EXPERIMENTAL STUDY OF LOW-SPEED GEARBOX FAULTS
USING VIBRATION AND ACOUSTIC EMISSION SIGNALS

by

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Abstract

With the goal of finding an effective condition monitoring tool for low-speed gearbox applications, vibration-based tests and Acoustic Emission (AE) tests have been performed on several gearbox sets in normal and faulty conditions.

Vibration based tests were conducted both on-site and off-line. Vibration signals from the gearboxes were mostly corrupted by background noise, as was demonstrated by the close proximity of dominant frequencies in the frequency spectrum. Among those, 121.5 Hz was observed as the most significant dominant frequency for both on-site and off-line test regardless of speed changes. This frequency response was speculated to be a property of electrical noise.

The AE tests were conducted on faulty and normal gearboxes at different shaft speeds from 3 to 35 rpm. Hit-based, time-driven and frequency domain AE parameters were used to compare their effectiveness. The count rate, absolute energy and signal strength were found to be good hit-based parameters in this application. The count rate was the best hit-based parameter with the largest parameter difference value at a shaft speed of 25 rpm. The absolute energy was observed to be the best time-driven parameter. Both frequency and time-frequency analysis results indicated that the faulty gearbox has higher spectral peaks in the lower frequency range, which was confirmed numerically by the frequency centroid difference.

AE was shown to be superior to vibration signal analysis in condition monitoring of low-speed gearbox faults with numerous AE parameters identified as showing significant differences between normal and faulty gearbox conditions.
Acknowledgements

This thesis would not have been completed without the support of a group of individuals. I would like to take this time to thank my supervisor Dr. Chris K. Mechefske, for the patient guidance, encouragement and advice he has provided from the beginning of my master study to end. I have been extremely lucky to have a supervisor who respond to my questions so promptly, and who provided countless reviews on my draft thesis.

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<td>AE</td>
<td>Acoustic Emission</td>
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<tr>
<td>CF</td>
<td>Crest Factor</td>
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<td>CI</td>
<td>Condition Indicator</td>
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<td>CM</td>
<td>Condition Monitoring</td>
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<td>CWT</td>
<td>Continuous Wavelet Transform</td>
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<td>DAQ</td>
<td>Data Acquisition</td>
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<td>DWT</td>
<td>Discrete Wavelet Transform</td>
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<td>FC</td>
<td>Frequency Centroid</td>
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<td>FFT</td>
<td>Fast Fourier Transform</td>
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<td>HDT</td>
<td>Hit Definition Time</td>
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<tr>
<td>HHT</td>
<td>Hilbert-Huang Transform</td>
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<td>HLT</td>
<td>Hit Lockout Time</td>
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<tr>
<td>MF</td>
<td>Mean Frequency</td>
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<tr>
<td>MSPS</td>
<td>Million samples per second</td>
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<tr>
<td>PAC</td>
<td>Physical Acoustic Corporation</td>
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<tr>
<td>PCI</td>
<td>Peripheral Component Interconnect</td>
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<tr>
<td>PD</td>
<td>Parameter Difference</td>
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<tr>
<td>PDT</td>
<td>Peak Definition Time</td>
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<tr>
<td>PZT</td>
<td>Lead zirconate titanate</td>
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<tr>
<td>RMS</td>
<td>Root Mean Square</td>
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<tr>
<td>rpm</td>
<td>Revolution per minute</td>
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<tr>
<td>SCR</td>
<td>Silicon Controlled Rectifier</td>
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<td>SSE</td>
<td>Sum of Squares Due to Error</td>
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<tr>
<td>SST</td>
<td>Sum of Squares About the Mean</td>
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<td>STFT</td>
<td>Short Time Fourier Transform</td>
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<td>WT</td>
<td>Wavelet Transform</td>
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<td>WVD</td>
<td>Wigner-Ville Distribution</td>
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List of Symbols

$A_i$  
The magnitude of the $i^{th}$ frequency component

$f_i$  
The $i^{th}$ frequency component

$m$  
The number of fitted coefficients

$n$  
The number of response values

$P_{Faulty}$  
The parameter value of the faulty signal

$P_{normal}$  
The parameter value of the normal signal

$v$  
The residual degree of freedom

$V_{max}$  
The maximum detected voltage

$w_i$  
The $i^{th}$ normalization coefficient

$x$  
The mean value of the signal

$x_i$  
The amplitude of the $i^{th}$ sampled signal

$x_p$  
The peak level of the signal

$x_{RMS}$  
The RMS level of the signal

$ar{y}$  
The mean response value

$y_i$  
The $i^{th}$ fit response value

$\hat{y}_i$  
The $i^{th}$ real response value
Chapter 1

Introduction

Gearboxes are widely used in industry applications to provide speed and torque conversions [1]. Unexpected gearbox failures can result in significant costs due to lost production and ruined products. Therefore, condition monitoring (CM) has drawn great attention and many methods have been developed for gearbox fault detection. Common CM methods include vibration analysis, lubricant analysis, thermography, physical conditions of materials, strain measurements, and Acoustic Emission (AE) monitoring [2]. Vibration analysis is one of most popular tools for detecting gearbox faults. Processing of vibration signals can be categorized into three approaches: time domain, frequency domain, and time-frequency domain. The RMS, kurtosis, crest factor and peak values are popular time-domain based indicators. The Fast Fourier Transform (FFT) method, demodulation method, and wavelet method are widely applied for frequency domain and time-frequency domain signal analysis [3][4].

AE monitoring was initially developed for nondestructive testing, however, it has recently been increasingly used for CM of rotating machinery, particularly in low speed applications [4]. The methods of processing AE signals include hit-based method, time domain method, and frequency domain method [5].

Many experimental studies have been performed to identify and compare the effectiveness of CM methods on gearboxes, however, few of them involve gearboxes for low-speed machinery, particularly of speeds less than 50 revolutions per minute (rpm). Machines with
operating speeds between 0.1 to 10 Hz (or 6 to 600 rpm) are customarily classified as low-speed machinery [5], and they are typically found in cooling tower, ball mill, wind turbine and roll forming machinery. These machines are critical to the production process and normally have high value.

As an example, roll forming is one of the most productive metal forming technologies, which process about 35% to 45% of all the flat steel produced by North American mills [6]. Although efforts have been made to increase roll forming efficiency, due to limitations of punching or cutoff applications, the average production rate for a roll forming line is lower than 180 feet per minute (60 metres per minute) [6]. Therefore, a gearbox-driven roll forming line that produces rolled metal strips, which is a typical low-speed machine was selected as the object of this study. This study is to assess the performance of vibration analysis and AE analysis on low-speed gearboxes with shaft speeds from 3 to 35 rpm.

For the selected forming line, a large number of rollers are used to form a series of incremental forming steps, which impart cross-sectional shape changes on the stock metal strip (see Figure 1-1(a)). Together these steps form a continuous process line with a flat stock metal strip entering and a shaped metal strip exiting. Each set of shaping rolls must exert the exactly correct amount of force to cause the required degree of forming and rotate at exactly the same line speed (25 metres per minute) in order to maintain constant throughput and avoid stretching or buckling of the strip. These rolls are all driven by individual gearboxes that are in turn driven by one main drive shaft. A series of worm gears spaced along the main drive shaft drive the gearbox input shafts that in turn drive parallel spur gears that then rotate the rollers (see Figures 1-1 (b) and (c)).
(a) Metal strip forming rolls

(b) Worm gears that drive the metal strip forming gearboxes

(c) Forming roll drive gears

Figure 1-1. Overview of the processing line and gearboxes
The experiments described in this thesis consisted of three parts and were carried out to collect vibration and AE signals: 1) On-site vibration testing - vibration signals were collected from the gearboxes operating as a part of the process line; 2) Off-line vibration testing - one normal and one faulty gearbox were transferred from the fabrication plant to a laboratory in McLaughlin Hall, Queen’s University, where the on-site operating conditions were simulated and vibration signals were collected from the gearbox housing; 3) Off-line AE testing - AE signals were collected using the same gearbox setup as noted above.

The rest of the thesis is structured as follows. Chapter 2 reviews the previous research on vibration and AE analysis. The experimental setup and testing results for both on-site and off-line vibration testing are discussed in Chapter 3. Chapter 4 presents the AE testing setup as well as the AE signal analysis. Finally, the conclusions are described and future work is discussed in Chapter 5.
Chapter 2

Literature Review

2.1 Vibration Analysis

Vibration is the motion of a machine or a part back and forth from its position of rest [7]. Vibration signals contain a wealth of gearbox condition information. Amplitude and/or phase modulation of vibration signals usually exist when the gearbox has a fault [8]. These signals are usually acquired from a gearbox via transducers mounted on the gearbox casing. Vibration analysis is a powerful tool for detecting typical gearbox faults such as gearbox distortion, fatigue cracking, tooth breakage and other common wear conditions.

2.1.1 Review of signal analysis methods

Sharma and Parey [3] reviewed the common condition indicators (CI) for gearbox fault diagnosis. The fault diagnosis methodologies applied on vibration signals can be categorized into three groups.

2.1.1.1 Time domain analysis

Time domain output represents the vibration signal changes over time. Four typical CIs are listed below:

(1) Peak Amplitude

The peak amplitude measures the maximum vibration level of the signal measured from zero or equilibrium to the maximum positive or negative value of the vibration signal.

(2) Root Mean Square (RMS)
The RMS is proportional to the energy of the signal in the time-domain. It is defined as the square root of the average of the sum of the squares of the signal samples as shown in equation 2.1. This indicator is suited for steady-state signals [4]. It is useful in tracking general fault progression [3].

\[
x_{RMS} = \sqrt{\frac{1}{N} \left[ \sum_{i=1}^{N} (x_i)^2 \right]}
\]

where \(x_i\) is the amplitude of sampled signal, \(i\) is the sample index, and \(N\) is the number of samples.

(3) Crest Factor (CF)

The crest factor is defined as the ratio of the peak level of vibration signal to the RMS level, as represented in equation 2.2:

\[
CF = \frac{x_p}{x_{RMS}}
\]

The CF is sensitive to the impulsive nature of vibration signals at the early stages of fault development when a small number of spikes appear, such as those caused by local tooth damage. However, the CF value will decrease if the impulsive nature begins to dominate the signal, because at this situation the RMS level increases to a high value while the peak value keeps at a relatively constant level. Therefore, CF is a good indicator of the early stage of gearbox fault development [3].

(4) Kurtosis
Kurtosis is the fourth moment of a signal distribution which measures the size of the tails of the given distribution. It is normally used as an indicator of the major peaks in a set of data[9] and the calculation is given by equation 2.3.

\[
K = \frac{N \sum_{i=1}^{N} (x_i - \bar{x})^4}{(\sum_{i=1}^{N} (x_i - \bar{x})^2)^2}
\]

where \( \bar{x} \) is the mean value of the given signal.

A kurtosis value of approximately 3 for Gaussian distributed noise is usually used as a reference value for gear wears or breakage which increase the vibration level and always result in a high kurtosis level [3].

2.1.1.2 Frequency domain analysis

Frequency domain analysis is the most widely used approach in gearbox fault detection. Previous developments on the FFT algorithm have made it easy and efficient to obtain the frequency spectra. In addition to the basic FFT method, a variety of further signal processing techniques have been proposed to improve spectra analysis. Windowing is a simple but efficient way to force the end points of a sampled data set to zero, which can prevent energy loss and false peaks (known as “leakage”) in the frequency domain [10]. Common types of window include Hanning, Hamming, Blackman and Bartlett-Hanning. Envelope detection is normally coupled with FFT analysis to extract the characteristic defect frequencies which are often overwhelmed by noise in the time domain signal [11].

Two frequency domain parameters that can further reduce the amount of information in the processed frequency spectra to a single number are listed below:

(1) Dominant Frequency
Dominant frequency components refer to the frequency components that contain maximum or near-maximum power in the frequency domain. These components are normally the frequency of most interest, which might be the shaft rotating frequency, gear mesh frequency and their multiples. Therefore, analysis of the dominant frequency is a good starting point for frequency analysis.

(2) Mean Frequency (MF)

Mean frequency represents the overall vibration energy in frequency domain, and it is calculated by:

$$\text{MF} = \frac{1}{N} \sum_{i=1}^{N} f_i$$

where $f_i$ is the $i^{th}$ measurement of the frequency spectrum of signal $x$, and $N$ is the total number of spectrum lines.

2.1.1.3 Time-frequency domain analysis

Although frequency domain analysis takes advantage of its sensitivity and is widely used for condition monitoring, it has a limitation when used for processing non-stationary signals. To handle this problem, time-frequency domain analysis was proposed as it could simultaneously represent both frequency and time information on a single chart.

The conventional time-frequency methods include Short Time Fourier Transforms (STFT), Wigner-Ville Distributions (WVD), and Wavelet Transforms (WT). Compared to the other two methods, one main advantage of the WT is its ability to produce a flexible resolution by employing different windows based on the frequency level [12]. Two basic forms of the WT are continuous wavelet transforms (CWT) and discrete wavelet transforms (DWT).
The common continuous wavelets include Morlet, Mexican hat, Poisson; while Haar, BNC, Daubechies wavelet are the popular discrete wavelets.

Even though the WT has better resolution with flexible windows, the uncertainty principle still limits the time and frequency resolution. A method proposed by Huang et al [13], named Hilbert-Huang Transform (HHT), only involves instantaneous frequency therefore is not constrained by the uncertainty limitations. The HHT method applies the Empirical Mode Decomposition (EMD) process to the signal, which decomposes the signal into the intrinsic mode functions (IMF), finally the energy-time-frequency distribution is obtained by employing the Hilbert transform on the IMFs [13]. This method also has the advantage of computing efficiency, which makes it suitable for signal analysis of large data sets [14]. Application of time-frequency techniques associated with time domain indicators have been proved to produce good fault diagnosis results by previous studies [15][16].

2.1.2 Limitation of vibration analysis for low-speed gearboxes

While vibration based techniques are well established for medium and high speed gearboxes, useful vibration monitoring tools for low-speed gearboxes have yet to be developed. There have been some attempts to develop vibration based techniques for slow speed gearboxes. Berry [17] stated that displacement is the best parameter for low-speed vibration measurement. Robison and Canada [18] described the slow speed technology developed by CSI. They concluded that application of low noise accelerometers of 500mV/g sensitivity, proper cables and portable data collectors with sufficient dynamic range can make vibration analysis meaningful for low-speed machinery. Wang et al [19] presented the instantaneous time-frequency spectrum constructed by local mean
decomposition. They also proposed the energy dispersion ratio as a new parameter which is sensitive to deterioration scenarios of low-speed helical gearboxes.

However, most of the published techniques are restrained to certain conditions. Vibration monitoring of slow speed gearbox is still difficult. Holroyed [20] summarized four reasons for the difficulty, which include 1) Decrease of the energy release rates; 2) Low defect frequencies that are usually overwhelmed by background noise; 3) Long time records required for further processing; 4) The large mass and stiffness of slowly moving structures. In addition, Mba et al. [21] pointed out that most data collectors, spectrum analyzers and sensors are equipped with high-pass filters at around 5 Hz to filter out the low frequency instrument noise, some important once-per-revolution fault features are left out due to this reason.

2.2 Acoustic Emission Analysis

While the development of new vibration based techniques for slowly rotating gearboxes is facing some challenges, a high frequency analysis technique named Acoustic Emission (AE) has increasingly been proposed as a powerful fault detection tool for low speed gearboxes.

2.2.1 Nature of Acoustic Emission

Acoustic Emission refers to the transient elastic waves within a material (typically metal), which are generated by the rapid release of localized stress energy caused by material deformation under mechanical loading [22]. This method was originally carried out by J. Kaiser in 1950, and B. H. Schofield initiated the term “AE” in 1954 [23]. It was initially
developed for nondestructive testing of static structures but, as the technique developed, AE became accepted as a CM method of rotating machines.

The basic AE wave is a stress pulse corresponding to a permanent displacement of grain boundaries or crack growth in a material. The source activities are normally completed in just a few microseconds; therefore, the stress pulse has a short duration, which results in the high frequency response of AE signals. Most of the released energy is within the 1 kHz to 1 MHz range, however, it is customarily accepted that the typical frequencies associated with AE activities range from 20 kHz to 1 MHz [22].

All AE processes begin with stress but can result from several source activities, such as slip and dislocation movements, melting, phase transformation, crack growth and corrosion. The primary source is the mechanical deformation in the originally fabricated material, which occurs when the material undergoes plastic deformation or is loaded at or near its yield stress. Generation and propagation of cracks are the most recognized sources of AE in metal, and the most detectable area is the crack-tip as the stress level shown in front of crack-tip can be several times higher than the surrounding area.

The AE wave radiates from the source in all directions and is reflected off the boundaries of the object. Due to wave propagation, the signal detected by a sensor is typically a combination of several parts of the initial waveforms. Because of attenuation, the intensity of a detected signal decreases as the AE wave travels through the structure. Three main reasons for attenuation are geometrical spreading, energy absorption and structural reflection. Measurements of the effects of attenuation can be made with a simulated AE source. The most widely used apparatus is a Hsu-Nielson Source. Another important aspect of wave propagation is velocity; it is required for source location determination. Lamb
waves are of the primary concern since they provide the best indication of wave propagation at distances larger than the material thickness [22].

The detected AE signal can be classified as burst signal and continuous signal types. Bursts are transient signals observed during the formation of damage, such as crack growth propagation within a material or during impact and breakage [24][25]. Continuous signals are generated when multiple transient waves repeatedly occur and overlap with each other, which make the envelope of the signal amplitudes become constant. Elastic noise, rubbing and plastic deformation in ductile materials can contribute to continuous AE waves [23][26].

2.2.2 Data collection equipment

As shown in Figure 2-1, a typical acoustic emission system contains a sensor, preamplifier, filter, and amplifier, along with the measurement, display, and storage equipment.

![Figure 2-1. A typical AE system setup][27]
Acoustic Emission sensors are mostly piezoelectric type sensors, which are made from ceramic like lead zirconate titanate (PZT). Sensor selection is based on the operating frequency, the sensitivity and the environmental characteristics. There are two types of AE sensor – resonant and broadband. Because most AE techniques only use AE features instead of the actual waveform, resonance sensors that are only sensitive to a certain frequency range are widely used in AE testing [27]. The preamplifier is connected right after the sensor to amplify the initial signal to the optimal level for measurement. Typical amplification gains are 40 or 60 dB. To minimize the electrical interference, the preamplifier is normally placed close to the sensor. Then the signal is passed through a band pass filter to eliminate the background noise. After that, the signal is amplified again and passed to the signal conditioning equipment before it is finally extracted and stored in a computer for further processing [27].

2.2.3 Review of AE signal analysis methods

Most commercial AE systems can record AE signals in two ways: hit-based and time-driven. The time-domain features extracted from hit-based and time-driven AE data are effective for fault detection. Frequency-domain as well as time-frequency domain methods are widely-used approaches as well.

2.2.3.1 Hit-based data analysis

A common detection strategy used in AE systems is to compare the AE signal against a certain threshold. The threshold is typically set on the positive side, slightly above the background noise level. It can be set to a fixed value, or float regularly if the noise level varies over time. A “hit” is detected once the signal level surpasses the threshold. The
determination of hit is associated with three parameters: the hit definition time (HDT) that defines the maximum time between threshold crossing; the hit lockout time (HLT) that specifies the time after HDT and before a new hit can be detected; and the peak definition time (PDT) that specifies the time allowed for peak determination after a hit is detected [24]. Figure 2-2 below represents a hit-based detection with typical AE features.

![Figure 2-2. The hit detection and typical AE features [24]](image)

According to the PCI-2 Based AE system User’s Manual [28], the common hit-based AE parameters are defined as follows.

1. **Peak Amplitude**

   The peak amplitude measures the maximum (positive or negative) AE signal level during an AE hit. It is the most often used parameter as it is closely related to the magnitude
of the detection event [23]. The peak amplitude expressed in dB is calculated by equation 2.5.

\[ dB = 20 \log \left( \frac{V_{\text{max}}}{1 \mu V} \right) - (\text{Preamplifier Gain in dB}) \]  

(2) Rise Time

Rise time is defined as the time between the start of an AE hit and the peak amplitude of the hit. It is related to the propagation of an AE wave between the source and the sensor, therefore, it is normally used as a criterion for the noise filter [29].

(3) Duration

Duration is the time difference between the first and last threshold crossing of the AE signal from the AE threshold. It is useful in identifying different types of sources.

(4) Counts

Counts, also known as “ring-down counts”, refers to the number of detected AE signal excursions over the threshold. The number of counts strongly depends on the threshold level and operating frequency. Previous study has proved that counts is a good parameter for the detection of small defects, however, it is unable to track the progression of the defect since it stops increasing when the defect reaches a certain size [30].

(5) “Energy”

“Energy” is derived from the integral of the rectified voltage signal over the duration of the AE hit and is gain related. As an example, the resolution of “energy” is 10 microvolt-seconds (mVs) at 0 dB instrument gain, while at 20 dB, the resolution becomes 1 mVs.
(6) Signal Strength

Signal Strength is also known as “relative energy” and “MARSE energy”, it has the same definition as energy, except that it is calculated over the entire AE signal dynamic range and is independent of gain. The resolution of signal strength is 3.05 picovolt-seconds (pVs) at 1 MHz or greater sampling rate.

2.2.3.2 Time-driven data analysis

Time-driven AE data presents the change of continuous AE activities during time, which provides a trending type of result. Common time-domain parameters including RMS, kurtosis and crest factor are widely applied.

Absolute energy is another time-based feature which measures the true energy as shown in figure 2-2. It is derived from the integral of the squared voltage signal divided by the reference resistance (10 k-ohm) over the duration of the AE signal. The resolution of absolute energy is 0.000931 attoJoules (aJ) when the sample rate is at or greater than 1 MHz. This feature is good for monitoring continuous signals as it is independent of the hit based activity.

2.2.3.3 Frequency analysis

Like in the vibration-based analysis, frequency analysis plays an important role in the AE-based analysis, which include FFT based frequency methods and time-frequency methods such as STFT and WT.
One common frequency-domain parameter is the frequency centroid (FC) defined in equation 2.6. It can provide information about the nature of the source and help in distinguishing different sources of AE.

\[
FC = \frac{\sum A_i f_i}{\sum A_i^2}
\]

where \( f_i \) is the \( i^{th} \) frequency component and \( A_i \) is the corresponding magnitude.

### 2.2.4 Application of AE on gearbox condition monitoring

The AE techniques have the advantage in identifying early stage defects since AE is produced at the microscopic level and is highly sensitive to the defect situation when compared to other CM techniques. In addition, AE mainly detects high frequency elastic waves, which means it is relatively insensitive to typical mechanical background noises [31]. These advantages make the AE technique a good choice for monitoring of gearboxes, particularly in low speed applications.

Several studies have been conducted to investigate the AE based techniques on gearbox condition monitoring. Singh et al [32] applied the AE based technique on a single tooth bending machine to detect tooth breakage of the tested spur gear. They concluded that AE offered early detection of crack initiation and growth over the traditional vibration-based method. They also performed an assessment of the transmissibility of AE signal within a gearbox and concluded that most of the strength loss occurs at the interfaces and can be used to predict the cumulative losses associated with different paths of propagation. Tan and Mba [33][34] studied the source mechanism of AE activities and the influence of gearbox operating parameters through a series of experiments. They concluded that the
asperity contact under sliding and rolling of the meshing gear teeth surfaces generate the AE activities. They also observed that temperature, speed and load have influence on levels of AE activity, while under isothermal conditions, the speed has a more significant influence on AE RMS levels relative to the load.

2.3 Condition monitoring of worm gears

As reviewed in the previous sections, vibration analysis is a most popular method for gearbox condition monitoring, but there are limitations in its application to slowly rotating gearboxes. The AE technique is effective in low speed range and has high sensitivity to early defect information.

However, most published work reports experimental studies on spur and helical gears to identify and assess the effectiveness of vibration and AE based CM methods and only few of them involve CM of worm drives. Peng and Kessissoglou [35] studied the correlation of wear debris analysis and vibration analysis by investigating different operating conditions of a worm gearbox. It was concluded that wear debris analysis provides information on the wear rate and mechanism of the gears, and vibration analysis is useful for the monitoring of the bearings. Elforjani et al [36] investigated the effectiveness of AE and vibration analysis in detecting defects on worm gears which operate at 150 or 300 rpm. They concluded that AE RMS and energy are more reliable, robust and sensitive to the detection of worm gear defects.
For a worm drive which is a gear arrangement consisting of a worm meshing with a worm gear (see figure 2-3), the sliding motion is the only transfer of power and it normally results in significant sliding friction. Many studies have been conducted to investigate the relationships between AE and sliding. Dornfeld and Handy [37] presented the sensitivity of AE to the onset of motion when slip occurs. The AE generated from the sliding contact was investigated by Jiaa and Dornfeld [38]. They proposed that AE is useful for tribological studies, and the increase of loading and sliding speed will increase the AE RMS level. Lingard et al [39] studied sliding wear using AE and concluded that AE signals are related to the wear rates, the frictional work inputs and established tribological contact variables. As sliding is the dominant motion of a worm drive, AE offers the opportunity for worm gear fault detection.

Besides sliding, high gear ratio is another common feature for most worm drives. It is possible that both slow speed and high speed rotating parts may exist in one worm gearbox.
Therefore, vibration analysis is also a possible method to monitor worm gearboxes even though limitations exist for its application in low-speed machines.

The aim of this thesis is to find a suitable fault detection tool for the low speed gearbox which consists of a worm drive. The effectiveness of the vibration and AE techniques were examined using experimental approaches. The detailed experimental set up and the signal analysis results for vibration and AE testing are presented in Chapter 3 and Chapter 4 respectively.
Chapter 3

Vibration Testing

The vibration experiments consisted of on-site experiments and laboratory experiments. On-site tests were conducted to collect the vibration data in a real operating situation, while the laboratory experiments simulated different, but similar operating conditions to fully test the performance of vibration-based methods. This chapter describes the on-site and laboratory vibration experimental setups and discusses the vibration results using time-domain and frequency-domain approaches.

3.1 On-site Experimental Setup

The first stage of vibration testing was the on-site vibration signal collection. As shown in Figure 3-1, the on-site data collection system consists of accelerometers, cables, prototype tools used to hold cables, data acquisition (DAQ) hardware (Photon II), and a computer that runs the signal analysis software (RT Pro) [40].

![Figure 3-1. On-site data collection and analysis system](image)

Figure 3-1. On-site data collection and analysis system
In this section, the descriptions of the on-site experimental setup, including measurement constraints, data collection instrumentation, and measurement methods are provided.

### 3.1.1 Measurement constraints and data collection system design

A safety screen is located around the gearboxes and must remain in place during the entire operating period for safety. It was not feasible to mount the transducer on the measuring surface of the gearbox directly, nor arrange the cable properly to avoid entanglement with the rotating shaft and gears.

Therefore, a steel rod with a sensor fixed at one end by cable ties and tape was used to hold the cable and mount the sensor (see Figure 3-2 (a)). Since the bearing housing and gearbox top surfaces are both ferromagnetic materials, a magnetic mount was used to take advantage of being easily removable. Moreover, a foam cushion was placed between the steel rod and the safety screen to reduce the effect caused by any screen vibration (see Figure 3-2 (b)).

![Prototype rod used to hold cable and sensor](image1)

![Installation](image2)

**Figure 3-2. Prototype data collection system and installation**
3.1.2 Instrumentation

3.1.2.1 Sensor

The most commonly used vibration transducers are of the piezoelectric type. Two sizes of accelerometers (Dytran 3035A and Dytran 353B33) (see Figure 3-3 (a)) have been used to take vibration measurements on several gearboxes. Table 1 below summarizes the properties of the two accelerometers.

**Table 1. Properties of Accelerometers [41][42]**

<table>
<thead>
<tr>
<th>Model</th>
<th>Sensitivity (mV/g)</th>
<th>Frequency Response (Hz)</th>
<th>Weight (g)</th>
<th>Size (in) Hex x Height</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dytran 3035A</td>
<td>100</td>
<td>0.5 to 10000</td>
<td>2.5</td>
<td>0.28x0.43</td>
</tr>
<tr>
<td>Dytran 353B33</td>
<td>100</td>
<td>0.7 to 6500</td>
<td>50</td>
<td>0.70x1.20</td>
</tr>
</tbody>
</table>

Comparing these two sensors, the Dytran 3035A has a smaller physical size and wider frequency response range, while the sensitivity is the same for both accelerometers. As the acceleration measurement was performed with a relatively massive gearbox, the weight of the accelerometer should only have a slight influence on the measurement. Instead, due to the existence of oil and dirt on the measuring surface, stability of the sensor installation is also a concern. The Dytran 3035A has the advantage of easy installation as it can easily go through the safety screen, while the Dytran 353B33 may offer more stable installation to the measuring surface as a larger magnet is used.
The hole size of the safety screen is not large enough to allow the Dytran 353B33 to go through. Two holes were combined to one by cutting the rail in between at four locations, which allowed the measurements of gearbox #2, 11, 19 and 21 using the Dytran 353B33. As shown in Figure 3-3 (b), the two accelerometers were mounted on the gearbox top surface to measure the vibration signals at the same time. Figure 3-4 shows the four selected sets of RMS values calculated from vibrations collected by these two types of accelerometers, where the x-axis records the gearbox set number and the y-axis presents the RMS value.

It was found that all measured vibration amplitudes were relatively low (at around $7 \times 10^{-4}$ g’s), therefore, even tiny changes in the measurement protocol may vary the result. From these preliminary measurements, there is no clear indication of which kind of accelerometer gives more accurate data. However, since they have uniform sensitivity (100 mV/g), the similar outputs indicate the comparable quality of installation for both kinds of accelerometers.
Due to the existence of the safety screen, using of the larger accelerometer (Dytran 353B33) is not recommended as it requires enlargement of the hole size for sensor installation, which will lower the protection level while increasing the cost. Therefore, the Dytran 3035A was selected for on-site vibration tests.

3.1.2.2 Data Acquisition Device (DAQ)

The data acquisition device used for the preliminary measurements was a Photon II produced by LDS DACTRON. This device has up to four input channels and one output channel. It offers a measurement dynamic range of 110 dB and a 84.2 kHz real-time data collection rate [40].

Uniform time sampling was used for collection of the preliminary measurements. The rotating speed and gear mesh frequency of each gearbox are relatively low (see Table 2), and according to the preliminary measurement results, the dominant frequency components for all measurements are less than 500 Hz. Therefore, the sampling rate was set as 5,120 Hz.
The collected raw vibration data can then be streamed to a PC to run the “RT Pro” software, where the vibration signals were saved as txt. files and then transferred into MATLAB to perform different analysis procedures.

**Table 2. Gearbox operating conditions**

<table>
<thead>
<tr>
<th>Line Speed (m/min)</th>
<th>Roller Diameter (m)</th>
<th>Shaft Rotating Speed (rpm)</th>
<th>Shaft Rotating Frequency (Hz)</th>
<th>Teeth Number of Gear</th>
<th>Gear Mesh Frequency (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>12</td>
<td>0.1524</td>
<td>25.06</td>
<td>0.42</td>
<td>38</td>
<td>15.8</td>
</tr>
</tbody>
</table>

**3.1.3 Selection of measurement location**

To find the most suitable location for data collection, the first several sets of vibration data were collected from as many different locations on each gearbox as were practical.

As per Figure 3-5 (a), the amplitudes of most vibration signals collected from the upper bearing housing in the vertical and horizontal directions are close to each other, while the data obtained from the lower bearing housing in the horizontal direction is slightly lower.

A possible reason for the amplitude difference is the distance between transducer and bearing. The upper part of the housing, which provides stronger signals, is preferred for data collection. Moreover, it was found that the frequency spectra obtained from vertical and horizontal signals after FFT contain similar spectral components, which all show that 364.5 Hz is the dominant frequency. Therefore, one measurement on the bearing side should be able to collect representative vibration signals that are sufficient for analysis. Considering the ease and stability of installation, the top surface of the upper bearing is the most suitable.
In addition to the bearings, the gearbox top surface is another good choice for data collection. Figure 3-5 (b) compares the vibration signals obtained from several bearing locations.
locations and the gearbox locations. Although the absolute difference of vibration data from various gearboxes is small regardless of measurement location, the percentage of variance for signals collected from the gearbox is relatively high due to the low vibration level. The time domain waveforms also show that the signal from each gearbox (especially the lower surface) contains more repeated impulses, which turn out to have high frequency (1,012 Hz or higher) components. However, since the operating speed is very low (0.42 Hz), and the gear mesh frequency is as low as 15.8 Hz, high frequency components are not of interest for fault analysis and are likely due to environmental disturbances. Based on such results, the bearing top surface was selected as the best measurement location.

Each gearbox set includes four single gearboxes (each gearbox driving one pair of rollers). As per a technician’s description, the whole set is typically removed for checking when some abnormal situation happens. In such cases, one measurement per set has been considered as one option to reduce the number of measurements required. However, the preliminary results show that the performance of each individual gearbox is somewhat independent from one another.

For example, Figure 3-6 (a) shows that gearbox #8 has higher vibration amplitude while all other three gearboxes have relatively low vibration levels. Also, Figure 3-6 (b) shows that gearbox #19 only contains low frequency components while all the other three gearboxes in that set have higher amplitudes and similar frequency components. Therefore, even if replacement and repair are typically performed on the entire gearbox set, it is necessary to take vibration signal measurements from each single gearbox.
3.2 Off-line (Laboratory) Experimental Setup

On-site measurement has the limitation of fixed operating speed, existence of safety screen and the measurement surface with oil and dirt, which may affect the vibration measurement.
and result in errors. Moreover, the replacement and repair of gearbox sets are not trackable, which reduces the comparability of data directly collected from the operating gearboxes. Therefore, one normal and one faulty gearbox were transferred from the fabrication plant to a laboratory in McLaughlin Hall, Queen’s University, where the on-site operating conditions were simulated and vibration signals were collected from the gearbox housing.

### 3.2.1 Experimental Facilities

![Figure 3-7. Overview of off-line experimental set-up](image)

Off-line experiment facilities were set up on a large table as shown in Figure 3-7. Two gearboxes (the left one is in normal condition, while the right one has faults) were mounted to the table by bolts and nuts. A 3/4 HP LEESON Motor (see Figure 3-8 (a)) was connected to run one gearbox for each test. The KBPC-240D SCR (Silicon controlled rectifier) was used with the motor to control the input power of 115 or 230 volts rectified AC (see figure
Steel plates with different height were placed under the motor, gearboxes and bearing housing to meet the connection requirement.

Since all gearboxes, motor and bearing housing were fixed to one table, the vibrations of each part were mixed and transferred to the table, which resulted in the table vibration that affected the signal collected from bearing housing or gearbox. To reduce the effect of table vibration, anti-vibration pads as presented in Figure 3-8 (c) were placed in between the table and motor/gearbox/bearing housing. In addition, nylon casings (see Figure 3-8 (d)) were placed over the roller in order to simulate the metal strip passing through two rollers.

(a) 3/4 HP LEESON 1750 rpm 56C TEFC 90VDC motor
(b) KBPC-240D SCR
(c) Anti-vibration pad
(d) Nylon cases

Figure 3-8. AE experimental components
All facilities except for the two gearboxes were removable. Appropriate installation of the removeable parts (include motor, bearing housing, and rollers) to each gearbox formed a complete unit for gearbox operation and vibration measurements.

3.2.2 Vibration signal collection method

The sensor used for off-line tests is a new version named Dytran 3035B, which has the same size and performance as Dytran 3035A. The measurement surfaces were cleaned before each test, and the measurement locations were marked by a marker to avoid any location change and therefore improve repeatability. The selected measurement locations include gearbox top, bearing housing top, upper bearing top, lower bearing top and motor plates as shown in Figure 3-9.

![Figure 3-9. Sensor installation locations](image)
It was observed that the RMS values of the vibration signals increase with the shaft rotating speed for both normal and faulty gearbox units. For the normal gearbox, there is no obvious difference between the signals collected from different bearing locations, while the strength of gearbox vibration signal is approximately half of the bearing signals. The vibration data collected from different bearing locations of the faulty unit are also close to each other and they are relatively higher than the data collected from the normal unit at similar rotating speed and location. Therefore, one measurement at the bearing side is sufficient for analysis, and the bearing housing top was selected since it provided the strongest vibration signals.

The difference between the normal and faulty unit is more obvious when comparing the gearbox signals. It was found from Figure 3-10 that the faulty RMS level is around twice that of the normal one when the speed increased to 18 rpm. Therefore, the gearbox top is a suitable position for vibration measurement as the collected vibration signals can be used to distinguish the normal and faulty units.

Two input lines were set up during each measurement, including one sensor mounted at the gearbox or bearing housing, with the other sensor attached to the motor plate. Considering that the vibration of the motor is one of the main disturbances to the collected gearbox vibration signal, such installation can ensure a simultaneous collection of motor vibration signal and gearbox/bearing housing signal for better data comparison.
The sampling rate was set to 5,120 Hz as the rotating speed as well as the dominant frequency are both low. The measuring time was fixed to 1 minute. Due to the facility limitations, the exact load applied to the rollers cannot be measured, two rollers were about
to touch each other during the tests to simulate the minimum loading condition. The rotating speed of the output shaft was adjusted from 5 rpm to 30 rpm by the SCR drive. Vibration signals were collected at different rotating speeds while the load applied was kept at the minimum level.

3.3 Time domain vibration signals and analysis

Time domain analysis was conducted on the on-site and off-line vibration results to investigate the performance of time-domain parameters. The time domain waveforms are presented and discussed in this section.

3.3.1 Time domain vibration signals

3.3.1.1 On-site results

As discussed before, the bearing housing top was selected as the measurement location for on-site vibration tests, and measurements were taken from each gearbox since the performance of each individual gearbox is independent from one another. Figure 3-11 below shows the time-domain waveforms of one set of gearboxes consisting of #13 to #16.

It was found that the vibration level was low at around $3 \times 10^{-3}$ g’s for all gearboxes, while gearbox #16 had slightly higher signal energy. However, no signs of a dominant underlying shaft rotational component can be directly observed from the time-domain waveforms.
3.3.1.2 Off-line results

Off-line tests were performed on normal and faulty units by measuring the vibration signals from the motor plate, bearing housing top, and gearbox top under different rotating speed. Time-domain waveforms of normal and faulty unit under rotating speed of approximately 25 rpm are presented in Figures 3-12 and 3-13.
As can be observed, the time domain waveforms for all measurement locations of both the normal and faulty units show a peak modulation with a repetition rate of approximately every rotation period. There are two possible explanations for this response. One is known as beating, which may be due to the interaction of two vibration sources with close frequency. It may also be an amplitude modulation caused by a single excitation that is being modulated by a significantly lower frequency signal [10].

Figure 3-12. Comparison of time domain waveforms of normal gearbox unit under rotating speed of 26.79 rpm (rotation period of 2.24 s)

Figure 3-13. Comparison of time domain waveforms of faulty gearbox unit under rotating speed of 24.51 rpm (rotation period of 2.45 s)
The modulations of the normal gearbox signals are relatively gentle but more obvious, while small spikes can be observed from the faulty gearbox waveforms. The difference is clear but cannot be used to distinguish the faulty gearbox directly. More time-domain indices and spectrum analysis are used to compare the signals.

### 3.3.2 Discussion of time domain vibration signals

#### 3.3.2.1 On-site signal analysis

It was assumed that most of the gearboxes in the processing line operated in a normal condition as the production line ran properly during each data collection period. Seven sets of data, five with samples of 1 minute from 23 gearboxes and two with samples from 20 gearboxes, were analyzed by computing three time-domain indices as shown in Figures 3-14 to 3-16 to determine the overall vibration level for the normal operating gearboxes under a rotating speed of around 25 rpm.

The RMS levels for seven sets of data are shown in Figure 3-14 (a), where the sets are distinguished by different colors. It was found that the RMS values for most samples are in the low range, while only a few of them jump to around 0.01 g’s. It is more intuitive by looking at the histogram with Pareto line in Figure 3-14 (b), around 75.5% of the RMS values fall within the range of 0.0005 to 0.0025 g’s. For such low-speed gearboxes, the impact of environmental and near machine electrical noise may affect vibration measurements. In addition, unstable sensor mounting or a small change in the measurement location may also result in the output deviation. However, the RMS range for the normal
condition can be narrowed to $0.0005 \sim 0.005 \text{ g's}$ as this range covers 90% of the collected samples.

(a) Comparison of the RMS values for seven sets of data

(b) The distribution of the RMS levels and the corresponding Pareto line

Figure 3-14. Comparison of the RMS level
The peak-to-RMS ratio is presented in Figure 3-15 (a) as Crest Factor, and the data distribution is summarized in Figure 3-15 (b). Approximately 55.5% of samples are almost equally distributed among the top three CF ranges from 3.5 to 6.5, while around 27.7% of samples fall into the range of 6.5 to 9.5. The appearance of random spikes may be one reason for the relatively wide variety in CF level.

(a) Comparison of the Crest Factors for seven sets of data

(b) The distribution of the Crest Factor levels and the corresponding Pareto line

Figure 3-15. Comparison of the Crest Factor level
It can be observed from Figure 3-16 (a) that most points are in the interval of 2 to 4 in the Kurtosis range. Figure 3-16 (b) shows that 51.6% of samples have kurtosis values between 2.8 and 3.8, and the range of 1.8 to 4.8 covers 83.9% of sample Kurtosis values. There are some outliers with the Kurtosis level larger than 32, and this indicates the existence of sharp peaks with a high value for some samples.

(a) Comparison of the Kurtosis levels for seven sets of data

(b) The distribution of the Kurtosis levels and the corresponding Pareto line

Figure 3-16. Comparison of the Kurtosis level
3.3.2.2 Off-line signal analysis

Seven sets of data, five collected from the faulty gearbox and two measured from the normal gearbox with samples of 1 minute from three measurement locations under different rotating speeds, were analyzed by computing the same time-domain parameters to distinguish between the normal and faulty gearboxes. Regression analysis was performed to draw fitting lines that indicate the trend of parameters level change with speed. The fitting models were selected based on the Adjusted R-square value, which presented the degree-of-freedom adjusted coefficient of determination defined by equation 3.4. An Adjusted R-square value closer to 1 indicates a better fitting [43].

\[
SSE = \sum_{i=1}^{n} w_i (y_i - \hat{y}_i)^2 \quad 3.1
\]

\[
SST = \sum_{i=1}^{n} w_i (y_i - \bar{y})^2 \quad 3.2
\]

\[
v = n - m \quad 3.3
\]

\[
Adjusted \ R - square = 1 - \frac{SSE \ast (n - 1)}{SST \ast (v)} \quad 3.4
\]

where SSE represents the Sum of Squares Due to Error, SST represents the Sum of Squares About the Mean, \( w_i \) is the \( i^{th} \) normalization coefficient, \( y_i \) is the \( i^{th} \) fit response value, \( \hat{y}_i \) is the \( i^{th} \) real response value, \( \bar{y} \) is the mean response value, \( v \) is the residual degrees of freedom, \( n \) is the number of response values, and \( m \) is the number of fitted coefficients.

(1) RMS
(a) The RMS of bearing housing top signals
(b) The RMS of gearbox top signals
(c) The RMS of motor plate signals collected together with bearing housing top
(d) The RMS of motor plate signals collected together with gearbox top
(e) The Motor-to-Bearing RMS ratio
(f) The Motor-to-Gearbox RMS ratio

Figure 3-17. Comparison of the RMS level between the faulty and normal gearboxes
Figures 3-17 (a) and (b) compare the RMS level of faulty and normal gearbox signals collected from bearing housing top and gearbox top. The red and green points represent faulty and normal gearbox results respectively.

It can be observed that the RMS of both faulty and normal signals at around 25 rpm shaft speed fall into the region of 0.0005 to 0.005 g’s which was obtained from on-site results as a normal level, while the faulty gearbox signals have larger RMS levels over the entire comparison and the gap increases with the increase of shaft rotating speed.

However, similar trends were found from Figures 3-17 (c) and (d) for the motor plate signals, which have relatively larger magnitude and obvious difference between the faulty and normal gearbox results. In addition, the motor-to-bearing housing top and motor-to-gearbox top RMS ratios were calculated and presented in Figures 3-17 (e) and (f). For both normal and faulty gearboxes, a constant ratio level was achieved when the shaft rotating speed increased beyond a certain level. This indicates the significant influence of motor vibration on the data collection from gearbox and bearing housing top, which means that the gap of RMS level might due to the motor vibration instead of the appearance of gear damage. Therefore, comparison of the RMS level is not sufficient for fault detection.

(2) Crest Factor and Kurtosis

Figures 3-18 and 3-19 compare the Crest Factor and Kurtosis level correspondingly. The data points of CF and Kurtosis do not have clear trend, as was demonstrated by the low Adjusted R-square values. A low value of CF and Kurtosis with some fluctuations can be observed in both faulty and normal gearbox sets of data. In The normal gearbox sets of data even tend to have relatively higher CF and Kurtosis values when speed increases,
which implies that more outliers are present in the normal gearbox signals. Therefore, the CF and Kurtosis of the vibration signals are not appropriate for the fault detection of low speed gearboxes.

(a) Bearing housing top signals  
(b) Gearbox top signals

Figure 3-18. Comparison of the Crest Factor between the faulty and normal gearboxes

(a) Bearing housing top signals  
(b) Gearbox top signals

Figure 3-19. Comparison of the Kurtosis between the faulty and normal gearboxes
3.4 Frequency domain vibration signals and analysis

Spectrum analysis was applied to the vibration signals to extract the frequency information of interest, such as the dominant frequency and the associated sidebands. The frequency-domain spectra obtained by FFT algorithm are presented and discussed in this section.

3.4.1 Spectrum plots of the on-site vibration signals

As mentioned in previous sections, the replacement and repair of on-site gearbox sets are not trackable, which reduces the comparability of data collected from the operating gearboxes. However, the frequency-domain spectra for each set of data are similar, four samples of each set were selected to present the frequency information as shown in Figure 3-20.

(a) Spectra for July 5th data
(b) Zoomed spectra for July 5th data
Figure 3-20. Frequency-domain spectra for four sets of on-site data.

(c) Spectra for July 28\textsuperscript{th} data

(d) Zoomed spectra for July 28\textsuperscript{th} data

(e) Spectra for Aug 25\textsuperscript{th} data

(f) Zoomed spectra for Aug 25\textsuperscript{th} data

(g) Spectra for Sep 30\textsuperscript{th} data

(h) Zoomed spectra for Sep 30\textsuperscript{th} data
The spectral plots with frequency up to the Nyquist frequency are presented in Figures 3-20 (a) (c) (e) and (g), while Figures 3-20 (b) (d) (f) and (h) show the zoomed spectra with frequency up to 600 Hz. Although the dominant frequency component as well as its amplitude varies for each set of data, they all fall into the range of 250 to 400 Hz, and the component at 364.4 Hz is of high interest. In addition, the appearance of the component at around 60 Hz and the impulses at approximately 120 Hz intervals may indicate the existence of motor related influence.

### 3.4.2 Spectrum plots of the off-line vibration signals

Recalling the peak modulation observed from time domain waveforms with a repetition rate of approximately every rotation period, the spectra were plotted to investigate the dominant (carrier) frequency and modulation frequency associated with the peak modulation. The spectra of signals under four speeds were compared for each measurement location as shown in Figure 3-21. The zoomed spectra were plotted in Figure 3-22 to analyze the frequencies of interests.

(a) Spectra of signals from bearing housing top of the faulty gearbox

(b) Spectra of signals from bearing housing top of the normal gearbox
The component at 121.5 Hz was observed to be the dominant frequency for all measurement locations regardless of the shaft rotating speed. In addition, the Harmonics of 60.7 Hz appear over the entire frequency range up to the Nyquist frequency (half the sampling frequency), and the lower amplitude frequency responses between 1500 and 2500 Hz are clearer for signals from the motor plate with the appearance of a peak at 2066 Hz. Although an increase of amplitude of the frequency components can be found with the
increase of speed, these peaks at 1X through to 8X line frequency are most likely to be produced by the electric motor instead of the gear pairs nor the bearings as they repeatedly appear for all rotating speeds. It also matches with the on-site results which contain motor related frequency components. Therefore, the obtained dominant frequency components are not useful indices for identifying the gearbox faults.

(a) Bearing housing top of faulty gearbox under 15.35 rpm (0.25 Hz)  
(b) Bearing housing top of faulty gearbox under 24.51 rpm (0.4 Hz)  
(c) Gearbox top of faulty gearbox under 24.51 rpm (0.4 Hz)  
(d) Motor plate of faulty gearbox under 24.51 rpm (0.4 Hz)
The period of a modulation equals the time interval between maximum peaks, which can be measured from the time-domain waveforms as shown in Figure 3-12 at the shaft rotating period. The corresponding modulation frequency can be calculated by dividing 1 by the period of modulation. This frequency component is not clearly evident on the frequency spectra as shown in Figure 3-21. Instead, the modulation frequency shows up clearly as sidebands on both sides of the carrier frequency for signals under all shaft rotating speeds. This indicates that the peak modulation shown in time-domain waveforms are possibly generated by a combination of amplitude and frequency modulation as more than two sidebands were observed. Two spectra of signals from bearing housing top of the faulty gearbox under two rotating speeds were chosen as examples in Figures 3-22 (a) and (b).

The sidebands were observed from the spectra for gearbox top and bearing housing top signals of both faulty and normal gearbox, while as presented in Figure 3-22 (d), the motor plate spectrum does not contain obvious sidebands which separate from the main carrier frequency response by a span of the modulation frequency. Therefore, the sidebands

Figure 3-22. Zoomed spectra of signals from different measurement locations

(e) Bearing housing top of normal gearbox under 26.79 rpm (0.45 Hz)  
(f) Gearbox top of normal gearbox under 26.79 rpm (0.45 Hz)
around 121.5 Hz are speculated to be produced by gear pairs or bearings, and are of more interest. By comparing Figures 3-22 (c) and (f), it is shown that the signals collected from the gearbox top of the normal gearbox have similar amplitude but wider spread of sidebands around 121.5 Hz than the faulty gearbox under similar rotating speed. While Figure 3-22 (e) shows a wide spread and higher amplitude in sidebands compared with the spectra of the signals from gearbox top of the faulty gearbox as shown in Figure 3-22 (b). Such results are in opposition to the general theory which states a positive relationship between the signal amplitude and the faulty level.

Bartelmus et al [44] studied the vibration diagnostic method for planetary gearboxes under varying external load, and concluded that the periodic variation of external load will result in the amplitude modulation and the associated speed variation will cause the frequency modulation. The slight unevenness of nylon cases may result in a load variation and the associated speed variation, which could be the main reason for the existence of multiple sidebands with such amplitude difference between the faulty and normal gearboxes. Therefore, the sidebands fail to reveal much of the gearbox fault information.

3.5 Chapter Summary

Vibration signals from the gearboxes were mostly corrupted by background noise, as was demonstrated by the similar RMS trends and the close proximity of dominant frequencies in the frequency spectrum. Among those, 121.5 Hz was observed as the most significant dominant frequency for the off-line tests regardless of speed changes, while the on-site results also contain impulses at around the 120 Hz interval. These frequency responses were speculated to be a property of electrical noise. Acoustic Emission testing was selected
as another fault detection method because of its advantage in eliminating noise effects. The experiment and results are discussed in Chapter 4.
Chapter 4

Acoustic Emission Testing

Acoustic Emission (AE) testing measures the transient elastic waves within a material which are generated by the rapid release of localized stress energy caused by structural alterations under mechanical loading. This chapter will describe the AE experimental activities and discuss the AE results based on the hit-based, time-driven and frequency-domain approaches.

4.1 AE experimental preparation

The AE tests were performed on the same normal and faulty gearbox units as the off-line vibration tests. The experimental facilities, settings, and the measurement methods are discussed in this section.

4.1.1 Experimental facilities

The PCI-II based AE system produced by Physical Acoustic Corporation (PAC) was used for AE signal collection. This system includes a sensor, preamplifier, cables, PCI-2 (Peripheral Component Interconnect) card and the AEwin software.

As shown in Figure 4-1 (a), the R15α used for AE tests is a narrow band resonant sensor with high sensitivity and low-frequency rejection. It weighs 48 grams and has dimensions of 19 mm OD x 22.4 mm H. The operating frequency range for the R15α is from 50 to 400 kHz, and the resonant frequency relative to 1V/μbar is 150 kHz [45]. The 1232-X-SMA cable was used to connect the sensor to the preamplifier. The connected preamplifier (see Figure 4-1 (b)) is 2/4/6 type with in-line differential and in-line signal ends. It is applied to
condition sensor outputs to be acceptable for inputs on the PCI-2. The PCI-2 (see Figure 4-1 (c)) is a 2 channel AE data acquisition and digital signal processing system on a single full-size PCI card. A desktop was fitted with the PCI-2 card to run the AEwin system which is capable of 18-bit A/D conversion and 40 MSamples/second acquisition with sample averaging and automatic offset control [28].

(a) The R15α sensor (b) The 2/4/6 preamplifier (c) The PCI-2 card

Figure 4-1. AE experimental system components

4.1.2 Experimental settings

The gain level of the preamplifier was set at 40 dB and the in-line single end was used. The sampling rate was set to 1 million samples per second (MSPM), which made a measurable frequency of up to 500 kHz.

The threshold is a reference line used to define an AE “hit”. The threshold level was set to 27 dB which ensured that only one or two events would be detected for every 5 seconds at stopped operating condition. For the hit-based data, the HDT, PDT and HLT were set at 600 μs, 300 μs and 1000 μs respectively, and around 7 seconds of signal was collected for each measurement. The time-driven rate, which controls the frequency of the time-driven parameters to be recorded, was set at 10 ms to record the time-driven data. The AE waveforms contain 10,000 samples each and were saved for further frequency analysis.
A high pass filter of 40 kHz was used to reject the unwanted parts of each signal for all AE tests. The raw AE signal collected from the gearbox top on the faulty unit under shafting rotating speed of 24.90 rpm is presented in Figure 4-2 (a). The signal filtered by the 40 kHz high pass filter is shown in Figure 4-2 (b). It is obvious that the filter successfully rejects the low-frequency but high-energy components. The low-frequency components under 40 kHz were speculated as relating to the gearbox vibration. This was supported by the sustained output shown in Figure 4-2 (c), which is the filtered signal using the 40 kHz low pass filter. Figures 4-2 (d) and (e) show the filtered signal using the 40-80 kHz and 80-120 kHz band pass filter respectively. Clear burst emissions were found in these two plots. The AE signals filtered by band pass filter with higher frequency range are presented in Figures 4-2 (f) to (h). These high frequency signals do not contain much AE activity but were kept as they may reveal useful information under other operating conditions.
Figure 4-2. Comparison of the AE signals using different filters
The frequency spectra of the raw and filtered AE signals collected from the gearbox top on the faulty unit under shafting rotating speed of 24.90 rpm were compared in Figure 4-3. It can be clearly seen that the low-frequency but high-energy parts shown on Figure 4-3 (a) between 0-40 kHz were filtered out by the high pass filter as shown on Figure 4-3 (b), thereby the peaks in the range of 40-100 kHz become obvious. This further indicates the good performance of 40 kHz high pass filter.

![Frequency spectra of raw AE signal](image1)
![Frequency spectra of filtered AE signal](image2)

**Figure 4-3. Comparison of frequency spectra of the raw and filtered AE signals**

### 4.1.3 AE signal collection method

As mentioned in Section 3.2.2, the exact loads applied between the two rollers driven by the gearbox were not measured due to facility limitations. For the AE tests, three loading conditions were simulated by adjusting the contact condition of the two rollers (fully separated, just contacting, and fully contact), which were achieved by changing the location of the upper shaft as indicated by a ruler attached to both gearbox and bearing housing. The rotating speed of the output shaft was adjusted from around 3 rpm to 35 rpm by the SCR
drive. The AE signals were collected at different rotating speeds for the normal and faulty gearbox under each loading condition.

Considering that AE signals are non-directional, use of one sensor is sufficient for all AE testing in this case. Attenuation of signals is one of the key problems requiring sensor installation to be as close to the source as possible. Therefore, the gearbox housing was selected as the measurement sensor location. Several tests were performed on three locations on the gearbox housing (see Figure 4-4) to find the best measurement location.

![Figure 4-4. The AE test measurement locations](image)

Figure 4-5 compares the RMS level of signals collected from different locations. Although stronger signals were collected from the right side of the gearbox housing, the clear deviations that occur in the high-speed range indicate a relatively low reliability. The signals collected from the gearbox side and top have gently increasing trends over shaft rotating speed and no unexpected deviation was observed. These two locations were deemed more reliable for data measurement. The gearbox top was selected for all subsequent tests because of the ease and stability of installation.
4.1.4 Base line test

Recalling that the gearbox vibration signals collected during vibration tests were mostly overwhelmed by the motor induced vibration, which was mainly transferred to the gearbox through the table (the foundation of all experimental facilities). Baseline tests were conducted to determine the effect of the drive motor on the measured AE signals.

To verify that the high frequency AE signals were generated by gear pairs rather than the structure vibration caused by the drive motor, several AE tests were conducted on the resting gearbox when the other gearbox was operating. The resting gearbox only has slight vibration transferred from the table, therefore can be treated as a base line condition to be compared with the signals collected from the operating gearbox. A clear difference was found (see Figure 4-6) between the resting gearbox, which keeps a steady and low RMS level at around 0.0004 V and the rotating gearbox, which has an increasing RMS level as
the speed increases. This shows that the AE tests have good performance at rejecting the low-frequency background noise.

![Graph showing comparison of RMS voltage level for resting and rotating gearboxes across varying shaft rotating speeds.](image)

**Figure 4-6. Comparison of the RMS level of the resting and rotating gearbox signals**

**4.2 Hit-based AE analysis**

AE hits are detected and recorded when the measured signal level crosses over the defined threshold. Hit-based analysis was conducted on the recorded data to study the performance of the hit-based AE parameters. Discussion of the hit-based data for the faulty and normal gearboxes in three roller loading conditions are presented in this section.

**4.2.1 Comparison of Hit-based AE data for gearbox in different roller loading conditions**

Three roller loading conditions (no load, minimum load, normal load) were simulated by changing the gap distance between two rollers (fully separated, just contacting, fully contact). Three sets of hit-based data with samples of around 7 seconds under different shaft rotating speeds were recorded for both faulty and normal gearboxes in each roller
loading condition. These data sets were analyzed by computing the hit-based parameters as shown in Figures 4-7 and 4-8 to compare the effect of load on AE hits. The averaged parameter value of samples collected under each shaft rotating speed are presented as circle dots, and the trending lines were plotted using the curve fitting function in MATLAB. The red, green and blue colors represent data collected under no load, minimum load and normal load conditions respectively.

![Graph showing RMS levels for different roller loading conditions](image)

(a) The faulty gearbox  
(b) The normal gearbox

**Figure 4-7. Comparison of the RMS levels for different roller loading conditions**

As shown in figure 4-7 above, an increase of RMS level with increasing speed was observed from all data sets. The signals collected from the normal gearbox have lower RMS level and its growth is episodic with overall slower rates when compared with the faulty gearbox data. For the faulty gearbox, no clear difference was found between the three roller loading conditions, the fitting line for minimum load condition is slightly higher than the other two fitting lines. While for the normal gearbox, the increase of RMS value for no load condition happens before the other two load conditions.
The results do not show the positive relationship between the RMS and load which has been stated by researchers [46][47]. However, Tan and Mba [34] concluded in their study that the increase of AE signal level is not dominantly influenced by load, but is mostly due to the increase of oil temperature. Their experimental results showed that the RMS value varied with time as the gearbox reached a stabilized temperature, parts of the signals under lower load condition presented higher RMS level as well before the stable temperature was achieved.

Based on this statement, one possible reason for the observed RMS difference is the variance of oil temperature. In the AE experiment, three load conditions only have tiny load difference because the motor was only able to drive the gearbox under small load. Also, each round of testing was undertaken at a different point in time with respect to the cold start. This might result in a variance of oil temperature which was not measured during each test. For the faulty gearbox with a high rate of wear and asperity deformation, the influence of oil temperature and load condition was not evident. However, for the normal gearbox under three close load conditions, a high temperature can result in thinner oil film thickness, which will affect the rate of wear and asperity deformation that generate the AE activity and therefore result in the RMS difference.

Figure 4-8 present the results of other hit-based parameters including the amplitude, count rate, signal strength, and absolute energy. The signal strength is derived from the integral of the rectified voltage signal over the entire AE signal dynamic range, and the absolute energy is derived from the integral of the squared voltage signal divided by the reference resistance (10 k-ohm). Similar differences were observed.
(a) The amplitude of faulty gearbox signals

(b) The amplitude of normal gearbox signals

(c) The count rate of faulty gearbox signals

(d) The count rate of normal gearbox signals

(e) The signal strength of faulty gearbox signals

(f) The signal strength of normal gearbox signals
The abs energy of faulty gearbox signals

Figure 4-8. Comparison of hit-based parameters for different load conditions

It is obvious that the turning point of indicator plots for the faulty gearbox is at around 12 rpm, while the quiet period is longer for the normal gearbox as the indicator levels start changing at around 20 rpm. The signals from the faulty gearbox have higher indicator levels than the normal signals when shaft rotating speed increases above 12 rpm. For the normal gearbox, the signals collected under no load condition again have higher level for all hit-based parameters. The signals collected under minimum load condition show slightly higher indicator levels for both faulty and normal gearboxes, except for the absolute energy level for the faulty gearboxes.

It was speculated that the variance of oil temperature is the main reason for the unusual indicator differences related to the three load conditions, especially for the normal gearbox under no load condition. Constant temperature tests should be performed in future to further examine the load effect on AE signals.

4.2.2 Discussion of Hit-based AE data for gearbox in roller just-contacting condition
As discussed before, the data from the gearboxes under minimum and normal load conditions have similar indicator levels. It was presumed that the oil temperature difference was small between these tests. Signals collected under minimum load condition are analyzed in this section to compare the performance of hit-based parameters in distinguishing the faulty and normal gearbox.

The plots shown in Figure 4-9 compare the faulty and normal data by using different hit-based parameters, including the RMS, peak amplitude, count rate, signal strength and absolute energy over the duration of 1 second. The parameter values for gearboxes rotating at 25 rpm were marked for comparison, the faulty and normal gearbox could be clearly distinguished by the parameter gaps.

(a) RMS Voltage vs. Shaft Rotating Speed  (b) Amplitude vs. Shaft Rotating Speed
To compare the performance of hit-based AE indicators, Parameter Difference (PD) was calculated using equation 4.1 to represent the relative difference of each parameter between the faulty and normal gearboxes.

\[
PD = 20 \log_{10} \frac{P_{\text{Faulty}}}{P_{\text{Normal}}} \text{ (dB)}
\]

4.1

Figure 4-9. Hit-based parametric comparison of AE signals from the faulty and normal gearbox
where $P_{\text{Faulty}}$ and $P_{\text{normal}}$ represent the parameter values of the faulty and normal signals respectively.

Since the shaft rotating speed was controlled by rotating the turn-button at SCR, not all faulty gearbox signals have the corresponding normal gearbox signal at the same shaft rotating speed. To handle this problem, the parameters were averaged per 1 rpm speed range before calculating the PD. The results are presented on Figure 4-10 below.

![Figure 4-10. Comparison of PD of hit-based AE parameters](image)

The absolute energy, signal strength and count rate are better hit-based AE parameters for fault detection as they present larger differences than the RMS and amplitude at all speeds. The PD value for these three parameters reaches the maximum (40 to 50 dB) at around 16 rpm, however, the value decreases steeply at speeds above 16 rpm. Instead, the PD of RMS increases steadily over the whole speed range. The RMS reaches around 10 dB of PD at 34 rpm, which already matches the PD level of signal strength. The results show that absolute energy, signal strength and count rate are good parameters for condition monitoring of
gearbox faults at speeds lower than 35 rpm, and count rate is the best parameter for condition monitoring of gearbox faults at 25 rpm as it shows the largest difference. It can be presumed that RMS will be a good indicator to distinguish the faulty gearboxes at high rotating speed.

4.3 Time-driven AE signals and analysis

Time-driven data was recorded to study the continuous AE signals in time-driven data rate intervals. The RMS and absolute energy were calculated and stored automatically for each interval. Other time-domain indicators such as Kurtosis and crest factor were computed by analyzing the recorded AE waveforms. The time-driven AE signals and parameters are discussed in this section.

4.3.1 Time-driven AE signals

The AE waveforms were recorded in every 10 ms within the period that the hit-based data was collected. Signals collected under the minimum load condition were selected for further analysis as per previous discussion. Due to computer storage limitation, two waveform files from each measurement were transferred to MATLAB for signal processing. To present the AE signals in different shaft rotating speeds, Figure 4-11 chooses two waveforms for each gearbox as examples.

Figures 4-11 (a) and (b) show the time-domain AE signals collected from the normal gearbox rotating at 21.167 rpm and 25.718 rpm respectively. Figure 4-11 (b) has slightly higher amplitude compared to Figure 4-11 (a). The increase in amplitude with speed is more clearly presented in Figures 4-11 (c) and (d), which show the AE signals from the faulty gearbox rotating at 20.919 rpm and 26.076 rpm respectively. In addition, as can be
observed by comparing Figures 4-11 (a) (c) or (b) (d), the faulty gearbox signals were obviously stronger than the signals collected from the normal gearbox rotating at the similar speeds.

![Comparison of AE signals of the faulty and normal gearboxes](image)

(a) Normal gearbox rotating at 21.167 rpm  (b) Normal gearbox rotating at 25.718 rpm

(c) Faulty gearbox rotating at 20.919 rpm  (d) Faulty gearbox rotating at 26.076 rpm

**Figure 4-11. Comparison of AE signals of the faulty and normal gearboxes**

### 4.3.2 Discussion of time-driven AE parameters

Figure 4-12 present the time-driven parameters including the RMS, absolute energy, Kurtosis and Crest Factor as a function of the shaft rotating speed. The RMS and absolute
energy were recorded by the AE system at every 10 ms, while the Kurtosis and Crest Factor were obtained by analyzing the waveforms using MATLAB.

\[ y = b_0 + b_1 x + b_2 x^2 + b_3 x^3 + \ldots \]

(a) RMS Voltage vs. Shaft Rotating Speed  
(b) Absolute Energy vs. Shaft Rotating Speed

(c) Kurtosis vs. Shaft Rotating Speed  
(d) Crest Factor vs. Shaft Rotating Speed

**Figure 4-12. Time-driven parametric comparison of AE signals from the faulty and normal gearbox**

As can be seen from Figures 4-12 (a) and (b), the faulty gearbox has higher RMS and absolute energy, and the difference becomes more obvious as speed increases. No clear trend lines were obtained for Kurtosis and Crest Factor plots as shown in Figures 4-12 (c) and (d), the Adjusted R-square value of the fitting lines are all below 0.5. In addition, no
significant difference was found between the faulty and normal signals. Both faulty and normal gearboxes tend to have stable and similar Kurtosis (around 3) and Crest Factor value (around 3.5) at speeds above 15 rpm.

The performance of each parameter can be more clearly observed from the PD plot as shown in Figure 4-13. The PD values were calculated by equation 4.1, where $P_{Faulty}$ and $P_{normal}$ represent the averaged parameter values of the faulty and normal signals per 1 rpm speed range.

![Figure 4-13. Comparison of PD of time-driven AE parameters](image.png)

The negative PD of Kurtosis and Crest Factor at the speed range from 5 to 15 rpm indicates that more outliers are presented in the normal gearbox signals under low rotating speed. The PD stays at around 0 for Kurtosis and Crest Factor when the rotational speed increases, therefore these two parameters are not ideal for gearbox fault detection. Instead, the RMS and absolute energy are good indicators as they have positive and increasing PD values.
over the whole speed range. It is obvious that the absolute energy is the best parameter for fault detection since it has the highest PD level at all measured speeds.

4.4 Frequency domain analysis of AE signal

A Fast Fourier Transform (FFT) algorithm was applied to the AE samples which were recorded over 10 ms. Frequency information of interest, such as peak frequency and frequency centroid are discussed in this section.

4.4.1 Spectrum plots of the AE signals

The spectrum plots of the AE signals collected from the faulty and normal gearboxes at all speeds (from around 3 rpm to 35 rpm) have similar distribution but different amplitudes. Four plots were chosen as examples and are shown in Figure 4-14. The concentration of high amplitude frequency components was found in the region of 50 to 100 kHz. Most frequency peaks occur at around 95 kHz. No clear dominant frequency component can be directly extracted from the frequency spectra. As stated by Mba and Rao [48] in their research, one possible reason is that the transient impulses associated with the relative motion of material excite a broad frequency range.

However, it was observed by comparing Figures 4-14 (a) (b) or (c) (d) that the amplitudes of the frequency components vary directly with the shaft rotating speed. A comparison of Figures 4-14 (a) (c) or (b) (d) shows a marked increase in amplitude of the frequency components with the appearance of gearbox wear. Therefore, the overall amplitude of the frequency responses in the peak frequency range is good at disguising the faulty and normal gearboxes.
Figure 4. Spectrum plots of AE signals from the faulty and normal gearboxes

4.4.2 Comparison of frequency centroid

The frequency centroid is the first moment of inertia of frequency, which measures the center of mass of the frequency spectrum. It can provide more information about the shape change of the frequency spectrum.

As can be observed from Figure 4-15, the centroid frequency level decreases with increasing shaft rotating speed for both the faulty and normal gearboxes, and the signals collected from the normal gearbox have higher frequency centroid values at all speeds. This indicates that more high frequency responses are present in the low frequency range.
for the faulty gearbox, which was consistent with the frequency spectra observations (Section 4.4.1). That is, the faulty gearbox has higher peaks in the lower frequency range (50 to 100 kHz), while both faulty and normal gearboxes have low frequency responses in the higher frequency range.

Figure 4-15. Frequency centroid of AE signals from the faulty and normal gearboxes

Figure 4-16. Plot of absolute PD of the frequency centroid
Negative PD values were obtained by computing equation 4.1 due to the higher $P_{normal}$ value compared to the $P_{faulty}$ value. The absolute PD values are plotted in Figure 4-16 to present the difference level of frequency centroid as a function of shaft rotating speed. The frequency centroid is a good indicator for fault detection as it can present the shape of the frequency spectrum numerically with the increasing absolute PD value over the whole speed range.

4.5 Time-frequency analysis of AE signals

To better illustrate the frequency and time information on a single chart simultaneously, the time-frequency domain spectra were computed by Continuous Wavelet Transform algorithm (CWT). The analytic morlet wavelet was employed in this study as it provides equal variance in time and frequency, which is good for transient localization. Four time-frequency spectra are shown in Figure 4-17 as examples to present the frequency characteristics over time.

It can be observed that all morlet wavelets contain frequency components between 64 and 128 kHz with the highest CWT coefficients. More transients show up with increasing shaft rotating speeds for both faulty and normal gearboxes. However, the signals from the normal gearbox (see Figures 4-17 (a) and (b)) have lower CWT coefficient magnitudes compared to the faulty signals at similar rotating speed as shown in Figures 4-17 (c) and (d). In addition, the magnitude increase due to speed increase can be more obviously observed from the faulty signals. The frequency range of components with high CWT coefficients is broader for the AE signals from the faulty gearbox than that for the normal signals. All these CWT analysis results were consistent with the FFT analysis results as discussed in
the previous section, which further demonstrates that the AE-based method is good at condition monitoring of gearboxes rotating at low speed.

![Morlet wavelets of AE signals from the faulty and normal gearboxes](image)

(a) Normal gearbox rotating at 21.167 rpm  (b) Normal gearbox rotating at 25.718 rpm

(c) Faulty gearbox rotating at 20.919 rpm  (d) Faulty gearbox rotating at 26.076 rpm

**Figure 4-17. Morlet wavelets of AE signals from the faulty and normal gearboxes**

### 4.6 Chapter Summary

An AE-based technique has been shown to be a good method for low-speed gearbox fault detection, as was demonstrated by the high PD values of hit-based, time-driven, and frequency domain parameters. Table 3 summaries the trends from different indicators and the PD values at the typical operating speed of the gearbox shafts during production (25 rpm).
Table 3. Summary of regression equations and PD values for different parameters

<table>
<thead>
<tr>
<th>Parameter Type</th>
<th>Parameter Type</th>
<th>Gearbox Condition</th>
<th>Regression Equation</th>
<th>Adjusted R-square value</th>
<th>Value at 25 rpm</th>
<th>PD value at 25 rpm (dB)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hit-based AE parameters</td>
<td>RMS Voltage (V)</td>
<td>Normal</td>
<td>When $21 &lt; x &lt; 30$: $y = 0.0006$</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>Faulty</td>
<td>$y = 1.8 \times 10^{-6}x^2 - 1.3 \times 10^{-6}x + 0.000315$</td>
<td>0.984</td>
<td>0.0014</td>
<td>7.36</td>
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<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Count Rate (Counts/sec)</td>
<td>Normal</td>
<td>$y = 3.79 \times 10^{-7}x^{6.78}$</td>
<td>0.9369</td>
<td>1127</td>
<td>30.10</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Faulty</td>
<td>$y = 26800 - 17810 \cos(0.11x) - 22230\sin(0.11x)$</td>
<td>0.9876</td>
<td>36060</td>
<td></td>
</tr>
<tr>
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<td>Peak Amplitude (dB)</td>
<td>Normal</td>
<td>$y = 29.31 + 0.63 \cos(0.21x) + 1.84\sin(0.21x)$</td>
<td>0.8819</td>
<td>28.16</td>
<td>2.01</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Faulty</td>
<td>$y = 34.09 - 2.34 \cos(0.12x) - 4.44\sin(0.12x)$</td>
<td>0.9506</td>
<td>35.5</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Absolute Energy (aJ)</td>
<td>Normal</td>
<td>$y = 3.94 \times 10^{-5}x^{5.4}$</td>
<td>0.9508</td>
<td>1413</td>
<td>23.13</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Faulty</td>
<td>$y = 1.85x^{2.89}$</td>
<td>0.9732</td>
<td>20250</td>
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<tr>
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<td>Signal Strength (pVs)</td>
<td>Normal</td>
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<td>1.43*10^6</td>
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<td></td>
<td></td>
<td>Faulty</td>
<td>$y = 3.9 \times 10^{4.134}x$</td>
<td>0.9408</td>
<td>1.04*10^7</td>
<td></td>
</tr>
<tr>
<td>Time-driven AE parameters</td>
<td>RMS Voltage (V)</td>
<td>Normal</td>
<td>When $21 &lt; x &lt; 30$: $y = 0.0006$</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>Faulty</td>
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<td>0.9839</td>
<td>0.0014</td>
<td>7.36</td>
</tr>
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<td>0.9866</td>
<td>47.78</td>
<td>13.41</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Faulty</td>
<td>$y = 0.0015x^{3.65} + 31.97$</td>
<td>0.9826</td>
<td>223.8</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Kurtosis</td>
<td>Normal</td>
<td>$y = 0.03x^2 - 0.2x + 6.19$</td>
<td>0.3004</td>
<td>3.1</td>
<td>-0.33</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Faulty</td>
<td>$y = 10.82e^{-0.35x} + 2.99e^{-0.00015x}$</td>
<td>0.4829</td>
<td>2.98</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Crest Factor</td>
<td>Normal</td>
<td>$y = 10.58x^{-0.32}$</td>
<td>0.4108</td>
<td>3.72</td>
<td>-0.57</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Faulty</td>
<td>$y = 14.27x^{-1.34} + 3.29$</td>
<td>0.4898</td>
<td>3.48</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Frequency Centroid (kHz)</td>
<td>Normal</td>
<td>$y = 2.72 \times 10^5e^{-0.012x} + 449.9e^{0.1x}$</td>
<td>0.8937</td>
<td>205.8</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>Faulty</td>
<td>$y = 2.57 \times 10^5e^{-0.03x} + 2.57 \times 10^4e^{0.018x}$</td>
<td>0.9691</td>
<td>160.8</td>
<td>-2.14</td>
</tr>
</tbody>
</table>
It was observed that three hit-based AE parameters including count rate, absolute energy and signal strength have more than 10 dB of PD at a shaft speed of 25 rpm. The count rate is the best hit-based AE parameter for condition monitoring of the gearboxes operating in the forming line as it shows the largest difference at a shaft speed of 25 rpm. For the time-driven parameters, Kurtosis and Crest Factor do not show clear differences between normal and faulty conditions at all speeds, and the absolute energy is the best parameter followed by the RMS value. The frequency centroid is a good frequency domain parameter to present the change of signal shape as the damage develops in the gear material.

However, oil temperature was not well controlled and recorded during the AE tests under different load conditions, which may affect the AE signal level and result in the unusual indicator differences. Constant temperature tests should be performed to fully define the relationship between AE signals and gearbox operational conditions.
Chapter 5

Conclusions and Future Work

Vibration-based and Acoustic Emission tests were performed on a set of low speed gearboxes that are part of the strip metal forming line. The vibration signals collected from on-site and off-line tests were analyzed in both the time and frequency domains. Hit-based, time-driven and frequency domain analysis were performed on the AE signals collected from off-line tests.

In Chapter 3, the analysis results of vibration signals collected on-site and off-line were discussed. The on-site vibration tests collected vibration data in a real operating situation with shaft rotating speed of 25 rpm. Recorded vibration levels were low, at around $3 \times 10^{-3}$ g’s, as may be observed in the time-domain waveforms, but no signs of a dominant underlying shaft rotational component were found. In the frequency domain, the appearance of dominant frequency components around 60 Hz and the impulses at approximately 120 Hz intervals indicated the existence of a motor related influence.

For the off-line tests, peak modulation with a repetition rate of approximately every rotation period was shown in all time domain vibration waveforms. In the time domain, the faulty gearbox signals have larger RMS levels compared to the normal gearbox signals. However, similar trends were observed from the motor plate waveforms. The constant motor-to-bearing housing top and motor-to-gearbox top RMS ratio indicate the influence of motor vibration. The Crest Factor and Kurtosis are not appropriate for fault detection in the case as the normal gearbox data had relatively higher CF and Kurtosis values.
In the frequency domain, all frequency spectra contain the dominant frequency component at 121.5 Hz with harmonics of 60.7 Hz over the entire frequency range. These frequency responses were speculated to be a property of electrical noise. The modulation frequency shows up clearly as sidebands on both sides of the carrier frequency. This indicates that the peak modulations shown on time domain waveforms are possibly generated by a combination of amplitude and frequency modulation which were caused by the slight unevenness of the nylon roller cases.

It was concluded that vibration signals from the gearboxes were mostly corrupted by background noise. No appropriate time-domain indicator nor frequency response was observed to effectively distinguish the faulty and normal gearbox. Acoustic Emission tests were therefore performed because of its advantage in eliminating noise effects.

In Chapter 4, the AE results were discussed based on the hit-based, time-driven and frequency-domain approaches. During the AE testing, three roller loading conditions were simulated by adjusting the gap distance between two rollers. For the normal gearbox, the signals collected under no load condition had higher levels for all hit-based parameters. Oil temperature variance was speculated to be the main reason for this unusual result.

In the hit-based analysis, the count rate, absolute energy and signal strength are good indicators as they have more than 10 dB of PD at speeds higher than 10 rpm. The count rate is the best hit-based AE parameter as it shows the largest difference at a shaft speed of 25 rpm.

As shown in the time-domain waveforms, the faulty gearbox signals were obviously stronger than the signals collected from the normal gearbox rotating at similar speeds. The
absolute energy, which was calculated and stored automatically by the AE system for each interval, was observed to be the best time-driven parameter followed by the RMS.

In the frequency domain, a concentration of high amplitude frequency components was found in the region of 50 to 100 kHz. The frequency centroid value decreases with shaft rotating speed and the normal gearbox signals have higher FC values at all speeds. This indicates that the faulty gearbox signals have higher peaks in the lower frequency range as was consistent with the frequency spectra observations.

In addition, morlet wavelets have shown the time-frequency information of AE signals. Frequency components between 64 and 128 kHz have the highest CWT coefficients for both faulty and normal gearboxes. The faulty gearbox signals have broader frequency range of components with higher CWT coefficient magnitudes. All these CWT analysis results were consistent with the FFT analysis results.

In summary, the AE technique has the potential to be a useful condition monitoring tool for low-speed gearboxes as it is superior to vibration-based method in rejecting background noise. The following areas are recommended to further study the performance of AE technique in low-speed gearbox fault detection.

1. Perform constant oil temperature tests under different load conditions to fully investigate the relationship between AE activities and load level.
2. Open the faulty gearbox to ascertain the fault type and location related to the AE signal results.
3. Simulate different fault types on the gearbox to study the performance of the AE based technique in single and combined fault conditions.
4. Develop physics-based or data-driven model to further investigate the performance of AE based technique in low-speed gearbox fault detection and diagnosis.
References


[34] C. K. Tan and D. Mba, “Identification of the acoustic emission source during a


Appendix

1. AE software setups

(a) Channel setup

<table>
<thead>
<tr>
<th>AE Channel</th>
<th>Threshold Type</th>
<th>Gain dill</th>
<th>Pre Amp dill</th>
<th>Analog Filter</th>
<th>Waveform Setup</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>REO-DO</td>
<td>27</td>
<td>8</td>
<td>0</td>
<td>40</td>
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<tr>
<td>2</td>
<td>REO-DO</td>
<td>45</td>
<td>4</td>
<td>0</td>
<td>40</td>
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</table>

(b) Timing parameter

<table>
<thead>
<tr>
<th>AE Channel</th>
<th>PDT (microseconds)</th>
<th>HDT (microseconds)</th>
<th>HLT (microseconds)</th>
<th>Max Duration (milliseconds)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>300</td>
<td>800</td>
<td>1000</td>
<td>10</td>
</tr>
<tr>
<td>2</td>
<td>200</td>
<td>300</td>
<td>1000</td>
<td>100</td>
</tr>
</tbody>
</table>

(c) Waveform streaming setup
2. MATLAB Code

(1) FFT analysis

```matlab
% Constants
clear; clc; close all
%% 1 Load Text
%% Sampling frequency
fSamp_sd = ['cd ';''];
Fs=5120;
Location = dir('*.txt');
for i=1:length(Location)
    eval(['load ' Location(i).name ' -ascii']);
end
for i=1:length(Location)
    Filename=Location(i).name;
    Acc = textread(Filename,' ', 'headerlines',33);
    Acci(:,i)= Acc(1:5007s,:);
    A=Acci(20*Fs:20*Fs,2,1);
    k(i,1)=kurtosis(A);  % Kurtosis
    r(i,1)=norm(A)/sqrt(length(A));  % RMS
    B=Acci(20*Fs:21*Fs,2,1);
    B1=abs(B);
    pl(i,1)=max(B1);
    CF(i,1)=pl/2;  % Crest Factor
end
```
(2) Frequency Centroid

function C = SpectralCentroid(signal,windowLength, step, fs)
signal = signal(:,1);
signal = signal / max(abs(signal));
curPos = 1;
L = length(signal);
numOfFrames = floor((L-windowLength)/step) + 1;
H = hamming(windowLength);
m = ((fs/(2*windowLength))'*[1;windowLength]);
G = zeros(numOfFrames,1);
for t=1:numOfFrames
    window = H.*(signal(curPos:curPos+windowLength-1));
    FFT = (abs(ifft(window,2))^2); FFT = FFT(1:windowLength);
    FFT = FFT / max(FFT);
    C(t) = sum(m.*FFT)/sum(FFT);
    if (sum(window.^2)<0.010)
        C(t) = 0.0;
    end
    curPos = curPos + step;
end
C = mean(C);