboxes contain a
magnet for purposes of attracting these loose pieces and preventing them from being incorporated into the gear mesh. This is not always possible and results in the question: does the lubricant from the oil bath and/or these metal filings affect the Hall Effect sensing ability? By inspection the rotation-to-rotation frequency from both sensors follows the same curve trends. There is however a large outlier present in Figure 3.23. This phenomena is not present in the metal filings data set recorded at 30 Hz (Appendix A) and was decided to be a statistical outlier.

Figure 3.22: Grease rotation-to-rotation frequency for an open loop control of 20 Hz
As shown in Figure 3.24, the statistical mean and standard deviation were found to better quantify the differences between these sensors. These values are also summarized numerically in Table 3.8.

Figure 3.24: Environmental factors mean speed with standard deviation error bars
The first comment/concern (as was evident before) is the deviation from the encoder data. Since the conditions are applied to the Hall Effect sensor only, this deviation should not be present. As mentioned before, this deviation is as a result of these tests being independent test cases where some disassembly and assembly was done on the rig between test cases. Before a differences chart is created (Table 3.9) some general comments on the trends should be addressed. Examining Figure 3.24 the order from largest to smallest frequency output is electrical tape, metal filings, masking tape, and then grease. This shows that these environmental factors do have an effect on the Hall Effect sensor in that order. However, to find out to what extent numerically they have on the sensor, Table 3.9 should be examined.
Table 3.9: Environmental factors difference of encoder and Hall Effect mean and standard deviation

<table>
<thead>
<tr>
<th>Test Case</th>
<th>Magnitude of Encoder - Hall 1</th>
<th>Magnitude of Encoder - Hall 2</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Mean (Hz)</td>
<td>Standard Deviation (Hz)</td>
</tr>
<tr>
<td>Electrical Tape 20Hz</td>
<td>0.055</td>
<td>-0.0034</td>
</tr>
<tr>
<td>Electrical Tape 30Hz</td>
<td>0.083</td>
<td>-0.0025</td>
</tr>
<tr>
<td>Masking Tape 20Hz</td>
<td>0.055</td>
<td>-0.0029</td>
</tr>
<tr>
<td>Masking Tape 30Hz</td>
<td>0.083</td>
<td>-0.0020</td>
</tr>
<tr>
<td>Grease 20Hz</td>
<td>0.055</td>
<td>-0.0034</td>
</tr>
<tr>
<td>Grease 30Hz</td>
<td>0.082</td>
<td>-0.0017</td>
</tr>
<tr>
<td>Metal Filings 20Hz</td>
<td>0.055</td>
<td>-0.0032</td>
</tr>
<tr>
<td>Metal Filings 30Hz</td>
<td>0.082</td>
<td>-0.0021</td>
</tr>
</tbody>
</table>

The mean differences when the encoder frequency is subtracted are all equal in value. This again points to the 0.28% of the operating frequency that has been reoccurring in each of the test cases.

To better analyze the sensitivity of the Hall Effect sensor by way of the standard deviation, these differences were graphed in Figure 3.25.
The largest standard deviation occurs for the metal filings test case while the lowest standard deviation occurs for the masking tape test case. The standard deviations calculated from both Hall Effect signals for the masking tape test case are roughly equal in value. As a result of this, the masking tape can be ruled out as having no significant effect on both the sensitivity and the overall mean rotational frequency measurement ability of the Hall Effect sensor. The metal filings large standard deviation was at first suspected to be as a result of the large spike in the rotation-to-rotation frequency that was noticed in Figure 3.23. However, if the results for the 30 Hz analysis are examined (Appendix A), the same large values are present.

The general trend that appears to be created can be verified with the results from the 30 Hz test samples located in Appendix A. The Hall Effect sensor is more sensitive to grease and metal filings and is less sensitive to physical obstruction to the sensing head. It should be noted at this time that these differences are small and of the order of a thousandth of a hertz. In the end, if various environmental factors impact the Hall Effect sensor, these result in minimal limitations to its ability to measure rotational frequency in comparison to an encoder. However, if ranking were to be applied to the test cases studied here it would result in: masking tape, electrical tape, grease, and then metal filings in order of smallest impact on sensor sensitivity.

3.6 Conclusion
A variety of tests were conducted in order to determine the measurement ability of a Hall Effect sensor compared to an encoder for a dynamic setting. These tests were conducted on a SpectraQuest test rig that consisted of an electric motor that rotated a shaft to a rotational frequency determined by an open loop controller. It was this open loop controller that was the first stage check stage in order to determine everything was recording properly. The rig was introduced to a variety of test conditions that are common to a gearbox and introducing different
environmental factors to the system. These included: steady state speeds, transient speeds, changing sensor position, and changing magnet sizes. The environmental factors were applied only to the Hall Effect sensor and consisted of obstructions to the sensor itself.

After repeating the findings for each of these test cases, some interesting trends were observed. The sensor is dependent on speed for measuring mean rotational frequency in comparison to the encoder. The Hall Effect sensors mean rotational frequency always varied the encoder by 0.28% of the operating frequency. Also, as rotational speed increased so did the sensitivity of the sensor as indicated by the standard deviation of the rotation-to-rotation frequency output. For the transient test case, the Hall Effect sensor was found to measure a larger rotational frequency during shaft acceleration and a smaller rotational frequency during shaft deceleration in comparison to the encoder. This difference peaked at 0.4 Hz and was designated as the error for transient test measurements conducted using the Hall Effect sensor.

Sensor mounting position can often be difficult due to where a gearbox is mounted and the overall machine orientation and/or configuration. This problem was explored by varying the distance between the sensor head and the magnet mounted on the end of the shaft. It was found that the sensor sensitivity was the best for the original position (0.035 inches). However, for the results from the two-spacer test case (0.043 inches) and the four-spacer (0.051 inches) test case, there was an increase in sensitivity relative to the original orientations.

The magnet held constant for all tests had a diameter of 3/8 inch. There were two other options of puck shaped magnets that could be used in combination with the Hall Effect sensor. These had diameters of ¼ inch and ½ inch. Here the results for sensitivity were all close to each other in value but if a magnet were to be suggested, the 3/8 inch magnet would be recommended. This is
due to it having the lowest measured standard deviation and also because it was the size of the magnet used by Taylor [1].

Gearboxes are often a complex sub-system and can be faced with harsh operational conditions. The last set of test cases was designed to explore these environmental factors. The environmental factors did not have a large effect on the overall measurements of mean and standard deviation recorded by the Hall Effect sensor. If a ranking were to be applied to the environmental factors the order from least sensitive to most sensitive would be: masking tape, electrical tape, grease, and metal filings. However, the value that these sensors deviate, with the relative encoder deviation scaled off, is approximately 0.003 Hz (roughly 0.015% of the rotational speed). This value is small in comparison to the operating rotational frequency and verifies the minimal affect these factors have on the Hall Effect sensor.

In conclusion, each of these individual test cases did not have a substantial impact on the measurement capability of the Hall Effect sensor in comparison to an encoder. Even though the analysis was conducted for full rotational frequency and the encoder has the ability to measure 1/360 of a degree, it is not deemed necessary to look at smaller fractions of a rotation. For purposes of the work to follow on steel gear fault analysis, the relative difference between two Hall Effect sensors is investigated and the degree of rotation is calculated in this way.
Chapter 4

Steel Gear Fault Diagnostics

The proposed method of gear fault diagnostics is based on using ‘Dynamic Transmission Error.’ To reiterate, dynamic transmission error is defined as the difference between the input and output shaft rotational position as a function of time. As proven via dynamic sensor calibration, the use of a Hall Effect sensor is a realistic alternative to an encoder for purposes of measuring shaft rotational speed. The Hall Effect sensor can also accurately track shaft position during a wide variety of running conditions encountered for collection of gearbox data.

The Hall Effect sensor used for this work has now been calibrated in both dynamic and static settings. The static configuration was completed by Taylor [1] where his objective was to optimize the sensor mounting configuration. This work was not tested in a dynamic setting, which initiated the research for dynamic sensor calibration. Through the exploration of different operating conditions, the Hall Effect sensor was compared to an encoder which could measure with an accuracy of 360 impulses per revolution. The overall conclusions from this work demonstrated that the Hall Effect sensor can be used accurately as a cost efficient alternative, in place of an encoder, for measuring shaft rotating speed.

Previous work was conducted on a plastic gear set which yielded promising results for the fault conditions of flat teeth and root cracks [1]. However, when this research was expanded to steel spur gears the results were inconclusive due to the inability to apply suitable torsional loads. It was these inconclusive results that triggered the present research into the application of dynamic transmission error via the use of rotational position calculated using a Hall Effect sensor.
4.1 Experimental Background
When designing a gearbox for any application, there are a number of key parameters that need to be considered in order to account for the running conditions faced. These parameters can range from running speed, lubricant, power transmission, stress factors, and others. All of these factors need to be considered in order to ensure the necessary steps for long trouble free operation life.

4.1.1 Spur Gear Selection
The first step when selecting gears was to make a selection based on the power transmission that is required. This will help to determine the number of teeth and the pitch radii. Next consideration was given to the stresses the gear teeth will face under loading. Simple calculations were made using AGMA equations 2.1 and 2.2 to determine these numbers and to further optimize the gear geometry selection [4]. A worked out sample calculation of gear tooth bending stress can be found in Appendix E.

4.1.2 Crack Simulation
Micro cracks and crack propagation is a common phenomenon in gears. If unnoticed this could lead to major gear damage and in turn catastrophic failure of the gearbox. The two common propagation paths a gear root crack will take are through the tooth hub or through the tooth base [12]. These two propagation paths, for purposes of this research, are referred to as tangential (across the tooth base) and radial (towards the shaft center) root cracks. Machine shop simulated micro cracks were created using wire electrical discharge machining (EDM). Using this type of machining, a crack having a width approximately equal to the diameter of the wire used can be created.

4.2 Experimental Apparatus and Testing Equipment
The experimental apparatus was located at Laurentian University in Sudbury, Ontario. The gearbox was mounted on a bench between two Baldor motors. These motors had power
capabilities of 25 and 50 horsepower. The 25 HP motor was used as the drive motor and the 50 HP was used as the load. The entire mounting apparatus is depicted in Figure 4.1.

![Image of experimental testing apparatus](image)

**Figure 4.1: Experimental testing apparatus**

Inside the gearbox, the healthy gear was mounted on the input shaft and the gear with the simulated fault was mounted on the output shaft. The four sensors measured for data analysis were two Hall Effect sensors (for the input and output shaft rotation), an accelerometer, and an optical sensor. The mounting orientation of all of these components can be seen in the block diagram below (Figure 4.2).
4.2.1 Spur Gears

The gears being explored were Martin steel spur gears having 32 teeth, 4 inch pitch diameter, with 1.25 inch in face width. Several gears were purchased in order to allow for each gear to have only one specific test fault and severity level. These spur gears had a rated torque value of 22 HP at 1200 RPM. These spur gears were wetted and run in 10W30 oil before each experiment commenced.

4.2.1.1 Faulted Spur Gears

The simulated faults were created using wire EDM having a diameter of 0.006 inches. For each of the gears, the tooth of interest was located in the same spot relative to the keyway. For the tangential root crack (across the tooth base), the severity cases were broken into three levels/depths. These three levels corresponded to 1/3 tooth width (approximately 1/16 inch), 2/3 tooth width (approximately 1/8 inch), and tooth completely missing (Figure 4.3). It should be noted that for each of the tests, the machine was run in the appropriate direction so that the forces were “opening” these cracks.
Figure 4.3: Fault depth sizes for the tangential crack condition (small to large)

The radial root cracks (towards shaft center) were created using the same wire thickness. One cut was made on either side of the tooth of interest and were at three severity levels as well. These three severity levels corresponded to depths of approximately 1/3 inch, 2/3 inch, and 1 inch (Figure 4.4).

Figure 4.4: Fault depth sizes for the radial crack condition (small to large)

4.2.2 Sensors and Instrumentation

4.2.2.1 Hall Effect Sensor

The Hall Effect sensor, in combination with a magnet, was the proposed sensor for measuring and determining the presence of gear faults. The sensor and its mounting bracket are depicted in Figure 4.5. The sensor to magnet lateral distance is 0.035 inches. Each of these sensors measured a sine and cosine signal.
4.2.2.2 Optical Sensor

The optical sensor shown below (Figure 4.6) is a light sensitive sensor. It was used as a once per revolution trigger to illustrate when the faulted tooth was in the gear mesh. This allowed for the creation of a flag indicator for later use in data analysis. A piece of reflective tape was placed on the input coupling at the point when the faulted tooth was in the gear mesh. The sensor read true when the piece of reflective tape became visible to the sensor.

Figure 4.6: Optical sensor mounting orientation
4.2.2.3 Accelerometer

A single axis accelerometer was mounted to the outside of the gearbox in the orientation illustrated in Figure 4.7. The purpose of this accelerometer was to collect vibration signals, generated by the gearbox, for the purpose of determining a rough starting point for filtering the data set. Similarly, the signal recorded by the accelerometer was briefly explored for the purposes of a more ‘traditional’ method of fault diagnostics.

![Accelerometer mounting orientation](image)

**Figure 4.7: Accelerometer mounting orientation**

4.2.2.4 FPGA Board

All sensor measurements were collected by way of an FPGA board. FPGA stands for field-programmable gate array. The Hall Effect sensors, the optical sensor, accelerometer, the output motor torque meter and input motor tachometer were wired to this board for data collection purposes.

4.3 Experimental Procedure and Data Collection

The faulted gears were all tested using the speed ramp profile depicted in Figure 4.8. All the faulted gears were also tested using the same healthy driver gear. Prior to each test, each of the gears was given an initial wear-in period at low speed and low loading. This wear-in period was between 10-15 minutes at 300 RPM and a load of 3 or 4 Nm. The purpose of this was to provide each gear an initial wear-in in order to smooth out the gear teeth and avoid any manufacturing
surface scales. The gears were also inspected prior to being mounted to the shaft, and a file was used to eliminate any surfaces burs visible by eye.

![Figure 4.8: Gear test speed profile](image)

Both the drive and load motor had built-in controllers which were wired to the LabVIEW collection program. These motors interpreted load and speed setpoints and outputted the corresponding RPM and load on their front panel display. The setpoint for the drive motor was tuned in order to achieve the three desired speeds shown in Figure 4.8. However, the loads were not tuned to even loads and were instead given a load setpoint of 1000, 2000 and 3000. These setpoints were equivalent to output loads of approximately 9.8, 21.8, and 33.4 Nm, respectively. Data was collected using the FPGA setup mentioned above at a sampling rate of 10 kHz. For each of the gear fault scenarios, there were five test runs to ensure a large data sample set.

The drive motor operating direction was set to clockwise or counter-clockwise based on the gear mounting orientation. This operating direction did not matter for the radial test case. However, for the tangential root crack, rotational direction mattered and was always in the direction of crack growth. Figure 4.9(A) illustrates an example of what direction would be applied if the gear was mounted on the shaft in this configuration. Here the drive motor would be given a clockwise direction in order to open the gear crack. Had the drive motor been given the opposite direction
this would not have been realistic for actual crack propagation. In order for the crack to have developed in such a manner the rotation needed to be clockwise. With a clockwise rotation a driving force to the left (Figure 4.9(B)) is created thus causing the tooth to try to force the crack open and propagate the crack in the manner depicted.

Figure 4.9: A) Gear interaction for a healthy gear driving a gear with a tangential root crack B) Transmitted contact ($F_c$) and normal ($F_N$) forces on a gear tooth

4.4 Data Analysis

As previously mentioned, the seven measured signals are input/output cosine/sine signals from the Hall Effect sensor, optical sensor, load torque meter, and drive tachometer. The load and drive tachometers were used to create the profile in Figure 4.8. The other five signals were carried through the process depicted in Figure 4.10. The MATLAB script can be found in Appendix D.

Figure 4.10: Data analysis flow chart
4.4.1 Cut Signal

The first step in the signal processing method was to cut the signal to the window of interest, otherwise known as segmentation of the signal. In this step the whole data sample was split into the segment/speed of interest. This process used the four Hall Effect signals, the fault indicator signal, and the acceleration data. The resultant signal for the Hall Effect data was now in the form of an ‘N by 4’ matrix and the other two signals were trimmed to have length N (where N is the number of data points in the cut signal of interest).

4.4.2 Discrete Fourier Transform

Vibrational analysis manipulates the accelerometer measurements. These accelerometer measurements were converted into the frequency domain. One of the methods used to complete this is by a Discrete Fourier Transform (DFT). Equations 2.2, 4.2, and 4.3 were used by MATLAB, by way of a Fast Fourier Transform algorithm, in order to achieve this conversion.

\[
X(k) = \sum_{j=1}^{N} x(j)\omega_N^{(j-1)(k-1)}
\]

4.1

\[
x(j) = \left(\frac{1}{N}\right) \sum_{k=1}^{N} X(k)\omega_N^{-(j-1)(k-1)}
\]

4.2

\[
\omega_N = e^{(-2\pi i)/N}
\]

4.3

The sample set is no longer represented by the acceleration amplitudes but is now represented by the frequencies which make up these vibrations and their amplitudes. These frequencies were examined for the characteristic frequencies that may indicate the presence of faults in gearboxes.

4.4.3 Shaft Synchronous Averaging

Due to the repeating phenomena of shaft gear rotations each point at every full rotation should be roughly equal in value. However this is not always the case and sometimes there is random error
associated with recorded measurements. Averaging each shaft rotation can eliminate this random error [5]. The resulting signals now represent an averaged shaft rotation.

4.4.4 Moving Average
Curve smoothing can be achieved in a number of ways. One of which is employing a moving average. A moving average is a simple formula that takes some number of points before and after the point of interest and averages them. This technique also helps to eliminate consistent per rotation random error. For purposes of this research, three points before and after the point of interest was chosen.

4.4.5 Normalization
The resulting Hall Effect signals are a measurement of the voltage output and were transformed to minimum and maximum amplitudes of -1 to 1. This normalization step was taken in order for the data to be passed into the arctangent function. Since the function has two inputs the input variables needed to illustrate a positive and negative change about a point. Thus the normalization process was completed.

4.4.6 The atan2 Combine Function
The input/output cosine and sine signals were combined into one input and output signal. This can be achieved by way of the atan2 function found in MATLAB software packages. This function combined two signals into one with the resulting signal having a domain of (-π, π]. Figure 4.11 illustrates how these two signals were used in achieving corresponding angles.

Figure 4.11: How the arctangent function works
This function utilizes the y and x Cartesian coordinates, determined by the sine and cosine measurements, and returns an angle in the appropriate coordinate. The need for both negative and positive measurements of the cosine and sine signal is highlighted once again.

### 4.4.7 Shaft Position Subtraction

Now that the input and output shafts are represented as an angle between \((-\pi, \pi]\) the difference between the input and output shaft can be calculated. This difference represented the radian difference of the input shaft compared to the output shaft. Since we are dealing with small angles, the radian difference multiplied by the pitch radii will give the dynamic transmission error over the course of one shaft revolution.

### 4.4.8 Data Filtering

Upon first inspection there was a large once per revolution attenuation in the dynamic transmission error results. It was concluded that this attenuation was as a result of mechanical looseness since the gearbox was assembled and reassembled frequently. A high pass filter at an odd multiple of the rotation frequency can be employed to remove the contribution from this. After the data was passed through this filter there still appeared to be high frequency noise present. This noise was attributed to the gears and their mesh higher order frequencies. It was determined that a low pass filter at twice the gear mesh frequency should be utilized to eliminate the high frequency noise [2].

### 4.4.9 Fault Window

The fault window was defined as the portion of time where the fault indicator was triggered. This represented the point per rotation that the faulted tooth was inside the gear mesh. This window was represented on the dynamic transmission error results as two vertical bars. The optical sensor was used as a verification method and is not critical to the method of fault diagnosis proposed. All dynamic transmission error graphs including the ‘fault window’ are included in Appendix C.
It should be noted, both the cut function and revolution synchronous averaging were applied to the optical sensor data.

### 4.5 Results and Analysis

For each of the tested faulty gears, the end goal was to find a peak dynamic transmission error which coincided with the created fault window and thereby showed the faulty tooth location. This value was calculated in millimeters and represents the difference between the output shaft and the input shaft as a result of the gear fault present. A correctly identified gear fault will be represented by a spike in the amplitude of the dynamic transmission error at the location when the faulted tooth is engaged in the gear mesh. The engagement and disengagement of a tooth is represented by the increasing and decreasing trend of the sinewave-like graph. However, since there are multiple teeth engaged at once, these peaks illustrate the addition of a tooth into the gear mesh thus there is always at least one tooth in contact.

The analysis depicted in Figure 4.10 is broken down step by step in the sub-sections to follow. Here the detailed steps are broken down and explained to illustrate how the dynamic transmission error was determined. This illustrates the validity of each data analysis step through the aid of the tangential root crack data at 600 RPM and the highest loading scenario (used through sections 4.5.2 to 4.5.8) as an example.

#### 4.5.1 Introduction to Vibrational Analysis

Vibrational analysis is the ‘traditional’ method for diagnosing faults for gearboxes. As previously mentioned, this can often be challenging due to sensor mounting difficulty, extremely complex raw vibration signals, and interference due to high levels of noise in the measured signal. Numerically this is typically translated to increased variability from test sample to test sample in peak amplitude. Peak amplitudes are critical for the purpose of condition monitoring because consistent amplitude measurements are required in order for changes to be tracked over time.
Figure 4.12 and Figure 4.13 illustrate the Discrete Fourier Transform (DFT) frequency spectra (obtained through the use of a Fast Fourier Transform Algorithm) for two separate test runs on the same gear fault at the same loading and speed. This specific test case was for the large tangential root crack at 600 RPM and the highest loading condition.

Figure 4.12: DFT for large tangential root crack at 600 RPM and largest loading test 1

Figure 4.13: DFT for large tangential root crack at 600 RPM and largest loading test 2
The peak amplitude frequencies from Figure 4.12 and Figure 4.13 are 0.293 g and 0.195 g respectively. They differ in value by 0.098 g which is approximately 40% amplitude difference for the exact same test case. This variance verifies the difficulty of collecting vibration signals from a gearbox for purposes of fault diagnostics and especially for condition monitoring.

These plots can however be used in determining the filtering coefficients that should be applied. Like the filtering that was applied in the dynamic sensor calibration chapter, the exact filtering parameter was determined through trial and error. Revisiting either Figure 4.12 or Figure 4.13, the peak value highlighted is equal to the gear mesh frequency. Using this value as a reference, by way of trial and error the window which included up to two times the gear mesh frequency was selected. This new DFT can be seen in Figure 4.14 and Figure 4.15. This shorter window was used in order to eliminate high frequency noise that affected the dynamic transmission error measurements.

**Figure 4.14:** DFT for filtered large tangential root crack at 600 RPM and largest loading test 1
Figure 4.15: DFT for filtered large tangential root crack at 600 RPM and largest loading test 2

In summary, this variability highlighted the difficulty in using vibrational analysis as a form of condition monitoring. Further vibrational methods can be applied such as cepstrum analysis, cyclostationary analysis, time synchronous averaging, and/or applying time-frequency analysis through wavelet transforms [13]. However, even these advanced signal processing techniques have their limitations.

4.5.2 Cut Signal

The recorded signals were measured over the course of the speed profile depicted in Figure 4.8. This profile was then broken down to the desired time period of study. Timing constants were created using both the drive tachometer and load torque meter. The timing for both the lowest and intermediate loads were equivalent. It was the timing constant for the largest load which affected the 300 RPM section since full speed was achieved before full load thus trimming this period of steady state shorter in comparison to the previous two loads. Figure 4.16 and Figure 4.17 show the full signal for one of the input signals measured from the Hall Effect sensor as well as the trimmed portion used for analysis of the 600 RPM segment.
Dynamic transmission error was defined as the difference between the input and output shaft over time. This time period was selected as one shaft rotation. Each coinciding data point which represented the same rotational position was averaged and recorded to create Figure 4.18. This
step was coined the ‘rotation synchronous averaging’ and represented the Hall Effect output for one averaged rotation.

![Rotation Synchronous Averaged - One Rotation](image)

**Figure 4.18:** Rotational synchronous averaged Hall Effect input cosine signal representing one shaft rotation

### 4.5.4 Moving Average

The data was then passed through the moving average function. This function created an averaged value three data points before and after the point of interest. Any statistical out layers that may have been present were eliminated by way of this function. The new averaged plot is depicted in Figure 4.19. Graphically the same pattern evident in Figure 4.18 is witnessed again.
4.5.5 Normalization

Before the input and output Hall Effect signals were combined, they were normalized between 1 and -1. This was achieved by first shifting the overall signal average value so it was approximately zero and then scaling the maximum and minimum values to 1 and -1, respectively. Figure 4.20 shows the newly normalized signal. This step was required in preparation for the combination of the input and output signals using the atan2 function.
4.5.6 Atan2 Combination

By way of the atan2 function found in MATLAB software packages, the cosine and sine signals for the input and output shaft were combined. As previously mentioned, this returned a graph on the domain of \((-\pi, \pi]\). Figure 4.21 illustrates the combined signal for the input shaft.

Figure 4.20: Normalized Hall Effect input cosine signal

Figure 4.21: Combined Hall Effect input signal
4.5.7 Shaft Subtraction
The relationship between the output shaft and input shaft was determined in order to calculate the dynamic transmission error. It was at this step that each of the signals was multiplied by the pitch radii. By multiplying the radian angle measurement by the pitch radius (when dealing with small angles) the arc length was determined. The difference of these two measurements was given the name ‘dynamic transmission error’. The raw dynamic transmission error is presented in Figure 4.22. The graph appeared to have a low frequency background component and resulted in the need for the data to be filtered.

![Dynamic Transmission Error Plot for One Rotation](image)

**Figure 4.22: Dynamic transmission error for one shaft rotation**

4.5.8 Data Filter
As a result of the low frequency component noticed in Figure 4.22, the data required additional filtering. A Butterworth filter was used in order to remove both high and low frequency components not related to the tooth mesh frequency. The low frequency was suspected to be the result of mechanical looseness while the high frequency components were related to higher orders of the gear mesh frequency. The low frequency was removed with a high pass filter and a low
pass filter removed the high frequency components. The resulting filtered data set is depicted in Figure 4.23.

![Filtered Dynamic Transmission Error for One Rotation](image)

**Figure 4.23: Filtered dynamic transmission error for on shaft rotation**

4.5.9 Dynamic Transmission Error

The dynamic transmission error was calculated for the two fault categories of tangential and radial root cracks, as well as for the fault-free case. The test faults were run through the experimental speed profile (Figure 4.8) at the three previously described loading conditions (9.8, 21.7, 33.4 Nm) and each of these recorded measurements were passed through the signal processing steps discussed in Figure 4.10. The following are the results of the dynamic transmission error recorded for all severity levels. For purposes of this section, only the 600 RPM dynamic transmission error was examined and the findings for the healthy gear set, 300 RPM, and 900 RPM can be found in Appendix C. The decision to exclude these three result sets was made off consistency of results and to avoid the extreme input torque loading cases. Similarly, only the results for two of the loading conditions are included, however the remainder can be found in Appendix C as well.
The following sub-sections organize the results into three categories. These categories were broken down into successful, moderately successful, and unsuccessful in order to highlight the Hall Effect sensors ability for fault detection. For each test, the dynamic transmission error plots were examined for the appearance of a peak dynamic transmission error value as well as a smooth engagement and disengagement profile. The graphs presented depict the ‘fault window’ by the two dashed red lines. Any case where the window starts at the end of the rotation and continues at the start has arrows depicting this.

4.5.9.1 Tangential Root Crack
The results shown in Figure 4.24, Figure 4.26, and Figure 4.28 were all measured at 600 RPM and the highest loading condition while Figure 4.25, Figure 4.27, and Figure 4.29 show results that were measured at the lowest loading conditions. The results were presented in a way to highlight the proposed method of tracking fault growth. This was achieved by first looking at the worst case scenario, a missing tooth, and working backwards in severity to the smallest faulted scenario.

![Graph](image)

**Figure 4.24:** Large tangential root crack under the largest load at 600 RPM
The presented results in Figure 4.24 clearly represent the engagement and disengagement of the tooth mesh between one and two teeth. This phenomenon is highlighted by the wave characteristic of the graph. There are 31 peaks and 31 valleys which were the expected values since the gears selected had 32 teeth. One tooth is always engaged because of the gear meshing geometry. This results in one tooth being engaged at the start and end of each rotation already.

Peak dynamic transmission error occurred after half of a revolution between 0.0484 and 0.0501 s. This contributed to a peak engagement value of 1.446 mm and a peak disengagement value of -1.6 mm. These values are quite close in magnitude and coincided with the fault window measured from the optical sensor. Part of the reason why there was such a peak negative value was because of the missing tooth. Since the gear was missing a tooth, this caused a dramatic shift in rotational position at engagement seen by the large negative value.

Due to the fact that the most severe tangential root crack was evident for the highest loading scenario, the load was reduced to the lowest loading and the signals examined. Figure 4.25 illustrates the dynamic transmission error for the smallest loading scenario with a missing tooth fault.
Figure 4.25: Large tangential root crack under the smallest load at 600 RPM

The peak dynamic transmission error was measured to be 1.43 mm which is almost identical to the measurement recorded at the highest loading case. This value was similar in magnitude compared to that measured in Figure 4.24. One thing that was noticed was that the largest dynamic transmission error that represented peak disengagement did not occur immediately following the peak error. This was as a result of the loading change on the gears. As a result of lower loading, a dramatic change from no tooth to tooth engagement is better seen when the loading on the gears is larger in magnitude. Due to the clean engagement and disengagement graph as well as the peak dynamic transmission error that agreed with the fault window for both the highest and lowest loading conditions, the proposed method of dynamic transmission error was deemed successful for the purpose of detecting the large tangential root crack (missing tooth) test case.

Figure 4.26 depicts the results for the medium tangential root crack loaded at the largest load and 600 RPM. Upon first inspection it was noted that as clean of an engagement and disengagement indication was not seen in comparison to the large crack. This could be the result of a variety of
reasons which will be highlighted all together at the end of the chapter. Nevertheless, there are roughly 31 engagement peaks just like the number presented in the previous results for the missing tooth test case.

![Graph showing dynamic transmission error over time with peak at 0.0649 s and 0.6127 mm]

**Figure 4.26: Medium tangential root crack under the largest load at 600 RPM**

The peak engagement measurement occurred after 0.0649 s. This peak dynamic transmission error was measured to be 0.6127 mm. The measured fault window was in agreement with this peak error. There was no peak disengagement value measured during this fault window. For the case of a missing tooth there was dramatic loading and unloading that took place as a result of the missing tooth. For this case however, there is a tooth present which absorbed some of the rotational energy transfer. With no tooth present this rotational energy was directly transformed into shaft rotational losses and contributed to spike dynamic transmission error measurement. The magnitude for the measured dynamic transmission error decreased in value by roughly 50% in this case compared to the missing tooth case.

If the load is reduced for the this same crack depth the dynamic transmission error measurements profile depicted in Figure 4.27 can be created.
After decreasing the load, the peak dynamic transmission error was measured at 0.3357 mm. The engagement profile that is created for both the loads at the medium crack depth failed to yield a clean engagement like that seen in the results for the missing tooth. There was also a decrease in measured dynamic transmission error as a result of a decreased loading situation like that seen in the no tooth condition. The presented results were deemed moderately successful because clear engagement and disengagement was not evident. However, the peak dynamic transmission error did occur during the fault window thus successfully identifying a fault.

The small tangential root crack measured at the highest loading scenario and at 600 RPM has a fairly good engagement and disengagement profile (Figure 4.28). The same number of engagement peaks of 31 is repeated once again.
The issue with this test case is that the peak dynamic transmission error has a value of 0.7791 mm but does not appear inside the fault window check. If the load is reduced once again (Figure 4.29), the peak dynamic transmission error is 1.036 mm however it also does not agree with the created fault window.
The immediate conclusion that can be drawn is that the proposed analysis cannot detect such a small crack depth. Although there is a peak transmission error value in the results presented, this is not as a result of the tangential root crack and could be a result of general gear meshing difficulty for this particular test set. Also, for the case of the lower load, there are several peaks with similar magnitude that lead to the speculation that some other factor may be affecting the results. Such a factor could include contaminants from the oil bath. The results here were deemed moderately successful since accurate engagement and disengagement were tracked, however the gear fault was not detected.

In summary, the proposed method of fault detection and diagnosis using a Hall Effect sensor and the defined dynamic transmission error was fairly successful in detecting tangential root cracks. The cases presented for large and small tangential root cracks could correctly track the engagement and disengagement of the teeth in the form of a smooth curve. On the other hand, the large and medium crack cases accurately illustrated a peak dynamic transmission error inside the defined fault window. Due to the fact that the accurate number of engagements/disengagements were detected for all test cases and that it was just the small tangential crack that could not be detected, the proposed method was deemed successful in detecting faults. It was the size of the small tangential root crack that resulted in the inability for a clear peak dynamic transmission error to be found that was directly related to the fault.

4.5.9.2 Radial Root Crack
Similarly to the results presented for the tangential root crack, the results are organized from the most severe to the least. This allowed for consistency and a visual representation of fault progression. The results for the deepest radial root crack under the largest loading condition and at an operational speed of 600 RPM are found in Figure 4.30. The peak dynamic transmission
error occurred during the start of the rotation relative to the zero position. This peak error had a value of 0.3205 mm and agreed with the fault window.

![Graph showing dynamic transmission error](image.png)

**Figure 4.30: Large radial root crack under the largest load at 600 RPM**

A clear consistent engagement and disengagement of the gear teeth was not evident. However, the results for the smallest loading condition, under the same fault level, show that there was a clearer engagement/disengagement profile (Figure 4.31). The peak dynamic transmission error agreed with the fault window and had a value of 0.2796 mm.
This engagement profile was smoother and consistent in comparison to Figure 4.30. The dynamic transmission error measurements are similar in comparison with the smaller load, having a slightly lower error. This lower dynamic transmission error was a result of the lower loading condition. The main issue highlighted was that the dynamic transmission errors are so small in magnitude that the resulting faulted engagement is not as clearly represented. As illustrated in Figure 4.30 and Figure 4.31 there are secondary peaks which are close in magnitude and had it not been for the fault window, determining which peak represented the fault would have been difficult to determine.

As a result of the difficulty of determining the fault for a radial root crack this method was listed as only moderately successful. The peak dynamic transmission error was determined for both the largest and the smallest load, however clear engagement and disengagement was not shown for both tests. Also, there were secondary, slightly lower, peaks evident for the lowest loading scenario which would have been difficult to distinguish without the aid of the fault window. Due to the fact that the criteria of a clear peak and a clean engagement pattern were not met for both
loading tests individually, the effectiveness in determining a large radial crack was deemed only moderately successful.

The fault severity was decreased to the medium severity level and the results for the largest load setting are listed in Figure 4.32. The peak dynamic transmission error for this case was found to be 0.4747 mm.

![Rad Crack Medium - Load Level 3 @ 600 RPM](image)

**Figure 4.32: Medium radial root crack under the largest load at 600 RPM**

Figure 4.32 has a clear peak dynamic transmission error which corresponded with the fault window. Like the case for the large radial crack, the medium crack failed to simulate the smooth engagement and disengagement profile. Therefore, the medium crack load level was reduced to the lowest load torque (Figure 4.33) for investigation.
Figure 4.33: Medium radial root crack under the smallest load at 600 RPM

The peak dynamic transmission error for the smaller loading case was found to be 0.5846 mm. Here there was a smoother engagement and disengagement profile created. This profile had approximately the number of engagements that was expected for two 32 tooth gears meshing together. There was not the issue of secondary peak values which caused difficulty in clear fault detection like that seen during the large radial root crack tests. As a result, the detection method of using dynamic transmission error was deemed moderately successful for the medium depth radial root crack due to the same reasons listed for the large crack depth.

After the load was decreased even further the results for the dynamic transmission error of the smallest radial root crack depth were plotted in Figure 4.34 and Figure 4.35. Here the largest loading scenario was not included since the results selected illustrated a better representation of the test method. These results can still be seen in Appendix C.
Figure 4.34: Small radial root crack under an intermediate load at 600 RPM

The largest dynamic transmission error for the intermediate load was found to be 0.7716 mm. This value appeared inside the recorded fault window. The engagement/disengagement is smooth and of repeating value. By further decreasing the load to the smallest, Figure 4.35 was created.

Figure 4.35: Small radial root crack under the lowest load at 600 RPM
Peak dynamic transmission error has a measured value of 0.7034 mm for the lowest loaded test. This value was slightly lower in magnitude compared to the medium load. This decreasing dynamic transmission error was therefore attributed to the decreased loading on the gears. This engagement profile was not a clear representation of each gear tooth. Due to the smooth engagement profile noticed for the medium loading condition and the correct location of peak dynamic transmission error, this allowed the detection method to be successful for the small radial root crack.

All in all, the proposed method of fault diagnostic when the fault of interest was a radial root crack was moderately successful. In most cases the peak dynamic transmission error occurred inside the fault window. However, often times there were quite a few, slightly lower, peaks which created difficulty in detection. The results proved to be least effective when the loading was at its largest and most effective when loading was at its lowest. The engagement/disengagement profile was not always a clear representation of each gear tooth entering and leaving the mesh. Also, as the magnitude of the dynamic transmission error increased, the root crack depth decreased which was an odd discovery. The dynamic transmission error did however decrease, with the exception of the medium depth, with the load decrease as was predicted. Overall, the detection ability for the method developed was sufficient to detect a gear fault most of the time (without external aid) but lacked consistency and the ability to accurately track the engagement profile all the time.

4.6 Conclusion
In conclusion, the proposed method of measuring dynamic transmission error, using a Hall Effect sensor, was generally successful for detecting the presence of a gear fault. The method was especially successful in detecting the presence of the tangential root cracks. For this fault case the peak dynamic transmission error was detected and corresponded to the expected fault location. This expected fault location was determined using the optical sensor to create a fault window.
Table 4.1 highlights the results of the dynamic transmission error for the tangential root crack tests measured at 600 RPM.

Table 4.1: Summary of dynamic transmission error results for the tangential root crack tests at 600 RPM

<table>
<thead>
<tr>
<th>Test Case</th>
<th>Dynamic Transmission Error (mm)</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Tangential Root Crack</td>
<td></td>
<td>Load 3</td>
<td>Load 1</td>
</tr>
<tr>
<td>Large</td>
<td>1.446</td>
<td>1.43</td>
<td></td>
</tr>
<tr>
<td>Medium</td>
<td>0.6127</td>
<td>0.3357</td>
<td></td>
</tr>
<tr>
<td>Small</td>
<td>0.7791</td>
<td>1.036</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Test Case</th>
<th>Peak DTE Inside Fault Window</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Tangential Root Crack</td>
<td></td>
<td>Load 3</td>
<td>Load 1</td>
</tr>
<tr>
<td>Large</td>
<td>Yes</td>
<td>Yes</td>
<td></td>
</tr>
<tr>
<td>Medium</td>
<td>Yes</td>
<td>Yes</td>
<td></td>
</tr>
<tr>
<td>Small</td>
<td>No</td>
<td>No</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Test Case</th>
<th>Smooth Engagement Profile</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Tangential Root Crack</td>
<td></td>
<td>Load 3</td>
<td>Load 1</td>
</tr>
<tr>
<td>Large</td>
<td>Yes</td>
<td>Yes</td>
<td></td>
</tr>
<tr>
<td>Medium</td>
<td>Yes</td>
<td>No</td>
<td></td>
</tr>
<tr>
<td>Small</td>
<td>Yes</td>
<td>No</td>
<td></td>
</tr>
</tbody>
</table>

When examining the large tangential root crack, at both the largest and smallest loading condition, the resulting peak dynamic transmission errors were close in magnitude. Any decrease in magnitude can be accounted for by the decreased loading condition. For the missing tooth test, there was also a clear representation of the engagement and disengagement profile of each gear tooth in the gear mesh. Once the fault crack depth was decreased to the intermediate depth, the smooth engagement profile was lost. However, this was not the case for the peak dynamic transmission error as this result was found inside the fault window. Upon further decreasing the fault size, the experimental method had a difficult time recording a peak dynamic transmission error.
error inside the simulated fault window for the smallest crack depth. Conversely, this smaller crack depth resulted in the smooth engagement profile returning.

Like the results for the tangential root crack, the radial root crack dynamic transmission error information is organized in Table 4.2.

**Table 4.2: Summary of dynamic transmission error results for the radial root crack tests at 600 RPM**

<table>
<thead>
<tr>
<th>Test Case</th>
<th>Dynamic Transmission Error (mm)</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Radial Root Crack</td>
<td>Load 3</td>
<td>Load 2</td>
</tr>
<tr>
<td>Large</td>
<td>0.3205</td>
<td>0.2796</td>
</tr>
<tr>
<td>Medium</td>
<td>0.4747</td>
<td>0.5846</td>
</tr>
<tr>
<td>Small</td>
<td>0.7716</td>
<td>0.7034</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Test Case</th>
<th>Peak DTE Inside Fault Window</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Radial Root Crack</td>
<td>Load 3</td>
<td>Load 2</td>
</tr>
<tr>
<td>Large</td>
<td>Yes</td>
<td>Yes</td>
</tr>
<tr>
<td>Medium</td>
<td>Yes</td>
<td>Yes</td>
</tr>
<tr>
<td>Small</td>
<td>Yes</td>
<td>Yes</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Test Case</th>
<th>Smooth Engagement Profile</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Radial Root Crack</td>
<td>Load 3</td>
<td>Load 2</td>
</tr>
<tr>
<td>Large</td>
<td>No</td>
<td>Yes</td>
</tr>
<tr>
<td>Medium</td>
<td>No</td>
<td>Yes</td>
</tr>
<tr>
<td>Small</td>
<td>Yes</td>
<td>No</td>
</tr>
</tbody>
</table>

In comparison to the tangential root crack, the results for the radial root crack are only moderately successful at detecting the presence of a fault. The main limitation was the inability to show a smooth engagement profile all the time. The method was able to detect a peak dynamic transmission error which is directly related to the presence of the radial root crack. This was noticeable for all depths of cracks at the load conditions presented but was difficult to see at times due to the presence of other secondary peak dynamic transmission error measurements. For the smallest and largest root crack there was a decrease in dynamic transmission error seen by the
reduction of the output load (see Table 4.2). This was not the case for the medium root crack as the dynamic transmission error increased with a decreasing load.

Unlike the results for the tangential root crack, the recorded dynamic transmission error appears to increase with a decreasing fault depth. This result was peculiar but in the end it is relative to the test case. For a given gear set, individual gear tooth to gear tooth meshing could be occurring more efficiently than another gear set. With this increased gear mesh efficiency would come a reduced dynamic transmission error measurement overall, which would reduce the peak dynamic transmission error.

In the end, the proposed gear fault diagnostics method using a Hall Effect sensor to measure shaft rotational position was somewhat successful in diagnosing the faults studied. A fault was deemed present if the peak dynamic transmission error was present at the correct location. This correct location was determined with the aid of the optical sensor measurements. As part of the criteria to determine the success of the sensors ability to accurately work, the dynamic transmission error for one shaft rotation was examined. The graph was examined for a well-defined peak dynamic transmission error as well as a smooth engagement/disengagement profile. If the graph possessed both of these, it was deemed successful and could accurately measure the fault at study. Due to the criteria requiring these two properties, the proposed method for radial root crack was somewhat successful in accurately determining a correctly located peak dynamic transmission error. These results were moderately successful because a clear engagement profile is not established all the time. Without this clear engagement profile, it proved to be difficult in accurately diagnosing and monitoring the radial crack. On the other hand, the proposed method was quite accurate in its ability to detect two out of the three tangential root crack depths. The third did not display a peak dynamic transmission error which corresponded to the fault window.
but accurately illustrated the engagement and disengagement profile. After examining the magnitude of the dynamic transmission error, a relationship between crack size as well as loading conditions was created. For the tangential root crack the dynamic transmission error was directly related to both the load size and crack depth. This verified the idea of using this method for condition monitoring due to the increased dynamic transmission error response as a result of increased crack size.
Chapter 5
Conclusions and Future Work

This thesis examined the use of a non-contact rotational position sensor for the fault detection and diagnosis ability in common steel gears. First, a direct comparison of the Hall Effect sensor’s measuring capability was made against an industry standard method of measuring shaft rotational position. Post calibration work, the proposed sensor was used for the detection and monitoring of a gearbox. Fault detection was made through measuring the Dynamic Transmission Error (DTE). Dynamic transmission error was defined as the difference between the input and output shaft rotational positions as a function of time.

Calibration of the Hall Effect sensor was accomplished by comparing the difference in the shaft rotational speed measurements provided by an incremental encoder. This encoder had the ability to measure at a per degree rate. The experiments measured the sensor’s operation at steady state speeds, transient speeds, changing sensor positions, a variety of magnet sizes, as well as the exposure to environmental factors. Environmental factors were defined as potential industry caused limitations that could affect the sensors recording ability (such as grease or debris). The sensitivity of the Hall Effect sensor was defined as the standard deviation of the recorded data. Despite minor differences, the recorded measurements between the two sensors were in good agreement and there were only slight differences in sensitivity found between each individual experiment.

With a correctly calibrated sensor, the dynamic transmission error for the purpose of machine condition monitoring and fault diagnostics could be explored. This method was applied to a parallel shaft gearbox which enclosed a steel spur gear set. The fault types explored were of
different severity levels and defined as tangential and radial roots cracks. For each of the dynamic transmission error results, the graphs were examined for both a clean engagement profile as well as an appropriate peak dynamic transmission error. It was found that the sensor’s ability to detect and track the presence of the tangential root crack was of notable success. The proposed method was able to detect two out of the three crack depths while forming a direct relationship between dynamic transmission error with load size and crack depth. This was all accomplished while recording a smooth engagement profile. The third was unable to illustrate a peak dynamic transmission error inside the fault window. On the contrary, the results for the radial root crack were successful at times in determining peak dynamic transmission error results. However, for these results there was not a clean engagement and disengagement profile nor was there any sort of relationship found between load and crack depth. If future work were conducted, this would be a great starting point.

As mentioned above, an area for future work would be to revisit these tests, or ones similar to them, in order to determine the limitations which caused the stated results. This could be accomplished by retesting the experiments in finer detail, in order to determine differences that are evident not only at a variety of consistent speeds but also in the transient state. A potential starting point would be exploring the direct impact of imperfect meshing and determining a method of reducing these measurements to better illustrate the smooth engagement profile that was desired. With the presence of a smooth profile, other characteristics of the signal can be examined as well. Such characteristics include kurtosis (peaked-ness) or crest factor (ratio of peak value to average value). Through the creation of a localized tooth dynamic transmission error baseline, pattern recognition could be adopted in order to track changes from gear tooth to gear tooth meshing by way of examining the signal properties. This hypothesis proposes a potential
solution to detecting and tracking the presence of a fault even if it is not the largest contribution to
the dynamic transmission error output.

After complete laboratory success is achieved using the dynamic transmission error approach, the
study could be expanded even further. Expansion could be conducted into the potential study of
multi-stage gear trains, or through the aid of the right industry partner, this proposed method
could be applied to a select piece of machinery.
Appendix A
Dynamic Sensor Calibration

The Hall Effect sensor measurements were quantified for every case, other than the speed dependency test case, using two rotational frequencies. These rotational frequencies were 20 Hz and 30 Hz and were set using the open loop controller. The results shown in the dynamic calibration section chapter were all for the 20 Hz test. The 30 Hz test case was omitted due to consistency of results and can be found in the appendix to follow.

The individual graphs for the speed dependency test case are presented first, followed by the results for the 30 Hz test case presented in a similar order as displayed in the thesis.

Figure A. 1: Speed dependency rotation-to-rotation frequency for an open loop control of 5 Hz
Figure A. 2: Speed dependency for 5 Hz rotational speed

Figure A. 3: Speed dependency rotation-to-rotation frequency for an open loop control of 10 Hz
Figure A. 4: Speed dependency for 10 Hz rotational speed

Figure A. 5: Speed dependency rotation-to-rotation frequency for an open loop control of 30 Hz
Figure A. 6: Speed dependency for 30 Hz rotational speed

Figure A. 7: Speed dependency rotation-to-rotation frequency for an open loop control of 40 Hz
Figure A. 8: Speed dependency for 40 Hz rotational speed

Figure A. 9: Speed dependency rotation-to-rotation frequency for an open loop control of 50 Hz
Figure A. 10: Speed dependency for 50 Hz rotational speed

Figure A. 11: Speed dependency rotation-to-rotation frequency for an open loop control of 60 Hz
Figure A. 12: Speed dependency for 60 Hz rotational speed

Figure A. 13: Transient speed rotation-to-rotation frequency for a full open loop control test profile (30 Hz maximum speed)
Figure A. 14: Transient speed mean difference comparison for the 30 Hz ramp case

Figure A. 15: Sensor position of 2 spacers (0.008 inches (0.2032mm)) rotation-to-rotation frequency for an open loop control of 30 Hz
Figure A. 16: Sensor position of 4 spacers (0.016 inches (0.4064)) rotation-to-rotation frequency for an open loop control of 30 Hz

Figure A. 17: Sensor position mean speed with standard deviation error
Figure A. 18: 0.25 inches (6.35 mm) diameter magnet rotation-to-rotation frequency for an open loop control of 30 Hz

Figure A. 19: 0.5 inches (12.7 mm) diameter magnet rotation-to-rotation frequency for an open loop control of 30 Hz
Figure A. 20: Magnet size mean speed with standard deviation error

Figure A. 21: Electrical tape rotation-to-rotation frequency for an open loop control of 30 Hz
Figure A. 22: Masking tape rotation-to-rotation frequency for an open loop control of 30 Hz

Figure A. 23: Grease rotation-to-rotation frequency for an open loop control of 30 Hz
Figure A. 24: Metal filings rotation-to-rotation frequency for an open loop control of 30 Hz

Figure A. 25: Environmental factors mean speed with standard deviation error
Figure A. 26: Environmental factors differences in standard deviation of encoder and Hall Effect sensor
Appendix B
Dynamic Sensor Calibration MATLAB Script

Attached is a copy of the MATLAB script file name ‘Calib_Test.m’. This script was used to determine how accurately the Hall Effect sensor was in comparison to a high accuracy encoder. Careful notice should be taken to variables that require changing from test case to test case. These changing variables included the file name, and low pass filter coefficient (which was determined by trial and error to be 20 Hz larger than the running frequency).

```matlab
% Calibration Testing Script: passes in a save .mat file
clear
clc
% Beginning of data analysis
% loads data
data = load('0.5 Diameter (20 Hz).mat');

% initializes variables
Hall1 = data.ConvertedData.Data.MeasuredData(1,4).Data;
Hall2 = data.ConvertedData.Data.MeasuredData(1,5).Data;
EncoderRaw = data.ConvertedData.Data.MeasuredData(1,3).Data;

% sets changing variables
samplerate = 85000;
lowpass = 40; % speed plus 20
dofoil = '0.5" Diameter Magnet 20 Hz';

% returns rotational frequency determined by way of the encoder
[EncoderFullSpeed EncoderHalfSpeed] = EncoderCount1(EncoderRaw, samplerate);

% filtering the signal
[filtHall1 filtHall2] = datafilt(Hall1,Hall2, samplerate, lowpass);

% normalization (taken from Taylor's normal function)
[normalHall1 normalHall2] = normalshift(filtHall1, filtHall2);

% % Use only for transient condition uncomment what applies
```
% % for transient speed 20 Hz:
% normalHall1 = normalHall1(217000:1070000);
% normalHall2 = normalHall2(217000:1070000);
%
% for transient speed 30 Hz:
% normalHall1 = normalHall1(280000:1395750);
% normalHall2 = normalHall2(280000:1395750);
%
% for transient speed 0-15-30-15-30-0 Hz:
% normalHall1 = normalHall1(239000:3700000);
% normalHall2 = normalHall2(239000:3700000);

% rotation counter function for Hall Effect signal data
[HallPeak1 HallPeak2 HallValley1 HallValley2] =
HallCount(normalHall1,normalHall2,samplerate);

% Hall Effect rotational frequency calculator
[ HallHalfSpeed1 HallHalfSpeed2 HallFullSpeed1 HallFullSpeed2 ] =
HallSpeed(HallPeak1,HallPeak2,HallValley1,HallValley2);

% trimming off the first and last rotation
HallFullSpeed1 = HallFullSpeed1(10:(length(HallFullSpeed1)-10));
HallFullSpeed2 = HallFullSpeed2(10:(length(HallFullSpeed2)-10));
HallHalfSpeed1 = HallHalfSpeed1(10:(length(HallHalfSpeed1)-10));
HallHalfSpeed2 = HallHalfSpeed2(10:(length(HallHalfSpeed2)-10));
EncoderFullSpeed = EncoderFullSpeed(10:(length(EncoderFullSpeed)-10));
EncoderHalfSpeed = EncoderHalfSpeed(10:(length(EncoderHalfSpeed)-10));

% full rotation
figure(1)
plot(1:length(EncoderFullSpeed),EncoderFullSpeed,1:length(HallFullSpeed1),HallFullSpeed1,
1:length(HallFullSpeed2),HallFullSpeed2)
title(dfile,'FontSize',22)
xlabel('Rotation Number','FontSize',14)
ylabel('Frequency','FontSize',14)
legend('Encoder','Hall Effect 1','Hall Effect 2')

% half rotation
figure(2)
plot(1:length(EncoderHalfSpeed),EncoderHalfSpeed,1:length(HallHalfSpeed1),HallHalfSpeed1,
1:length(HallHalfSpeed2),HallHalfSpeed2)
title(dfile,'FontSize',22)
xlabel('Half Rotation Number','FontSize',14)
ylabel('Frequency','FontSize',14)
legend('Encoder','Hall Effect 1','Hall Effect 2')

% statistics
EncoderHalfSTD = std(EncoderHalfSpeed);
EncoderHalfMean = mean(EncoderHalfSpeed);
EncoderFullSTD = std(EncoderFullSpeed);
EncoderFullMean = mean(EncoderFullSpeed);

HallHalfSpeed1STD = std(HallHalfSpeed1);
HallHalfSpeed1Mean = mean(HallHalfSpeed1);
HallHalfSpeed1STD = std(HallHalfSpeed1);
HallFullSpeed1Mean = mean(HallFullSpeed1);

HallHalfSpeed2STD = std(HallHalfSpeed2);
HallHalfSpeed2Mean = mean(HallHalfSpeed2);
HallFullSpeed2STD = std(HallFullSpeed2);
HallFullSpeed2Mean = mean(HallFullSpeed2);

% statistics for the encoder and two hall effect signals
encoderdata = [EncoderHalfMean EncoderHalfSTD EncoderFullMean EncoderFullSTD]
hall1data = [HallHalfSpeed1Mean HallHalfSpeed1STD HallFullSpeed1Mean HallFullSpeed1STD]
hall2data = [HallHalfSpeed2Mean HallHalfSpeed2STD HallFullSpeed2Mean HallFullSpeed2STD]

datatata = [EncoderHalfMean EncoderHalfSTD EncoderFullMean EncoderFullSTD; HallHalfSpeed1Mean HallHalfSpeed1STD HallFullSpeed1Mean HallFullSpeed1STD; HallHalfSpeed2Mean HallHalfSpeed2STD HallFullSpeed2Mean HallFullSpeed2STD];
Appendix C
Steel Gear Fault Diagnostics

The speed profile was used for all testing scenarios. By doing this, measurements for the 300 RPM, 600 RPM, and 900 RPM could be collected under one test window for each of the desired loads. In the steel gear fault diagnostics chapter, only the 600 RPM data samples were included. These graphs demonstrated the measured dynamic transmission error without the mentioned fault window included. In this appendix, the same graphs that are presented in the steel gear fault diagnostics chapter as well as all remaining test speeds are included. Here all the graphs have the measured fault window included. The findings for those graphs that appear in the body of the thesis are listed first, followed by the 300 RPM, 900 RPM and finally the healthy measurements.
Figure C.1: Large tangential root crack under the largest load at 600 RPM with fault window

Figure C.2: Large tangential root crack under the intermediate load at 600 RPM with fault window

Figure C.3: Large tangential root crack under the smallest load at 600 RPM with fault window

Figure C.4: Medium tangential root crack under the largest load at 600 RPM with fault window

Figure C.5: Medium tangential root crack under the intermediate load at 600 RPM with fault window

Figure C.6: Medium tangential root crack under the smallest load at 600 RPM with fault window
Figure C. 7: Small tangential root crack under the largest load at 600 RPM with fault window

Figure C. 8: Small tangential root crack under the intermediate load at 600 RPM with fault window

Figure C. 9: Small tangential root crack under the smallest load at 600 RPM with fault window

Figure C. 10: Large radial root crack under the largest load at 600 RPM with fault window

Figure C. 11: Large radial root crack under the intermediate load at 600 RPM with fault window

Figure C. 12: Large radial root crack under the smallest load at 600 RPM with fault window
Figure C. 13: Medium radial root crack under the largest load at 600 RPM with fault window

Figure C. 14: Medium radial root crack under the intermediate load at 600 RPM with fault window

Figure C. 15: Medium radial root crack under the smallest load at 600 RPM with fault window

Figure C. 16: Small radial root crack under the largest load at 600 RPM with fault window

Figure C. 17: Small radial root crack under the intermediate load at 600 RPM with fault window

Figure C. 18: Small radial root crack under the smallest load at 600 RPM with fault window
Figure C. 19: Large tangential root crack under the largest load at 300 RPM with fault window

Figure C. 20: Large tangential root crack under the intermediate load at 300 RPM with fault window

Figure C. 21: Large tangential root crack under the smallest load at 300 RPM with fault window

Figure C. 22: Medium tangential root crack under the largest load at 300 RPM with fault window

Figure C. 23: Medium tangential root crack under the intermediate load at 300 RPM with fault window

Figure C. 24: Medium tangential root crack under the smallest load at 300 RPM with fault window
Figure C. 25: Small tangential root crack under the largest load at 300 RPM with fault window

Figure C. 26: Small tangential root crack under the intermediate load at 300 RPM with fault window

Figure C. 27: Small tangential root crack under the smallest load at 300 RPM with fault window

Figure C. 28: Large radial root crack under the largest load at 300 RPM with fault window

Figure C. 29: Large radial root crack under the intermediate load at 300 RPM with fault window

Figure C. 30: Large radial root crack under the smallest load at 300 RPM with fault window
Figure C. 31: Medium radial root crack under the largest load at 300 RPM with fault window

Figure C. 32: Large radial root crack under the intermediate load at 300 RPM with fault window

Figure C. 33: Large radial root crack under the smallest load at 300 RPM with fault window

Figure C. 34: Small radial root crack under the largest load at 300 RPM with fault window

Figure C. 35: Small radial root crack under the intermediate load at 300 RPM with fault window

Figure C. 36: Small radial root crack under the smallest load at 300 RPM with fault window
Figure C. 37: Large tangential root crack under the largest load at 900 RPM with fault window

Figure C. 38: Large tangential root crack under the intermediate load at 900 RPM with fault window

Figure C. 39: Large tangential root crack under the smallest load at 900 RPM with fault window

Figure C. 40: Medium tangential root crack under the largest load at 900 RPM with fault window

Figure C. 41: Medium tangential root crack under the intermediate load at 900 RPM with fault window

Figure C. 42: Medium tangential root crack under the smallest load at 900 RPM with fault window
Figure C. 43: Small tangential root crack under the largest load at 900 RPM with fault window

Figure C. 44: Small tangential root crack under the intermediate load at 900 RPM with fault window

Figure C. 45: Small tangential root crack under the smallest load at 900 RPM with fault window

Figure C. 46: Large radial root crack under the largest load at 900 RPM with fault window

Figure C. 47: Large radial root crack under the intermediate load at 900 RPM with fault window

Figure C. 48: Large radial root crack under the smallest load at 900 RPM with fault window
Figure C. 49: Medium radial root crack under the largest load at 900 RPM with fault window

Figure C. 50: Medium radial root crack under the intermediate load at 900 RPM with fault window

Figure C. 51: Medium radial root crack under the smallest load at 900 RPM with fault window

Figure C. 52: Small radial root crack under the largest load at 900 RPM with fault window

Figure C. 53: Small radial root crack under the intermediate load at 900 RPM with fault window

Figure C. 54: Small radial root crack under the smallest load at 900 RPM with fault window
Figure C. 55: Healthy gear under the largest load at 300 RPM

Figure C. 56: Healthy gear under the intermediate load at 300 RPM

Figure C. 57: Healthy gear under the smallest load at 300 RPM

Figure C. 58: Healthy gear under the largest load at 600 RPM

Figure C. 59: Healthy gear under the intermediate load at 600 RPM

Figure C. 60: Healthy gear under the smallest load at 600 RPM
Figure C. 61: Healthy gear under the largest load at 900 RPM

Figure C. 62: Healthy gear under the intermediate load at 900 RPM

Figure C. 63: Healthy gear under the smallest load at 900 RPM
Appendix D

Steel Gear Fault Diagnostics MATLAB Script

Attached is a copy of the MATLAB script file name ‘Get_Data.m’. This script was used to create the dynamic transmission error analysis. Careful notice should be taken to variables that require changing from test case to test case. These changing variables are listed near the start under the ‘testing parameters’ section.

% Converts TDMS and Structures it in an Array. Then cuts test segment
% of interest, averages the data, and uses the arctan function to combine sine and cosine
% signal. Returns three plots: Dynamic Transmission Error, Dynamic % Transmission Error (with fault window), and a FFT

clear

clc

%Temporary conversion from TDMS file with file location prompting.
data = convertTDMS(0);

%Extracts Data file for Fault Index from temporary .mat file
FaultIndex = data.Data.MeasuredData(1,3).Data;

%Extracts Data file for Acceleration from temporary .mat file
Acceleration = data.Data.MeasuredData(1,4).Data;

%Extracts Data file for Motor Set Point from temporary .mat file
MotorSetPoint = data.Data.MeasuredData(1,5).Data;

%Extracts Data file for Load Set Point from temporary .mat file
LoadSetPoint = data.Data.MeasuredData(1,6).Data;

%Extracts Data file for Drive Tach 25HP from temporary .mat file
DriveTach25 = data.Data.MeasuredData(1,7).Data;

%Extracts Data file for Load Tach 50HP from temporary .mat file
LoadTach50 = data.Data.MeasuredData(1,8).Data;

%Extracts Data file for Input Sine from temporary .mat file
InputSine = data.Data.MeasuredData(1,9).Data;

%Extracts Data file for Input Cosine from temporary .mat file
InputCosine = data.Data.MeasuredData(1,10).Data;

%Extracts Data file for Output Sine from temporary .mat file
OutputSine = data.Data.MeasuredData(1,11).Data;
%Extracts Data file for Output Cosine from temporary .mat file
OutputCosine = data.Data.MeasuredData(1,12).Data;

%Beginning of Data analysis
%timing variables
%for Load 1
time1 = [4.1;11.2;14.3;11.3;14.1;11.2;14.3;32.8];
%for Load 2
time2 = [4.1;11.2;14.3;11.3;14.1;11.2;14.3;32.9];
%for Load 3
time3 = [4;15.5;10;11.3;14.2;11.2;14.3;32.7];

%Sets time to time of test case. Change depending on load
time = time1;
%time points for 300, 600, 900 RPM (2/3 pair, 4/5 pair, 6/7 pair)
tp1 = 4;
tp2 = 5;

%testing parameters that require changing depending on test case
dfile = 'Tan Crack Medium - Load Level 1 @ 600 RPM'
RPM = 600; %initializes RPM
sample_rate = 10000; %sample frequency
nt = 32; %number of teeth
Freq_low = RPM/60*9; %mechanical looseness frequency
Freq_high = RPM/60*nt*2; %gear mesh frequency (6 at first)
in_rad = 50.8; %input pitch radii (mm)
out_rad = 50.8; %output pitch radii (mm)

%for unfiltered data
[CutSignal CutFaultIndex CutAcceleration] = cutsignal1(
InputCosine,InputSine,OutputCosine,OutputSine,time,FaultIndex,Acceleration,sample_rate,tp1,tp2);

%time sync average data
[TimeAvgSignal TimeAvgFaultIndex] = timeavg( CutSignal, time, CutFaultIndex, RPM, sample_rate,tp2);

%Moving average of 'x' points
AvgSignal = average(TimeAvgSignal);

%normalize the signal between 1 and -1
NormalSignal = normal(AvgSignal);

%combines the cosine and sine values together
CombineSignal = combine(NormalSignal,in_rad,out_rad);
%Subtracts the output signal from the input signal to find overall TE
(from Taylor Functions)
[C TEerror] = subtract_pos2(CombineSignal);

%filtering final data
FilteredTE = datafilter1(TEerror, Freq_low, Freq_high, sample_rate);

%sets data points to second intervals ie 1/sample_rate. with fault window
figure(1)
plot(1/sample_rate:1/sample_rate:length(FilteredTE)/sample_rate,FilteredTE,1/sample_rate:1/sample_rate:length(TimeAvgFaultIndex)/sample_rate,TimeAvgFaultIndex*7)
title(dfile,'FontSize',22)
xlabel('Time (s)','FontSize',14)
ylabel('Dynamic Transmission Error (mm)','FontSize',14)
ylim([min(FilteredTE)*3 max(FilteredTE)*3])

%without fault window
figure(2)
plot(1/sample_rate:1/sample_rate:length(FilteredTE)/sample_rate,FilteredTE)
title(dfile,'FontSize',22)
xlabel('Time (s)','FontSize',14)
ylabel('Dynamic Transmission Error (mm)','FontSize',14)
ylim([min(FilteredTE)*3 max(FilteredTE)*3])

%FFT
laccel = length(CutAcceleration);
power2 = 2^nextpow2(laccel);
y_fft = fft(CutAcceleration,power2)/laccel;
x_fft = sample_rate/2*linspace(0,1,power2/2+1);
figure(3)
bar(x_fft,2*abs(y_fft(1:power2/2+1)));
title('Raw FFT')
xlabel('Frequency (Hz)')
ylabel('Amplitude')
Appendix E

Gear Bending Stress Sample Calculation

Attached is an example of worked out calculations for gear bending stress. Here a spur gear having 32 teeth and a 4 inch pitch diameter was selected. For purposes of these calculations the transmitted horsepower was 25 at an operating speed 600 RPM.
Bending Stress Calculations - Example

\[ d_p = \frac{N_p}{P_d} = \frac{32}{8} = 4 \text{ in} \Rightarrow \text{pitch diameter} \]

\[ V = \frac{\pi d_p P_n}{12} = \frac{\pi (4) 600}{12} = 628.3 \text{ ft/min} \]

\[ \omega = \frac{33,000 \times 12}{628.3} = 1313.0 \text{ lbft} \Rightarrow \text{transmitted force} \]

\[ K_o = 1 \Rightarrow \text{uniform loading} \]

\[ K_{u,r} = \left( \frac{A + \sqrt{V}}{A} \right)^8 \]

\[ = \left( \frac{59.77 + \sqrt{628.3}}{59.77} \right)^{0.0535} \]

\[ A = 50 + 56(1 - 0.825) = 59.77 \]

\[ = 1.335 \]

\[ K_s = 1.192 \left( \frac{F \sqrt{V}}{P_d} \right)^{0.0535} \]

\[ = 1.192 \left( \frac{1.25 \times 0.265}{8} \right)^{0.0535} \]

\[ = 1.051 \]

\[ K_b = 1 \]

\[ K_m = 1 + C_{m_e} (C_{p_e} C_{p_m} + C_{m_a} C_e) \]

\[ C_{p_e} = \frac{F}{10(4)} - 0.375 + 0.0175 \]

\[ = 0.9375 \]

\[ C_{m_e} = A + B F + C F^2 \]

\[ = 0.247 + 0.0167(1.75)^2 \]

\[ = 0.268 \]

\[ J_{\text{bending}} = \omega + K_o K_u K_s \frac{d_p K_m K_b}{F} \]

\[ = 1313.0(1) 1.335 (1.051) 2.125 (1.227) \]

\[ = 51042.8 \text{ in}^4 \]

\[ \Rightarrow \text{for 25 HP transmission @ 600 RPM} \]
References


