HEALTH MONITORING APPLICATIONS FOR SMALL WIND TURBINES

A Thesis in
Aerospace Engineering

by
Brenton Forshey

© 2011 Brenton Forshey

Submitted in Partial Fulfillment
of the Requirements
for the Degree of

Master of Science

August 2011
The thesis of Brenton Forshey was reviewed and approved* by the following:

Karl M. Reichard  
Research Associate, Applied Research Laboratory  
Assistant Professor, Graduate Program in Acoustics

Dennis K. McLaughlin  
Professor of Aerospace Engineering

Susan Stewart  
Research Associate, Aerospace Engineering & Architectural Engineering

George A. Lesieutre  
Professor of Aerospace Engineering  
Head of the Department of Aerospace Engineering

*Signatures are on file in the Graduate School
ABSTRACT

Costly failures have plagued industrial-scale wind turbines in recent years, and these failures have spawned the development of systems designed to provide continuous data on the health of these machines. Though they vary in implementation, the common goal of these systems is to detect potential faults before they lead to catastrophic failures and extended downtime of the turbine. The most basic condition monitoring systems typically employ vibration sensors placed at key locations in the machine that can detect faults that develop within the main bearings. Torsion and bending sensors on the tower are also used to identify and distinguish various types of blade imbalances, be they aerodynamic or mechanical in nature. The purpose of this investigation was to develop methods for fault detection on a residential-scale wind turbine. Two separate fault detection experiments are presented in this thesis.

The purpose of the first experiment was to detect and qualify the severity of an inertial imbalance seeded on a Whisper 500 turbine in the field. Such imbalances are typically the result of ice buildup during normal operation. Through the observation of passive vibration spectra from a tower-mounted accelerometer, two separate imbalance cases were able to be successfully detected and differentiated from the baseline case.

A second test was conducted on a dynamometer with the purpose of detecting a seeded fault within one of the wind turbine’s main bearings. It was shown through the use of envelope analysis that lab-grade accelerometers could successfully detect vibration frequencies that indicated the presence of a seeded outer race fault in the Whisper 500’s rearmost bearing.
# TABLE OF CONTENTS

LIST OF FIGURES ........................................................................................................... vii

LIST OF TABLES ................................................................................................................ xiii

NOMENCLATURE ............................................................................................................... xiv

ACKNOWLEDGEMENTS ................................................................................................. xv

Chapter 1 INTRODUCTION ............................................................................................ 1
  1.1 History of Wind Energy ............................................................................................. 2
  1.2 Wind Turbine Failures ............................................................................................. 7
  1.3 Health Monitoring of Wind Turbines ....................................................................... 11
  1.4 Performance-based Methods of Health Monitoring ................................................ 14
  1.5 Vibration-based Methods of Health Monitoring ...................................................... 18
  1.6 Scope and Objectives ............................................................................................... 22

Chapter 2 THEORY ......................................................................................................... 24
  2.1 Introduction ............................................................................................................. 24
  2.2 Wind Turbine Performance ...................................................................................... 25
    2.2.1 Power in the Wind ............................................................................................ 25
    2.2.2 Actuator Disk Theory .................................................................................... 25
    2.2.3 The Betz Limit ............................................................................................... 27
    2.2.4 Blade Element Momentum Theory .................................................................. 28
  2.3 Wind Turbine Performance Monitoring .................................................................. 32
    2.3.1 Long-Term Observation ............................................................................... 32
    2.3.2 Short Term Methods ...................................................................................... 32
  2.4 Vibration Monitoring of Wind Turbines .................................................................. 33
    2.4.1 Basic Vibration Spectrum ............................................................................. 33
    2.4.2 Envelope Curve Analysis .............................................................................. 34
    2.4.3 Cepstrum Analysis ....................................................................................... 37

Chapter 3 FACILITIES ................................................................................................... 39
  3.1 Introduction ............................................................................................................. 39
  3.2 Whisper 500 Wind Turbine Site ............................................................................. 40
  3.3 Whisper 500 System ............................................................................................... 42
    3.3.1 Wind Turbine ................................................................................................. 44
    3.3.2 Generator .................................................................................................... 46
    3.3.3 Controller .................................................................................................... 47
    3.3.5 Battery Bank ................................................................................................. 49
    3.3.6 Diversion Load ............................................................................................. 50
Chapter 4 EXPERIMENTAL METHODS ......................................................68

4.1 Inertial Imbalance Experiments ..................................................68
4.1.1 Normal Operation .................................................................69
4.1.2 Inertial Imbalance .................................................................71
4.2 Inertial Imbalance Data Processing ..........................................78
4.2.1 Goals of Data Processing .....................................................78
4.2.2 Data Selection ..................................................................... 79
4.2.3 Data Evaluation ................................................................. 83
4.2.4 Power Data Processing .........................................................86
4.2.5 Vibration Data Processing ....................................................89
4.2.6 Power Data Feature Identification ........................................91
4.2.7 Vibration Data Feature Identification ....................................93
4.2.8 Consolidated Power Processing Code ....................................94
4.2.9 Consolidated Vibration Processing Code .................................96
4.3 Bearing Test ......................................................................... 98
4.3.1 Original Bearing Test ...........................................................99
4.3.2 New Bearing Test ...............................................................101
4.3.3 Faulted Bearing Test ..........................................................103
4.4 Bearing Test Data Processing ...................................................105

Chapter 5 RESULTS AND DISCUSSION .............................................107

5.1 Inertial Imbalance Test Results ................................................108
5.1.1 Power Signal Results ...........................................................108
5.1.2 Power Signal Discussion ......................................................115
5.1.3 Vibration Signal Results ......................................................116
5.1.4 Vibration Signal Discussion ...............................................123
5.2 Bearing Bench Test Results .......................................................126
5.2.1 Lab-Grade Accelerometer Results .......................................127
5.2.2 Microphone Results ...........................................................137
5.2.3 Wireless Accelerometer Results ..........................................141

Chapter 6 CONCLUSIONS AND FUTURE WORK ..............................143
6.1 Summary and Conclusions ................................................................. 143
6.2 Future Work .................................................................................. 145

Bibliography ......................................................................................... 148

Appendix A  RPM_PLOT.m ................................................................. 151
Appendix B  FFT_Average.m ................................................................. 154
Appendix C  Power_Vibration_Processing.m ........................................ 157
LIST OF FIGURES

**Figure 1-1:** Early windmills employed flat blades and a simple gear mechanism. (3) .......3

**Figure 1-2:** The MOD-2 was used in the investigation of wind turbine dynamics. (6) .......5

**Figure 1-3:** Major components of a wind turbine nacelle include mechanisms for adjusting yaw orientation and blade pitch as well as the gearbox and generator. (6) .....6

**Figure 1-4:** Distribution of failures in Swedish wind turbines between 2000 and 2004 (8) ..8

**Figure 1-5:** Distribution of downtime per system in Swedish turbines between 2000 and 2004 (8) .......................................................................................................................9

**Figure 1-6:** The ‘bath-tub’ curve predicts the highest failure rates during burn-in and wear out. (11) ..................................................................................................................12

**Figure 1-7:** Condition-based maintenance balances corrective and schedule-based methods. (11) ..................................................................................................................13

**Figure 1-8:** A simple fault detection algorithm would detect if the wind turbine was producing a power level outside of the normal range for a given wind speed. (14) .......15

**Figure 1-9:** The power spectra of a wind turbine can provide information on internal components: 1) Rotor Frequency 2) Blade-pass Frequency 3) Rotation of the First Gearbox Stage 4) Rotor Blade Eigenfrequency 5) Rotation of the Second Gearbox Stage 6) First Harmonic of Gearbox Second Stage 7) Rotation of the Third Gearbox Stage 8) Electrical Asymmetries 9) Tooth Mesh Frequency of First Gearbox Stage (15) ..................................................................................................................16

**Figure 1-10:** a) Power signal for normal operation, rotor imbalance, and aerodynamic imbalance and b) time series of the p-amplitude for normal operation, rotor imbalance, and aerodynamic imbalance. (16) .........................................................................................18

**Figure 1-11:** a) Spectrum and b) envelope curve spectrum of the vibration signal of a generator bearing with surface pitting (14) .........................................................................................21

**Figure 1-12:** Spectral analysis of gearbox vibration using a) power density spectrum and b) cepstrum of spectral harmonics (19) ..........................................................................................22

**Figure 2-1:** The freestream expands and slows down as it passes through the actuator disk. (20) ........................................................................................................................................26

**Figure 2-2:** At a V2/V1 value of 1/3, the maximum CP value is 0.59. ..................................27

**Figure 2-3:** The blade is analyzed in discrete sections to evaluate turbine performance.......29
Figure 2-4: Power Coefficient is plotted versus Tip Speed Ratio for various pitch angles. (21) ........................................................................................................................................30

Figure 2-5: Local power coefficient is plotted with and without tip loss. The red line indicates the local power coefficient without 3-D effects, and the black line indicates the local power coefficient with 3-D effects. ........................................................................................................................................31

Figure 2-6: The enveloping process includes two stages. First, a band pass filter is applied and then the signal is rectified. A Fourier transform of the rectified signal is then used to identify bearing frequencies. ..................................................................................................................36

Figure 2-7: The cepstrum is found by taking the Fourier transform of the logarithm of the magnitude of the vibration spectrum ........................................................................................................................................38

Figure 3-1: Obstacles on the landscape can create large regions of turbulent flow. As a result, wind turbines should be placed far from any potential obstructions. ......................41

Figure 3-2: The wind turbine is located in an area of highly varied terrain. The light blue lines denote 5 ft elevation changes. Green lines outline groupings of trees. Medlar Field is located in the upper left corner, and is shown in dark blue. (26) .........................41

Figure 3-3: The Whisper 500 system consists of a generator, controller, battery bank, diversion load, and an external brake switch. (26) ..................................................................................................................43

Figure 3-4: The Southwest Whisper 500 is a two-bladed, direct-drive wind turbine that utilizes a passive yaw system. (27) ..................................................................................................................44

Figure 3-5: The power curve for the Whisper 500 is published by Southwest, and rated power is predicted to be 3 kW at a wind speed of 24 mph. (27) .................................................................45

Figure 3-6: Major components of the Whisper 500 generator assembly include the rotor, stator, blade mounting plate, and spindle and bearing assembly. (28) .................................46

Figure 3-7: The three-phase AC current from the generator is rectified to DC using a delta connection in 24-volt battery charging applications. (28) .................................................................47

Figure 3-8: The EZ-Wire controller is tasked with regulating the battery charge and rectifying the AC power from the generator into DC power. (26) .................................................................48

Figure 3-9: The diversion load consists of four 0.75 Ohm resistors that are configured as two sets of parallel resistors wired in series. (26) .........................................................................................50

Figure 3-10: The diversion load resistors are stored in a metal box that allows heat to escape while protecting the resistors from interaction with other equipment. ..................51

Figure 3-11: The Whisper 500 brake switch is installed in a lockable box. ..........................52

Figure 3-12: An accelerometer was mounted to the wind turbine tower to measure vibration produced by the generator. ..................................................................................................55
Figure 3-13: The NIDAQ software was used to collect and save data from the USB-4432 data acquisition system.

Figure 3-14: The NIDAQ Data Quick View program is useful for examining data traces and converting the data to .wav format for analysis.

Figure 3-15: A direct-drive coupler was used to connect the drive motor to the Whisper 500’s generator.

Figure 3-16: The forward accelerometer was mounted to a flange on the leading edge of the armature so as to monitor the forward bearing.

Figure 3-17: The rear accelerometer was mounted behind the windings so as to capture vibrations produced by the rear bearing.

Figure 3-18: A microphone was placed in the rear of the stator to detect noise created by the rear bearing. During the actual test runs, acoustic foam was placed around the microphone stand.

Figure 3-19: The KCF sensor was mounted in a fixture near the rear lab-grade accelerometer so as to monitor the vibrations produced by the rear bearing.

Figure 3-20: The KCF accelerometer node (left) communicates data to the KCF USB receiver (right).

Figure 3-21: KCF’s data acquisition interface allows the range, sample rate, collection axes, and burst interval to be adjusted.

Figure 4-1: Vibration in the Y and Z axes was documented.

Figure 4-2: A steel plate was mounted to the blade plate to create an inertial imbalance.

Figure 4-3: A second counterweight was used to characterize the turbine’s vibration with additional counterweight.

Figure 4-4: A counterweight was mounted on the blade plate to create an inertial imbalance. (28)

Figure 4-5: The counterweight was mounted to the blade plate using the existing blade bolts.

Figure 4-6: The large imbalance was bolted to the blade plate using the existing blade bolts.

Figure 4-7: Phase Voltage data was used to create a plot of the rotation rate over time.

Figure 4-8: RPM_Plot.m outputs a plot of the turbine’s rotation rate over time.

Figure 4-9: Minimum variability in the turbine rotation rate is required for a data section to be useable for analysis.
Figure 4-10: Sections of data where current was being produced for none or part of the three seconds were rejected for analysis.................................................................84

Figure 4-11: Periods of high-amplitude, low-frequency vibration occur when the turbine yaws quickly. Data captured during these events, such as the one between 97 and 100 s, was not used in analysis........................................................................86

Figure 4-12: The FFT_Average.m code inputs a data file and outputs an averaged FFT plot. ......................................................................................................................................................................................88

Figure 4-13: A modified version of the FFT_Average.m code was used to produce FFT plots of the accelerometer output. .............................................................................................................................89

Figure 4-14: Total vibration magnitude is the magnitude of the resultant of the Y and Z vectors........................................................................................................................................................................90

Figure 4-15: Averaged FFT plots of the vibration magnitude were produced using a modified version of the FFT_Average.m code.................................................................91

Figure 4-16: The power and rotor frequency peaks were identified in the Power Spectrum. The AC power signal cycles eight times per revolution of the turbine........93

Figure 4-17: The turbine rotor and blade pass frequencies were identified in the single-channel and vibration magnitude data.................................................................94

Figure 4-18: Power_Processing.m was used to produce plots of the average magnitude of voltage signal at the shaft frequency vs. the shaft frequency. ..............................96

Figure 4-19: Vibration_Processing was used to produce plots of the average magnitude of vibration at the shaft frequency vs. the shaft frequency. .................................98

Figure 4-20: The original rear bearing was removed after the first set of tests, and corrosion of the rear surface of the bearing was noted. ..................................................101

Figure 4-21: The spindle assembly was removed using a slide hammer which was bolted to the spindle using the spindle’s existing bolt holes. This picture was taken after the rear bearing had been removed from the spindle. ...............................................102

Figure 4-22: Prior its reinstallation in the stator, the new bearing was installed on the spindle.................................................................103

Figure 4-23: A fault was seeded in the outer race of the rear bearing by drilling a hole through the bearing’s outer wall. Care was taken to avoid drilling into the roller elements or the cage.................................................................104

Figure 4-24: The McInerney and Dai amplitude demodulation code is provided with a GUI that allows the user to load data, apply a bandpass filter, perform a Hilbert Transform, and plot a spectrum of the transformed data. ..................................................106
**Figure 5-1:** A typical power signal spectrum shows a large peak at eight times the shaft frequency and a small peak at the shaft frequency..........................109

**Figure 5-2:** Spectra of the power signal from the three imbalance cases are compared for 5.4 Hz approximate shaft frequency. .................................................................110

**Figure 5-3:** Spectra of the power signal from the three imbalance cases are compared for a 6.5 Hz approximate shaft frequency.................................................................112

**Figure 5-4:** Average peak magnitude values at the shaft frequency are plotted against a range of shaft frequencies for the three imbalance cases...................................................114

**Figure 5-5:** A typical three-second FFT of the vibration magnitude shows a rotor frequency peak between 3 and 10 Hz depending on the turbine’s operating RPM. ......117

**Figure 5-6:** Spectra of the vibration magnitudes are compared between the three imbalance cases for a 5.4 Hz approximate shaft frequency......................................................118

**Figure 5-7:** Spectra of the vibration magnitudes are compared between the three imbalance cases for a 6.5 Hz approximate shaft frequency......................................................120

**Figure 5-8:** Average vibration magnitude values at the shaft frequency are plotted against a range of shaft frequencies for the three imbalance cases........................................122

**Figure 5-9:** Average vibration magnitude values at the shaft frequency are plotted against a range of shaft frequencies for the three imbalance cases.................................125

**Figure 5-10:** Vibration spectra from the rear accelerometer are compared for the 600 RPM case. ..................................................................................................................128

**Figure 5-11:** Vibration spectra from the forward accelerometer are compared for the 600 RPM case. ..................................................................................................................129

**Figure 5-12:** Examination of the high frequency from the rear accelerometer spectra indicated signs of low frequency modulation at the outer race fault frequency.........131

**Figure 5-13:** Examination of the high frequency spectra from the forward accelerometer indicated signs of low frequency modulation at the outer race fault frequency...........132

**Figure 5-14:** The envelope spectrum for the new bearing displays modulation at the shaft frequency, but no significant activity at the inner or outer race fault frequencies or their harmonics.................................................................133

**Figure 5-15:** The envelope spectrum for the original bearing displays modulation at the shaft frequency, but additional activity was observed at the roller element fault frequency and its harmonics.................................................................134

**Figure 5-16:** The envelope spectrum for the faulted bearing displays significant activity at the outer and inner race fault frequencies and their respective harmonics.................134
Figure 5-17: Outer race fault frequencies are readily identified at a shaft frequency of 6.67 Hz.

Figure 5-18: At 5 Hz shaft frequency, the harmonics of the outer race fault frequency are still visible above the noise floor, but the primary fault frequency is not readily distinguishable from the other structural frequencies.

Figure 5-19: A comparison of the microphone output over the low frequency spectra between the three cases shows little evidence to suggest a significant ability to distinguish between the damaged and undamaged cases.

Figure 5-20: The high frequency spectra produced from the microphone data shows little evidence to suggest significant differences between the damaged and undamaged cases.

Figure 5-21: The envelope spectrum of the microphone data from the faulted bearing case shows minor activity at the inner race fault frequency and its harmonics.

Figure 5-22: The envelope spectrum of the microphone data from the new bearing case shows no activity at the inner race fault frequency or its harmonics.

Figure 5-23: Few significant differences between the three cases were observable in the spectra created from the wireless accelerometer data.
**LIST OF TABLES**

Table 1-1: Gearbox failures are broken down by subcomponent for Swedish Turbines (8)...........................................................................................................................................10

Table 4-1: The steel plates resulted in a change in the rotational inertia of the blade and produced 10.549 N and 21.099 N at 400 rpm respectively........................................................................75

Table 5-1: Shaft Frequency Peak magnitudes are compared for similar rotation rates over the three imbalance cases........................................................................................................111

Table 5-2: Shaft Frequency Peak magnitudes are compared for similar rotation rates over the three imbalance cases........................................................................................................112

Table 5-3: Vibration magnitudes at the shaft frequency are compared for similar rotation rates over the three imbalance cases. ........................................................................................................119

Table 5-4: Vibration magnitudes at the shaft frequency are compared for similar rotation rates over the three imbalance cases. ........................................................................................................120

Table 5-5: Fault frequencies for the 6207-2RS were of particular interest during the analysis........................................................................................................................................127

Table 5-6: The highest observed outer race fault frequency peak magnitude was observed for the faulted bearing case. The original bearing displayed a similar peak magnitude while the new bearing’s observed magnitude was significantly lower. .......128

Table 5-7: Again, the highest observed outer race fault frequency peak magnitude was observed for the faulted bearing case. The original bearing displayed a similar peak magnitude while the new bearing’s observed magnitude was significantly lower. .......130
NOMENCLATURE

\[ a \]
axial flow induction factor

\[ a' \]
tangential flow induction factor

\[ A \]
rotor swept area

\[ A_u, A_w \]
Upstream and downstream stream-tube cross-sectional areas

\[ c \]
blade chord; Weibull scale parameter

\[ C_{\alpha} \]
lift curve slope

\[ C_P \]
power coefficient

\[ C_T \]
thrust coefficient

\[ F \]
force

\[ F_x \]
force in x (downwind) direction

\[ F_y \]
force in y direction

\[ F(\mu) \]
function determining the radial distribution of induced velocity normal to the plane of the rotor

\[ I \]
current

\[ L \]
lift force

\[ N \]
number of blades

\[ p \]
static pressure

\[ P \]
aerodynamic power; electrical real (active) power

\[ r \]
radius of blade element or point on blade; correlation coefficient

\[ R \]
blade tip radius; electrical resistance

\[ T \]
rotor thrust

\[ t \]
time (sec)

\[ X_{TSR} \]
tip speed ratio

\[ \alpha \]
angle of attack – i.e., angle between air flow incident on the blade and the blade chord line; wind-shear power law exponent

\[ \mu \]
non-dimensional radial position, \( r/R \)

\[ \theta \]
phase angle between voltage and current of AC electrical signal

\[ \rho \]
air density

\[ \sigma \]
blade solidity

\[ \omega \]
angular frequency (rad/s)

\[ \Omega \]
rotational speed or rotor
ACKNOWLEDGEMENTS

I would like to begin by thanking my advisors, Dr. Karl Reichard, Dr. Dennis McLaughlin, and Dr. Susan Stewart for their invaluable guidance throughout the course of this research. Each has provided me with a unique set of skills that I am sure will prove essential in my career. I would also like to extend thanks to Brian Wallace, Mark Turner, Nathan Lasut, and Richard Auhl for providing their technical support to this project. Each of these gentlemen devoted their time to assisting the research efforts described in this thesis, and without them, the completion of this project would not have been possible. Additional thanks are extended to K.C.F. Technologies for providing their wireless accelerometer system for use in the dynamometer tests. Finally, I would like to thank the Applied Research Laboratory at Penn State for funding this research through the Exploratory and Fundamental Research Program.
Chapter 1

INTRODUCTION

On November 24, 2008, the country of Spain produced enough power from its wind farms to cover over 40% of its hourly demand. In all of 2008, Spanish wind turbines were able to meet 11.5% of the country’s total energy consumption (1). Wind energy has been utilized by man for over 3000 years in one form or another (2), but these statistics serves to highlight the fact that wind energy is quickly taking its place as a major means for energy production. Wind turbines present an attractive solution to the problem of increased energy demand with the desire for decreased carbon emissions. This growth of importance of wind energy has led to the desire for further advancements in wind turbine technology. Over the past several years, research in wind power has led to larger and more efficient machines, but with this increase in size and complexity has come with an increased cost of repair and maintenance of said machines. This issue is further compounded by the fact that wind turbines are often located in remote areas or even offshore. This problem has led to the development of health monitoring and fault detection systems for wind turbines. Though they vary in implementation, the common goal of these systems is to detect potential faults before they lead to catastrophic failures and extended downtime of the machine. The most basic condition monitoring systems typically employ vibration sensors placed at key locations in the machine that can detect faults that develop within the bearings as well as damage that is present in the gear teeth. Performance monitoring systems can also be employed to detect faults, be they aerodynamic or mechanical in nature. So far, however, this type of comprehensive health monitoring has not been applied to residential-scale
(1 – 10 kW) machines in any great number. The goal of the research described in this paper was to develop such a system for a residential-sized wind turbine.

### 1.1 History of Wind Energy

Because windmills of some type have existed for several thousand years, the origins of the first designs have likely been lost to history. Despite this, one of the first notable documents from ancient history depicting a windmill comes from Heron of Alexandria’s *Pneumatica* from the first century A.D. (3) The more commonly-known horizontal-axis European windmill, however, began to appear during the twelfth century, and by the fourteenth century, they had become commonplace as a source of power in England, France, Belgium, and Holland. (3) Though there is debate as to the nature of the evolution of the horizontal-axis design, it is generally agreed upon that the design became favored over vertical-axis designs due to its increased efficiency. Most of these mills were similar in design, with flat wooden blades and a gear mechanism to transfer power down to the mill. See Figure 1-1.
The classic American windmill appeared during the nineteenth century, and its popularity was driven by the need for a reliable means of water pumping during the growth of agriculture in the American Midwest. The American windmills mirrored the horizontal-axis mills of Europe but also added some crucial improvements. Daniel Halliday, an American mechanic, is credited with introducing the first over-speed protection mechanism to the windmill that helped solve the problem of damage due to high winds. By furling the blades in high winds, the rotational rate of the turbine was regulated, therefore saving it from potential damage. Halliday also added a tail which kept the windmill facing the prevailing winds at all times. The ability to have a machine operate without human intervention was a huge step forward for the progress of wind energy, and Halliday’s mechanisms are still used on residential-scale wind turbines today.

During the late nineteenth and early twentieth century, wind turbines began to take on a new role as electrical power generators. As with the water-pumping windmills, the electrical power-producing turbines found uses in remote agricultural applications. These variable-speed
turbines were typically pushed into service to generate DC power that was used to charge batteries. It was this era of wind turbine development that saw the first popular residential-scale turbines. Between the 1920’s and 1950’s, Marcellus and Joseph Jacobs sold thousands of affordable residential-scale wind turbines mostly to those who had yet to be connected to an AC power grid. With the eventual growth of the national power grid, variable-speed DC turbines declined in popularity. (5)

One of the first notable attempts at building a wind turbine that could connect to an AC grid came in 1941. The Smith-Putnam wind turbine produced a remarkable 1250 kW and boasted a 174 ft blade diameter. More significantly, the turbine solved the problem of grid connection by fixing the rotational rate of the rotor through the use of variable-pitch blades. An active yaw system and flapping blades meant that the Smith-Putnam wind turbine was far ahead of its time, and a comparable machine would not be built until the mid-1970’s. (2) Following the start of the energy crisis in 1973, the U.S. Federal Wind Energy Program was created as part of the Department of Energy. This agency was the driving force behind a number of experiments and construction projects related to wind energy in the United States. (6) For example, the MOD family of turbines, constructed during the 1980’s, were used as test beds to facilitate advances in the modeling of wind turbine aerodynamics. (7) See Figure 1-2. Throughout the late 1970’s and 1980’s, wind farms began to spring up across the U.S., especially in California where tax incentives for renewable energy were plentiful.
Figure 1-2: The MOD-2 was used in the investigation of wind turbine dynamics. (6)

Today, wind farms have spread across the United States, and according to the Department of Energy, the U.S. has installed 40,180 MW of wind energy as of 2010, accounting for 2.9% of the country’s total power production. By 2030, the U.S. plans to reach 20% of total demand through the use of wind farms. Most wind turbines used for power generation in the U.S. range from between 1 and 3 MW in peak output with the average output for a machine installed in 2010 being 1.79 MW (1).

Industrial-size wind turbines have largely converged on a common fundamental design in recent years. Beginning with the basics, these turbines are almost always of the horizontal-axis variety. A generic industrial wind turbine typically has three long, slender blades that connect to nacelle which sits atop a monopole tower. Most feed power to a grid rather than battery storage,
and as a result, almost all of these turbines include mechanisms for altering the pitch of the blades in order to maintain a relatively constant rotational rate. A gearbox housed within the nacelle regulates the rotation to about 10 - 20 rpm, and wind direction sensors together with an active yaw system keep the nacelle pointing toward the wind. See Figure 1-3. The blade diameter of a modern wind turbine can exceed 100 m with tower heights in excess of 100 m. (1)

![Diagram](image)

**Figure 1-3:** Major components of a wind turbine nacelle include mechanisms for adjusting yaw orientation and blade pitch as well as the gearbox and generator. (6)

Though industrial-sized machines dominate the public view in the area of wind energy, residential-sized machines have also been gaining traction. These small systems are typically purchased and installed by private entities in order to offset energy consumption. Residential turbines are typically rated to produce a peak output between 1 and 200 kW and have blade diameters ranging between 3 and 50 ft. While industrial-sized turbines have largely converged...
on a common configuration, residential-sized machines still vary greatly in design, employing both upwind and downwind configurations and anywhere from 2 to 6 blades. In general, residential-sized systems take a simpler approach to power production, typically forgoing gearboxes, active yaw control, and active pitch control systems in favor of direct drive, passive yaw, and fixed pitch designs.

1.2 Wind Turbine Failures

The modern wind turbine is a complex machine with a myriad of systems and subsystems, and with that complexity comes the potential for failure. Because of the high cost of maintenance and repair of wind turbines, efforts have been put forward to catalogue the frequency of system failures within the machines. By pinpointing the areas most prone to failure, an effective condition monitoring system can be developed to catch faults before they worsen. A survey carried out by Johan Ribrant of KTH School of Electrical Engineering presents failure statistics for Swedish, Finish, and German wind turbines between 2000 and 2004 (8).
Figure 1-4: Distribution of failures in Swedish wind turbines between 2000 and 2004 (8)

Figure 1-4 shows the distribution of failures in Swedish turbines between 2000 and 2004. Note that most failures occur within the electrical and sensor systems. Figure 1-5 presents the distribution of downtime per failure type in the same turbines. It is interesting to note that while failures in the gear system only accounted for 9.8% of failures, they accounted for a larger share
of the downtime than any other system.

\[ \text{Figure 1-5: Distribution of downtime per system in Swedish turbines between 2000 and 2004 (8)} \]

Ribrant goes on to highlight the fact that replacement of a gearbox is a particularly time-consuming affair, requiring the turbine to be shut down for a period of 250 to 290 hrs on average. Within the category of gearbox failures, subcomponent failure statistics are provided for the Swedish turbines. See Table 1-1. The two columns on the far right are failures which are explicitly specified as failures due to wear.
Table 1-1: Gearbox failures are broken down by subcomponent for Swedish Turbines (8)

<table>
<thead>
<tr>
<th>Type of reported failure code</th>
<th>Component</th>
<th>Number of failures [n]</th>
<th>Average downtime [h] (min-max)</th>
<th>Number of failures, Cause: B1 [n]</th>
<th>Average downtime, Cause: B1 [h]</th>
</tr>
</thead>
<tbody>
<tr>
<td>I-1</td>
<td>Bearings</td>
<td>41</td>
<td>562 (15-2067)</td>
<td>36</td>
<td>601</td>
</tr>
<tr>
<td>I-2</td>
<td>Gearwheels</td>
<td>3</td>
<td>272 (57-383)</td>
<td>2</td>
<td>379</td>
</tr>
<tr>
<td>I-3</td>
<td>Shaft</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>I-4</td>
<td>Sealing</td>
<td>8</td>
<td>52 (2-218)</td>
<td>4</td>
<td>30</td>
</tr>
<tr>
<td>I-5</td>
<td>Oil system</td>
<td>13</td>
<td>26 (1-63)</td>
<td>5</td>
<td>36</td>
</tr>
<tr>
<td>I-other</td>
<td>Not specified</td>
<td>44</td>
<td>230 (9-1248)</td>
<td>19</td>
<td>299</td>
</tr>
</tbody>
</table>

It is clear from the data provided that bearing failure is a major contributor to gearbox failure and subsequent downtime of Swedish wind turbines during the survey period.

Further evidence of the wind industry’s concern with gearbox failures is found in the formation of the National Renewable Energy Lab’s Gearbox Reliability Collaborative. Musial and Butterfield’s paper entitled “Improving Wind Turbine Gearbox Reliability” (9) underscores the fact that the higher-than-expected rate of gearbox failures is adding to the overall cost of wind energy. Not only are gearbox replacement costs being passed on to customers, but the risk of gearbox failure has also led to increases in contingency funds of both the manufacturers and operators. In the same paper, the authors put forth a number of general observations about wind turbine gearbox failures that mirror Ribrant’s findings closely. It is noted that a majority of gearbox failures initiate within the bearings even though said bearings have been designed using the most advanced methods available. Also important is the fact that flaws in the manufacturing process were only found to be the cause of the fault in about 10% of gearbox failures, suggesting that deficiencies in other areas are likely to blame. (9) These areas could include, but are not limited to deficiencies in the installation, materials, and design of the bearings.
Unfortunately, there has been little investigation into the failures of small-scale residential wind turbines. Since they are generally simpler machines that typically lack gearboxes and active pitch and yaw control systems, these systems are generally thought to be more reliable than their large-scale counterparts. Paul Gipe, author of *Wind Power: Renewable Energy for Home, Farm, and Business* warns potential operators of small wind turbines that their machines will still require regular maintenance. Anecdotally, he mentions the case of a direct-drive turbine that suffered bearing failure and debonding of the permanent magnets from the rotor. He also encourages owners to check that all of the blades are uniform. Anomalies caused by missing blade material or foreign matter on the blade can cause serious damage to the system. (10)

1.3 Health Monitoring of Wind Turbines

Determining the optimal time to replace components has consistently been a challenge of engineers and mechanics. In many cases, the probability of failure of a part can be expressed by a ‘bath tub’ curve where the likelihood of failure is highest during the initial burn-in period and when the part begins to reach the end of its operational lifespan. See Figure 1-6. Failures during the burn-in period are generally due to defects in manufacturing or incorrect installation while failures late in the life cycle are attributed to wear.
The ‘bath-tub’ curve predicts the highest failure rates during burn-in and wear out. (11)

The challenge of the design engineer and operator of the machine is to determine the correct time to replace the part. Performing corrective maintenance after the part has failed maximizes the life of the part, but the unexpected failure of the part could lead to extended downtime and damage to other components in the machine. A second alternative is regularly scheduled preventative maintenance. This approach calls for maintenance at fixed intervals during the life of the machine. While scheduled maintenance is useful for preventing costly breakdowns, it usually requires parts to be replaced before they have reached the end of their useful life and require increased system downtime. Condition-based monitoring (CBM) is a third alternative that seeks to balance corrective and scheduled maintenance. Under CBM, specific performance parameters within the machine are monitored. When these parameters cross a pre-set threshold, the part is replaced. By directly monitoring components within the system, parts can be utilized for more of their useful life, but they can also be replaced before they fail completely. (11) See Figure 1-7.
Figure 1-7: Condition-based maintenance balances corrective and schedule-based methods. (11)

Though the methods of damage detection vary from system to system, the overarching goals of health monitoring can be qualified into four separate levels. The first goal is to determine that damage is present within the machine. The second goal is to determine the exact location of damage within the machine, and the third goal is to quantify the severity of the damage. The fourth and final step is to predict the remaining life of the faulty component. (12)

Specific methods for condition-based monitoring vary greatly depending on the system or subsystem that is being monitored, but a number of methods lend themselves well to wind turbines gearboxes:

1) **Electrical Performance Monitoring:** One of the most direct methods of determining the presence of faults in a wind turbine is to monitor its performance over a period of time. The relationship between wind speed and power output can be observed, and the operator can determine if the wind turbine is under or overperforming. Furthermore, the
voltage and current signals from the turbine generator can be directly observed and analyzed. Because of their effectiveness and ease of implementation, these methods will be discussed further in section 1.4.

2) **Acoustic Monitoring:** Microphones and acoustic sensors can be applied in wind turbine condition monitoring to detect anomalies in rotating components such as gearboxes and bearings. These are generally used in conjunction with vibration sensors.

3) **Vibration Sensors:** Vibration sensors are commonly employed in condition monitoring of rotating machinery, and are often used to keep tabs on wind turbine gearbox and bearing health. (13) Because of their usefulness in monitoring these high-trouble areas, vibration-based methods are discussed further in section 1.5.

### 1.4 Performance-based Methods of Health Monitoring

Perhaps the most straightforward method of health monitoring of a wind turbine is accomplished through monitoring the performance of the machine. The purpose of a wind turbine is electrical power production, and most manufacturers of residential wind turbines provide a plot of their turbine’s output with respect to the measured wind speed. If the turbine is properly sited, the machine’s performance should theoretically match these output figures. Caselitz applies the simple method of monitoring the average power output of a 600 kW wind turbine for given wind speeds. See Figure 1-8. A condition monitoring system would generate a warning if a data point fell outside of the dotted line and a full alarm if a data point fell outside of the solid line.
A simple fault detection algorithm would detect if the wind turbine was producing a power level outside of the normal range for a given wind speed. (14)

While direct performance monitoring can be useful for the detection of major problems within a wind turbine system, more specific information on the health of the machine can be obtained through analysis of the signals generated by power output. Furthermore, it is more attractive from an operator’s standpoint to detect problems before they have an affect on wind turbine performance. Examinations of the spectra of the power output can provide a surprising amount of information about individual components within the system. Caselitz provides the spectrum of electrical power output of an Aeroman wind turbine under partial load.
Figure 1-9: The power spectra of a wind turbine can provide information on internal components: 1) Rotor Frequency  2) Blade-pass Frequency  3) Rotation of the First Gearbox Stage  4) Rotor Blade Eigenfrequency  5) Rotation of the Second Gearbox Stage  6) First Harmonic of Gearbox Second Stage  7) Rotation of the Third Gearbox Stage  8) Electrical Asymmetries  9) Tooth Mesh Frequency of First Gearbox Stage  (15)

In a later paper, Caselitz was able to use the signal generated by the power production of the Aeroman wind turbine and detect a pair of induced faults in the machine. The first of these faults was a simple rotor imbalance produced by attaching a weight to one of the blades. The second fault was an aerodynamic asymmetry produced via the application of tape to one of the blades. First, the spectra of the power output signals produced during normal operation, rotor imbalance, and aerodynamic imbalance were compared. See Figure 1-9. Each of these faults
produced a notable increase in the p-amplitude ($f_{\text{rotor}} = 1.4$ Hz) over the amplitude recorded for normal operation.

**Figure 1-9:** The spectra of the electric power output for a) normal operation b) rotor imbalance, and c) aerodynamic imbalance. (16)

Caselitz concluded that the spectrum of the power output was an effective tool for determining imbalance in the rotor system, but he noted that this type of analysis was not necessarily as effective for determining the type of imbalance present. To solve the problem of fault identification, Caselitz examined the time series of the p-amplitude. It was found that when compared with normal operation, a rotor imbalance was identified by large fluctuations in the p-
amplitude, and an aerodynamic asymmetry was identified by an increased but stable value for p-amplitude. See Figure 1-10.

Figure 1-10: a) Power signal for normal operation, rotor imbalance, and aerodynamic imbalance and b) time series of the p-amplitude for normal operation, rotor imbalance, and aerodynamic imbalance. (16)

1.5 Vibration-based Methods of Health Monitoring

Vibration methods are widely used in the health monitoring of rotating machinery including wind turbine gearboxes and bearings. This method is a multi-step process that, when applied correctly, can present an effective solution to the problem of fault detection. Farrar and Duffey (17) provide a helpful breakdown of the damage detection process, beginning with an operational evaluation. The goal of this evaluation is to answer a series of questions about the machine that is to be monitored:

1) How is damage to the system defined?

2) Under what operational and/or environmental conditions will the system operate?

3) What limitations are imposed by the environment on data collection?
Once the parameters of the system have been defined, the next step is data acquisition and cleansing. During this step in the process, the types and locations of sensors are defined along with the rate, transmission, and storage of data. Furthermore, normalization of data should be considered with respect to environmental or operation factors. Farrar and Duffey note that sources of variability should also be identified and eliminated where possible. In vibration-based monitoring, data acquisition is typically carried out by accelerometers. Accelerometers typically enjoy a broad frequency range and Farrar notes that they are particularly useful for monitoring roller bearings and gear trains. (17)

The third and arguably most important step in vibration-based condition monitoring is the identification of data features which separate a damaged component from one that is operating within its normal range. One of the most popular and useful tools for feature extraction and analysis is simply past experience. By applying engineered flaws similar to those experienced in actual operation to a mechanical system and measuring the response, important data features can be identified. By comparing the responses to known forms of damage, the engineer or operator can then match the effects of damage during operation to the damage applied during the simulation.

Time domain methods are the most basic vehicles for vibration monitoring because they often require little transformation of the raw data. These methods are useful for detecting singular events such as a system shock or the removal of a piece of material from the contact surface of a bearing. The drawback to monitoring the time domain is that all of the data must be examined because singular events will not manifest in the observed trends. Other methods of analysis include monitoring of the RMS amplitude, peak amplitude, and kurtosis analysis. Frequency domain methods are probably one of the most popular methods used to detect
damage. One of the most important qualitative features for analyzing the health of rotating machinery is the presence of peaks in the acceleration spectra. These peaks typically occur at multiples of the rotational frequency and tracking their growth and/or movement over time is an effective method of damage identification. (18) Further analysis can be completed using envelope curve methods which analyze the modulation of high frequency oscillations by low frequency excitations that are caused by faults in the system. Caselitz successfully used envelope curve analysis to identify a faulty bearing within the turbine. See Figure 1-11. The large peak in b) at 103 Hz is caused by the impulses emitted from balls in the bearing passing over a pitted area in the bearing face. This peak is less noticeable in the simple spectrum shown in a), but it is clearly dominant in the envelope curve spectrum given in b). (16)
Figure 1-11: a) Spectrum and b) envelope curve spectrum of the vibration signal of a generator bearing with surface pitting (14)

In a later paper, “Applications of Condition Monitoring Systems in Wind Energy Converters,” Caselitz goes on to highlight the usefulness of cepstrum analysis. Cepstrum is the inverse Fourier transform of the logarithmic power spectrum, and it is useful for identifying periodic behavior of the power spectrum. (19) Caselitz was able to use cepstrum analysis to identify the strong harmonics produced by a defect in the planetary stage of a wind turbine gearbox, and the usefulness of this method is compounded by the fact that monitoring of only one fault frequency is required. See Figure 1-12.
Figure 1-12: Spectral analysis of gearbox vibration using a) power density spectrum and b) cepstrum of spectral harmonics (19)

1.6 Scope and Objectives

Section 1.2 contains a review of documented wind turbine failures. Though the majority of documentation has focused around the costly failures of industrial-scale turbines, some attention has been paid to the maintenance of small-scale systems. Paul Gipe, author of Wind Power, Renewable Energy for Home, Farm, and Business uses his extensive experience to highlight potential trouble spots in residential wind turbines. He advises owners and operators to
periodically check their machines for asymmetries among the blades, suggesting that missing blades sections and/or foreign matter on a rotor blade can both pose a serious problem. In addition, he mentions that bearing replacement may be required as part of regular maintenance. (10) As is the case with their large-scale cousins, bearing failure can also be a source of concern for operators as it can lead to costly downtime and extensive repair. In light of this information, a series of goals were set for the research presented in this thesis.

The first goal in the development of a fault detection system for a small wind turbine should center on the detection of inertial blade imbalance as this problem can lead to rapid degradation of other components within the machine. A serious imbalance could even threaten the structure of the tower, so catching such an event in its early stages is paramount. Sections of the following chapters are devoted to discussing the theory, facilities, activities, and results relevant to an inertial imbalance detection methodology for the Whisper 500 turbine.

A second SHM goal for a small wind turbine is the detection of bearing failure. Not only is bearing failure in small wind turbines an issue, but advances in the detection of such failures could have applications for industrial-sized machines as well. For the purposes of this thesis, a dynamometer facility was constructed with the express purpose of aiding in the development of a bearing fault detection system. The theory, facilities, procedures, and results related to these bearing fault detection experiments are also discussed in the following chapters.
Chapter 2

THEORY

2.1 Introduction

Condition monitoring of a wind turbine relies heavily on the ability to acquire, process, and present data in a way that is relevant and understandable for the operator. The theory behind both performance and vibration-based methods will be discussed in this section. Section 1.4 discussed past efforts that have been put into condition monitoring through the use of performance-based methods. This requires an understanding of the theory behind wind turbine power production and the guidelines for acquiring and processing said data. Section 1.5 discussed a number of vibration-based methods for fault detection. A similar understanding is required of the principles of oscillating motion and the methods for gathering and processing of vibration data.
2.2 Wind Turbine Performance

2.2.1 Power in the Wind

If one is to consider the concept of harvesting power from a resource, one must first consider the nature of the resource itself. The power contained within the wind is given as a function of the mass flow, \( \dot{m} \), through the rotor disk.

\[
\dot{m} = \rho AV_{\infty}
\]

Equation 2-1

Power contained within the wind is then given by:

\[
P = \frac{1}{2} \dot{m} |V_{\infty}|^2 = \frac{1}{2} \rho A |V_{\infty}|^3
\]

Equation 2-2

Equation 2-2 serves to underscore the importance of wind turbine siting since power varies with the cube of wind speed. It is also important to note that the power contained within the wind differs from the actual power extracted from the wind by a turbine. (20) This concept will be explained further with the derivation of the Betz limit.

2.2.2 Actuator Disk Theory

The overriding premise of momentum theory is that mass flow must be maintained across the rotor disk. Since the velocity downstream of the turbine, \( V_2 \), is lower than that of the freestream, \( V_1 \), the area of the stream downstream of the turbine, \( A_2 \), must be proportionally larger than the area upstream, \( A_1 \). (6) See Figure 2-1.
Figure 2-1: The freestream expands and slows down as it passes through the actuator disk. (20)

Power through the actuator disk can then be written as:

\[ P = \frac{1}{2} \dot{m}(V_1^2 - V_2^2) \tag{2-3} \]

Substituting mass flow in terms of rotor disk area into equation 2-2 yields:

\[ P = \frac{1}{2} \rho A(V_1^2 - V_2^2)(V_1 + V_2) \tag{2-4} \]

In order to find the power coefficient, \( C_P \), of a wind turbine, the power must be non-dimensionalized in terms of a reference power value, \( P_0 \). In this case, \( P_0 \) is the total amount of power contained in the wind stream. See Equation 2-2. The expression for \( C_P \) is given in equation 2-5 (6).

\[ C_P = \frac{\frac{1}{2} \rho A(V_1^2 - V_2^2)(V_1 + V_2)}{\frac{1}{2} \rho AV_1^3} \tag{2-5} \]
2.2.3 The Betz Limit

For a wind turbine to extract all of the available power from a given stream of wind, the velocity of the wake, $V_2$, would have to fall to zero. Since this is impossible, it follows that some theoretical maximum must exist for the power coefficient. This limit is known as the Betz limit, and it can be illustrated by plotting the power coefficient with respect to $\left(\frac{V_2}{V_1}\right)$. Rearranging equation 2-5 and writing it in terms of $\left(\frac{V_2}{V_1}\right)$ gives yields:

$$C_p = \frac{1}{2} \left[ 1 - \left(\frac{V_2}{V_1}\right)^2 \right] \left[ 1 + \left(\frac{V_2}{V_1}\right) \right]$$

Plotting equation 2-6 for $\left(\frac{V_2}{V_1}\right)$ values ranging from 0 to 0.5 yields Figure 2-2:

**Figure 2-2:** At a $V_2/V_1$ value of 1/3, the maximum CP value is 0.59.
A maximum power coefficient value of 0.59 is obtained meaning that a wind turbine may only extract a maximum of 59% of the power contained within a given wind stream. This limit of efficiency is known as the Betz limit.

2.2.4 Blade Element Momentum Theory

Blade Element Momentum Theory (BEMT) for a wind turbine is very closely related to that of a typical rotorcraft. By calculating the forces over spanwise sections of the rotor blade and summing those forces, the performance of a wind turbine with a given geometry can be quantified. Like BEMT for a helicopter rotor, it is assumed that each blade section acts as a 2-D airfoil and that there is no flow in the radial direction. See Figure 2-3.
Figure 2-3: The blade is analyzed in discrete sections to evaluate turbine performance.

For each blade section, an induction factor, $a$, can be found:

$$a(r, X_{TSR}) = -\sqrt{\left(\frac{\sigma X_{TSR} c_i a}{16} + \frac{1}{2}\right)^2 - \frac{\sigma X_{TSR} c_i a (X_{TSR} \theta r + 1)}{16} + \left(\frac{\sigma X_{TSR} c_i a}{16} + \frac{1}{2}\right)}$$

2-7

where $X_{TSR} = \frac{\Omega y}{V_\infty}$ is known as the tip speed ratio. The thrust and power coefficients are found using this induction factor:
When comparing the performance of different wind turbines, power coefficient values are typically plotted as a function of tip speed ratio. For example, various values of $\Theta_{tip}$ are compared in Figure 2-4.

\[
C_T = 8 \int_0^1 (1 - a) \ar dr
\]

\[
C_P = C_T (1 - a) + \frac{\alpha X_{TSR}}{4} C_{do}
\]

As with BEMT for rotorcraft, BEMT for a wind turbine also includes provisions for tip loss effects. One of the most popular techniques is Prandtl’s correction factor which can account
for the number of blades, blade taper, and blade twist. As is the case with rotorcraft, momentum theory analysis predicts the maximum local power coefficient at the blade tips. In actuality, 3-D effects at the tips mean that the local power coefficient drops to zero close to the tips. In Figure 2-5, this local power coefficient is plotted over a blade with and without Prandtl’s tip loss factor.

(20)

![Thrust Gradient vs Radian Position With and Without Tip Loss Effects](image)

**Figure 2-5:** Local power coefficient is plotted with and without tip loss. The red line indicates the local power coefficient without 3-D effects, and the black line indicates the local power coefficient with 3-D effects.
2.3 Wind Turbine Performance Monitoring

2.3.1 Long-Term Observation

The methods for monitoring the performance of a wind turbine vary greatly depending on the goals of such monitoring. Most owners and operators are interested in long-term power production which heavily influences return on investment. Though most methods of health monitoring rely on more direct forms of fault detection, a system based on simple adherence to a pre-determined power output expectation does have merit, and such a setup is discussed in Section 1.4. NREL’s methods for acquiring and processing data are outlined in van Dam and Meador’s report on the performance of the Bergey Excel-S/60. Their method calls for gathering data on power output and wind speed at an acquisition rate of 1 Hz over a long period of operation. Specifically, van Dam and Meador’s report included data for 241.8 hours of wind turbine operation taken over a period of 20 days. Power output and meteorological data was averaged over 10 minute blocks of time, and wind speed data was calibrated based on barometric pressure, air density, and temperature at the site. Equation 2-10 gives wind speed as a function of a calibration factor, Γ, which is calculated based on the averaged meteorological data. (22)

\[ V_{\text{Turb}} = \Gamma_{\text{Site}} V_{\text{Met}} \]  

2-10

2.3.2 Short Term Methods

One of the most useful and direct methods of health monitoring via performance data is the examination of the spectra of the power output. Past efforts in this area are discussed in
Section 1.4. By looking at the frequency content of the power signal, various features of the signal can be linked to the periodic motion of components of the machine. Health monitoring through performance data is particularly advantageous because it does not typically require additional sensors beyond those already employed to track the machine’s performance. To gather sufficient data to perform spectral analysis, data on the power output of the machine needs to be gathered at an acquisition rate that is twice that of the highest frequency of interest. It is possible to calculate the $n^{th}$ Fourier coefficient for $rT_p$ seconds where $T_p$ is the period of the signal being captured. The $n^{th}$ coefficient is then given by equation 2-11 (23).

$$c_n = \frac{1}{rT_p} \int_0^{rT_p} x(t) e^{-\frac{2\pi nt}{rT_p}} \, dt$$

By observing the growth of peaks in the frequency domain over time, an operator can determine whether faults are present in a given component. Two of the simplest faults experienced by a small wind turbine are an inertial rotor imbalance and an aerodynamic rotor imbalance, both of which would likely be given by an increase in the peak magnitude at the shaft frequency of the turbine.

2.4 Vibration Monitoring of Wind Turbines

2.4.1 Basic Vibration Spectrum

Vibration is defined generally as the periodic motion about an equilibrium position, and it can be observed in almost every type of rotating machinery. Common causes include imbalance, shaft misalignment, component looseness, and component defects. The key to the diagnosis of
faults within machinery is the ability to correlate the frequencies of observed vibrations to various components and faults in the machine. The previous section briefly discussed the basic theory behind transferring a signal from the time to frequency domain by use of a Fourier transform. This type of transform is also used heavily in vibration monitoring. The most basic failure modes for a residential turbine, rotor imbalance, would likely be evidenced by an increase in vibration at the same frequency as the rotation rate of the machine. For a machine that rotates at a constant rate, the identification procedure is fairly straightforward. Unfortunately, the rotation rate of most residential turbines varies with wind speed, and this presents a challenge in the analysis phase. It is therefore important to perform spectral analysis on data taken that while the machine’s rotation rate is varying as little as possible.

2.4.2 Envelope Curve Analysis

An important tool for roller bearing fault detection is envelope curve analysis or amplitude demodulation which can be used to extract modulating components from a vibration signal. Faults in the race of a roller-element bearing will typically produce burst signals at a frequency depending on the rotation rate of the machine and the geometry of the bearing. See Section 1.5. Unfortunately, these bursts are difficult to detect because they are damped quickly, and therefore they are not always detectable by basic Fourier analysis. It is possible to detect bearing faults, however, by examining the modulation of high frequency vibrations by low frequency excitations. The first step in detecting bearing faults using enveloping is usually to apply a band pass filter. This will serve to separate the high-frequency bearing signals from the lower frequency vibrations of the rest of the machine. Next, the filtered signal is sent through a
rectifier which smoothes the high-frequency pulses to facilitate extraction of their repetition rate.

In the case of the analysis presented in Chapters 4 and 5, a Hilbert Transform was employed to rectify the filtered data. The Hilbert Transform is best understood in the context of the analytic signal which is formed from a real and imaginary signal. The real signal is the original signal \( x(t) \), and the imaginary signal \( \hat{x}(t) \) is the Hilbert Transform of the original signal. The analytic signal is given by:

\[
a_x(t) = x(t) + j\hat{x}(t)
\]

The analytic signal can also be expressed as

\[
a_x(t) = A_x(t)e^{j\phi_x(t)}
\]

where

\[
A_x(t) = \sqrt{x^2(t) + \hat{x}^2(t)}
\]

is the magnitude of the analytic signal and the envelope of the real signal. Finally, a Fourier transform is performed on the rectified signal to extract the frequencies and harmonics produced by the bearing fault. See Figure 2-6.
Figure 2-6: The enveloping process includes two stages. First, a band pass filter is applied and then the signal is rectified. A Fourier transform of the rectified signal is then used to identify bearing frequencies.
2.4.3 Cepstrum Analysis

A second method that can be quite effective for bearing fault detection is cepstrum analysis. It is particularly useful for determining periodic harmonics within the frequency spectrum caused by faults in gear teeth and is, at its most basic level, the Fourier transform of the logarithm of the vibration spectrum. The real cepstrum, as opposed to the complex cepstrum, is given by Equation 2-12.

$$c_x = \frac{1}{2\pi} \int_{-\pi}^{\pi} \log |X(e^{i\omega})| e^{i\omega n} d\omega$$

Rao points out that the advantage of cepstrum analysis is that periodic harmonics can be detected even when they are shrouded in background noise. (24) The cepstrum is plotted in the quefrency domain and has a gamitude which is analogous to the magnitude of usual spectral components. The gamitude is given in decibels. As a general rule, peaks in the cepstrum will grow in gamitude as bearing wear increases over time. In light of this observation, it may be useful to monitor the gamitude of cepstral components over time so that changes from the signature values can be effectively observed and identified. This technique is comparable to the
observation of the growth of peaks in the vibration spectrum over time. (25) See Figure 2-7.

Figure 2-7: The cepstrum is found by taking the Fourier transform of the logarithm of the magnitude of the vibration spectrum.
Chapter 3
FACILITIES

3.1 Introduction

The development of a health monitoring system for a wind turbine is greatly facilitated
by a wind turbine on which testing can be carried out. Fortunately, the Penn State University
Aerospace Department has been in possession of a Whisper 500 residential wind turbine since
the Fall of 2008. The wind turbine was initially installed by the Penn State Center for
Sustainability in 2005 to support an off-grid living experiment. After the experiment was
concluded, the turbine fell into disrepair and was handed over to the Aerospace Department for
restoration. As part of the restoration, the turbine was lowered and the generator nacelle was
removed for repair. The controller, battery bank, and diversion load were also removed from the
site during the restoration process. The wind turbine restoration covered almost every
subsystem. First, the generator’s rotor and blade plate were replaced, and the copper wire on the
stator was re-sealed. The diodes that form an integral part of the rectifying circuit in the
controller were replaced, along with the metal-oxide-semiconductor field-effect transistors
(MOSFETs) which control the diversion load circuit. Finally, the batteries were replaced, and
the turbine was reinstalled at the site during the spring of 2009.

The Whisper 500 wind turbine presents an excellent opportunity for testing of a wind
turbine health monitoring system for a number of reasons. While it would certainly be possible
to develop a standalone system in a laboratory, building a health monitoring system for use on an
actual wind turbine provides an opportunity for direct testing of said monitoring system. Results
gained from the condition monitoring tests are then directly applicable to a similar machine. A second advantage of using Penn State’s Whisper 500 turbine is that faults can easily be introduced and removed from the system. Because the machine is no longer being relied upon for power generation, lowering the machine, introducing faults, and raising the machine would not pose an inconvenience to any third parties. Furthermore, transferring the generator to the laboratory for additional tests would also be a possibility.

3.2 Whisper 500 Wind Turbine Site

One of the most important factors in determining the performance of any wind turbine is the site at which it is installed. The abundance of the wind resource and the proximity of obstacles have almost complete control over the power output a turbine system. Penn State’s Whisper 500 wind turbine is located on a remote part of campus that is adjacent to Medlar Field. Unfortunately, this location does not necessarily lend itself well to a wind turbine installation. In his book, Wind Power: Renewable Energy for Home, Farm, and Business, Paul Gipe recommends that the entire rotor disk should be placed at least 30 feet above an obstruction that is located within 300 feet of the site. This is to ensure that the wind turbine is safely above the region of highly turbulent flow created by obstructions. (10) See Figure 3-1. A site evaluation was conducted by Brian Wallace, and the results are given in Table 3-1. The current Whisper 500 site fails all but one criterion. A map of the area surrounding the Whisper 500 site highlighting a number of the obstructions is given in Figure 3-2.
Figure 3-1: Obstacles on the landscape can create large regions of turbulent flow. As a result, wind turbines should be placed far from any potential obstructions.

Figure 3-2: The wind turbine is located in an area of highly varied terrain. The light blue lines denote 5 ft elevation changes. Green lines outline groupings of trees. Medlar Field is located in the upper left corner, and is shown in dark blue. (26)
<table>
<thead>
<tr>
<th><strong>Criterion</strong></th>
<th><strong>Description</strong></th>
<th><strong>Distance</strong></th>
<th><strong>Sector</strong></th>
<th><strong>Test Site Condition</strong></th>
<th><strong>Pass/Fail</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Maximum slope of best fit plane &lt; 3%</td>
<td>&lt; 2L</td>
<td>360</td>
<td>14.9%</td>
<td>Fail</td>
</tr>
<tr>
<td>2</td>
<td>Maximum variation from best fit plane &lt; 0.08 D</td>
<td>&lt; 2L</td>
<td>360</td>
<td>--</td>
<td>Fail</td>
</tr>
<tr>
<td>3</td>
<td>Maximum slope of best fit plane &lt; 5%</td>
<td>2-4L</td>
<td>Inside prel. meas. sector</td>
<td>13 %</td>
<td>Fail</td>
</tr>
<tr>
<td>4</td>
<td>Maximum variation from best fit plane &lt; 0.15 D</td>
<td>2-4L</td>
<td>Inside prel. meas. sector</td>
<td>--</td>
<td>Fail</td>
</tr>
<tr>
<td>5</td>
<td>Steepest slope maximum &lt; 10%</td>
<td>2-4L</td>
<td>Inside prel. meas. sector</td>
<td>See Crit. 3</td>
<td>Fail</td>
</tr>
<tr>
<td>6</td>
<td>Maximum slope of best fit plane &lt; 10%</td>
<td>4-8L</td>
<td>Inside prel. meas. sector</td>
<td>13%</td>
<td>Fail</td>
</tr>
<tr>
<td>7</td>
<td>Maximum variation from best fit plane &lt; 0.15 D</td>
<td>4-8L</td>
<td>Inside prel. meas. sector</td>
<td>--</td>
<td>Fail</td>
</tr>
<tr>
<td>8</td>
<td>No neighboring and operating turbines</td>
<td>&lt; 2D&lt;sub&gt;n&lt;/sub&gt;</td>
<td>360</td>
<td>none</td>
<td>Pass</td>
</tr>
<tr>
<td>9</td>
<td>No obstacles</td>
<td>&lt; 2D&lt;sub&gt;e&lt;/sub&gt;</td>
<td>360</td>
<td>1.66 (Obs. 1)</td>
<td>Fail</td>
</tr>
<tr>
<td>10</td>
<td>Preliminary measurement sector within available measurement sector</td>
<td>n/a</td>
<td>n/a</td>
<td>No (Obs. 3)</td>
<td>Fail</td>
</tr>
</tbody>
</table>

*L is distance between the turbine and the meteorological tower (11.25 m @ location B [Figure 3-2])

D<sub>n</sub> is the rotor diameter of a neighboring turbine (n/a for test site)

D<sub>e</sub> is the equivalent rotor diameter of obstacles (defined here as obstacle height)

**Table 3-1:** A site evaluation was conducted based on existing elevation data on the Center for Sustainability site. (26)

### 3.3 Whisper 500 System

The Southwest Whisper 500 wind turbine is a residential-sized wind turbine designed to be operated by a home or small business with the aim of offsetting overall power consumption. The machine can be configured for either grid-tie or battery charging applications. Penn State’s machine was originally set up for battery charging to support an off-grid residence. The overall system contains several components that will be discussed in further detail. These components
include the Whisper 500 generator, the controller/rectifier, diversion load, battery bank, and brake switch. See Figure 3-3.

**Figure 3-3:** The Whisper 500 system consists of a generator, controller, battery bank, diversion load, and an external brake switch. (26)
3.3.1 Wind Turbine

The Southwest Whisper 500 is a two-bladed residential wind turbine that utilizes an upwind configuration and a passive yawing, or “weather vane” system for directional control. See Figure 3-4.

Figure 3-4: The Southwest Whisper 500 is a two-bladed, direct-drive wind turbine that utilizes a passive yaw system. (27)

The fiberglass rotor blades span 15 feet in diameter, and the turbine is mounted atop a guy-wired, monopole tower that is 30 feet tall and constructed from 5 inch Schedule 40 steel pipe. Unlike most industrial-sized wind turbines, the Whisper 500 is a direct-drive unit and does not employ a gearbox. Further separating it from its industrial-sized cousins is the absence of
pitch control. The rotational rate of the turbine varies greatly with wind speed, as does instantaneous power output. The general construction of the Whisper 500 is fairly simple. The nacelle consists simply of the direct-drive generator weighing approximately 155 lb and a tail assembly. The rated power of the Whisper 500 is 3 kW in a 24 mph wind with a peak power of 3.2 kW produced in a 27 mph wind. The Whisper 500 turbine has a cut-in wind speed of 7.5 mph, and the published power curve from Southwest is given in Figure 3-5. In order to protect the turbine from damage in high-winds, the Whisper 500 employs a side-furling mechanism that points the rotor disk out of the freestream in very high winds.

**POWER**

Figure 3-5: The power curve for the Whisper 500 is published by Southwest, and rated power is predicted to be 3 kW at a wind speed of 24 mph. (27)
3.3.2 Generator

Power from the Whisper 500 wind turbine is produced by a 3 kW, three-phase generator. See Figure 3-6. The two blades attach to the rotor, part 1, via the blade plate, part 29. The rotor contains eight permanent magnets which interact with the copper wire windings on the stator, (part 3) to produce power. Because the rotor contains eight permanent magnets, the AC voltage and current signals cycle eight times per revolution. The windings are connected in a junction box, and the three-phase AC power is sent down the tower to the controller. Also shown in Figure 3-6 are the blade plate (part 29) and spindle and bearing assemblies (parts 2, 12, 13, and 16).

Figure 3-6: Major components of the Whisper 500 generator assembly include the rotor, stator, blade mounting plate, and spindle and bearing assembly. (28)
The stator mounts to a yaw housing that connects to the tower. Three-phase AC power is sent through a slip ring so that the yaw housing can be free to point in the direction of the prevailing wind. The tail for the passive yawing system also mounts to the yaw housing.

3.3.3 Controller

The EZ-Wire controller is sold alongside the Whisper 500 wind turbine by Southwest and is responsible for rectifying the three-phase AC power into 24 volt DC power so that it can be stored in the battery bank. The EZ-Wire controller can also support a hybrid system and includes inputs for solar cells. These inputs are not currently utilized in Penn State’s particular system. The Whisper 500 generator is wired such that its three-phases are connected to the diodes in the controller in a delta configuration. It should be noted that for battery charging applications higher than 36 volts, a wye connection may be used. See Figure 3-7.

Figure 3-7: The three-phase AC current from the generator is rectified to DC using a delta connection in 24-volt battery charging applications. (28)
The use of a delta connection has two significant implications. First, the voltage in each phase is ideally equal to the other two phases. Secondly, there is a 120° phase difference between each phase in the generator.

To rectify the AC power to DC, the controller makes use of six diodes. Three Model 70HF10 diodes on the positive side and three Model 70HFR10 diodes on the negative side convert the AC power to DC for battery storage, but it should be noted that the DC still contains traces of AC “ripple.” See Figure 3-8.

Figure 3-8: The EZ-Wire controller is tasked with regulating the battery charge and rectifying the AC power from the generator into DC power. (26)
Battery voltage regulation is handled by eight Model IRF540N MOSFETs and a charge regulator with a float setting that can be adjusted by the operator. The float setting may be adjusted between 2.20 volts per cell and 2.80 volts per cell. The voltage per cell or total system voltage can be monitored via a digital readout on the EZ-Wire controller. If the battery charge level exceeds the set value, the MOSFETs will trigger and divert power to the diversion load circuit. It should be noted that rather than simply switching to the diversion load and remaining at that setting, the circuit regulation triggers on and off at a relatively high frequency. The trigger rate depends on the rotational speed of the turbine, and has typically been observed to operate at a rate of approximately once per revolution. This rapid triggering between the charging and diversion circuits undoubtedly alters the load experienced by the generator, and therefore data taken while the controller is triggering the diversion load must be examined with care. The reaction of the wind turbine in a given wind is not necessarily guaranteed while the diversion load is being triggered.

### 3.3.5 Battery Bank

After the AC power has been converted to DC by the rectifier, it is sent to the batteries for storage. Though several power configurations of the Whisper 500 are available, including a grid-tie option, Penn State’s Whisper 500 sends its power to two 12V Diehard Deep Cycle Marine/RV 29HM batteries that are wired in series for a total of 24 V. Each battery is rated for 115 Amp-hours at a 20 hour rate. When the batteries are being charged, current flows from the rectifying circuit into the cells. If the batteries are not being charged, however, there is a small amount of backflow due to the fact that the EZ-Wire controller display requires a small amount
of power to function. It should be noted that current does not actually flow to the battery until the turbine experiences sustained wind speeds above the cut-in speed of 7.5 mph.

### 3.3.6 Diversion Load

The purpose of the diversion load is to provide an adequate avenue for power dissipation to protect the battery system. If the batteries are fully charged, but the wind turbine is experiencing an extended period of high winds, the diversion load may be triggered so that the batteries are not overcharged. The diversion load consists of four 0.75 Ohm resistors that are configured as two sets of parallel resistors wired in series. See Figure 3-9. The equivalent resistance of the diversion load is 0.75 Ohms. The resistors have been sized so that they can dispense the peak power output of the Whisper 500 though basic heat dissipation, but the Owner’s Manual does note that if the battery bank is changed to a value beyond 24 volts, the diversion load circuit will need to be adjusted accordingly. In the interest of safety, the resistor bank is installed in a protective case that allows heated air to escape but also keeps other equipment from coming in contact with the resistors. See Figure 3-10.

![Diagram of diversion load](image)

**Figure 3-9**: The diversion load consists of four 0.75 Ohm resistors that are configured as two sets of parallel resistors wired in series. (26)
3.3.7 Brake Switch

Before the 3-phase power is sent to the rectifying circuit, it passes through the brake switch. See Figure 3-3. The Whisper 500 does not employ a friction brake, but rather to stop the turbine, all three AC phases are shorted together. This reduces the effective resistance seen by the generator to zero and causes the back EMF to increase drastically. This increased back EMF acts as a brake and stops the wind turbine. The Whisper 500 Owner’s Manual warns that when shutting off the turbine in high winds, care should be taken to avoid causing damage to the machine. If the brake fails to stop the turbine after 30 seconds, the brake should be deactivated,
and the operator should wait until lower winds to try again. For safety reasons, the brake switch is mounted in its own lockable box. See Figure 3-11.

![Figure 3-11: The Whisper 500 brake switch is installed in a lockable box.](image)

### 3.4 Field Test Data Acquisition System

#### 3.4.1 System Goals

The health monitoring data acquisition system used to collect measurements on the Whisper 500 was largely designed and built around the limitations imposed by the use of a National Instruments NI USB-4432 5-channel data acquisition system. Though this system does not allow for the monitoring of as many parameters as the 8-channel National Instruments DAQPad-6020E system used by Brian Wallace, (26) the 5-channel DAQ allows for the use of ICP powered sensors. Priority when measuring the parameters of any generator system is given
to characterizing power output. In the specific case of a wind turbine, it is also prudent to measure the wind speed so that the machine’s operational state can be determined effectively. By measuring the voltage and current of one of the three phases, the total AC power output of the generator can be determined using knowledge of the Whisper 500’s delta wiring configuration. Two of the five available channels are required to gather voltage and current data for one of the three phases, and a third channel is required to gather data from an AC anemometer. This leaves two channels for additional data. Given the amount of attention that has been paid to vibration monitoring in the area of wind turbine fault detection, the decision was made to devote the final two channels to vibration monitoring. This leaves the channel lineup as follows:

1) Accelerometer, y-axis
2) Accelerometer, z-axis
3) Wind Speed
4) AC Voltage
5) AC Current

3.4.2 Data Acquisition System

Data acquisition duties were handled by a National Instruments NI USB-4432. This 5-channel analogue input USB data acquisition system was chosen because of its ability to provide IEPE (2.1 mA constant current) signal conditioning for accelerometers as well as its ease of use. The USB-4432 samples simultaneously at rates up to 102.4 kS/s, allows for 101 dB of dynamic range, and has a 24-bit resolution. Input voltages of ±40 V are accepted by the USB-4432.
3.4.3 Performance Sensors

Performance monitoring of the Whisper 500 was conducted purely on the AC power output from the generator, because this data is most relevant to monitoring the health of the machine. LEM LV-25P voltage transducers were used to take readings of the AC voltage difference between the three phases (V12, V23, V13). See Figure 3-12. These transducers require an excitation voltage of 15 VDC which was provided by a purpose-built power supply constructed by personnel at the Applied Research Lab. This power supply was configured to draw power from a conventional 120 VAC electrical outlet. Current from the generator was measured using LEM HAL 100-S Hall-Effect sensors. These sensors are configured in a closed loop, and the cable through which the current is being measured is placed through the sensor. The Hall-Effect sensor then determines the current in the cable based on the measured electric field.

3.4.4 Vibration Sensors

Vibration monitoring was accomplished through the use of a PCB Piezotronics 356M254 tri-axial accelerometer. The PCB accelerometer requires ICP power which is provided by the National Instruments USB data acquisition system. The accelerometer was calibrated by the support team at the Applied Research Lab, and these calibrations were applied to the configuration input file used by the data acquisition software. The accelerometer was originally mounted to the tower using an aluminum mounting plate. This plate was replaced with a plastic plate when it was discovered that AM radio waves were being picked up in the vibration data due to ground loops. See Figure 3-12.
Figure 3-12: An accelerometer was mounted to the wind turbine tower to measure vibration produced by the generator.

The x-axis of the accelerometer measures vibration in the vertical direction while the y and z axes measure vibration in the lateral directions. To reduce system cost, the accelerometer was mounted to the tower rather than the wind turbine. Because the wind turbine is free to yaw on top of the tower, mounting the accelerometer directly to the generator would have required a method for transmitting the data down the tower to the data acquisition system. This would have either required the addition of a slip ring to transmit the data directly, or a wireless transmission system. Due to cost restrictions, these solutions were not deemed to be feasible.

3.4.5 Data Acquisition Software

The PSU-ARL System Operations and Automation Division’s NIDAQ Version 3.1.7 software was used to collect and save data on a laptop computer. This program allows the user
to configure specific parameters to fit the desired application. These parameters are applied by loading a configuration file of .ini format into the program. This input file contains information on the number of channels to be sampled, the sample rate, and applied calibrations. In the interest of resolving relatively high frequency vibrations, data was sampled at rates varying between 1 kHz and 20 kHz. Due the limitations of MATLAB, the decision was made to limit the data snap time based on the sample rate which would result in data files with 1.2 million data points, near the limit for MATLAB files sizes. Snap times would range from 1 minute for a 20 kHz snap to 20 minutes for a 1 kHz snap.

The data acquisition software allows for both a manual and automatic mode. In automatic mode, the user may set the snapshot length and interval. See Figure 3-13. Snapshot length settings are retained when the user switches to manual mode. In manual mode, each snapshot is individually triggered by the user.

A “quick look” at the data can be obtained using the window in the lower portion of the screen. A Fourier transform of the gathered data may also be viewed. Data is saved to a user-specified location on the hard drive. Within the root directory, data from individual channels is saved to individual folders that are named for the channels that they represent. For instance, AC voltage data is saved to a folder named “V.” Files from each snap are saved with the name of the folder with an incrementally increasing number. For example, data from the 37th snap of the voltage channel is saved in a file called “V00037.V.” Channel aliases are given as follows:

1) Accelerometer, y-axis – Ay
2) Accelerometer, z-axis – Az
3) Wind Speed – Wx
4) AC Voltage – V
5) AC Current – I
The NIDAQ software was used to collect and save data from the USB-4432 data acquisition system.

Additional data viewing was conducted with the use of ARL’s NIDAQ Data Quick Viewer Version 1.4.6. The viewer allows the user to view up to eight data traces at a time, and the program is also capable of performing a Fourier transform of the data traces that are being viewed. See Figure 3-14. To view data files, the user simply has to drag the file from the folder to one of the color-coded boxes on the left side. Once a time trace is being displayed, the user can zoom and pan using the buttons above the viewing window. One interesting feature of the data viewer is its ability to convert data files to .wav format. In .wav format, the files can be played like any other audio file, and they can also be imported into MATLAB for analysis.
3.6 Bench Test Facility

3.6.1 Bench Test Facility Goals

While the Whisper 500 field facility was useful for a portion of the experiments, more complex monitoring of the generator necessitated the fabrication of a dynamometer system. Conducting tests in the lab also opened up more options for vibration monitoring of the Whisper
generator. The ability to place lab-grade accelerometers directly on key locations on the generator meant that the ability to detect bearing faults could be explored in depth. A second advantage of a bench test facility was the ability to drive the generator at a constant rate. In the field, the turbine’s rotation rate responds to fluctuations in wind speed, and this uncertainty complicates the fault detection process. A constant rotation rate allows for more data to be gathered at a given setting and higher quality spectra to be produced.

The goals of the bench test system were simple. First, the motor used to drive the Whisper 500 generator had to be sized such that it could drive the generator through the entire operating range of the turbine. In the Whisper 500’s case, this range spans from 0 to 600 RPM. The second requirement of the facility was that it needed to be designed in such a way as to allow for monitoring of the performance and vibration parameters of the generator. The final goal of this system was to achieve the smoothest possible operation. In order to accurately measure vibrations produced by the generator, vibrations produced by the other sources (drive motor, coupling system, alignment issues, external sources) had to be minimized.

### 3.6.2 System Setup

A three-phase electric motor was employed to drive the Whisper 500 generator for the bench test experiments. This motor was a Baldor Super E EMT3665T rated at 5 hp at 1760 RPM. See Figure 3-15. Power to the motor was supplied and controlled by a Saftronics PC10 controller that allows the motor speed to be controlled by setting the desired rotation rate. The controller itself requires three-phase power.
In an effort to minimize extraneous vibration associated with a belt-drive system, a direct drive setup was used. A set of Lovejoy L-Type couplers were used to link the drive motor to the generator. For the duration of the bench tests, the Whisper 500’s blades and blade plate were removed, and the coupler was attached using the bolts that would normally be used to attach the blade plate. The upper section of the armature was separated at the furling hinge from the lower armature. The stands used to support the Whisper 500 generator and Super E drive motor were fabricated specifically for the purposes of this test. After the motor and generator shafts were aligned, the motor and generator stands were permanently fixed to a sheet of plywood to ensure that they would not migrate during the course of the tests.
In order to eliminate the possibility of the generator running under varied loading conditions, as it does as the batteries charge and discharge, it was decided to run the generator unloaded through the course of the bench tests.

### 3.7 Bench Test Data Acquisition System

#### 3.7.1 System Goals

The overarching goal of the bench test experiments was the detection of faults in the Whisper 500 bearings. This goal, coupled with the desire to reuse the 5-channel data acquisition system and corresponding software from the field test, drove the design of the bench test data acquisition system. The Whisper 500 generator rotor rotates via a spindle that is supported by three 6207-2RS bearings (See Figure 3-6.). The 6207-2RS bearing is a double sealed, deep groove, single row ball bearing with a 35 mm inner diameter, a 72 mm outer diameter, and a 17 mm depth. To best observe vibrations created by faults in these bearings, it was decided that a single-channel lab-grade accelerometer should be devoted to monitoring the vibration near the front bearing, and a second lab-grade accelerometer should be devoted to monitoring vibration near the rear bearing. It was also decided that the voltage and current of a single phase of the generator should continue to be monitored as it was during the field tests. The equipment used to monitor the generator’s performance in the lab was the same as that used during the field tests. This left one channel of acquisition system to be filled by a microphone that was used to make airborne acoustic measurements. The channel lineup used during the bench tests was as follows:

1) AC Voltage
2) Accelerometer, Rear Bearing
3) Accelerometer, Forward Bearing
4) Microphone
5) AC Current

A secondary goal of the bench tests was to compare the output of the lab-grade accelerometers and a wireless accelerometer intended for use in the field. This system is described in Section 3.7.3.

### 3.7.2 Vibration and Acoustic Sensors

Vibration monitoring during the bench tests was accomplished through the use of two PCB Piezotronics 353B16 single-axis accelerometer. The PCB accelerometers require ICP power which is provided by the National Instruments data acquisition system. Each accelerometer was calibrated prior to the test using a PCB 394C06 1g, 159.2 Hz calibrator, and these calibrations were applied to the configuration input file used by the data acquisition software. The accelerometers were mounted to the Whisper 500 generator in locations where they would have best chance of detecting faults in the forward and rear bearings. See Figures 3-16 and 3-17. The mounting pads for the accelerometers were attached using super glue. The ICP-powered microphone, a PCB 378B02, was positioned in the opening at the rear of the stator in the hope that it would pick up noise created by the rear bearing. See Figure 3-18.
Figure 3-16: The forward accelerometer was mounted to a flange on the leading edge of the armature so as to monitor the forward bearing.

Figure 3-17: The rear accelerometer was mounted behind the windings so as to capture vibrations produced by the rear bearing.
A microphone was placed in the rear of the stator to detect noise created by the rear bearing. During the actual test runs, acoustic foam was placed around the microphone stand.

**3.7.3 KCF Technologies System**

A second vibration sensor and data acquisition system were also employed during the bench testing. KCF Technologies provided one of their VMS V2 wireless tri-axial accelerometers that was mounted in a weather-proof casing near the rear lab-grade accelerometer. The purpose of this accelerometer, like its lab-grade counterpart, was to detect vibrations produced by the rear bearing. See Figure 3-19.
The KCF sensor was mounted in a fixture near the rear lab-grade accelerometer so as to monitor the vibrations produced by the rear bearing.

The configuration of the generator and the size of the wireless nodes precluded the possibility of mounting a wireless accelerometer to monitor the forward bearing. The accelerometer communicates with a VMS V2 USB receiver that connects to a laptop computer. See Figure 3-20. The data is saved using KCF’s Vibration Monitoring System Version 6.1 data acquisition software which allows the acceleration range, acquisition rate, collection axes, and burst intervals to be adjusted. The sensors are limited to acquiring 1899 data points. See Figure 3-21. For the purposes of this test, the accelerometer was set to record data at 200 Hz on its Z-axis, and it then transmitted that data to the receiver where it was saved in a .txt file for post processing.
Figure 3-20: The KCF accelerometer node (left) communicates data to the KCF USB receiver (right).
Figure 3-21: KCF’s data acquisition interface allows the range, sample rate, collection axes, and burst interval to be adjusted.
Chapter 4

EXPERIMENTAL METHODS

This chapter presents the experimental procedures used during the inertial imbalance and bearing fault tests mentioned in Section 1.6. Sections 4.1 and 4.2 are devoted to discussing procedures and processing code relevant to the inertial imbalance tests, and Sections 4.3 and 4.4 are devoted to information regarding the bearing fault experiments.

4.1 Inertial Imbalance Experiments

Chapter 3 was devoted to describing the turbine and sensing equipment, and Chapter 4 will discuss how this equipment was used to detect faults within the wind turbine system. In order to determine the effects that a specific fault has on the wind turbine system, a simple testing procedure was developed. Phase 1 of the procedure called for testing the turbine under normal operating circumstances. The goals of this phase of testing were two-fold. First, acquiring data under normal operation would serve as a baseline for future tests. A secondary goal of this phase of testing would be to use the data being generated to develop and refine the processing codes that will be used to analyze data generated later in testing. In Phase 2 of the experiment, specific faults would be seeded within the system, and the turbine would be operated. Phase 3 would involve comparing the data from turbine with and without the seeded faults. Specific markers within the data were indentified that indicate the presence and severity of the seeded faults.
4.1.1 Normal Operation

In order to establish a baseline for later tests, the Whisper 500 turbine was operated under normal conditions before faults were seeded. The data acquisition setup outlined in Chapter 3 was used for these tests. Five channels of data were acquired at an acquisition rate of 2 kHz. This acquisition rate was used to ensure that frequencies up to 1 kHz could be observed. To get the most accurate picture of how the wind turbine was operating, it was important to gather data on wind speed as well as turbine power output. To this end, it was decided that one channel of the data acquisition system should be devoted to gathering wind speed data. Two more channels were required to gather AC voltage and current data. This left two channels for vibration measurements. Movement in the tower’s vertical direction was deemed less likely to occur than tower bending, so data for vibration in the Y and Z directions were included. See Figure 4-1.
Figure 4-1: Vibration in the Y and Z axes was documented.

The five channels monitored are then given as follows:

6) AC Voltage – V
7) Accelerometer, y-axis – Ay
8) Accelerometer, z-axis – Az
9) Wind Speed – Wx
10) AC Current – I
Channels 2 and 3 required ICP power from the data acquisition system, and the current and voltage sensors draw ±15 V from an ARL-designed power supply. At 2 kHz, ten minutes of data will produce 1.2 million data points, so in order to keep data files at a manageable size; data snap lengths were limited to ten minutes. Five ten-minute snaps were taken with the wind turbine operating between 200 and 400 rpm in 8 to 17 mph of wind. The snaps were taken consecutively with a 10 second interval between the end of one snap and the beginning of the next. During the process of capturing the data, the waveforms for each channel were checked using the “quick look” feature of the NIDAQ software to ensure that data was being collected correctly.

4.1.2 Inertial Imbalance

In order to understand the reaction of the wind turbine to an inertial imbalance, a series of counterweights were fabricated and bolted to the blade plate. The counterweights were sized such that they would create a detectable imbalance. The challenge was to also size the counterweights such that running the turbine with the imbalances would not cause permanent damage to the system. Two methods were used to size the counterweights. First, the change in rotational inertia was calculated. Caselitz’s imbalance tests were run with a counterweight that added about 10% to the rotational inertia. (15) It is important to note, however, that Caselitz’s tests were run on a pitch-controlled wind turbine that rotated at a fixed rate of 84 rpm. The Whisper 500 turbine is not pitch regulated, and the rotation rate can reach as high as 600 rpm during normal operation. It was therefore important to calculate not only the change in rotational
inertia but also the additional centrifugal force produced at higher rotational rates. Rotational inertia was calculated using Equation 4-1.

\[ I = \sum mr^2 \]  

4-1

The wind turbine blade is 3.39 kg and its center of mass is located 53.3 cm from the center of rotation. Similarly, the blade plate contributes 3.39 kg at 14.0 cm to the rotational inertia of each blade. Centripetal force was found using Equation 4-2 where \( m \) is the mass of the imbalance, \( r \) its distance from the center of rotation, and \( \omega \) is the rotation rate.

\[ F = mr\omega^2 \]  

4-2

The counterweights were sized such that they would add less than 12 N of force at 400 rpm. By keeping the actual centrifugal force on the turbine relatively low, the risk of damaging the turbine is minimized while still maintaining detectability of the fault. The first counterweight consisted of one 1/8 inch thick steel plate measuring 8.6 cm by 6.9 cm. See Figure 4-2. This plate added a total of 0.061 kg of mass at a distance of 9.8 cm from the center of rotation of the rotor. This additional mass increased the rotational inertia of one of the blades by 0.06%, and it increased the centripetal force produced by the blade by 10.6 N at 400 rpm. See Table 4-1 and Figure 4-2.
Figure 4-2: A steel plate was mounted to the blade plate to create an inertial imbalance.

In order to characterize the turbine’s reaction to counterweights of various sizes, a second plate was fabricated. Like the first plate, the second plate was made from 1/8\textsuperscript{th} inch steel, but the dimensions were increased to 8.6 cm by 12.8 cm. Subsequently, the mass of the second plate increased to 0.122 kg and provided 21.1 N of centripetal force on the turbine at 400 rpm and increasing the inertia of the weighted blade by 0.12%. See Figure 4-3 and Table 4-1.
Figure 4-3: A second counterweight was used to characterize the turbine’s vibration with additional counterweight.
Table 4-1: The steel plates resulted in a change in the rotational inertia of the blade and produced 10.549 N and 21.099 N at 400 rpm respectively.

<table>
<thead>
<tr>
<th>Counterweight</th>
<th>Mass (kg)</th>
<th>Center of Mass distance from Center of Rotation (cm)</th>
<th>Centripetal Force at 200 rpm (N)</th>
<th>Centripetal Force at 400 rpm (N)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Counterweight 1</td>
<td>.061</td>
<td>9.80</td>
<td>2.637</td>
<td>10.549</td>
</tr>
<tr>
<td>Counterweight 2</td>
<td>.122</td>
<td>9.80</td>
<td>5.271</td>
<td>21.099</td>
</tr>
</tbody>
</table>

Figure 4-4: A counterweight was mounted on the blade plate to create an inertial imbalance. (28)
The first counterweight was mounted to the blade plate using the existing blade bolt. In order for the counterweight to be installed, the wind turbine had to be lowered so that access could be gained to the nacelle. Once the counterweight was installed, the wind turbine was raised for testing. See Figure 4-4 and Figure 4-5.

Figure 4-5: The counterweight was mounted to the blade plate using the existing blade bolts.

The procedure for lowering and raising the Whisper 500 wind turbine is given in Southwest’s Whisper 500 Owner’s Manual. The turbine was then operated between approximately 180 and 410 rpm in winds ranging from 8 to 19 mph. Results are presented in Chapter 5. As was the case during the Normal Operation, five channels were monitored:

1) AC Voltage – V
2) Accelerometer, y-axis – Ay
3) Accelerometer, z-axis – Az
4) Wind Speed – Wx
5) AC Current – I

The testing procedure was the same for the Inertial Imbalance case as it was for the Normal Operation case. See Section 4.2.1.

To test the second counterweight, the wind turbine was lowered, and the small counterweight was replaced with the larger one. See Figures 4-3 and 4-6. The turbine was tested with the second counterweight using the same procedure employed with the first counterweight. The turbine was then operated between 160 and 420 rpm in winds ranging from 7 to 18 mph. Results are presented in Chapter 5. As was the case during the Normal Operation Case, five channels were monitored:

1) AC Voltage – V
2) Accelerometer, y-axis – Ay
3) Accelerometer, z-axis – Az
4) Wind Speed – Wx
5) AC Current – I
4.2 Inertial Imbalance Data Processing

4.2.1 Goals of Data Processing

So that useful data could be viewed and interpreted by the user of a wind turbine, the data that was collected needed to be processed. Processing of data for this particular project followed three main steps. Each of these steps will be discussed in more detail in the following sections. The first step of data processing was to separate meaningful data from data that was less...
meaningful. For example, data taken while the wind turbine was not spinning was not useful and was therefore not used to detect faults. More specifically, data taken while the turbine was operating at a relatively constant rotation rate was of the greatest interest. Since one of the goals is to examine the frequency content of the vibration signals, acquiring data at a constant rotation rate would ensure that peaks in the vibration and power spectra would be as well-defined as possible. The second step in processing was to extract important features from the data. As an example, an important feature of vibration data is frequency content which is found by performing a Fourier Transform of the time trace. The final goal of processing is to present the data in a way that is meaningful to the viewer. This typically means plotting data in such a way that important features are easily visible. The procedure for processing is outlined in the following sections.

4.2.2 Data Selection

Data from the tests outlined in Section 4.1 was processed with the goals outlined in the previous section in mind. The first step was to identify useful sections of data and highlight them for later investigation. In order to identify areas where the turbine was operating at a constant rotation rate, a MATLAB code, RPM_Plot.m was written to plot the rotation rate given the voltage signal. See Appendix A. The AC phase voltage signal typically takes the form of a sine wave that varies with amplitude and frequency as the rotation rate increases and decreases. See Figure 4-7.
Figure 4-7: Phase Voltage data was used to create a plot of the rotation rate over time.
The voltage signal cycles eight times per rotor revolution turbine because there are eight poles in the Whisper 500 generator. RPM_plot.m contains an algorithm that identifies and counts level crossings in the voltage signal. The code then computes the rotation rate and outputs a plot of rotation rate over time. See Figure 4-8.

**Figure 4-8:** RPM_Plot.m outputs a plot of the turbine’s rotation rate over time.

With the eventual goal in mind of creating plots of the vibration spectra from the turbine, a decision was made on the appropriate length of data to analyze. Ideally, an FFT would contain as many data points as possible, but the high variability of the wind turbine’s rotation rate means that finding periods of constant rotation is quite a challenge. It was decided that three-second
sections of data would be long enough to provide enough data points to generate an acceptable spectra but short enough so as to be realistically found within the turbine data. For a three-second section of data to qualify to be analyzed, the rotation rate within that section of data could vary by no more than 10 RPM. Allowing the rotation rate to vary by no more than 10 RPM would ensure that peaks would deviate by no more than 0.17 Hz. The segment between 150 s and 153 s in Figure 4-9 was used for analysis because the rotation rate varies by less than 3 RPM over that particular three-second window. See Figure 4-9.

**Figure 4-9:** Minimum variability in the turbine rotation rate is required for a data section to be useable for analysis.
4.2.3 Data Evaluation

After a three-second data section was selected, the data from that section was evaluated for use. For data to be relevant for analysis, it had to meet to two conditions. First, the turbine had to be producing power (voltage and current) during the selected interval. It is possible for the generator to be producing a voltage while producing no current, so after a three-second data segment was selected for analysis, the current signal for that section of data was checked to ensure that current was being produced for the entire three seconds. Sections of data where current was being produced for none or part of the three seconds were rejected. See Figure 4-10.
Figure 4-10: Sections of data where current was being produced for none or part of the three seconds were rejected for analysis.

The second condition that was required for data to be considered useable for spectrum creation involved the nature of the vibration signal. The Whisper 500 turbine is prone to periodic low-frequency, high-amplitude vibrations that occur when the nacelle yaws quickly.
This vibration has been observed to last between one and three seconds and is a side effect of the Whisper 500’s two-bladed design. When the turbine is operating, the yawing inertia of the turbine changes depending on the orientation of the blades. When the blades are horizontal, the yawing inertia is at a maximum, and when the blades are vertical, yawing inertia is at a minimum. This switch between maximum and minimum inertia occurs twice per revolution of the turbine and creates a high-amplitude vibration at twice the shaft frequency along with harmonics at other frequencies. See Figure 4-11. Because the interest was in data recorded during normal operation, data sections that captured these transient events were not included in the analysis.
Figure 4-11: Periods of high-amplitude, low-frequency vibration occur when the turbine yaws quickly. Data captured during these events, such as the one between 97 and 100 s, was not used in analysis.

4.2.4 Power Data Processing

When processing a three-second section of data, an averaged FFT of the power spectrum was produced first. The power was calculated simply by multiplying the voltage and current signals. Because of the nature of the generator’s delta connection, the AC power produced can be obtained using Equation 4-3.

\[ P = 3IV \]
Multiplying the current and voltage signals together produced a power signal with a frequency that was double that of voltage and current signals. When the spectrum of the power signal was plotted, the data had to be plotted against a frequency vector whose values were half of their original values. The primary goal of the power signal spectrum was to verify the average rotation rate over the data section. It is known that the voltage and current signals cycle eight times per revolution, so by identifying the primary peak in the power FFT, the average rotation rate could be confirmed. The secondary goal of generating a power FFT is for use with identifying faults in a similar manner as outlined by Caselitz. (16) See Section 1.4.

After the data sections were selected and evaluated for their use, they were processed using FFT_average.m. See Appendix B. This program was written to produce averaged FFT’s of three second sections of data. The code inputs a data file and isolates the desired 3-second segment. A Butterworth filter is applied to the 3-second segment and then that data segment is broken down into three 1-second segments. A 4096-point FFT is then performed on each of the 1-second segments, and finally, the three FFT’s are averaged to create one averaged FFT. See Appendix B. A block diagram for the code is given in Figure 4-12.
Figure 4-12: The FFT_Average.m code inputs a data file and outputs an averaged FFT plot.
4.2.5 Vibration Data Processing

Two channels of a tower-mounted accelerometer were monitored throughout the course of the testing. The accelerometer’s make, model, and location are discussed in Section 3.3.4. Vibration spectra over three-second intervals were plotted from the individual output of each accelerometer channel using the FFT_average.m code discussed in the previous section. A block diagram for the code used to produce these spectra is given in Figure 4-13.

![Block Diagram]

**Figure 4-13**: A modified version of the FFT_Average.m code was used to produce FFT plots of the accelerometer output.
In addition to producing spectra of the individual vibration channels, three-second FFT’s of the total vibration magnitude were also produced. Because the nacelle changes direction to meet the prevailing wind, there is no guarantee that relevant vibrations will be fully captured by either the Y or Z channels. Furthermore, the current data acquisition system does not monitor to the direction that the nacelle is pointed at any given time. Vibration magnitude was found geometrically by treating the output from the Y and Z accelerometer channels as orthogonal vectors. See Figure 4-14. The magnitude of the resultant vector was found using Equation 4-4.

\[
\text{Magnitude} = \sqrt{Y^2 + Z^2}
\]  

Figure 4-14: Total vibration magnitude is the magnitude of the resultant of the Y and Z vectors.

The block diagram for the code used to produce the FFT plots of the vibration magnitude is given in Figure 4-15.
4.2.6 Power Data Feature Identification

Feature identification is paramount to any SHM activity. As is common with rotating machinery, the most important vibrations occur at multiples of the shaft frequency. Therefore, the first task was to determine the average rotation rate for each data segment and use that information to find the shaft frequency. The average rotation rate was found using the FFT of
the power signal obtained using the procedure outlined in Section 4.3.4. Two peaks of interest were identified in each power signal FFT. First was primary frequency of the AC power output, which cycles eight times per revolution of the turbine. The second peak to be identified was the rotor frequency which, as predicted, was found at a frequency $1/8^{th}$ that of the AC power peak. The shaft frequency peak is particularly relevant to identifying an inertial rotor imbalance. Since an inertial imbalance typically causes increased vibration at the shaft frequency, it was hypothesized that an imbalance would also manifest as an increased magnitude of the shaft frequency peak in the power signal. A sample frequency spectrum of the power signal is shown Figure 4-16. For the sample shown, the power peak cycled at 42.48 Hz, and the rotor peak was identified at 5.19 Hz, putting the average rotation rate of the turbine during the sample at 311 RPM. For each sample, the magnitudes of the power and rotor frequency peaks were noted.
Figure 4-16: The power and rotor frequency peaks were identified in the Power Spectrum. The AC power signal cycles eight times per revolution of the turbine.

4.2.7 Vibration Data Feature Identification

Identification of data features within the vibration spectra followed a similar procedure as the power spectra. After the rotor frequency was identified within the power spectra, the turbine rotor frequency was identified and labeled in the spectra generated from the Y and Z accelerometer channels as well as in the vibration magnitude spectra. Since inertial imbalance typically manifests as increased vibration at the shaft frequency, (24) the magnitude of the
vibration at the rotor frequency was specifically noted. See Chapter 5 for results. Also noted was the blade pass frequency which was found at twice the rotor frequency. See Figure 4-17.

![Average FFT for Az, Run 80 from 30 to 33 (with imbalance)](image)

**Figure 4-17:** The turbine rotor and blade pass frequencies were identified in the single-channel and vibration magnitude data.

### 4.2.8 Consolidated Power Processing Code

In order to process large amounts of data quickly, a consolidated processing code was developed. The primary goal of this code was to plot the magnitude of the shaft frequency component of the power signal over a range of shaft frequencies for different imbalance cases. This would allow the effects of various imbalances to be observed and compared over the
operational range of the wind turbine. To accomplish this goal, the RPM_Plot.m code was combined with the FFT_Average.m code to create Power_Processing.m. A block diagram is given in Figure 4-18. For each separate run, the code uses RPM_Plot.m to produce a plot of the RPM vs Time from the voltage data. See Section 4.3.2. Next, an algorithm finds sections of time where the rotation rate of the turbine varies by less than 10 RPM for at least three seconds. For each of these sections, the FFT_Average.m code is run, and a peak finder identifies the frequency and magnitude of the shaft frequency component of the power signal. See Section 4.2.6 and Figure 4-16. The frequency and magnitude of the peak is then stored. The entire operation is performed for each separate 10 minute run. Finally, the frequency magnitude values are averaged at each shaft frequency and plotted over the range shaft frequencies. See Section 5.1 for results.
Figure 4-18: Power_Processing.m was used to produce plots of the average magnitude of voltage signal at the shaft frequency vs. the shaft frequency.

4.2.9 Consolidated Vibration Processing Code

The consolidated vibration code is very similar to the consolidated voltage processing code discussed in the previous section. The primary goal of this code was to plot the vibration magnitude at the shaft frequency over a range of shaft frequencies for different imbalance cases. This would allow the effects of various imbalances to be observed and compared over the operational range of the wind turbine. As with the voltage processing code, the RPM_Plot.m code was combined with the FFT_Average.m code to create Vibration_Processing.m. The
functionalities of Power_Processing.m and Vibration_Processing.m were then combined into a single code, Power_Vibration_Processing.m, which is given in Appendix C. A block diagram is given in Figure 4-19. For each separate run, the code uses RPM_Plot.m to produce a plot of the RPM vs Time from the voltage data. See Section 4.3.2. Next, an algorithm finds sections of time where the rotation rate of the turbine varies by less than 10 RPM for at least three seconds. For each of these sections, the FFT_Average.m code is run, and a peak finder identifies the frequency and magnitude of vibration of the shaft frequency peak. The frequency and magnitude of the peak is then stored. The entire operation is performed for each separate 10 minute run. Finally, the vibration magnitude values are averaged at each shaft frequency and plotted over the range shaft frequencies. See Section 5.1 for results.
4.3 Bearing Test

One of the most common failure points in industrial and residential wind turbines is the bearings. Section 1.2 outlines the scope of this problem within the context of large scale power generation. There is also some evidence to suggest that bearing failures are an issue for residential turbine operators as well, and this fact is discussed in the introduction to this chapter. In light of this, a test was devised with the goal of detecting differences in the vibration profiles between old, new, and faulted bearings. A secondary goal of this bench test was to compare the

**Figure 4-19:** Vibration Processing was used to produce plots of the average magnitude of vibration at the shaft frequency vs. the shaft frequency.
detection capabilities of the wireless sensor to those of the wired lab-grade units. Since the wireless sensor can be used in the field, it was important to gauge its ability to detect faults in comparison to the lab-grade accelerometers.

The bearings present in the turbine during the inertial imbalance test have been installed in the machine since it was acquired by the Penn State Aerospace Department in 2008 and are likely the original bearings. A test was designed consisting of three phases to test the vibration characteristics of the old, new, and faulted bearings. The test facility used to complete these experiments is described in Section 3.6. Phase 1 involved conducting bench tests with wired and wireless accelerometers and a microphone with the original rear bearing. For Phase 2, the spindle and bearing assembly was removed. The inside of the stator was cleaned thoroughly, and the spindle was then reinstalled with a new rear bearing. A second bench test was then conducted. For the third phase of testing, a rear bearing with an outer race fault was installed, and a final set of tests were conducted. The results of the three tests were compared with the goal of identifying features within the spectra vibration signal that would indicate bearing wear. Specific attention was paid to activity at frequencies that would indicate faults in the inner race, outer race, cage, or roller elements.

4.3.1 Original Bearing Test

The first phase of the bearing test involved driving the Whisper 500 generator on the dynamometer in order to collect vibration data on the bearings. The bench test facility and instrumentation are outlined in Sections 3.6 and 3.7. In order to ensure that bearing activity up
to 10 kHz could be resolved, an acquisition rate of 20 kHz was chosen. The generator was
driven at 0, 100, 200, 300, 400, 500, and 600 RPM, and 30 second data snaps were taken at each
rotation rate. The five channels monitored are then given as follows:

1) AC Voltage – V
2) Rear Accelerometer – RA
3) Forward Accelerometer – FA
4) Microphone – M
5) AC Current – I

The two accelerometers and the microphone were provided with ICP power from the data
acquisition system, and the voltage and current sensors drew power from the ARL-provided
power supply discussed in Section 4.1.1.

Data from the KCF Technologies system outlined in Section 3.7.3 was also recorded
during the test at each of the seven rotation rates. Snaps of 1899 data points were recorded at an
acquisition rate of 200 Hz for each rotation rate.

It is important to note the observed condition of the original bearing as it is relevant to the
results presented in Section 5.2. Because the rear face of the rear bearing is exposed to the inside
of the stator, it accumulated some corrosion during its estimated five years in the field. Though
the bearing’s interior was well-greased, it is still possible that the observed corrosion was not
strictly limited to the exterior. The forward two bearings showed no signs of corrosion or
observable wear. See Figure 4-20.
4.3.2 New Bearing Test

Prior to conducting Phase 2 of the bench test, the spindle and bearing assembly was removed from the stator using a slide hammer, and the rear bearing was removed from the spindle using a bearing puller. See Figure 4-21. After the original rear bearing was removed, a replacement 6207-2RS from VXB Ball Bearing was reinstalled along with the original spindle. So that the new bearing could more easily be pressed onto the spindle, the spindle was placed in a freezer for approximately 12 hrs prior to mounting the bearing. A new bearing was then pressed on the spindle using a drill press. See Figure 4-22. Prior to the spindle’s reinstallation,
the inside of the stator was cleaned thoroughly. A second set of tests were then conducted using the same setup, parameters, and procedures outlined in Section 4.3.1.

Figure 4-21: The spindle assembly was removed using a slide hammer which was bolted to the spindle using the spindle’s existing bolt holes. This picture was taken after the rear bearing had been removed from the spindle.
Figure 4-22: Prior its reinstallation in the stator, the new bearing was installed on the spindle.

4.3.3 Faulted Bearing Test

In order to definitively characterize the acquisition system’s ability to detect a specific bearing fault, a third test was conducted. Following the New Bearing Test, the spindle assembly was removed, and a faulted bearing was installed in the rear position. A fault was seeded in the outer race of the rear bearing by drilling a 1/16\textsuperscript{th} inch hole through the outer wall of a fresh 6207-2RS bearing. See Figure 4-23.
Figure 4-23: A fault was seeded in the outer race of the rear bearing by drilling a hole through the bearing’s outer wall. Care was taken to avoid drilling into the roller elements or the cage.

Special care was taken to ensure that the drill bit did not contact any of the roller elements or the cage. After the hole was drilled, the bearing seals were removed, the inside of the bearing was cleaned thoroughly, the interior of the bearing was greased, and the seals were replaced. The bearing was then reinstalled in the stator, and a series of tests were conducted using the procedure outlined in Section 4.3.1.
4.4 Bearing Test Data Processing

As with the inertial imbalance field tests, FFT plots were created from the accelerometer and microphone data. To create these FFT plots, the FFT_average.m code discussed in Section 4.2.5 was modified to accept 30 second segments of data recorded at a rate of 20 kHz. A diagram of the code is given in Figure 4-13, and a copy of the code is given in Appendix B.

In order to more effectively detect bearing faults within the vibration spectra, envelope analysis was performed using an amplitude demodulation code. Envelope analysis is discussed in Section 2.4.2. The particular code that was employed was developed specifically for detecting bearing faults by McInerny and Dai at the University of Alabama (29). The code is operated via a graphical user interface (GUI) that allows the user to load data in .txt format, apply a bandpass filter for a variety of frequency ranges, apply a Hilbert Transform, and produce a spectrum of the transformed signal. See Figure 4-24. The code also contains a utility for plotting the theoretical bearing fault frequencies based on bearing geometry for the purposes of comparison.
Figure 4-24: The McInerny and Dai amplitude demodulation code is provided with a GUI that allows the user to load data, apply a bandpass filter, perform a Hilbert Transform, and plot a spectrum of the transformed data.
Chapter 5
RESULTS AND DISCUSSION

This chapter presents the results obtained from the experiments outlined in Chapter 4. First, the results from the inertial imbalance field test on the Whisper 500 turbine are presented in Section 5.1. Signal properties from the normal operation case are compared with those from the two faulted cases in an effort to pinpoint differences that could be useful in identifying the seeded fault. Specifically, spectra of the power and voltage signals are compared across the three cases with the goal of matching Caselitz’s ability to detect an inertial imbalance using the frequency content of the power signal. Data from the vibration sensor mounted on the Whisper 500 is compared in a similar fashion. The effectiveness of the two methods is then discussed in the context of their usefulness to a potential user.

The results from the bearing test are presented in Section 5.2. Signal spectra from the various sensors are compared between the old, new, and faulted bearing cases with the goal of identifying markers that would indicate the presence of a bearing fault. The results of the envelope spectra are also presented. The effectiveness of each of the sensors for detecting bearing faults on the Whisper 500 is compared, and finally, an investigation into the sensors’ threshold of fault detection is presented.
5.1 Inertial Imbalance Test Results

The procedures for the inertial imbalance tests conducted in the field are outlined in Section 4.1. The goal of these tests was to be able to detect and quantify the severity of an inertial imbalance by examining changes in the features of the power output and vibration of the machine. Specifically relevant to the detection of inertial imbalances is the behavior of the power and vibration signal at the shaft frequency.

5.1.1 Power Signal Results

Fourier transforms of the power and voltage data were processed and compared for selected 3-second segments using the FFT_Average.m code and the procedure outlined in Sections 4.2.4 and 4.2.6. A sample plot is given in Figure 5-1.
Figure 5-1: A typical power signal spectrum shows a large peak at eight times the shaft frequency and a small peak at the shaft frequency.

Figure 5-1 shows a typical spectrum of the power signal. The voltage and current signals both cycle eight times per revolution, creating the dominant peak at 8 times the shaft frequency. The second and third harmonics of the fundamental frequency can also be seen at approximately 86 and 128 Hz respectively. Similar to the results observed by Caselitz, the shaft frequency peak can also be seen at 5.37 Hz.

In an effort to observe a change in the shaft frequency peak between the three imbalance cases, power signal spectra were compared for similar rotation rates. The data was selected in
accordance with the procedure outlined in Section 4.2.3. Three spectra of three-second data segments are compared across different runs in Figure 5-2. Each of the three spectra was taken at a rotation rate of roughly 320 rpm.

**Figure 5-2:** Spectra of the power signal from the three imbalance cases are compared for 5.4 Hz approximate shaft frequency.

<table>
<thead>
<tr>
<th>Imbalance Type</th>
<th>Shaft Frequency (Hz)</th>
<th>Rotation Rate (RPM)</th>
<th>Shaft Frequency Peak Magnitude (dB)</th>
</tr>
</thead>
<tbody>
<tr>
<td>No Imbalance</td>
<td>5.37</td>
<td>322.3</td>
<td>-53.04</td>
</tr>
<tr>
<td>Small Imbalance (0.061 kg)</td>
<td>5.49</td>
<td>329.4</td>
<td>-55.76</td>
</tr>
<tr>
<td>Large Imbalance (0.122 kg)</td>
<td>5.25</td>
<td>315.0</td>
<td>-46.71</td>
</tr>
</tbody>
</table>
Shaft frequency peak magnitudes for the three spectra are compared in Table 5-1. Though the magnitude of the shaft peak is the highest for the large imbalance case, the small imbalance case produced the lowest peak magnitude. This is contrary to the theory that the two imbalance cases would produce higher shaft frequency peak magnitudes than the no imbalance case.

In order to gain a better picture of the shaft frequency peak’s behavior, three more 3-second spectra were compared at a higher shaft frequency with the rationale being that the imbalance cases would be more visible as the turbine’s rotational rate increased. See Figure 5-3 and Table 5-2.
Figure 5-3: Spectra of the power signal from the three imbalance cases are compared for a 6.5 Hz approximate shaft frequency.

Table 5-2: Shaft Frequency Peak magnitudes are compared for similar rotation rates over the three imbalance cases.
As with the first set of power signal FFT’s, the magnitude of the shaft frequency peak does not follow the prediction that the peak magnitude would increase with imbalance size. Though the magnitude of the shaft frequency peak is highest for the large imbalance case, the smallest peak magnitude was observed for the small imbalance case rather than for the no imbalance case. These individual power signal results make it difficult to draw meaningful conclusions about the power signal’s usefulness in diagnosing inertial imbalance.

In order to gain a clear look at the shaft frequency peak’s reaction to imbalance cases over the entire operational range of the turbine, the Power_Processing.m code was used. See Section 4.2.8. The goal of this code is to plot the frequencies and magnitudes of the shaft frequency peaks from many power signal spectra taken at multiple points from multiple runs. The idea is that behavioral trends of the power peak can then be easily observed and differences in those trends between the imbalance cases can be noted. The Power_Processing.m code was run for two 10-minute runs from each imbalance case:

1) No Imbalance
    a. Run 39
    b. Run 40
2) Small Imbalance (0.061 kg)
    a. Run 80
    b. Run 81
3) Large Imbalance (0.122 kg)
    a. Run 117
    b. Run 120

The runs were selected based on the criteria discussed in Section 4.2.3. Of the 10-min run taken with each imbalance case, the selected runs contained the most amounts of data taken while the wind turbine was producing both voltage and current. The results of the Power_Processing code are given in Figure 5-4.
Figure 5-4: Average peak magnitude values at the shaft frequency are plotted against a range of shaft frequencies for the three imbalance cases.

The results given in Figure 5-4 largely reflect the results obtained from examining the individual FFT’s. Though a trend toward higher peak magnitudes with increasing shaft frequencies is visible, there is almost no ability to distinguish the power signal’s behavior between the three imbalance cases. Therefore, it can be concluded that monitoring the magnitude of the shaft frequency peak of the power signal is not an effective method for detecting an inertial imbalance on a Whisper 500 wind turbine.
5.1.2 Power Signal Discussion

There are a few possible explanations for the inability to detect the inertial imbalances through the power signal. The first possibility is that the shape of the signal is not simply dependent on the turbine’s rotation rate. The level of charge of the battery likely plays a direct role in current production of the generator. Since power is proportional to the product of voltage and current, a higher battery charge level could allow for less current to be produced. This would likely affect the power signal greatly. Note the wildly varying noise floor levels in Figures 5-2 and 5-3. Specifically, the difference in the noise floor level between the large and small imbalance cases is roughly 20 dB despite the fact the turbine is operating at a similar rotation rate in both cases.

A second explanation could lie in the generator’s design. See Figure 3-6. The rotor (part 1) fits over the stator (part 3) and is allowed to rotate by the spindle and bearing assembly (parts 2, 16, 12, 13). The interaction between the windings on the stator and the magnets within the rotor are responsible for creating the AC power that is sent to the controller. The copper windings on the stator are wired into three phases in a junction box located on the nacelle (part 8). Though a shaft frequency peak is observable in the spectrum of the power signal, there is no guarantee that shaft frequency peak is a direct result of a mechanical trait of the generator. The peak may simply be the result of one strand of the windings producing more or less current than the rest of the windings in the stator. Future efforts could be devoted to obtaining a better understanding of the power signal’s production, and the subject of fault detection using the power signal could be revisited.
5.1.3 Vibration Signal Results

Data from the tower-mounted accelerometer was processed using the procedure outlined in Sections 4.2.5 and 4.2.7. As with the power signal FFT’s, three-second segments of data were examined and processed using the FFT_Average.m code. The output from the accelerometer was calibrated to output acceleration in g’s. For the sake of comparison, the vibration data presented in this section is from the same three-second segments as the power data examined in Section 5.1.2. By using data from the same runs, the usefulness of the vibration data for detecting an inertial imbalance can be effectively compared with that of the power data. A sample three-second FFT of the vibration magnitude is given in Figure 5-5.
Figure 5-5: A typical three-second FFT of the vibration magnitude shows a rotor frequency peak between 3 and 10 Hz depending on the turbine’s operating RPM.

As is typical with all FFT plots of the vibration magnitude, the rotor peak is seen at the shaft frequency. Also observable is a peak created by the second harmonic of the shaft rotation / blade passage. Third and fourth harmonics are also visible.

In an effort to observe a change in the shaft frequency peak between the three imbalance cases, vibration magnitude spectra were compared for similar rotation rates. The data was
selected in accordance with the procedure outlined in Section 4.2.3. Three spectra of three-second data segments are compared across different runs in Figure 5-6. Each of the three spectra was taken at a rotation rate of roughly 320 rpm.

Figure 5-6: Spectra of the vibration magnitudes are compared between the three imbalance cases for a 5.4 Hz approximate shaft frequency.
<table>
<thead>
<tr>
<th>Imbalance Case</th>
<th>Shaft Frequency (Hz)</th>
<th>Rotation Rate (RPM)</th>
<th>Shaft Frequency Peak Magnitude (dB)</th>
</tr>
</thead>
<tbody>
<tr>
<td>No Imbalance</td>
<td>5.37</td>
<td>322.3</td>
<td>-53.44</td>
</tr>
<tr>
<td>Small Imbalance</td>
<td>5.37</td>
<td>322.3</td>
<td>-47.09</td>
</tr>
<tr>
<td>Large Imbalance</td>
<td>5.37</td>
<td>322.3</td>
<td>-45.28</td>
</tr>
</tbody>
</table>

Table 5-3: Vibration magnitudes at the shaft frequency are compared for similar rotation rates over the three imbalance cases.

The vibration magnitudes at the shaft frequency for each imbalance case are presented in Table 5-3. As expected, the vibration magnitude increases with increased imbalance mass. The largest jump in vibration magnitude is between the no imbalance and small imbalance case, and a smaller jump is seen between the large and small imbalance cases.

In order to gain a better picture of the vibration magnitude’s behavior at the shaft frequency, three more 3-second spectra were compared at a higher shaft frequency with the rationale being that the imbalance cases would be even more visible as the turbine’s rotational rate increased. Vibration data for these three cases was taken at the same time as the data presented in Figures 5-2 and 5-3. See Figure 5-7 and Table 5-4.
Figure 5-7: Spectra of the vibration magnitudes are compared between the three imbalance cases for a 6.5 Hz approximate shaft frequency.

<table>
<thead>
<tr>
<th></th>
<th>Shaft Frequency (Hz)</th>
<th>Rotation Rate (RPM)</th>
<th>Shaft Frequency Peak Magnitude (dB)</th>
</tr>
</thead>
<tbody>
<tr>
<td>No Imbalance</td>
<td>6.23</td>
<td>373.8</td>
<td>-47.16</td>
</tr>
<tr>
<td>Small Imbalance (0.061 kg)</td>
<td>6.71</td>
<td>402.6</td>
<td>-44.10</td>
</tr>
<tr>
<td>Large Imbalance (0.122 kg)</td>
<td>6.23</td>
<td>373.8</td>
<td>-40.10</td>
</tr>
</tbody>
</table>

Table 5-4: Vibration magnitudes at the shaft frequency are compared for similar rotation rates over the three imbalance cases.
Vibration magnitudes at the shaft frequencies are presented in Table 5-4 for each of the three imbalance cases at similar operating RPM’s. As was the case for the data presented in Table 5-3, the vibration magnitude at the shaft frequency increases as the mass of the imbalance is increased. It is also worth noting that vibration levels can be seen to increase with rotation rate within each imbalance case. This trend is predicted by the relationship between rotation rate and centripetal force given in Equation 4-2.

In order to gain a clear look at the shaft frequency peak’s reaction to imbalance cases over the entire operational range of the turbine, the Vibration_Processing.m code was used. See Section 4.2.9. The goal of this code is to plot the frequencies and magnitudes of the shaft frequency peaks from many vibration magnitude spectra taken at multiple points from multiple runs. The idea is that behavioral trends of the vibration magnitude at the shaft frequency can then be easily observed and differences in those trends between the imbalance cases can be noted. The Vibration_Processing.m code was run for two 10-minute runs from each imbalance case. These were the same runs used to create Figure 5-4.

1) No Imbalance  
   a. Run 39  
   b. Run 40  
2) Small Imbalance (0.061 kg)  
   a. Run 80  
   b. Run 81  
3) Large Imbalance (0.122 kg)  
   a. Run 117  
   b. Run 120  

The runs were selected based on the criteria discussed in Section 4.2.3. Of the 10-min run taken with each imbalance case, the selected runs contained the most amounts of data taken
while the wind turbine was producing both voltage and current. The results of the Vibration_Processing.m code are given in Figure 5-8.

![Figure 5-8: Average vibration magnitude values at the shaft frequency are plotted against a range of shaft frequencies for the three imbalance cases.](image)

The results given in Figure 5-9 largely reflect the results obtained from examining the individual vibration magnitude spectra. For a given rotation rate, the vibration level at the shaft frequency can be seen to increase by an average of at least 2 dB between the no imbalance and imbalance cases. This observation is extremely useful in the context of fault diagnosis because a separation between the vibration levels between the imbalance cases means that a potential fault can be diagnosed based on observed vibration of the turbine at the shaft frequency. Furthermore,
a consistent difference of 1 to 2 dB in the vibration magnitude can be observed between the small and large imbalance cases.

### 5.1.4 Vibration Signal Discussion

The ability to observe differences in the turbine’s vibration level based on seeded imbalances is extremely useful in the context of fault detection and condition monitoring. According to Rytter, the first phase of an effective fault detection system is the ability to identify the presence of a fault. (12) By observing the vibration magnitude of the turbine at the shaft frequency, it has been shown by the separation in vibration levels between the faulted and non-faulted cases, that the presence of an inertial imbalance can be easily identified. The claim can then can be made that observation of the vibration magnitude of the top of the Whisper 500 tower is an effective method of identifying the presence of an inertial imbalance.

The second phase of fault detection, as defined by Rytter, involves identification of the fault’s location within the machine. It is difficult in this case to draw concrete conclusions about the specific type of fault through exclusive monitoring the vibration magnitude at the shaft frequency because several types of faults can exhibit this type of symptom. Davies notes several faults that could cause increased vibration at the shaft frequency including misalignment of bearings, a bent drive shaft, or an aerodynamic imbalance, but he goes on to assert that the most common cause of vibration at the shaft frequency is inertial imbalance. (25) If a change in the vibration at the shaft frequency is noted by the operator, it can then be concluded that one of this group of potential faults has likely occurred within the turbine. Therefore, while it is difficult to
identify one specific type of fault using only this method of monitoring, it is possible to narrow
down the cause of an observed fault to a small group of potential failures.

In an effort to rule out faults such as a bearing misalignment and an aerodynamic imbalance, the vibration level at twice the shaft frequency (the blade-pass frequency) was plotted over the operational range of blade-pass frequencies. According to Davies, both bearing misalignment and aerodynamic imbalance can create increased vibrations at this frequency. A significant observed difference in the vibration at the blade-pass frequency between the three cases could indicate the presence of one of these two failures. A plot of the average vibration level at the blade-pass frequency versus a range of blade-pass frequencies is given in Figure 5-9. To create this plot, the Vibration_Processing code was run for two 10-minute runs from each imbalance case. These were the same runs used to create Figures 5-4 and 5-8.

1) No Imbalance
   a. Run 39
   b. Run 40
2) Small Imbalance (0.061 kg)
   a. Run 80
   b. Run 81
3) Large Imbalance (0.122 kg)
   a. Run 117
   b. Run 120

The average vibration magnitude at the blade-pass frequency can be seen to remain relatively constant over the range of blade-pass frequencies, and there is little discernable difference in the vibration magnitude between the faulted and non faulted cases. These results are consistent with Davies’ assertion that an inertial imbalance typically only causes a consistent increase in vibration at the shaft frequency. (25) By not observing an increase in vibration at the
blade-pass frequency, bearing misalignment and aerodynamic imbalance are much easier to rule out.

![Average Vibration vs Shaft Frequency](image.png)

**Figure 5-9**: Average vibration magnitude values at the shaft frequency are plotted against a range of shaft frequencies for the three imbalance cases.

The third phase fault detection involves qualification of the severity of the fault. According to Equation 4-2, the amplitude of vibration should increase with the mass of a given imbalance. Figure 5-8 shows a clear difference between the shaft frequency vibration levels for the 0.061 kg and 0.122 kg imbalance cases, especially between shaft frequencies of 4 and 5.5 Hz. The ability to observe a difference in the vibration level at the shaft frequency means that the relative severity of a given imbalance can be estimated by the user. It can therefore be concluded
that measurement of the vibration magnitude at the shaft frequency is an effective method of qualifying the severity of an inertial imbalance.

The fourth and final phase of fault detection and condition monitoring is the ability to estimate the remaining life of affected part. It is possible that the presence of an inertial imbalance could lead to the degradation of a number of parts within the Whisper 500 generator, but direct monitoring of those parts would likely be required to determine their remaining life. It is plausible, however, that an observed increase in vibration over time could indicate a gradual deterioration of one or more of the generator’s components, but an estimation of the remaining life of the part would likely require running the machine until failure. Furthermore, the investigation of an inertial imbalance’s long term effect on the health of the machine was not within the scope of this project.

5.2 Bearing Bench Test Results

The procedures and processing codes for the bearing bench tests conducted in the lab facility are outlined in Sections 4.3 and 4.4. The goal of these tests was to be able to detect and identify the presence of a bearing fault in the Whisper 500 turbine through the identification of specific features within the vibration spectra. The results obtained from the wired and wireless sensors and microphone are presented for relevant rotation rates within the operational range of the generator. Of particular interest are the fault frequencies for the 6207-2RS bearing. These frequencies are given for a range of operating speeds of the generator. See Table 5-5.
### Table 5-5: Fault frequencies for the 6207-2RS were of particular interest during the analysis.

<table>
<thead>
<tr>
<th>RPM</th>
<th>100</th>
<th>200</th>
<th>300</th>
<th>400</th>
<th>500</th>
<th>600</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shaft Freq (Hz)</td>
<td>1.67</td>
<td>3.33</td>
<td>5.00</td>
<td>6.67</td>
<td>8.33</td>
<td>10.00</td>
</tr>
<tr>
<td>Inner Race (Hz)</td>
<td>9.05</td>
<td>18.10</td>
<td>27.15</td>
<td>36.20</td>
<td>45.25</td>
<td>54.30</td>
</tr>
<tr>
<td>Outer Race (Hz)</td>
<td>5.95</td>
<td>11.90</td>
<td>17.85</td>
<td>23.80</td>
<td>29.75</td>
<td>35.70</td>
</tr>
<tr>
<td>Roller (Hz)</td>
<td>7.68</td>
<td>15.37</td>
<td>23.05</td>
<td>30.73</td>
<td>38.42</td>
<td>46.10</td>
</tr>
</tbody>
</table>

5.2.1 Lab-Grade Accelerometer Results

Data from the two lab-grade accelerometers taken using the procedures outlined in Section 4.3 is presented in this section. Processing procedures for the data presented are outlined in Section 4.4. A comparison of the vibration spectra over a low frequency range for the rear accelerometer is presented in Figure 5-10. The 600 RPM case most clearly shows the differences between the three cases. The peaks at the outer race fault frequency have been highlighted, and they are also given in Table 5-6.
Figure 5-10: Vibration spectra from the rear accelerometer are compared for the 600 RPM case.

Table 5-6: The highest observed outer race fault frequency peak magnitude was observed for the faulted bearing case. The original bearing displayed a similar peak magnitude while the new bearing’s observed magnitude was significantly lower.

As expected, the peak magnitude for the faulted bearing case was observed to be the highest at the outer race fault frequency. The original bearing’s fault frequency was observed to be approximately 2 dB lower than that of the faulted bearing. Furthermore, the new bearing’s fault frequency was observed to be approximately 13 dB lower than that of the faulted bearing.
In addition to the difference at the fault frequencies, the noise floor for the faulted bearing case was observed to be approximately 12 dB higher between 20 and 45 Hz than the other two cases. A similar plot for the forward accelerometer was produced. See Figure 5-11.

**Figure 5-11**: Vibration spectra from the forward accelerometer are compared for the 600 RPM case.
The outer race peak magnitudes for each case are given in Table 5-7.

<table>
<thead>
<tr>
<th></th>
<th>Peak Magnitude at Outer Race Fault Frequency (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>New Bearing</td>
<td>-70.74</td>
</tr>
<tr>
<td>Original Bearing</td>
<td>-65.90</td>
</tr>
<tr>
<td>Faulted Bearing</td>
<td>-65.20</td>
</tr>
</tbody>
</table>

Table 5-7: Again, the highest observed outer race fault frequency peak magnitude was observed for the faulted bearing case. The original bearing displayed a similar peak magnitude while the new bearing’s observed magnitude was significantly lower.

As was the case with the rear accelerometer, the forward accelerometer measured the highest outer race fault frequency peak magnitude for the faulted case. The next highest by only 0.70 dB was the original bearing. The new bearing’s peak magnitude was approximately 5 dB lower than the other two cases. The larger differences between the peak magnitudes observed by the rear accelerometer are likely explained by that sensor’s greater proximity to the damaged bearing. As was the case with the rear accelerometer, a notable difference in the noise floor of approximately 12 dB was observed between the faulted case and the other two cases.

Though there is an observable difference between the faulted and non-faulted cases in the low frequency spectra, more definitive evidence was desired. A common symptom of bearing failure is low frequency modulation within the high frequency vibration. Specifically, an outer race fault may cause peaks in the high frequency spectrum to be modulated by the bearing fault frequency. In order to identify this modulation, plots of the high frequency vibration recorded by the rear and forward accelerometers were produced. See Figures 5-12 and 5-13.
Figure 5-12: Examination of the high frequency from the rear accelerometer spectra indicated signs of low frequency modulation at the outer race fault frequency.
Figure 5-13: Examination of the high frequency spectra from the forward accelerometer indicated signs of low frequency modulation at the outer race fault frequency.

Close examination of the high frequency peaks indicates evidence of modulation by the outer race fault frequency in the spectra from both accelerometers. As expected, the peaks that show this modulation appear approximately 5 dB higher in the rear accelerometer’s spectrum than in the forward accelerometer’s spectrum.

Also worth noting is the observed differences in the noise floors between the three cases. The noise floor for the faulted case is approximately 15 dB higher than that of the new bearing case. The original bearing case falls between the two, and its noise floor is approximately 6-8 dB higher than that of the new bearing.
A more efficient method for observing low frequency modulation of high frequency
signals is via envelope analysis. The theory behind envelope analysis is discussed in Sections
1.5 and 2.4.2. To produce plots of the envelope spectrum, the McInerny and Dai code discussed
in Section 4.4 was employed. The envelope spectrum for the new, original, and faulted bearing
cases from the rear accelerometer are given in Figures 5-14, 5-15, and 5-16 respectively. A
bandpass frequency range of 2.5-5 kHz was used, and the results presented are for a shaft
frequency of 10 Hz.

**Figure 5-14:** The envelope spectrum for the new bearing displays modulation at the shaft
frequency, but no significant activity at the inner or outer race fault frequencies or their
harmonics.
Figure 5-15: The envelope spectrum for the original bearing displays modulation at the shaft frequency, but additional activity was observed at the roller element fault frequency and its harmonics.

Figure 5-16: The envelope spectrum for the faulted bearing displays significant activity at the outer and inner race fault frequencies and their respective harmonics.

The results produced by the envelope analysis definitively display the differences in the vibration characteristics between the three bearing cases. The new bearing’s envelope spectrum shows modulation at the shaft frequency and its harmonics, but no activity at the inner, outer, or roller element fault frequencies is visible. The original bearing also shows the expected activity
at the shaft frequency harmonics, but additional activity was noted at the roller element fault frequency and its harmonics. This suggests the presence of a roller element fault in the original bearing perhaps caused by the presence of corrosion on one of the roller elements. Finally, the faulted bearing displays significant activity at the fault frequencies and harmonics of both the inner and outer race. The magnitudes of the outer race fault frequency peaks are only slightly higher than those of the inner race. While the cause for the presence of the inner race fault frequency peaks in the faulted bearing spectra is unknown, it is may be the result of the presence of foreign matter left over seeding the outer race fault. Regardless, the results of the envelope analysis clearly demonstrate the distinction between the new, original, and faulted bearings, and it can be concluded that envelope analysis is the most effective tool for identifying the presence of common bearing faults.

Since envelope analysis has proven to be such an effective method for identifying the presence of bearing faults, it is relevant to investigate the generator operation range over which this method is capable of providing useful results. Envelope analysis was performed for the rear accelerometer data at shaft frequencies of 5 Hz and 6.67 Hz, and the results for the faulted bearing are given in Figures 5-17 and 5-18.
Figure 5-17: Outer race fault frequencies are readily identified at a shaft frequency of 6.67 Hz.

Figure 5-18: At 5 Hz shaft frequency, the harmonics of the outer race fault frequency are still visible above the noise floor, but the primary fault frequency is not readily distinguishable from the other structural frequencies.

The threshold for the definitive detection of bearing faults using envelope analysis appears to fall between 5 and 6.67 Hz shaft frequency. Figure 5-18 shows that the outer race fault frequency harmonics are clearly visible above the noise, but for the 5 Hz shaft frequency case shown in Figure 5-18, the fault frequency peak magnitudes decrease drastically. The primary fault frequency peak at 17.8 Hz is barely visible above the noise floor. It can therefore
be concluded that the effective threshold for bearing fault detection using envelope analysis at 5 Hz shaft frequency. In making this observation, it is also important to note that the vibration data presented in this section was taken in a controlled environment. It is likely that effective damage detection in the field environment will require a higher shaft frequency than 5 Hz. In the case of the Whisper 500, a shaft frequency of 5 Hz correlates to a sustained wind speed of approximately 6 m/s while a shaft frequency of 6.67 Hz correlates to a sustained wind speed of approximately 8.5 m/s. (27)

5.2.2 Microphone Results

Results from the microphone discussed in Section 3.7.2 are presented in this section. A comparison of the acoustic results between the three bearing cases for the low frequency range is given in Figure 5-19.
Figure 5-19: A comparison of the microphone output over the low frequency spectra between the three cases shows little evidence to suggest a significant ability to distinguish between the damaged and undamaged cases.

Though there are some observable differences between the acoustic spectra of the damaged and undamaged cases, there is little evidence in the faulted bearing’s spectrum to suggest the presence of an outer race fault. The high frequency spectra are presented in Figure 5-20.
The high frequency spectra produced from the microphone data shows little evidence to suggest significant differences between the damaged and undamaged cases. Unlike, high frequency accelerometer results presented in the previous section, there is also little evidence to suggest low frequency modulation of the high frequency data within the spectrum of the damaged bearing.

In order to more thoroughly investigate the presence of low frequency modulations, envelope analysis was performed on the microphone data from the faulted bearing case. See Figure 5-21. For comparison, the envelope spectrum from the new bearing case is given in Figure 5-22.
Figure 5-21: The envelope spectrum of the microphone data from the faulted bearing case shows minor activity at the inner race fault frequency and its harmonics.

Figure 5-22: The envelope spectrum of the microphone data from the new bearing case shows no activity at the inner race fault frequency or its harmonics.

A comparison of the envelope spectra of the microphone data between the faulted and new bearing cases shows a minor difference between the spectra. The faulted case shows only minor activity at the outer race fault frequency and its harmonics. Though some differences are observable between the two cases, the fault frequencies are not nearly as evident as they are in
the envelope spectra created from the lab-grade accelerometer data. It can therefore be
concluded that even though some differences are observable in the microphone data, that
particular detection method is not as useful for determining the presence of bearing faults as
dedicated lab-grade accelerometers.

5.2.3 Wireless Accelerometer Results

Data from the wireless accelerometer discussed in Section 3.7.3 is presented in this
section. A comparison of the vibration spectra between the three bearing cases for the low
frequency range is given in Figure 5-23.

![Wireless Accelerometer, 10 Hz Shaft Frequency](image)

**Figure 5-23:** Few significant differences between the three cases were observable in the spectra created from the wireless accelerometer data.
Because the wireless accelerometer acquired data at 200 Hz, the resulting spectra were limited to a Nyquist frequency of 100 Hz. These spectra did not show significant differences between the three cases, and a peak indicating the outer race fault frequency was not observable. In light of this, it would be useful to conduct a transmission test to examine the relative sensitivity between the wireless and lab-grade accelerometers. It is likely that the method by which the sensor was mounted played a part in its detection capabilities.

The highest possible acquisition rate for the wireless accelerometer is 3200 Hz which provides for a Nyquist frequency of 1600 Hz. Regardless of the acquisition rate, the sensor acquires 1899 data points, and as a result, the data snap time decreases from roughly 9.5 seconds for an acquisition rate of 200 Hz to 0.59 seconds for an acquisition rate of 3200 Hz. While an envelope spectrum of data taken at the highest acquisition rate could theoretically be produced, the data snap length of 0.59 seconds is only able to provide a frequency resolution of 1.69 Hz. As a result of the sample rate and data snap length limitations, the particular wireless sensor used in these experiments is not as useful for creating relevant envelope spectra, and it is therefore not as useful for detecting bearing faults as the lab-grade accelerometers in this particular application. It should be noted that a wireless sensor with the acquisition abilities similar to those of the lab-grade units used in these tests could be useful for bearing fault detection. It also should be noted that the wireless accelerometer used in these experiments would be useful for more accurately detecting the presence of an inertial or aerodynamic imbalance as these faults are observable in the low frequency spectrum.
Chapter 6

CONCLUSIONS AND FUTURE WORK

6.1 Summary and Conclusions

Wind turbines present a viable solution to the desire for low-emission power production in today’s world, but unfortunately, a trend toward premature failures in some machines’ components has caused the cost of operation of these large turbines to remain higher than anticipated. As a method for detecting problems before they lead to catastrophic failures, condition monitoring and fault detection systems have been applied to industrial-scale wind turbines with the primary goal of detecting bearing and gearbox failures. The implementation of these systems has been relatively successful, but so far these health monitoring systems have not been applied to residential-scale machines in any great number.

Presented in this thesis are the methods, procedures, and results of two experiments related to detecting faults present in a residential wind turbine. The first of these experiments was an investigation of a tower-mounted accelerometer’s ability to passively detect an inertial imbalance seeded on the turbine in the field. Vibration and performance data was taken for three separate imbalance cases. An attempt was made to detect the presence of the inertial imbalance using the spectra produced by the power output in an effort to obtain similar results to those found by Caselitz (15). Unfortunately, the inertial imbalance was undetectable through strict observation of the electrical power spectra. Attention was then turned to the vibration measurements obtained from the tower-mounted accelerometer. By examining the magnitude of the tower’s vibration at the shaft frequency over a 10-minute period of operation, the presence of
a 0.061 kg and 0.122 kg additional mass imbalance were detected and distinguished from the baseline case.

A second set of experiments were conducted on a dynamometer facility with the goal of determining and qualifying the differences in the vibration characteristics between a new bearing, the original (worn) bearing, and a bearing with a seeded outer race fault. Lab-grade accelerometers were mounted in two locations on the stator. A wireless accelerometer and microphone were also employed with the secondary goal of gauging their fault detection abilities against those of the lab-grade accelerometers. Examination of the low frequency spectra from the lab-grade accelerometers revealed limited ability to distinguish between the damaged and non-damaged bearings. Further examination of the high frequency vibration spectra showed the presence of modulation of the high frequency peaks by the bearing fault frequencies. In order to more effectively visualize this modulation, envelope analysis was performed on the lab-grade accelerometer data. Using this analysis method, the presence of the seeded bearing fault was easily discernable. Furthermore, envelope analysis revealed evidence of the presence of a roller element fault in the original bearing that could be the cause of non-uniform affects of corrosion. At the time of writing, the existence of this fault had not been confirmed through visual inspection.

Data obtained from the microphone was not as conclusive. Significant differences between the three cases were not observed in the low or high frequency spectra, but limited evidence of the outer race fault was observed in the envelope spectrum. It was therefore concluded that airborne acoustic measurement is only a semi-useful method for identifying seeded bearing faults.
Finally, low frequency spectra from a wireless sensor were compared across the three test cases. Do to the snap length and acquisition rate limitations of the sensor, high frequency and envelope spectra were not able to be produced from the data acquired from this sensor. Unfortunately, no significant differences between the spectra of the damaged and non-damaged cases were observed in the low frequency spectra. It was therefore concluded that the particular wireless sensor employed in the experiments could not detect seeded bearing faults.

6.2 Future Work

Though the experiments described in this thesis have met with a certain measure of success, there is still a significant amount of activity that would benefit the eventual goal of installing a comprehensive health monitoring system on the Whisper 500 turbine. Furthermore, much of this work could be completed using the facilities and capabilities outlined in this thesis.

The first of the proposed experiments is an extension of the inertial imbalance tests conducted at the field facility. Following the model of Caselitz’s experiments (15), the next step in testing would involve the application of aerodynamic roughness to one of the wind turbine’s blades to simulate a buildup of foreign matter on the blade surface. The goal of this test would be to detect the presence of this roughness and differentiate it from the vibration signature produced from an inertial imbalance. This difference would likely lie in the presence of increased vibration at the blade pass frequency. In the case of the inertial imbalance, increased vibration at the blade pass frequency was not observed, so if an increase in vibration at this frequency was observed, that could indicate the presence of an aerodynamic imbalance. In an effort to more accurately distinguish between faulted and baseline cases, the installation of a
vibration sensor such as the K.C.F. wireless accelerometer used in this project directly to the Whisper 500 generator is recommended.

The second proposed set of experiments could be completed as a follow-up to the bearing fault detection work outlined in this thesis. The goal of detecting an outer race fault in one of the main bearings was accomplished successfully, but bearings are susceptible to multiple types of faults. Developing a database of the vibration signatures for different bearing faults would be advantageous to the overall goal of constructing a comprehensive fault detection system for the Whisper 500. Future faults to be tested could include an inner race fault, a roller element fault, a cage fault, and general corrosion of the bearing internals. Furthermore, the effect of the position of the faulted bearing could also be tested. The tests carried out over the course of this thesis research limited the application of damage to the rear bearing, but it would be prudent to test the system’s ability to detect faults present at the other two bearing positions.

A third set of proposed experiments would involve the implementation of the bearing fault detection capabilities at the Whisper 500 field facility. This task would likely require the use of a wireless accelerometer; however, the wireless system employed in the tests described in this thesis was not shown to effectively detect the seeded bearing fault. Therefore, it would be prudent to acquire a wireless sensor with the capability to acquire data at an acquisition rate greater than or equal to that of the lab-grade units (20 kHz) for an amount of time that is sufficient to effectively resolve low frequency excitations.

A fourth and final set of experiments could coincide with the installation of a new set of redesigned blades on the Whisper 500. At the time of writing, these blades and the hub to accommodate them were being fabricated by a team of students at Penn State. These blades could be installed with embedded vibration sensors in an effort to characterize the vibration to
which these blades are subjected. Furthermore, the blades could be equipped with strain gages to characterize the stress and strain incurred their operation.

The experiments presented in this thesis did see successful outcomes, but the activity described in this section would greatly increase the fault detection capabilities and useful experience of the wind turbine research team within Penn State’s Aerospace Department.
Bibliography


Appendix A

RPM_PLOT.m

%RPM_PLOT.m
%Brenton Forshey
%21 July 2010
%This program is intended to find level crossings in wind turbine current
%data and convert those crossings into a plot of turbine RPM. The program
%first applies a lowpass filter to the data set to remove high frequency
%noise that could create higher-than-accurate RPM readings.

%Read in voltage data file
 clear all
 data = wavread('V95.wav');

 sample_rate = 2000;
 X = data;

 n = 1;

 %Find Level Crossings at -0.1
 for i = 1:length(X) - 1
   if X(i) > -.1 && X(i+1) < -.1
     n = n+1;
     crosspos(n) = i;
   elseif X(i) < -.1 && X(i+1) > -.1
     n = n+1;
     crosspos(n) = i;
   elseif X(i) == -.1
     n = n+1;
     crosspos(n) = i;
   elseif X(i+1) == -.1
     n = n+1;
     crosspos(n) = i;
 end
 end

 %Find space between level crossings
for k = 1:length(crosspos)-2
crossspace(k) = crosspos(k+1) - crosspos(k);
end

%Find data points per rev
m = 1;
for i = 1:length(crossspace)
    if mod(i,18) == 0
        revcount(m) = crossspace(i)+crossspace(i-2)+crossspace(i-3)+crossspace(i-4)+crossspace(i-5)+crossspace(i-6)+crossspace(i-7)+crossspace(i-8)+crossspace(i-9)+crossspace(i-10)+crossspace(i-11)+crossspace(i-12)+crossspace(i-13)+crossspace(i-14)+crossspace(i-15)+crossspace(i-16)+crossspace(i-17);
        m = m+1;
    end
end

%Create time vector
m = length(revcount);
for i = 1:length(revcount)
    time_per_rev(i) = revcount(i)/sample_rate;
    if i > 1
        time(i) = time_per_rev(i)+time(i-1);
    else
        time(i) = time_per_rev(i);
    end
end

%Create rpm vector
for i = 1:length(revcount)
rpm(i) = 1/(revcount(i))*sample_rate*60;
end

%Find areas where RPM varies by < 5 RPM over 3 seconds
r=1;
checklength = 3;
for i = 1:length(time)-40
    time_start = time(i);
    time_end = time_start+checklength;
    q=1;
clear section_save
    clear levelcatch
    for k = 1:40
        time_stop = time(i+k);
        if time_stop > time_end
            section_save(q) = i+k;
            q=q+1;
            hi = section_save;
        end
    end
    section = min(section_save);
    for j=i:section
if rpm(j) > rpm(i)+5 || rpm(j) < rpm(i)-5;
    levelcatch(j-i+1) = 1;
else
    levelcatch(j-i+1) = 0;
end
end
if sum(levelcatch) == 0;
    levelpos(r) = time(i);
    r=r+1;
end
end

%Plot results
figure
plot(time,rpm)
title('RPM vs Time')
xlabel('Time(s)')
ylabel('Turbine RPM')
Appendix B

FFT_Average.m

% Brenton Forshey
% FFT_average
% 10 Feb 2011
% Updated 13 April 2011 to include vibration magnitude analysis.

% The purpose of this program is to create an averaged FFT of a section
% of vibration data. First, the data is run through a butterworth filter.
% Next, a section of the total data snap is extracted
% based on the 'Start' and 'Stop' parameters. Next, the given section is
% divided into three smaller sections, and an FFT of each section is taken.
% The three FFT's are then averaged to produced a single average FFT.

clear V
clear I
clear Ay
clear Az
clear data

%Read in data files
V = wavread('V95.wav');
I = wavread('I95.wav');
Ay = wavread('Ay95.wav');
Az = wavread('Az95.wav');

Vib = wavread('Az95');

% data = sqrt(Ay.^2+Az.^2);
% % data=(Ay+Az)/2;
% data=Vib;
% data=I.*V;
data=V;
% mag_data = data;

% Apply Butterworth filter
[b,a]=butter(5,800/1000,'low');
y = filtfilt(b,a,data);
% freqz(b,a,4096);
clear data
data = y;

% Data section is extracted
Start = 45;
Stop = 48;
sample_rate = 2000;
data1 = data;
clear data;
data = zeros((Stop*sample_rate)-(Start*sample_rate),1);
for i=1:(Step*sample_rate)-(Start*sample_rate));
   data(i) = data1(i+(Start*sample_rate)-1);
end

% Apply Hanning Window
win = window(@hann,length(data));
data = data.*win;
clear data1

% Divide the data trace into three equal sections
ChopLength = round(((Stop*sample_rate)-(Start*sample_rate))/3);
for i=1:ChopLength;
   data1(i) = data(i);
end
for j=ChopLength+1:(2*ChopLength);
   data2(j-ChopLength) = data(j);
end
for k=(2*ChopLength)+1:(3*ChopLength);
   data3(k-(2*ChopLength)) = data(k);
end

% Take an FFT of each section
NFFT = 2^12;
win1 = window(@hann,length(data1));
data1 = data1.*transpose(win1);
Y1 = fft(data1,NFFT)/length(data1);
f1 = sample_rate/2*linspace(0,1,NFFT/2+1);

win2 = window(@hann,length(data2));
data2 = data2.*transpose(win2);
Y2 = fft(data2,NFFT)/length(data2);
f2 = sample_rate/2*linspace(0,1,NFFT/2+1);

win3 = window(@hann,length(data3));
data3 = data3.*transpose(win3);
Y3 = fft(data3,NFFT)/length(data3);
f3 = sample_rate/2*linspace(0,1,NFFT/2+1);

% Average the three FFT's
Y = ((abs(Y1).^2)+(abs(Y2).^2)+(abs(Y3).^2))/3;

% Calculate the average FFT of the data section and frequency vector
f = sample_rate/2*linspace(0,1,NFFT/2+1);
FFT_plot = 10*log10(abs(Y(1:NFFT/2+1)));
FFT_plot_aux = 10*log10(abs(Y3(1:NFFT/2+1)));
for i = 1:10
    Peakfind(i) = FFT_plot(i+5);
end
Rotorpeak = max(Peakfind);
for i = 1:20
    if Rotorpeak == FFT_plot(i)
        peak_spot = i;
    end
end

% Plot FFT
figure
plot(f,FFT_plot)
title('Average FFT ')
xlabel('Frequency (Hz)')
ylabel('Magnitude in dB')
% Power_Vibration_Processing.m
% 13 May 2011
% Brenton Forshey

% In order to process large amounts of data quickly,
% a consolidated processing code was developed.  
% The primary goal of this code was to plot the magnitude of 
% the shaft frequency component of the power signal over a 
% range of shaft frequencies for different imbalance cases. 
% This would allow the effects of various imbalances to be observed 
% and compared over the operational range of the wind turbine. 
% To accomplish this goal, the RPM_Plot code was combined with the 
% FFT_Average code to create Power_Processing.m. For each separate 
% run, the code uses RPM_Plot to produce a plot of the RPM vs Time 
% from the voltage data. Next, an algorithm finds sections of time 
% where the rotation rate of the turbine varies by less than 10 RPM 
% for at least three seconds. For each of these sections, the 
% FFT_Average code is run, and a peak finder identifies the frequency 
% and magnitude of the shaft frequency component of the voltage signal. 
% The frequency and magnitude of the peak is then stored. The entire 
% operation is performed for each separate 10 minute run. Finally, the 
% frequency magnitude values are averaged at each shaft frequency and 
% plotted over the range shaft frequencies.

clear
clc
close
o = 1;
v = 1;
c = 1;

% Select files to be processed
Vfiles = {'V117.wav';'V120.wav';'V80.wav';'V81.wav';'V39.wav';'V40.wav'};
Ifiles = {'I117.wav';'I120.wav';'I80.wav';'I81.wav';'I39.wav';'I40.wav'};
Ayfiles = 
{'Ay117.wav';'Ay120.wav';'Ay80.wav';'Ay81.wav';'Ay39.wav';'Ay40.wav'};
Azfiles = 
{'Az117.wav';'Az120.wav';'Az80.wav';'Az81.wav';'Az39.wav';'Az40.wav'};

for g = 1:length(Vfiles)
    % Read in data files for the current run
    data = wavread(char(Vfiles(g)));
sample_rate = 2000;

clear X
clear time
clear rpm
clear revcount
clear crossspace
clear crosspos
clear time_per_rev
clear levelpos
clear section
clear section_save
clear level_catch

X = data;

n = 1;

%Find Level Crossings at -0.1
for i = 1:length(X) - 1
    if X(i) > -.1 && X(i+1) < -.1
        n = n+1;
        crosspos(n) = i;
    elseif X(i) < -.1 && X(i+1) > -.1
        n = n+1;
        crosspos(n) = i;
    elseif X(i) == -.1
        n = n+1;
        crosspos(n) = i;
    elseif X(i+1) == -.1
        n = n+1;
        crosspos(n) = i;
    end
end

%Find space between level crossings
for k = 1:length(crosspos)-2
    crossspace(k) = crosspos(k+1) - crosspos(k);
end

%Find data points per rev
m = 1;
for i = 1:length(crossspace)
    if mod(i,18) == 0
        revcount(m) = crossspace(i)+crossspace(i-2)+crossspace(i-3)+crossspace(i-4)+crossspace(i-5)+crossspace(i-6)+crossspace(i-7)+crossspace(i-8)+crossspace(i-9)+crossspace(i-10)+crossspace(i-
% Create time vector
m = length(revcount);
for i = 1:length(revcount)
    time_per_rev(i) = revcount(i)/sample_rate;
    if i > 1
        time(i) = time_per_rev(i)+time(i-1);
    else
        time(i) = time_per_rev(i);
    end
end

% Create rpm vector
for i = 1:length(revcount)
    rpm(i) = 1/(revcount(i))*sample_rate*60;
end

% Find sections of > 3 seconds where the RPM varies by < 5 RPM
r=1;
checklength = 3;
for i = 1:length(time)-60
    time_start = time(i);
    time_end = time_start+checklength;
    q=1;
clear section_save
clear levelcatch
for k = 1:40
    time_stop = time(i+k);
    if time_stop > time_end
        section_save(q) = i+k;
        q=q+1;
        hi = section_save;
    end
end
section = min(section_save);
for j=i:section
    if rpm(j) > rpm(i)+5 || rpm(j) < rpm(i)-5;
        levelcatch(j-i+1) = 1;
    else
        levelcatch(j-i+1) = 0;
    end
end
if sum(levelcatch) == 0;
    levelpos(r) = time(i);
    r=r+1;
end
% Check for overlap with previously processed FFT
for q = 1:length(levelpos)
    if q>1
        previous = previous;
    else
        previous = -3;
    end

    if levelpos(q) < previous + 3;
        q = q + 1;
    else
        Start = levelpos(q);
        previous = Start;
        Stop = levelpos(q)+3;
        clear Az
        clear V
        clear I
        clear PWR
        clear dataz
        clear datai
        clear datapwr
        clear Ay
        clear datay
        clear data1z
        clear data1y
        clear data1pwr
        clear FFT_ploty
        clear FFT_plotz
        clear FFT_plotpwr
        clear f
        clear fpwr

        %Read in data files
        Az = wavread(char(Azfiles(g)));
        Ay = wavread(char(Ayfiles(g)));
        V = wavread(char(Vfiles(g)));
        I = wavread(char(Ifiles(g)));

        dataz=Az;
        PWR = (Az.*12.666).*(I.*12.5);
        datapwr = PWR;

        % Apply Butterworth filter
        [b,a]=butter(5,800/1000,'low');
        z = filtfilt(b,a,dataz);
        % freqz(b,a,4096);
clear dataz
dataz = z;

% Data section is extracted
sample_rate = 2000;
data1z = dataz;
clear dataz;
dataz = zeros((Stop*sample_rate)-(Start*sample_rate),1);
for i=1:((Stop*sample_rate)-(Start*sample_rate));
    dataz(i) = data1z(round(i+Start*sample_rate-1));
end
if length(dataz)<6000
    dataz(6000) = 0;
end

clear data1z

% Divide the data trace into three equal sections
ChopLength = round(((Stop*sample_rate)-(Start*sample_rate))/3);
for i=1:ChopLength;
    data1z(i) = dataz(i);
end
for j=ChopLength+1:(2*ChopLength);
    data2z(j-ChopLength) = dataz(j);
end
for k=(2*ChopLength)+1:(3*ChopLength);
    data3z(k-(2*ChopLength)) = dataz(k);
end

% Take an FFT of each section
NFFT = 2^13;
win1 = window(@hann,length(data1z));
data1z = data1z.*transpose(win1);
Y1z = fft(data1z,NFFT)/sum(win1);
f1 = sample_rate/2*linspace(0,1,NFFT/2+1);

win2 = window(@hann,length(data2z));
data2z = data2z.*transpose(win2);
Y2z = fft(data2z,NFFT)/sum(win2);
f2 = sample_rate/2*linspace(0,1,NFFT/2+1);

win3 = window(@hann,length(data3z));
data3z = data3z.*transpose(win3);
Y3z = fft(data3z,NFFT)/sum(win3);
f3 = sample_rate/2*linspace(0,1,NFFT/2+1);

% Average the three FFT's
Yz = ((abs(Y1z).^2)+(abs(Y2z).^2)+(abs(Y3z).^2))/3;
% Plot the average FFT of the data section
f = sample_rate/2*linspace(0,1,NFFT/2+1);
FFT_plotz = 10*log10(abs(Yz(1:NFFT/2+1)));
% FFT_plot_aux = 10*log10(abs(Y3z(1:NFFT/2+1)));
for i = 1:10
  % Peakfind(i) = FFT_plot(i+5);
end
Rotorpeak = max(Peakfind);
for i = 1:20
  if Rotorpeak == FFT_plot(i)
    peak_spot = i;
  end
end

%% % Perform the same operation on the Y-axis vibration data
clear data
clear y
datay=Ay;
mag_datay = datay;

% Apply Butterworth filter
[b,a]=butter(5,800/1000,'low');
y = filtfilt(b,a,datay);
% freqz(b,a,4096);
clear datay
datay = y;

% % Data section is extracted
datayl = datay;
clear datay;
datay = zeros((Stop*sample_rate)-(Start*sample_rate),1);
for i=1:((Stop*sample_rate)-(Start*sample_rate));
  datay(i) = datayl(round(i+(Start*sample_rate)-1));
end
if length(datay)<6000
  datay(6000) = 0;
end

% % Apply Hanning Window
% win = window(@hann,length(datay));
% datay = datay.*win;
clear datayl

% % Divide the data trace into three equal sections
ChopLength = round(((Stop*sample_rate)-(Start*sample_rate))/3);
for i=1:ChopLength;
  datayl(i) = datay(i);
end
for  j=ChopLength+1:(2*ChopLength);
    data2y(j-ChopLength) = datay(j);
end
for  k=(2*ChopLength)+1:(3*ChopLength);
    data3y(k-(2*ChopLength)) = datay(k);
end

% Take an FFT of each section
NFFT = 2^13;
win1 = window(@hann,length(data1y));
data1y = data1y.*transpose(win1);
Y1y = fft(data1y,NFFT)/sum(win1);
f1 = sample_rate/2*linspace(0,1,NFFT/2+1);

win2 = window(@hann,length(data2y));
data2y = data2y.*transpose(win2);
Y2y = fft(data2y,NFFT)/sum(win2);
f2 = sample_rate/2*linspace(0,1,NFFT/2+1);

win3 = window(@hann,length(data3y));
data3y = data3y.*transpose(win3);
Y3y = fft(data3y,NFFT)/sum(win3);
f3 = sample_rate/2*linspace(0,1,NFFT/2+1);

% Average the three FFT's
Yy = ((abs(Y1y).^2)+(abs(Y2y).^2)+(abs(Y3y).^2))/3;

% Calculate average FFT of the data section
f = sample_rate/2*linspace(0,1,NFFT/2+1);
FFT_ploty = 10*log10(abs(Yy(1:NFFT/2+1)));

% FFT_plot = (FFT_plotz+FFT_ploty)/2;
% FFT_plot = -sqrt((FFT_plotz.^2)+(FFT_ploty.^2));
FFT_plot = 10*log10(Yz(1:NFFT/2+1)+ Yy(1:NFFT/2+1));

%%
% Perform the same operation on the power data
%
% Apply Butterworth filter
[b,a]=butter(5,800/1000,'low');
power = filtfilt(b,a,datapwr);
clear datapwr
datapwr = PWR;

% Data section is extracted
\begin{verbatim}

sample_rate = 2000;
data1pwr = datapwr;
clear datapwr;
datapwr = zeros((Stop*sample_rate)-(Start*sample_rate),1);
for i=1:((Stop*sample_rate)-(Start*sample_rate));
    datapwr(i) = data1pwr(round(i+Start*sample_rate-1));
end
if length(datapwr)<6000
    datapwr(6000) = 0;
end
clear data1pwr

% Divide the data trace into three equal sections
ChopLength = round(((Stop*sample_rate)-(Start*sample_rate))/3);
for i=1:ChopLength;
    data1pwr(i) = datapwr(i);
end
for j=ChopLength+1:(2*ChopLength);
    data2pwr(j-ChopLength) = datapwr(j);
end
for k=(2*ChopLength)+1:(3*ChopLength);
    data3pwr(k-(2*ChopLength)) = datapwr(k);
end

% Take an FFT of each section
NFFT = 2^13;
win1 = window(@hann,length(data1pwr));
data1pwr = data1pwr.*transpose(win1);
Y1pwr = fft(data1pwr,NFFT)/sum(win1);
f1 = sample_rate/2*linspace(0,1,NFFT/2+1);

win2 = window(@hann,length(data2pwr));
data2pwr = data2pwr.*transpose(win2);
Y2pwr = fft(data2pwr,NFFT)/sum(win2);
f2 = sample_rate/2*linspace(0,1,NFFT/2+1);

win3 = window(@hann,length(data3pwr));
data3pwr = data3pwr.*transpose(win3);
Y3pwr = fft(data3pwr,NFFT)/sum(win3);
f3 = sample_rate/2*linspace(0,1,NFFT/2+1);

% Average the three FFT's
Ypwr = ((abs(Y1pwr).^2)+(abs(Y2pwr).^2)+(abs(Y3pwr).^2))/3;

% Plot the average FFT of the data section
fpwr = sample_rate/2*linspace(0,1,NFFT/2+1);
FFT_plotpwr = 10*log10(abs(Ypwr(1:NFFT/2+1)));
\end{verbatim}
% Peak finder locates the shaft frequency peak in vibration
% data
for i = 1:20
    Peakfind(i) = FFT_plot(i+10);
end
for i = 1:10
    Peakfind(i) = FFT_plot(i+5);
end
Rotorpeak = max(Peakfind);
for i = 1:50
    if Rotorpeak == FFT_plot(i)
        peak_pos = f(i);
        if (peak_pos > 7) && (abs(FFT_plot(i)) > 50)
            peak_pos = f(i)/2;
        end
        peak_mag = FFT_plot(i);
    end
end

% Peak finder locates the shaft frequency peak in power
% data
w=1;
ShaftPos = peak_pos;
Shaftminus = ShaftPos-1;
for i = 1:50
    if fpwr(i) >= Shaftminus
        spot(w) = i;
        w=w+1;
    end
end
SearchStart = min(spot);
for i = 1:5
    plotlook(i) = FFT_plotpwr(SearchStart + i);
end
Shaftmag = max(plotlook);
for i = 1:50
    if Shaftmag == FFT_plotpwr(i)
        pwr_peak_pos = fpwr(i)
    end
end
pwr_peak_mag = Shaftmag;

% Plot Results
if (g == 1) || (g == 2)
pwr_vib_pos3(o) = pwr_peak_pos;
pwr_vib_mag3(o) = pwr_peak_mag;
vib_pos3(o) = peak_pos;
vib_mag3(o) = peak_mag;
o = o + 1;

else if (g == 3) || (g == 4)
pwr_vib_pos2(v) = pwr_peak_pos;
pwr_vib_mag2(v) = pwr_peak_mag;
vib_pos2(v) = peak_pos;
vib_mag2(v) = peak_mag;
v = v + 1;
else if (g == 5) || (g == 6)
pwr_vib_pos1(c) = pwr_peak_pos;
pwr_vib_mag1(c) = pwr_peak_mag;
vib_pos1(c) = peak_pos;
vib_mag1(c) = peak_mag;
c = c + 1;
end
end
end
end
end

end

%%% Calculate average vibration magnitude for each shaft frequency

for i = 1:length(vib_pos1)
b=1;
vib_mag_ave(b) = vib_mag1(i);
for j = 1:length(vib_pos1)
    if vib_pos1(i) == vib_pos1(j) && i~=j
%         vib_pos_ave1(b) = vib_pos(i);
vib_mag_ave(b+1) = vib_mag1(j);
b=b+1
    end
end
vib_mag_ave1(i) = mean(vib_mag_ave);
vib_mag_max1(i) = max(vib_mag_ave);
clear vib_mag_ave
end

for i = 1:length(vib_pos2)
b=1;
vib_mag_ave(b) = vib_mag2(i);
for j = 1:length(vib_pos2)
    if vib_pos2(i) == vib_pos2(j) && i~=j
%         vib_pos_ave1(b) = vib_pos(i);
vib_mag_ave(b+1) = vib_mag2(j);
b=b+1
    end
end
vib_mag_ave2(i) = mean(vib_mag_ave);
vib_mag_max2(i) = max(vib_mag_ave);
clear vib_mag_ave

end

for i = 1:length(vib_pos3)
b=1;
vib_mag_ave(b) = vib_mag3(i);
for j = 1:length(vib_pos3)
    if vib_pos3(i) == vib_pos3(j) && i~=j
        vib_pos_ave1(b) = vib_pos(i);
        vib_mag_ave(b+1) = vib_mag3(j);
        b=b+1
    endif
end
vib_mag_ave3(i) = mean(vib_mag_ave);
vib_mag_max3(i) = max(vib_mag_ave);
clear vib_mag_ave
end

dd
for i = 1:length(pwr_vib_pos1)
b=1;
pwr_vib_mag_ave(b) = pwr_vib_mag1(i);
for j = 1:length(pwr_vib_pos1)
    if pwr_vib_pos1(i) == pwr_vib_pos1(j) && i~=j
        pwr_vib_mag_ave1(b) = pwr_vib_pos(i);
        pwr_vib_mag_ave(b+1) = pwr_vib_mag1(j);
        b=b+1
    endif
end
pwr_vib_mag_ave1(i) = mean(pwr_vib_mag_ave);
pwr_vib_mag_max1(i) = max(pwr_vib_mag_ave);
clear pwr_vib_mag_ave
end

for i = 1:length(pwr_vib_pos2)
b=1;
pwr_vib_mag_ave(b) = pwr_vib_mag2(i);
for j = 1:length(pwr_vib_pos2)
    if pwr_vib_pos2(i) == pwr_vib_pos2(j) && i~=j
        pwr_vib_mag_ave2(b) = pwr_vib_pos(i);
        pwr_vib_mag_ave(b+1) = pwr_vib_mag2(j);
        b=b+1
    endif
end
pwr_vib_mag_ave2(i) = mean(pwr_vib_mag_ave);
pwr_vib_mag_max2(i) = max(pwr_vib_mag_ave);
clear pwr_vib_mag_ave
end

for i = 1:length(pwr_vib_pos3)
b=1;
pwr_vib_mag_ave(b) = pwr_vib_mag3(i);
for j = 1:length(pwr_vib_pos3)
if pwr_vib_pos3(i) == pwr_vib_pos3(j) && i~=j
    vib_pos_ave1(b) = vib_pos(i);
pwr_vib_mag_ave(b+1) = pwr_vib_mag3(j);
b=b+1
end
end
pwr_vib_mag_ave3(i) = mean(pwr_vib_mag_ave);
pwr_vib_mag_max3(i) = max(pwr_vib_mag_ave);
clear pwr_vib_mag_ave
end

% Plot Results
figure
plot(vib_pos3,vib_mag_ave3,'linestyle','*');
hold on
plot(vib_pos2,vib_mag_ave2,'linestyle','o','color','red');
plot(vib_pos1,vib_mag_ave1,'linestyle','+','color','black');
title('Average Vibration vs Shaft Frequency')
xlabel('Shaft Frequency (Hz)')
ylabel('Vibration Level at Shaft Frequency (dB)')
legend('0.122 kg Imbalance','0.061 kg Imbalance','No Imbalance')

figure
plot(vib_pos3,vib_mag_max3,'linestyle','*');
hold on
plot(vib_pos2,vib_mag_max2,'linestyle','o','color','red');
plot(vib_pos1,vib_mag_max1,'linestyle','+','color','black');
title('Maximum Vibration vs Shaft Frequency')
xlabel('Shaft Frequency (Hz)')
ylabel('Vibration level at Shaft Frequency (dB)')
legend('0.122 kg Imbalance','0.061 kg Imbalance','No Imbalance')

figure
plot(vib_pos3,vib_mag3,'linestyle','*');
hold on
plot(vib_pos2,vib_mag2,'linestyle','o','color','red');
plot(vib_pos1,vib_mag1,'linestyle','+','color','black');
title('Vibration vs Shaft Frequency')
xlabel('Shaft Frequency (Hz)')
ylabel('Vibration level at Shaft Frequency (dB)')
legend('0.122 kg Imbalance','0.061 kg Imbalance','No Imbalance')

figure
plot(pwr_vib_pos3,pwr_vib_mag_ave3,'linestyle','*');
hold on
plot(pwr_vib_pos2,pwr_vib_mag_ave2,'linestyle','o','color','red');
plot(pwr_vib_pos1,pwr_vib_mag_ave1,'linestyle','+','color','black');
title('Average Vibration vs Shaft Frequency')
xlabel('Shaft Frequency (Hz)')
ylabel('Vibration Level at Shaft Frequency (dB)')
legend('0.122 kg Imbalance','0.061 kg Imbalance','No Imbalance')
figure
plot(pwr_vib_pos3,pwr_vib_mag_max3,'linestyle','*');
hold on
plot(pwr_vib_pos2,pwr_vib_mag_max2,'linestyle','o','color','red');
plot(pwr_vib_pos1,pwr_vib_mag_max1,'linestyle','+','color','black');
title('Maximum Vibration vs Shaft Frequency')
xlabel('Shaft Frequency (Hz)')
ylabel('Vibration level at Shaft Frequency (dB)')
legend('0.122 kg Imbalance','0.061 kg Imbalance','No Imbalance')

figure
plot(pwr_vib_pos3,pwr_vib_mag3,'linestyle','*');
hold on
plot(pwr_vib_pos2,pwr_vib_mag2,'linestyle','o','color','red');
plot(pwr_vib_pos1,pwr_vib_mag1,'linestyle','+','color','black');
title('Vibration vs Shaft Frequency')
xlabel('Shaft Frequency (Hz)')
ylabel('Vibration level at Shaft Frequency (dB)')
legend('0.122 kg Imbalance','0.061 kg Imbalance','No Imbalance')