Investigation of Vibration Related Signals
for
Monitoring of Large Open-Pit Rotary Electric
Blasthole Drills

By

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Abstract

This thesis investigated the monitoring and analysis of signals related to the vibration of large rotary blasthole drills. The research focused on a machine with electric drilling actuators: such machines are used as primary production equipment in the drill-and-blast cycle of the surface (open-pit) mining process. The performance of such machines is limited by the onset of severe vibrations, which can arise due to the interaction of geology, bit, drill string, machine structure, and control settings. Experimental data for the thesis were obtained during field periods at an iron ore mine in Minnesota’s Mesabi Iron Range. The data acquisition and signal analysis techniques which were utilized are presented, including smoothing of signals and calculated variables such as specific energy. Ambient vibration sources and vibration aliasing issues are investigated. Results from analyzing structural response tests indicate that, as expected, the natural frequency of the drill mast decreases with increasing bit depth — although the mounting position of accelerometers distorts this trend. The pull down force (weight on bit) is shown to have no appreciable impact on the mast’s natural frequency, nor on the mast’s damping ratio. A strong relationship between rotary speed and the dominant vibration frequency peaks at 3x and 6x rotary speed is demonstrated, and a physical explanation of the 6x vibration peak is postulated. The rotary motor current is shown to consistently exhibit frequency peaks at 3x and 6x rotation speed, indicating that this variable is a good candidate for use either as a substitute for accelerometer feedback, or as an auxiliary signal to detect down-the-hole vibration when it is not manifested by the mast mounted accelerometers. System identification is used to demonstrate that the dynamic relationship between vibration and rotary current, while it can be modeled locally, varies with depth and geology and hence is essentially a time-varying process. This results in the amplitude of rotary current not being usable as a proxy for vibration amplitude. Nonetheless, it is demonstrated that the root-mean-square (RMS) of the low frequency current oscillations, in a nonlinear combination with the RMS of the current signal as a whole, may be able to serve as a proxy for the RMS of the vibration signal.
Acknowledgements

The work done towards the completion of this thesis has been the cornerstone of my education experience here at Queen’s University. The opportunity to learn and apply skills in a real world environment was a key contributor to my continued enthusiasm and optimism throughout my degree. Without the guidance and support of my supervisor, Dr. Laeeque K. Daneshmend, I would not have had the excellent opportunities that were instrumental to my success. My time working with Dr. Daneshmend has been rewarding both educationally and personally as he is an indispensable mentor in all areas.

I would also like to acknowledge Dr. Jonathan Peck for his contributions to my learning experience, his time and his advice throughout my time at Queen’s University; he too was instrumental in constantly renewing my resolve and ensuring my success.

I would finally like to thank the people in the office of the Mechanical and Materials engineering department as well as those in the department of Mining engineering. For questions quickly answered and help so freely given, they made things possible and guided me in the nuances of the University environment. Their help is sincerely appreciated.
Dedications

This thesis is dedicated to my family and friends and most especially to my wife, Amy Hewitt, for her perpetual and unwavering encouragement in this endeavor.
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Nomenclature and Abbreviations

cfm  Cubic Feet per Minute

F    Force (N)

ft-lbs.  Foot-pounds

$g$  Gravity (9.81 m/s$^2$)

kPa  Kilopascals

L    Litres

lbs  Pounds

Hz   Hertz
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<tr>
<td>Nm</td>
<td>Newton-metres</td>
</tr>
<tr>
<td>psi</td>
<td>Pounds per square inch</td>
</tr>
<tr>
<td>R</td>
<td>Penetration rate (m/s)</td>
</tr>
<tr>
<td>RMS</td>
<td>Root-Mean Squared</td>
</tr>
<tr>
<td>RPM</td>
<td>Revolutions per minute</td>
</tr>
<tr>
<td>SE</td>
<td>Specific Energy (synonymous with SFE)</td>
</tr>
<tr>
<td>SFE</td>
<td>Specific Fracture Energy (synonymous with SE)</td>
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<tr>
<td>T</td>
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Chapter 1: Introduction

Open pit mining is accomplished in several parts, from exploration to design, production and finally closure and remediation. In the production phase, there are several different pieces of equipment and machinery that work together to form an intricate production system that starts with intact rock and ends with refined product. The first step in this production system is the drilling & blasting process, whereby a series of blastholes are drilled, filled with explosives and detonated to liberate and fragment the rock mass. The process of open pit drilling has been continually improved over time and now boreholes as large as 22 inches (56 cm) in diameter can be drilled. Large boreholes allow for mines to take advantage of economies of scale and when applied correctly can reduce overall drilling and blasting costs. Drilling larger boreholes has necessitated more powerful drills capable of higher rotary torques and greater pulldown pressures. This up scaling has presented greater machine vibration issues than before necessitating a control system that can monitor and control severe vibrations up to 0.3g. These vibrations reduce the lifespan of drill bits (which make up half of the operational and maintenance costs of the drill) substantially, thus increasing the overall cost of drilling. Such vibration control systems are the focus of this thesis; their intended use as well as their efficacy. This thesis will look at the vibrations involved in the drilling process itself to determine exactly what variables can be observed and what information can be used to help mitigate damaging vibrations both for machine health as well as to maximize productivity.

1.1 The Machine

Large scale rotary blasthole drills weigh over 150 tonnes (165 tons), are crawler driven machines capable of drilling holes in rock up to 22 inches (56 cm) in diameter and up to 65 feet (19.8 m) deep in a single pass, meaning there is no need to add pieces of drill pipe during the
drilling of a hole. The drill is also capable of multi-steel drilling, however all of the data in this thesis is for single-pass drilling. These drills are over 100 feet (30.5 m) tall and can be seen in Figure 1-1 with full size service trucks to provide scale.

![Large rotary blast hole drill with bit at maximum depth](image)

**Figure 1-1 – Large rotary blast hole drill with bit at maximum depth**

These large scale drills are electrically powered with hydraulic propel motors and electric rotary and hoist motors that can apply up to 150,000 lbs (68,000 kg) of thrust on the bit with up to 25,000 ft-lbs. (33,895 Nm) of rotary torque. They use 3,500 cfm (100,000 L/m) air compressors to deliver an average 50 psi (345 kPa) to the bit while drilling. These drills can commonly run in manual and ‘autodrill’ modes, both of which can regulate variables such as pulldown force, penetration rate, RPM, bit air pressure, water injection as well as monitor voltages, currents and vibration levels.

**1.2 The Drill Bit**

The boring device used to penetrate the rock is a tri-cone bit, shown in Figure 1-2.
The cones are covered in rows of ‘teeth’ or inserts made of tungsten carbide. Inside the cone are rows of large roller bearings used to transfer the heavy load from the drill and allow the cone to roll along the rock surface as the inserts push into the rock, creating chips or cuttings. The bit also has three orifices through which water and air are blown out at high pressure to both clear the cuttings and cool the bit.

Many factors influence the parameters used in drilling. The biggest controlling factor is geology, which determines the rotary speed and the weight on the bit required to break the rock. The bit design is also geology dependant with longer, more aggressive inserts for softer rock and shorter, more durable inserts for harder rock. Aside from geology, mine planning (which includes the slope angle of the walls and grade control) will decide whether to use large or small diameter bits, which will in turn alter the drilling parameters as well. The bits considered here are between 10 5/8” (27 cm) diameter and 16” (40.6 cm) diameter and are used in harder rock types such as silicated chert.
1.3 Outline of Thesis
A literature review encompassing relevant aspects of rotary drilling and drill vibration, as well as some related topics, is presented in Chapter 2. The experimental setup used for obtaining the data analyzed in this thesis is outlined in Chapter 3.

The next three chapters present the results obtained from analyzing the acquired data. Results from some structural vibration tests are presented in Chapter 4. The results obtained from analyzing drill vibration are presented in Chapter 5, while Chapter 6 focuses on the correlation of vibration with motor variables.

Finally, Chapter 7 presents the conclusions and recommendations for future work. Appendices include the technical specifications of the blasthole drill, data acquisition system, and accelerometers used in the thesis research.
Chapter 2: Literature Review

2.1 Tri-cone Bits and the Rock Breakage Mechanism
The tri-cone bit is the current standard in bit manufacturing for open pit drilling. The design of two-cone rotary bits was patented in 1909 by Howard Hughes Sr. The design, originally called the Hughes Rollerbit, used two unpowered cones covered in metal teeth that rolled along a rock surface, penetrating and breaking away pieces of material. After the invention of the two-cone bit, very little changed in terms of the mechanics of rotary drill bits. In 1933 The Hughes Tool Company evolved the design and created the first tri-cone rotary drill bit whose basic design is still in use today. (P. H. Brown and P. H. Broeske, 1996)

The tri-cone bit is a rotary, non-percussive bit which relies on axial force and adjacent impact locations to break rock. The three cones are covered in cutters or indenters made primarily of tungsten carbide (Figure 1-2). Tungsten carbide is used due to its high melting point and extreme hardness (Deketh, 1995; Hartley, 1994). These hard cutters are used to break rock through the combination of the applied pressure supplied by the hoist motor and the weight of the drill string itself. The method of rock failure can be seen in Figure 2-1 where the insert applies a force that deforms and ultimately overcomes the tensile strength of the rock, causing fractures which propagate towards a free face, releasing a rock chip (Maurer, 1962; Liu and Karen-Yin, 2001).

![Figure 2-1 - Crater Formation Mechanism (after Maurer)](image-url)
The chip size or crater volume varies to the square of penetration depth, while the impact energy required varies linearly with crater volume beyond the crater initiation threshold energy (Maurer, 1962).

Multiple indenters impacting adjacently to one another are more efficient as the cracks begin to interconnect and produce several chips between indenters (Atlas Copco, 2005).

In an effort to maximize production, rotary bits have had to drill faster, penetrate deeper and increase their diameters to satisfy mine planning and design. These bigger bits have necessitated larger, more powerful drills to reach production targets. These larger machines have incorporated automatic controls, electronic interfaces and sensor packages to handle the larger machines (Hodgins, 1992). This evolution has led to the current state of the art wherein the bit design, in terms of its ability to tolerate higher torques and cutter loads, has become the limiting factor in drilling.

### 2.2 Primary Drilling Variables

Drilling is a complex process. The four major factors in rotary drilling are penetration rate, pulldown force, rotary speed and rotary torque (Cunningham, 1978). A change to any one of these variables affects the others and there are optimal combinations for different geological and lithological conditions.

Pulldown force is the axial component that applies vertical force to the bit as it engages the ground. Through increased pulldown force, the individual carbides on the bit penetrate further into the rock. On each successive carbide strike, as described in section 2.1, a crushed zone appears and causes spalling of the rock or ‘chips’ to form through tensile cracks around the crushed zone (Atlas Copco, 2005). The depth of this crushing action is increased through greater axial force.
Rotary Speed (RPM) is the rate at which the bit body is turning in the hole and is achieved through applied rotary torque. Modern electric drills are speed controlled and vary their torque to produce a constant RPM. As such, while rotary speed hovers very near a given target, rotary torque can vary significantly as the motor controller adjusts the torque to meet the demands and maintain speed. The large swings in torque result largely from geology (including discontinuities and fault zones) as well as potential imbalance issues. As rock is heterogeneous, the level of resistance changes throughout a hole. In addition, any mechanical rotational imbalances in the system, either due to poor installation or damage in the field (such as bent drill steel) will cause eccentricities in the rotary motion that will cause the torque to continually adjust to maintain a set rotary speed.

Penetration rate is the by-product of pulldown force and RPM. A bit at maximum pulldown force and zero RPM does not penetrate, nor does a bit at full RPM and zero axial force. The combination of the two produces a penetration rate, with the level of pulldown force determining how much rock is broken and the rotation speed determining how quickly the indenters progress. Penetration rate is also limited by the manufacturing specifications for force on the bit as well as by the excessive machine vibrations that occur due to the events at the bit-rock interface. This creates a ceiling for pulldown force and an overall vibrational limit. This vibrational limit is either the limit the manufacturer has imposed for the health of the machine or the limit that the operator can withstand, whichever is lower. In general, vibrations of 0.1g are considered unpleasant and vibrations of 0.5g are intolerable (Wowk, 1991) while the drill experiences vibrations up to 0.3g. Within these constraints and with adequate flushing of the rock chips from properly selected bit air pressure, there exists an optimal setting for rotary speed and pulldown force which will maximize the penetration rate depending on the geological conditions present.
2.3 Models of the Drilling Process

Ideally, the optimal set of primary drilling variables should always be used in order to maximize drilling rate. In order to do this, the drilling process must be understood. The fundamental mechanics described in section 2.1 suggest that continually increasing axial force will continually increase chip size and subsequently increase the penetration rate, however physical limitations (carbide length – i.e. the height of the insert) prevent this. The carbide inserts on a bit have a typical length of anywhere between 6 mm (1/4 inch) to 25 mm (1 inch), though they can be beyond this range as well. This creates a limitation due to the maximum depth the carbide can penetrate and can be referred to as the maximum penetration per revolution.

Hard rock formations require the use of bits with shallow, dome-shaped inserts that rely exclusively on the breakage mechanism described previously. For softer geological formations, the inserts become more ‘aggressive’, meaning they become more ‘cone’ or ‘tooth’ shaped. These soft-rock bits are able to use their inserts to do some gouging or scraping as well as simply applying axial pressure to release the rock. Due to the softer rock, the chance of pulling out carbides is reduced through decreased resistance and longer, more aggressive inserts can help to expedite the drilling process (Lyons, W.C., 1996).

Further limiting the maximum carbide depth is a phenomenon called rock swell (The Institute of Quarrying, 2006). As the chips spall away from the bit-rock interface, their effective volume increases due to the voids generated between rock chips. Yet further limiting the maximum carbide depth is the necessary room required for the clearing of chips by the bit air. These constraints result in an optimal cutter depth or penetration per revolution. Regardless of the length of the insert, the drill may only penetrate a percentage of that length (the exact length is dependent on lithology) and still maintain adequate clearing of the cuttings. Increasing the pulldown force will reduce the necessary clearing space and cause the bit to plug, thus greatly
increasing the rotary torque as the bit begins to grind the chips. This unnecessary grinding wears the bit excessively and is a common cause of premature bit failure.

The bit air itself is an important factor as well. The compressed air (and water for bit cooling and dust suppression) is pumped down the centre of the drill steel and through the orifices of the bit shown in Figure 1-2 with the chips exiting the hole through the annular region between the hole wall and the drill steel. The drill steel has a smaller diameter than the bit itself to allow this process to occur. The bit air should be as high as possible to achieve maximum clearing of the chips, however the velocity of the escaping air and chips should be considered to not cause excess wear on the drill steel itself.

To select the optimal combination of variables, a new metric, penetration-per-revolution, is required. Dividing the penetration rate by the rotary speed represents the instantaneous progress in mm/rev or in/rev. Through on site experimentation, an optimal value is derived for a given bit type and geological context. This value can be used to maximize bit productivity without sacrificing bit life due to unnecessary grinding. It should be noted that exceeding this value and grinding the chips can actually reduce productivity as the chips cannot be cleared to expose fresh rock. As such, maintaining the optimal carbide penetration extends bit life and increases productivity.

The relationship between penetration rate, pulldown force and rotary speed is covered at length by Workman and Calder (1996). Penetration rate is seen to vary according to the formula below (Workman and Calder, 1996):

\[
R = (61 - 28 \log_{10} S_c) \times \frac{W}{d} \times \frac{RPM}{300}
\]

*Equation 2.1 - Penetration Rate*
Where \( R \) = Penetration Rate, ft/hr

\[ S_c = \text{Uniaxial compressive strength expressed in thousands of pounds/sq.inch} \]

\[ W/d = \text{pulldown weight per inch of bit diameter expressed in thousands of lbs/inch.} \]

\[ \text{RPM} = \text{revolutions of the drill pipe per minute} \]

This formula shows that penetration rate is linear with both pulldown and RPM. This linear relationship only applies within the constraints previously described, including optimal cutter depth and clearing of the rock chips. Having already established a limit to the maximum allowable pulldown, managing the rotary speed is the next step toward improved penetration rate.

### 2.4 Vibration in Drilling

#### 2.4.1 Understanding Causes of Vibration

Vibration is another limiting factor in the drilling process. Too much vibration can damage the drill itself as well as become detrimental to operator health and comfort. As a result, drill manufacturers equip their machines with vibration sensors and set limits in the controllers to scale back rotary speed and pulldown force in the event of excessive vibrations.

While in consistent geology, an increase in rotary torque is necessary to increase RPM. The resistance of the bit on the borehole wall also increases with increases in RPM. This increase in speed and resistance leads to an increase in vibration. Drill manufacturers do not maximize speed within an established vibrational limit but instead use conservative set points for RPM with control mechanisms that reduce or halt the rotary motion should excessive vibrations arise.
The drill operator controls vibration by selecting a lower rotary speed set point to better match the drilling conditions.

Vibration itself can be used as a control mechanism much like penetration rate. While current systems use excessive vibration data as a trigger to reduce RPM, productivity can be optimized by using instantaneous and/or averaged vibration as part of a feedback system to control drill parameters. By controlling the drilling parameters in a dynamic rather than preset fashion, drilling can proceed below vibration limits without stopping-and-starting as is the current practice. By controlling vibration in a dynamic way, the vibrations do not have a chance to amplify in a resonant state, which can be harmful to the drill.

Understanding the causes of excessive vibration can also help to mitigate their damaging effects to machine and operator health. Vibration is caused by events at the bit-rock interface and the actions or inactions in adjustments of the drilling parameters. Rock is a heterogeneous material and significant variation while drilling is commonplace. Cracks, inclusions, transition zones, voids and several other factors cause the optimum drilling parameters to change constantly (Bailey and Finnie, 1960). Subsequently, the interaction between bit and rock is time-varying, resulting in variations in the vibrations that subsequently propagate through the drill.

### 2.4.2 Resonance and Excitation Mechanisms

Every structure vibrates freely when it is displaced from its static-equilibrium state and subsequently released (Wowk 1991; Aboujaoude, 1997). This vibration occurs at a specific frequency that depends on the material’s mass and elasticity. These vibrations oscillate sinusoidally and decay over time due to a damping element in the system (Wowk, 1991). In the case of drill steel, there is a natural frequency at which the steel will resonate just like a guitar string or a tuning fork. As in any structure, there is a built in dampening effect that dissipates
the energy of this vibration over time, but if the driving force of the vibration oscillates at the natural frequency, this vibration energy builds and the motion of the structure amplifies. This phenomenon is called resonance and in structures such as steel the amplification can multiply the initial force as much as 50 times (James et. al., 1989). With the amount of energy transferred to the bit rock interface during drilling, multiplications of up to 50 times are of great concern.

### 2.4.3 Axial, Torsional and Lateral Vibration

Low frequency (0-20 Hz) vibrations arise from many sources; the impact of each carbide cutter, the individual cones glancing over inclusions and cracks, rubbing of the bit or steel against caving rock as well as other physical obstacles. High frequency vibration (greater than 20Hz) can be caused by rubbing of the bit or even rotating elements on the drill itself such as the motors or the compressor (Aboujaoude, 1997). Any or all of these vibrations can be present during normal drilling and are normally non-destructive, but when drilling variables are sub-optimal, the magnitude of one or more of these causes can increase rapidly. Low frequency (sub 20 Hz) vibrations caused by the interactions at the bit-rock interface must be closely monitored to prevent rapid escalation as the amount of machine movement caused at low frequencies could be particularly damaging to both machine and operator health. With bit speeds below 2 Hz (120 RPM maximum = 2 rotations per second) significant displacements can occur at high vibration conditions. As such, the importance of understanding and adapting to geology is crucial to vibration mitigation.
2.5 Geology and Lithology

Geology, or the structure of the earth, includes within its broad definition lithology and stratigraphy (American Geological Institute, 1984). Lithology is defined as the type of rock, its chemical and physical structure on a micro-scale as well as any weathering or alteration that distinguishes a given rock from any other. Stratigraphy is the layering of rock, particularly sedimentary rock. These distinguishing characteristics within a rock mass have a great impact on drilling performance. As the drill moves throughout the strata it can encounter different bands or stratified layers of rock. Each rock type has its own physical characteristics; its own lithology. On an inch by inch basis, the drilling conditions can change due to lithological changes while geological changes usually refer to larger zones such as various benches or areas of the mine.

As such, lithology and lithological changes due to stratigraphy are of paramount importance over the course of a single hole. Strength and elasticity properties of different rock types will affect how the fractures form in the rock and what shapes the broken pieces will take (Tarbuck, E.J. and Lutgens, F.K., 1993). On a smaller scale, the chemical structure will determine how easy it is for the atoms within the rock to separate from one another. This is best explained by Liu and Karen-Yin (2001):

“*In addition to structure properties, rock strength and fracture properties and its abrasiveness have significant influence on the drilling process. The brittleness of rock and its ability to release deformation energy by crack propagation will affect the results of drilling and blasting.*”

The nature of the crystalline structure, the shape of liberated fragments and the abrasion characteristics of the material all help determine the requirements for bit cutter strength, shape
and durability. The resulting interaction between the bit and the fragmenting rock produces different vibrations from one rock type to another. Even rock that is the same material but is altered, weathered, or previously fractured in some way will respond to drilling in a new and different manner.

Lithology and stratigraphy can change throughout a drilling zone. The drilling plan as well as the size and type of drill bits are geology and grade control dependent. Spacing and placement of boreholes, while typically in a grid arrangement, may need to be altered to accommodate varying rock types. In order to reduce overall costs, the goal of a drill and blast crew is to achieve an acceptable fragment size with as uniform a size distribution as possible. This uniform fragment size significantly reduces costs in the crushing and grinding circuit of the processing plant. If the rock mass varies significantly over the bench with uniform borehole spacing the fragmentation distribution will become large and greater cost will be incurred. Furthermore, bits that may be perfectly acceptable in one rock type may be poor candidates for another rock type. Balancing all the costs in a drill and blast scenario requires a level of understanding about the geology that is to be drilled.

2.6 Bit Life and Drilling Cost
As mines strive to reduce drilling and blasting costs, larger bits have become more and more common. The use of larger diameter bits results in a greater required pulldown force in order to achieve the desired pressure over the drilling face. This force increases to the square of the increased diameter, meaning for a bit of 16” (40.6 cm) diameter, the force required will be four times that of a 8” (20.3 cm) diameter bit as the drilled ‘face’ (area of the circle being drilled) will quadruple in area. The larger the diameter of the bit, the more robust it must be to withstand the exponentially increasing force. The size of the internal bearings in the bit increases linearly with the diameter of the bit and subsequently they do not last as long as those in smaller bits of
comparable quality. Using larger bits reduces cost by necessitating fewer holes. Despite the decreased lifespan and greater capital cost of large bits, this approach reduces the cost of drill bits on the whole. This can be described in a drilling cost equation. However, first the cost of blasting must be briefly addressed.

For a given volume of rock (bench) consisting of several boreholes, a certain amount of explosive force is required to adequately fragment the entire volume. This metric is referred to as the Powder Factor (The Institute of Quarrying, 2006) and varies between rock types but can be represented as kg/m^3 (lbs/ft^3) or kilograms of explosive required to break one cubic metre of a given rock (pounds per cubic foot). Each borehole will hold a given amount of explosive and the larger the borehole, the greater volume of rock it should adequately fracture. As the volume of the borehole goes up exponentially to the diameter of the borehole, so does the amount of fractured rock. Thus if 8” (20.3 cm) boreholes are spaced 10 feet (3.048 m) apart from one another, 16” (40.6 cm) boreholes may be spaced 20 feet (6.096 m) apart from one another. In this example the borehole contains four times the explosive and fragments four times the volume of rock. The overall amount of explosive used for that bench is the same as is the volume of fractured rock, however the amount of drilling (in linear feet) drops to 25% as shown in Figure 2-2.
Figure 2.2 - Borehole Spacing Differences

This relationship is evidenced in the drilling cost equation below:

$$C_{m^3} = \frac{C_m}{BA} = \frac{C_{bit} + C_{time}(T_{drill} + T_{tram})}{D + BS^2}$$

Equation 2.2 - Drilling Cost Equation (after Lyons)

Where: $C_{m^3}$ = Cost per m$^3$ of broken rock

$C_m$ = Cost per m drilled

$BA$ = Blasted Area = $BS^2$

$BS$ = Borehole Spacing (m)

$C_{bit}$ = Cost of bit ($)
\[ C_{\text{time}} = \text{Cost of drill operation ($/hour)} \]

\[ T_{\text{drill}} = \text{Drilling time for the given bit (hours)} \]

\[ T_{\text{tram}} = \text{Tramming time (moving and setup between holes) for the given bit (hours)} \]

\[ D = \text{Linear distance drilled (m)} \]

As a result of this equation the increased cost of larger bits, with their shorter lifespans and their somewhat slower drilling speeds (per metre) are balanced against the reduction in number of holes drilled per bench and the reduction in overall drilling and tramming time. This equation must be run by each mine as there are many geological formations where the rock is simply too hard to use a very large diameter bit. In these scenarios the penetration rate may be so low that the drilling cost actually goes up with larger bits. In these cases mines must experiment with different bits to find the right combination of bit life, bit cost and penetration rate that still minimizes the number of holes drilled and therefore minimizes overall drilling cost.

### 2.7 Vibration Analysis
Vibrations on open pit drills are recorded with accelerometers attached to the drill mast. In this study, there are two sensors; one oriented vertically, the other horizontally. The combination of these two sensors enables the drill to gather vibration information whether axial or lateral. While open pit drills use the vibration sensors as a feedback device to monitor the buildup of vibration energy, the raw signal data can be used to compare against the primary drilling variables.
In order to analyse the various frequencies present in a vibration signal, Fast Fourier transforms (FFT) are used as an efficient way of computing the discrete Fourier Transform. The concept of this process was first discovered in 1805 by Carl Freidrich Gauss and was rediscovered over time and has seen extensive use since 1965 through computing applications (J.W. Cooley and J. W. Tukey, 1965). The process decomposes a discrete signal into components of different frequencies. The resulting spectral density plot showcases the dominant frequencies present in a given signal with spikes of higher power at the causal frequencies (in this case these are typically the frequencies of the rotating components on the machine). This allows the apparently noisy vibration signal to be broken up into its component periodic signals and the amplitudes plotted relative to their frequency.

The process of generating a representative Fourier transform plot works best when using longer sampling intervals and/or high sampling rates. The Fourier transform will plot frequency along the x-axis from zero hertz to the Nyquist frequency; half the recording frequency (Wowk, 1991). The data are binned evenly along the x-axis with the number of bins varying exponentially with the number of samples in the signal window. The number of bins is the largest base-2 value that is smaller than the number of samples in the set. For example, with 1200 samples in the window, 1024 bins ($2^{10}$) are allocated for the x-axis on the spectrum plot. All FFT analysis in this thesis uses the Matlab FFT code which employs the Cooley-Tukey FFT algorithm. An example of a Matlab spectral density plot can be seen below in Figure 2-3.
Vibration Excitation Mechanisms

There are many excitation mechanisms (sources) that cause vibration to occur in drilling. Much of the research on the topic has been conducted in the oil and gas industry, but several of the results can be applied to the mining field. Some of these sources include mass imbalances, misalignment of the drill string, buckling of the drill string, tri-cone bit effects and rotational walk of the drill string. These sources occur at multiples of the rotation speed.

Mass imbalances, such as bent pipes, oscillate laterally at 1 x rotation speed with sympathetic oscillations in the axial direction (2 x rotation speed) and torsionally at 1 and 2 x rotation speed (Besaisow and Payne, 1986). The torsional and axial components are the result of the primary lateral excitation. Misalignments and buckling of the drill string cause lateral oscillations at 1 x rotation speed primarily, with axial and torsional oscillations at 2 x rotation speed as well.
Asymmetric hole bottoms can also induce lateral oscillations at 2 x rotation speed as well due to moment of inertia variations. Tri-cone bit effects cause axial and torsion oscillations at 3 x rotation speed (Deily et. al., 1986; Besaisow and Payne 1986).

Rotational walk causes vibrations that are linearly related to rotation speed, but depend on the hole diameter and the drill steel diameter to determine the correct ratio. Once the drill steel has touched the borehole wall, it will ‘walk’ backwards (counter rotation) to the direction of the drill’s rotation (Vandiver et. al., 1989). In ideal, non-slip conditions, the formula for determining the oscillation speed of the rotational walk is:

$$\frac{D}{(D - d)} \times Rotation\ speed$$

Equation 2.3 - Rotational Walk Oscillation Speed

This phenomenon is more common in the oil and gas industry where the long, narrow drill steel can more easily bend and touch the borehole wall, however it can occur in single pass drilling as well.

2.8 System Identification

System identification is the process of assigning a model or formula to a given set of inputs and trying to produce an acceptable output which matches observed outputs (Ljung, L. 1987). The process as described by Ljung, requires three basic entities; data, a set of possible models and a method of assessing the validity of those models. The data component requirements are simply one or more inputs and one or more outputs. Data can be time-based or frequency based.

Selection of models is an intricate process, there are two methods of choosing models; blindly (black box) or with some knowledge of the system and subsequent input into model
characteristics (grey box). The black box method requires less knowledge of the system dynamics and utilizes mathematical formulae to predict model outputs.

The assessment of the models is crucial to know how well one model compares to another. The metric that determines how accurate each model is will vary from system to system. Most often, system identification is an iterative process whereby the model will be adjusted until the optimal set of variables is found. If the model output is an acceptable representation of the system, the process is finished. If not, other models or combinations of models are assessed, while the possibility exists that there will be insufficient computing power to test models with enough complexity to achieve desired results in a reasonable time frame.

2.9 Specific Energy
The concept of specific energy in drilling was introduced in 1965 by R. Teale. The specific energy is a measure of the energy required per unit volume excavated. The specific energy varies by rock type and condition, much in the same way that uniaxial compressive strength (UCS) varies by rock type and condition (Teale 1965). Specific energy differs from UCS in that it is a measure of the minimum amount of energy required to excavate a given volume of rock in situ, whereas UCS is a measurement undertaken in a laboratory setup to determine the failure point of the specimen. In the field there are many things that prevent the accurate prediction of the minimum energy required to excavate a column of rock. Sub-optimal drilling parameters, for example, fracture the rock more than is necessary. Friction, either mechanical or between the drilling components and the hole wall, further distorts the measurement of specific energy. Having no absolute standard against which to compare, specific energy is calculated as the amount of energy utilized to break a given volume of rock, losses included.
The specific energy equation is made up of axial and rotary components as seen below:

\[ e = \left( \frac{F}{A} \right) + \left( \frac{2\pi}{60} \right) \left( \frac{NT}{AR} \right) \]

**Equation 2-4 - Specific Energy (after Teale)**

Where:
- \( e \) = Specific Energy (Pa)
- \( F \) = Axial (Vertical) Force (N)
- \( A \) = Area
- \( N \) = Rotary Speed (RPM)
- \( T \) = Torque (Nm)
- \( R \) = Rate of Penetration (m/s)

Specific energy is measured in pascals (Pa), a unit of pressure. This shows the linear relationship between the applied force and the area of the drilled rock face. This relationship is the same in UCS testing where the units are also pressure, often given in megapascals (MPa). This equation can also be presented in imperial units but caution must be exercised to include appropriate constants to ensure the correct result.
Chapter 3: Experimental Setup

3.1 Experimental Environment
The experiments performed and data collected for this thesis took place in an open pit iron mine in north-eastern Minnesota in the Mesabi Iron Range shown in Figure 3-1.

![Figure 3-1 - Mesabi Iron Range](image)

The Mesabi Iron Range is 60 miles (100 km) north-west of Duluth and is primarily taconite iron ore. The geological formations are considered extremely hard and are made up of varieties of silicated chert.

The drill used for all experiments herein is the P&H 120A large rotary blasthole drill. The 120A is an electric drill that runs on 3 phase, 7200 volt alternating current. The drill has 2 direct current electric motors to provide the hoist/pulldown force and rotary torque. The hoist motor, via a
rack and pinion system, can attain a maximum no-load feed rate of 80 fpm (24.4 mpm) and supply up to 106,000 lbs. (48,090 kg) of force to the drill string, plus carriage weight of 44,000 lbs. (19,960 kg) for a total of 150,000 lbs (68,050 kg). The rotary motor can apply torque up to 25,000 ft-lbs. (33,900 Nm) and attain a maximum no-load speed of 120 RPM. The bit-air is supplied by a Gardner-Denver rotary screw type compressor rated at 3,500 cfm (100,000 L/m). The bit-depth is measured using an absolute linear encoder with an average resolution of 0.1 feet. The machine vibrations are measured by quartz-based accelerometers manufactured by IMI Sensors with a measurement range of ± 50g and a broadband resolution (1 to 10000 Hz) of 1000μg. The system incorporates signal conditioning units that impart a 10 second lag to the system, however the raw signals were recorded for all vibration data in this thesis.

The electrical control system used on the 120A is an Allen Bradley PLC 5/40. The motor armature voltages and currents are read by the PLC via a signal cable attached to the motor drives. The bit-air, bit-depth and vibration signals are also connected to the PLC rack as shown in Figure 3-2. A DATAQ data logger [DI-718Bx, Dataq Instruments Inc.] was used to record drill variables for post-processing. Specifications for the drill, the accelerometers and the DATAQ data logger can be found in the appendices.

The testing periods took place in 2006 and 2007 and spanned several weeks. The author was responsible for interfacing the DATAQ to the PLC signals, validating data acquisition and signal scaling, observing drilling tests, and gathering all the data used for this thesis. It should be noted that experimentation at an operational mine site is extremely challenging and time consuming. A production drill was used for the experimentation, and hence drilling tests had to be scheduled around operational requirements and scheduled maintenance down times. Logistical hurdles, such as sensor failure of the head position resolver, were also magnified due to the
difficulty in accessing drill components (e.g. the resolver is mounted on the rotary head, which is often inaccessible).

3.2 Signal Acquisition

In order to gather data from the various experiments performed on the drill no additional equipment, sensors or devices were required other than the DATAQ data logger. All signals recorded were taken from existing signals used inside the programmable logic controller (PLC) cabinet. Manufacturer installed accelerometers were used to detect vibrations and are standard equipment on large rotary drills.

The PLC cabinet pictured in Figure 3-2 controls all aspects of the drill and is connected to the operator screen in the cab. The thick blue wires in Figure 3-2 and Figure 3-3 are the leads from the PLC to the DATAQ data logger and were easily connected in parallel with existing connections.

Figure 3-2 - Upper PLC Cabinet with Blue Dataq Connection Wires
3.2.1 DATAQ Data Acquisition

3.2.1.1 Hardware

The DATAQ DI-718Bx-S data logger shown in Figure 3-4 is a 16 channel, optically isolated unit capable of measuring voltages, currents and a number of other signals, such as thermocouple inputs, up to a combined throughput frequency of 14400 samples per second. The DI-718Bx-S uses 14-bit analog-to-digital signal conversion.
For the purpose of vibration monitoring, eleven signals were collected as listed in Table 3-1.

<table>
<thead>
<tr>
<th>Channel</th>
<th>Signal</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Horizontal Vibration</td>
<td>Accelerometer signal</td>
</tr>
<tr>
<td>2</td>
<td>Vertical Vibration</td>
<td>Accelerometer signal</td>
</tr>
<tr>
<td>3</td>
<td>Bit Air Pressure</td>
<td>Pressure Transducer current signal</td>
</tr>
<tr>
<td>4</td>
<td>Hoist Current</td>
<td>Current value from hoist drive</td>
</tr>
<tr>
<td>5</td>
<td>Rotary Current</td>
<td>Current value from rotary drive</td>
</tr>
<tr>
<td>6</td>
<td>Hoist Voltage</td>
<td>Back EMF from hoist motor</td>
</tr>
<tr>
<td>7</td>
<td>Rotary Voltage</td>
<td>Back EMF from rotary motor</td>
</tr>
<tr>
<td>8</td>
<td>Hoist Voltage Request</td>
<td>Target hoist motor voltage</td>
</tr>
<tr>
<td>9</td>
<td>Rotary Voltage Request</td>
<td>Target rotary motor voltage</td>
</tr>
<tr>
<td>10</td>
<td>Depth</td>
<td>Linear resolver signal in feet</td>
</tr>
<tr>
<td>11</td>
<td>Hoist Current Limit</td>
<td>Operator selected value</td>
</tr>
</tbody>
</table>

Table 3-1 – Dataq Signal List

The signals inside the PLC of greatest importance to the monitoring of vibration signals are the motor currents and voltages as well as the accelerometer signals, while the other signals are important for interpretation and sorting of the data, as well as possible future research.
3.3 Recording methodology

The data sampling rate used for the majority of the experimentation in this thesis was 400 Hz per channel. When testing for aliasing, a 2000 Hz recording frequency was used, while large volume data collection during periods when the data logger was left running unattended was recorded at 50 Hz. The adoption of 400 Hz was due to aliasing concerns at lower sampling rates balanced against file size and recording capacity of higher rates. When analyzing physically disruptive vibrations, higher frequencies do not sufficiently influence operator comfort or structural integrity of the drill. These higher frequencies do not cause excessive movement of the drill, further validating the 400 Hz recording frequency choice. Further discussion of recording and aliasing can be found in chapter four.

3.3.1 Signal mapping

Information given by the manufacturer about the full range values for each channel allowed the ±10V signals inside the PLC to be converted to their real world equivalent values of volts, amps, g, psi, L/s and metres. Using proprietary DATAQ® software (WINDAQ, DATAQ®, Version 2.51, 2006) the recorded files were converted first into *.DAT files and subsequently scaled to real world units and analyzed using Matlab® (Matlab®, The Mathworks, Version 7.4.0.287, 2007).
Chapter 4: Structural Vibration Responses

In order to establish baseline values for ambient vibrations, the natural frequency of the drill was tested through a series of structural vibration tests. The tests included striking both the drill mast and drill steel and recording the resultant vibrations at a sampling rate of 2000 Hz. In both cases the accelerometers used to measure the subsequent vibration were located on the mast. Mounting a temporary accelerometer on the drill string was considered and the approach was deemed inappropriate for this study due to the curved and irregular surface of the drill steel which inhibits a solid connection through either adhesive or magnet mounts. Stud mounting to the drill steel or the mast itself was not a possibility due to safety regulations at the mine site.

4.1 Impact Testing

The impact testing was carried out by striking first the drill mast and then the drill steel in both the lateral and longitudinal directions as seen in Figure 4-1. By repeating these processes at different depth intervals, the changing nature of the natural frequency can be analyzed for different conditions that can be seen while drilling. During the testing, the drill operator would drill to a specified depth, stop rotating the drill steel and apply a static load in increasing steps, under different weight-on-bit conditions to determine the natural frequency.
The tests were performed under varying bit loads or pulldown pressures as well as at different depths. The table of tests performed can be seen in Table 4-1.

<table>
<thead>
<tr>
<th>Depth</th>
<th>Weight on Bit</th>
</tr>
</thead>
<tbody>
<tr>
<td>0 Feet @</td>
<td>44,000 Lbs</td>
</tr>
<tr>
<td>10 Feet @</td>
<td>44,000 Lbs</td>
</tr>
<tr>
<td>20 Feet @</td>
<td>44,000 Lbs</td>
</tr>
<tr>
<td>30 Feet @</td>
<td>44,000 Lbs</td>
</tr>
<tr>
<td>40 Feet @</td>
<td>44,000 Lbs</td>
</tr>
</tbody>
</table>

Table 4-1 - Impact Testing Parameters

The resulting vibration plot shows a very clear accelerometer response (blue and green lines) for the strikes to the mast but no identifiable readings for the strikes to the drill steel. This indicates that all of the mechanical connections between the drill steel and the sensors on the mast were
sufficiently attenuating the signal as to make any natural frequency tests while drilling impossible. An example of the impact testing plot can be seen in Figure 4-2.

![Impact Testing (10 Feet)](image)

**Figure 4-2 - Impact Test at 10 Foot Depth**

With the steel at depth, no response from the drill steel can be seen, however with the bit at zero depth the strikes to the steel can be seen just above the ambient noise as shown in Figure 4-3.
While difficult to see with the naked eye, these steel strikes can be analyzed and a natural frequency determined. In Figure 4-4 we can see the frequencies present before the steel strikes and in Figure 4-5 we can see the frequencies present just after the steel strikes. There are multiple peaks that appear, however the lowest ones are in the 70-85 Hz range and are quite a bit higher than our area of interest. (The frequency range of interest for vibration is discussed in section 4.3.)
Figure 4-4 - Frequency Domain Before Steel Strikes
Figure 4-5 - Frequency Domain After Steel Strikes
Previous research (Aboujaoude, C., 1997; Aboujaoude, C., 1991) has shown that as the length of drill string in the hole increases, the natural frequency decreases. While many open pit mines will use multiple steels in multi-pass drilling, the large diameter borehole drills focused on here are single pass machines. As a result the natural frequency can be assumed high enough that drilling induced vibrations from the bit-rock interface will not induce resonance in the drill string.

The frequencies that were seen from the mast strikes are summarized in Figure 4-6 and Table 4-2. As shown, the level of force applied to the bit does very little to change the natural frequency of the mast while depth has a more significant influence.

![Mast Natural Frequency Diagram](image)

**Figure 4-6 - Mast Natural Frequency**
The natural frequency is seen to steadily decrease as depth increases, with the exception of the values at 40 feet (12.2 m). This illustrates a potential problem with frequency measurements on this model of drill. The mount for the accelerometer is at the midway point of the mast (Figure 4-7) and once the carriage has passed it the sensor is no longer in the stressed zone between the motor and the bit-rock interface. This could prove to be problematic for data acquisition, however the majority of holes drilled on site are between 40 and 45 feet (12.2 and 13.7 m) deep and no vibration data is presented for analysis from the lower portion of any hole.

<table>
<thead>
<tr>
<th>Weight on Bit (lbs)</th>
<th>Depth (feet)</th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0</td>
<td>10</td>
<td>20</td>
<td>30</td>
<td>40</td>
</tr>
<tr>
<td>44,000</td>
<td>403.9</td>
<td>356.9</td>
<td>342.1</td>
<td>329</td>
<td>409.6</td>
</tr>
<tr>
<td>60,000</td>
<td>405.6</td>
<td>356.7</td>
<td>340.3</td>
<td>334</td>
<td>409</td>
</tr>
<tr>
<td>80,000</td>
<td>405.6</td>
<td>355.3</td>
<td>335</td>
<td>337.2</td>
<td>405.3</td>
</tr>
<tr>
<td>100,000</td>
<td>404.6</td>
<td>355.4</td>
<td>335.1</td>
<td>342.5</td>
<td>403.9</td>
</tr>
</tbody>
</table>

Table 4-2 - Mast Natural Frequency (Hz)
4.2 Damping Ratio
In addition to analyzing the natural frequency as it changes under load and depth, the damping ratio of the mast was analyzed from the hammer strikes in section 4.1. The damping ratio is normally calculated by looking at adjacent peaks in a decaying sinusoid, however this requires that the system in question be a 2\textsuperscript{nd} order system. With the all the complexity of the drill mast and components attached to it, the impulse response is non-ideal for 2\textsuperscript{nd} order analysis (Dareing, D.W. and Livesay, B.J., 1968). Nonetheless, averaging over several peaks instead of just one pair of adjacent peaks can give the effective damping ratio. The vibration signals was low-pass filtered with a cut-off frequency of 10 Hz using a fifth-order Butterworth filter. The filtered vibration signal can be seen (with peaks) in Figure 4-8.
By isolating the maximum value and the time when the values fall back below the background level, both an amplitude difference and a time period are specified. By knowing the dominant frequency (by Fast Fourier analysis) we can manipulate the following formula to find the damping ratio:

$$\frac{\ln\text{(tolerance fraction)}}{\omega_n \ast \zeta}$$

Equation 4.1 - Time Period

Where:

Ts = Time Period (seconds)

Tolerance fraction = maximum value/minimum value

$\omega_n$ = Natural Frequency
\[ \zeta = \text{Damping ratio} \]

Isolating the damping ratio gives us:

\[ \zeta = \frac{\ln(\text{tolerance fraction})}{\omega_n \cdot T} \]

**Equation 4-2 - Damping Ratio**

Since we only have the recorded natural frequency, \( \omega_d \), and not actual natural frequency, \( \omega_n \), we use this formula to substitute:

\[ \omega_n = \frac{\omega_d}{\sqrt{1 - \zeta^2}} \]

**Equation 4-3 - Natural Frequency**

Where:

\( \omega_d = \text{Damped Natural Frequency} \)

By substituting this and rearranging we arrive at the following equation:

\[ \sqrt{\frac{\zeta^2}{1 - \zeta^2}} = \frac{\ln(\text{tolerance fraction})}{\omega_d \cdot T} \]

**Equation 4-4 - Damping Ratio Solution**

Using this equation, we can solve the right hand side and subsequently solve for \( \zeta \). A table of the damping ratios is seen in Table 4-3.
Weight on Bit (lbs) & Depth (feet) \\
<table>
<thead>
<tr>
<th>(lbs)</th>
<th>0</th>
<th>10</th>
<th>20</th>
<th>30</th>
<th>40</th>
</tr>
</thead>
<tbody>
<tr>
<td>44000</td>
<td>0.00330</td>
<td>0.00421</td>
<td>0.00435</td>
<td>0.00353</td>
<td>0.00339</td>
</tr>
<tr>
<td>60000</td>
<td>0.00668</td>
<td>0.00343</td>
<td>0.00448</td>
<td>0.00485</td>
<td>0.00411</td>
</tr>
<tr>
<td>80000</td>
<td>0.00344</td>
<td>0.00491</td>
<td>0.00553</td>
<td>0.00349</td>
<td>0.00454</td>
</tr>
<tr>
<td>100000</td>
<td>0.00469</td>
<td>0.00379</td>
<td>0.00792</td>
<td>0.00260</td>
<td>0.00372</td>
</tr>
</tbody>
</table>

Table 4-3 - Damping Ratios for the Mast System (zeta values)

As can be seen in Figure 4-9, there is no obvious association between damping ratio and depth as there was for the natural frequency. More importantly, the extremely low values of zeta indicate that the recorded oscillation frequencies are within 0.01% of the actual natural frequencies and are therefore treated as such for the remainder of this thesis. When damping ratios are low as is common for steels (Wowk, 1991), it indicates there is very little damping in the system to attenuate an impulse. This is very important to note as it means there is little in the system to help mitigate vibration transmission.
4.3 Vibration Excitation Mechanisms

Another method of analyzing the structure is through finite element analysis and computer simulation. Modelling of the drill string and all of its associated component parts was not conducted due to the changing nature of the drill steel and drill-sub. As the steel turns in the hole, it gradually wears away until it needs to be replaced. This erosion of material changes the vibrational properties over its life. This phenomenon occurs in a non-uniform fashion and is inconsistent from steel to steel as the drills are used in different areas of the mine. As each mine is different and observes different drilling practices, attempts to model this progressive wear are impractical and were not pursued.

Modelling the mast itself, while complex, is possible but unnecessary. Observing the range of natural frequencies in the preliminary tests, all the frequencies lie near the 300-400 Hz range. For the drilling-induced vibrations that are of concern, these high frequencies are of minimal consequence to the much lower frequency, high-displacement vibrations that cause damage to the drill. The same principle also applies to the drill steel itself. Previous work has shown (C. Aboujaoude, 1997) that single drill steels have resonant frequencies above 80 Hz, which is outside of our range of interest.

Due to the vibrational excitation mechanisms discussed in chapter 2, it is known that the drill string, and consequently the drill, will be excited at multiples of the rotation frequency. Since the maximum rotary speed of the drill is 120 RPM or 2 Hz, and the focus is on 1x, 2x and 3x rotation speed excitation mechanisms, induced frequencies should be below 6 Hz. By focusing on the lower frequencies (0-20 Hz) of the vibrational spectrum, vibrations can be adequately
recorded using the manufacturer installed accelerometers and can bypass the computer modeling of the structural components of the mast and the drill itself. Using the 0-20 Hz range, it is also possible to detect harmonics above 6 Hz adequately as well.

Since the natural frequencies of the drill steel and mast are much higher than the excitation mechanisms that cause vibration, vibrations recorded at lower frequencies are not being amplified by resonance, but rather are the actual magnitudes of vibrations that represent the various effects of the process at the bit-rock interface.

### 4.4 Ambient Vibration

Whenever a blasthole drill is powered up there are subsystems that contribute background or ambient vibration that can be detected by the accelerometers on the mast. The process of determining the ambient frequencies present is crucial to identify them later as background noise when looking for vibrations caused by drilling. Figure 4-10 shows the fast Fourier transform plot of the horizontal vibration recorded at 2057 Hz with all systems powered and ready, but not drilling. Conversely, Figure 4-11 shows the same horizontal vibration signal moments later after the drill has begun drilling.

![Fast Fourier Transform](image)
Figure 4-11 - Fast Fourier Transform of Horizontal Vibration

While the amplitudes all increase during drilling (vertical axes do not match in order to better show the peaks) several of the high frequency peaks match in both figures, indicating that these frequencies are not due to the drilling action itself but rather the onboard systems themselves. The peaks in this figure are all from high-frequency sources. Such high frequencies come from other components on the drill such as pumps and fans. The drills themselves incorporate large compressors for pumping air to the bit in order to adequately clear the hole. These high-speed, multi-vane compressors can deliver 3,500 cubic feet per minute (100,000 L/m) and are incredibly loud. In addition to the air-compressor there is also a complex hydraulic sub-system with pumps and motors that operate the auxiliary devices on the drill. Their frequencies are within the range of human hearing (20 – 20000 Hz) and are audible while in proximity to the component in question. Of particular note, the air compressor produces the most audible noise in the 200-250 Hz range and can be heard distinctly from over a hundred metres (330 feet) away. These high frequency signals have the potential to affect data collection of lower frequency data if sampling rates are low and aliasing is not taken into account.
4.5 Aliasing
As a result of all these higher frequency peaks, aliasing must be considered when recording at slower rates. In order to avoid aliasing the recording frequency has to be twice the highest frequency or twice the Nyquist frequency (Nyquist, H., 1928; Figliola, R.S. and Beasley, D.E., 2000). The recording frequency of 2057 Hz was chosen as the closest option available to 2000Hz on the DATAQ® unit. The recording modules inside the DATAQ® have a cut-off frequency of 1000 Hz, therefore a recording frequency of 2000 Hz is sufficient to ensure no aliasing is present in the signal.

The use of 400 Hz as the recording frequency for analysis was done knowing the possible existence of higher frequencies that could impact the data. It was nonetheless chosen as a balance between hardware/software limitations and data accuracy. Figure 4-12 shows a region of high vibrations recorded at 400 Hz.

Figure 4-12 - High Vibrations Recorded at 400 Hz

In the above figure there is a lot of apparent vibration energy between 60 and 90 Hz as well as 160 and 180 Hz. These values are reflections or aliasing of higher signals. Figure 4-11 shows activity near 225 Hz that is reflected in Figure 4-12 as 175 Hz. Activity at 475 Hz reflects at 75
Hz. These signals reflect at the Nyquist frequency, in this case 200 Hz, therefore signals at 200, 600 and 1000 Hz register as 200 Hz and 0, 400 and 800 Hz frequencies register at 0 Hz. As the frequency increases, it reflects back and forth between 0 and the Nyquist frequency (Figliola, R.S. and Beasley, D.E., 2000). Due to the limited number of high frequency signals and their observed static nature (with the exception of the mast natural frequency) they are easy to identify and all are outside of the area of interest below 20 Hz.

By keeping the bottom end of the spectrum free of aliased signals, vibrations induced by the drilling process can be accurately separated from background noise. The lower, sub 20 Hz range is of particular importance as previous work has shown that much of the vibrations recorded while drilling arise from multiples of the rotation frequency (C. Aboujaoude, 1997; Deily, 1968), specifically at 2, 3 and 6 times the rotation frequency. Frequency spikes are often seen as a result of down-the-hole occurrences such as inclusions or voids, cracks, veins or other inhomogeneities.

In contrast, Figure 4-13 shows how aliasing overshadows the fast Fourier transform of a file recorded at 50 Hz as compared to the same range recorded at 400 Hz. The amplitude of the aliased file is higher as more information is being interpreted as a single frequency while the real information is overshadowed. By reducing the Nyquist frequency to 25 Hz, the size of the reflection ‘window’ is very small and the data produced are not useful.
Aliasing can be avoided by low-pass filtering however the control system does not employ such a filter as it is primarily concerned with vibrational energy, not frequency. The lower frequency range is typically responsible for the bulk of this energy due to the sometimes substantial displacement of the drillstring resulting from bit-rock interaction. To illustrate the importance of the lower end of the frequency spectrum, vibration velocity plots can be used instead of vibration acceleration plots. This method of looking at vibration obscures higher frequencies unless they are powerful enough to contribute significant energy into the system. These plots are further discussed in chapter 5.

Figure 4-13 - Aliasing in the 50 Hz Vibration Signal as compared to the 400 Hz Vibration Signal
Chapter 5: Vibration While Drilling

This chapter presents the analysis of the vibration signal while drilling both in normal and vibration exception conditions (where excessive vibrations cause parameter changes in the drill’s autodrill mode). The effect of geology on drilling parameters is presented. Vibration measurement is shown in acceleration plots as well as velocity plots. Specific energy and kinetic energy imparted to the drill are addressed. Finally, the importance of lower frequencies on drilling vibration is presented.

5.1 The Effect of Geology on Vibration

Vibration in rotary drilling is unavoidable but can become uncomfortable and even destructive if drilling parameters are not altered to accommodate changes in geology. Vibration can increase quickly as the drill encounters any number of geological scenarios such as changes in lithology, fractured ground, boulders or when the drilling parameters are unsuitable for the rock type being drilled.

Geological changes throughout a hole that are not accompanied by a subsequent change in drilling parameters often end up in higher vibration due to poorly selected values for pulldown pressure and rotary speed. As the bit is more or less engaged in the rock face, burying or skipping of the bit (lack of penetration) can occur. As fractures are encountered, resistance can increase due to material sloughing, bit ‘stick-slip’ (the repetitive stopping and rapid rotary acceleration of the bit) or insufficient clearing of the cuttings due to air pressure drop. Hard boulders or tri-lobate surfaces occurring at the hole bottom can cause severe oscillations at multiples of the rotation frequency. These occur most typically at multiples of 3 and 6 times the rotation frequency due to the three cones on the bit itself. Some inclusions/events occur near
the hole wall and generate vibrations once per revolution for each cone, while other events (such as fractures) that bisect the hole bottom cause each cone to strike twice per revolution, resulting in the 6-times rotation frequency oscillations.

All of the above scenarios require adjustment of the drilling parameters to mitigate severe vibration, either through computer controlled algorithms or operator experience. The nature of the drilling environment is such that any combination or variation of the described scenarios can occur at almost any time and it is impractical to design a control system that can recognize them all.

5.2 Torsion while Drilling

One of the events that can occur in drilling is a stick-slip (start-stop) rotation seen visually from the operator cab. As discussed, this is due to geological constraints at the bit-rock interface, however the question exists as to whether or not the stoppage is due to stalling of the motor or twisting of the drill steel. In oil drilling, where several drill pipes are attached end to end, there is quite a lot of research devoted to drill string deviation, drill steel whirl and oscillating torques. As the drill steel gets longer and longer, these all become serious concerns. However, with only one drill steel 65 feet in length, the angle of twist is not very large.

Occasionally while drilling, the drill steel can be seen to stop and start at regular intervals at three times the rotary speed. This visual observation demonstrates that the downhole events are having an effect but in order for the stoppage to be due to pipe twist, the degree of twist will have to be several degrees from the bottom to the top of the steel in order to be visually observed. To calculate the angle of twist due to torsion, the following equations are presented:
\[ \theta = \frac{TL}{JG} \]

Equation 5-1 - Angle of Twist

Where:
- \( J \) = Polar Moment of Inertia (m\(^4\))
- \( G \) = Shear Modulus (Pa)
- \( T \) = Applied Torque (Nm)
- \( L \) = Length of Cylindrical Pipe (m)
- \( \Theta \) = Angle of Twist (radians)

The formula for the polar moment of inertia is given for a pipe:

\[ J_{(pipe)} = \frac{\pi}{2}(r_o^4 - r_i^4) \]

Equation 5-2 - Polar Moment of Inertia for Pipe

Where:
- \( r_o \) = Outer Radius (m)
- \( r_i \) = Inner Radius (m)

The exact dimensions of drill pipe vary as different pipes are used for different bit diameters. This is done to maintain air pressure by regulating the annular space between drill steel and the hole wall itself. As the drill steel wears, it becomes progressively thinner. This thickness is regularly monitored as it changes quickly and a minimum thickness must be maintained for proper clearing of the cuttings to prevent plugging of the bit. This ongoing variation in thickness means there are no exact measurements to use for pipe diameter. Therefore we will choose conservative pipe measurements with an outer radius of 14 cm (5.5 inches) and a wall thickness of 2 cm (3/4 inch). Using these measurements, we obtain a polar moment of inertia of 0.0001768 m\(^4\). Using a shear modulus of 79.3 GPa (11,500,000 psi) (Crandall et. al., 1959),
maximum applicable torque of 33,900 Nm (25,000 ft.lbs.) and pipe length of 27 m (88.5 ft) we calculate the following:

\[ \theta = \frac{(33900 \text{ Nm})(27 \text{ m})}{(0.001768 \text{ m}^4)(79300000000 \text{ Pa})} \]

\[ \theta = 0.06528 \text{ rad} = 3.74^\circ \]

With a maximum of only 3.74\(^\circ\) of twist, there must be another cause for the visually perceptible start-stop phenomenon. The above equations show the drill pipe to be quite rigid even when using very conservative figures. The only other possible cause of the start-stop motion is motor oscillations which are discussed in Chapter 7.

### 5.3 Vibration Characteristics

To form a basis for improved computer control of vibration while drilling, vibration levels were recorded in different geological conditions in both regular drilling and drilling ‘exceptions’ where the programmable logic controller reacted to severe vibrations. Different depths of drilling were also considered to see any variation with depth. Figure 5-1 shows the difference between regular and high-vibration drilling.
As shown in the above figure, the magnitude of vibrations can increase substantially in certain cases. Figure 5-1 shows a vibration response induced by a fracture or inclusion that crosses the hole, generating a three-times and six-times rotary speed oscillation. Knowing the rotation speed of the drill is crucial to determining the cause of vibration. A large spike at values other than multiples of rotary speed indicates a different source of vibration. Frequency spikes that are not as clear can indicate plugged or buried bits that are regrinding material. Most often however, this does not occur and vibrations that generate a spike at a distinct frequency are related to rotary speed.

The three- and six-times rotary speed oscillations occur due to the three cones on the tri-cone bit. The interactions the cones have with the drilling face generate oscillations at multiples of the rotation frequency as they move over obstacles. These types of oscillations show up frequently in the drilling logs because the majority of the energy input to break the rock is reflected through the rotary motor signals themselves.
5.3.1 Specific Energy

The energy required to break rock by drilling is commonly known as specific energy as defined in chapter 2 and it is the amount of energy required to break a given volume of rock. The two components that make up breakage are the rotary and hoist components. The most accurate way of measuring specific energy is to look at the power consumption of the motors for a specific volume. Since power equals voltage times current (P=VI), the power can be calculated from the voltage and current signals already monitored. The diameter of the hole is known as is the depth of the bit at any given time. With these variables specific energy can be calculated throughout a drill hole over its instantaneous volume.

Figure 5-2 shows the specific energy while drilling a typical hole. The green lower line represents the energy input by the hoist motor and the blue line represents the energy input by the rotary motor. The summation of the two (red line) gives the overall energy input while drilling. The first two metres of drilling show very little hoist component and quite a lot of rotary energy as a result of broken ground at the top of each hole from the previous bench blast.

![Figure 5-2 - Specific Energy Plot](image-url)
As seen in the above figure the rotary component is dominant, contributing 80-90 percent of the energy required to break the rock. As this energy increases, the drilling conditions usually become rougher and vibration levels increase. An increase in input energy results in an increase of energy released both in acoustic energy as well as vibrational energy. Specific Energy itself is analyzed further in section 5.7.

5.3.2 Dominant Frequencies

Even in normal drilling there is some drilling induced vibration at levels which are acceptable, both from an operator comfort standpoint and structural integrity standpoint. In order to find relationships with the vibration signal, the drill was run at different speeds in small steps between 30 and 117 RPM (120 RPM attempted but halted due to safety concerns). These frequency domains were plotted and a relationship was found between the vibrational frequencies and RPM. The results showed a distinct peak in the vibration signal at 6-times the rotary speed. At lower speeds, the dominant vibration peaks lost correlation with the rotary speed as the overall amplitude of vibration was low and the peaks blended into the ambient background noise. A plot of the data can be seen in Figure 5-3.
The above figure shows the dominant vibration spikes as well as secondary spikes where there was a vibration frequency at 6-times the rotary speed but the frequency amplitude was not the highest in the plot. This is observed in the lower half of the data set between 45 and 60 RPM, however drilling occurs at speeds typically between 60 and 90 RPM and therefore the loss of correlation at slower speeds is not of operational importance. Below 45 RPM, the dominant peaks in the frequency spectrum of the vibration signal trend around 7-8 Hz as the overall values of vibration are lower and the background noise can be seen. This decrease in rotary induced vibration energy can be seen in Figure 5-4 which compares the RMS vibration magnitude to the rotary speed.
As shown above, as the drill rotates faster, the interactions between the rock face and the drill bit occur more rapidly and increase the vibration amplitude, however the variable nature of the geology is such that a good correlation is difficult to demonstrate.

5.4 Velocity and Vibration

The amount of kinetic energy imparted to the drill as a result of these vibrations depends on the magnitude of the velocity of vibration; the integral of the acceleration signal. Figure 5-5 shows the acceleration and velocity plots of the same region of high vibration shown in Figure 5-1.
The velocity plot shows speeds as high as 1.35 cm/s at 3.615 Hz and 7.23 Hz. The acceleration plot also shows that the magnitude of vibrations at 7.23 Hz are strong and not just harmonics of the 3.615 Hz peak. The energy transferred to the drill is dependent on the mass of the moving components and their velocity. The velocity plot shows the importance of lower frequencies and how peaks at higher frequencies in the acceleration plot trend asymptotically toward zero.
in the velocity plot. It should also be noted that the beginning of the velocity plot shows a very sharp rise off the scale of the plot. As velocity is the integral of acceleration, the values towards the low end of the frequency scale trend towards infinity. These are erroneous and not part of the drilling action.

5.5 Vibration at Multiples of RPM

As mentioned in section 5.4, the tricone bit produces vibrations not only at 3-times the rotary speed but at 6-times the rotary speed as well. Due to the heterogeneous nature of the drilled strata, there are several inclusions that can cause a 3-times vibration as each cone rolls over or impacts the inclusion, however this does not explain the 6-times vibration that is seen.

Fractures can explain the 6-times vibration quite well. As each cone makes one revolution, a fracture will be encountered twice by each cone. This doubling causes 6 strikes of the fracture or crack. This effect can be seen in the vibration signal as the fracture moves across the borehole.

A consistent fracture, shown as a plane in Figure 5-6, will intersect the borehole at some angle between 0 and 90 degrees. As the bit progresses through the face, the fracture will ‘move’ from one side of the hole to the other. As it begins to progress, each cone will encounter and leave the fracture once per revolution, however as the fracture progresses (as the bit gets deeper in the borehole) each cone will actually clear the fracture and reencounter it before completing a full revolution; the cones each strike the fracture twice. As the fracture continues to progress, the reverse occurs.
Figure 5-6 - Borehole Intersected by Fracture

By plotting FFTs of the vibration signal in successive steps, this process can be seen in Figure 5-7. All FFTs contain peaks at both 3 and 6 times the rotary speed with their magnitudes being the only difference. This is due to the progressive nature of the fracture and the manner in which the cones cross it. While the cones meet the fracture parallel to their direction of travel, once the fracture is in the middle of the hole, the fracture is perpendicular to the direction of travel. As this is happening, the carbides are ‘falling’ into the void of the crack and then ‘climbing’ out again, even when the fracture is at the edge of the hole.
Figure 5-7 - Borehole Intersection FFT Plots
This means that while the cone may only contact the fracture once per revolution, as it progresses that ‘one contact’ means both an ‘entering’ and ‘exiting’ as the carbides go deep in the fracture and then climb out.

The complex combination of progressing fracture, changing angle of incidence for the cones, interaction of carbides going in and out of the fracture and the varying geology makes the identification of this phenomenon difficult. In addition, Figure 5-7 shows that both the 3-times and 6-times rotary speed-induced vibrations are hard to distinctly identify because the vibrations show up once the fracture has already started to cause an issue. This situation is identifiable however as there are separate scenarios in the data where only distinct 3-times vibrations can be seen. Distinct 6-times vibration events (that excluded 3-times vibration) were not found that were not either led by or followed by a 3-times vibration event. Further exploration of this phenomenon would require a controlled laboratory setting to try to mitigate changing geological effects. In the geological context of this thesis, several fractures (both healed and/or open) were encountered yielding several 6-times rotary speed induced vibrations.

5.6 Vibration with Depth

The structural dynamics of the drill are constantly changing as the depth increases and the bit moves further away from the drill itself. In oil drilling, the process is rendered even more complicated with the constant addition of drill pipe sections, increasing the drill string tendency to deviate, bend, buckle, twist, whirl and many other potential issues that arise through the use of drill strings of several hundred metres in length. For production drilling in mining, however, the depths are very short and the deviation and torsion induced vibrations observed in oil
drilling do not present themselves. The following sections show any changes of vibration with depth.

5.6.1 Normal Drilling Conditions

While the natural frequency of the drill changes predictably as shown in chapter four, the vibration frequency throughout the drilling process does not change in any measureable fashion. Figure 5-8 and Figure 5-9 show the vibration frequencies, both acceleration and velocity, at depth in ten foot increments all at levels below 0.025 g. In general, frequencies below 5 Hz have lower amplitude than frequencies seen during vibration events which show more activity in the same frequency range. Again, as with Figure 5-5, Figure 5-9 has high peaks at the very low end of the frequency scale. These peaks are erroneous and come from the integration of the acceleration signal.
Figure 5-8 - Typical Vibration Levels in Normal Drilling (Acceleration)
Figure 5-9 - Typical Vibration Levels in Normal Drilling (Velocity)
5.6.2 Vibration Exception Drilling Conditions

Vibration frequency plots during severe vibrations show a marked difference from normal drilling but are very different from one another due to the unique geological circumstances of each event. Figure 5-10 and Figure 5-11 show two different cases of severe vibration from the same hole at different depths. In Figure 5-10 several harmonics are seen that are not present in Figure 5-11. Despite being from the same hole in one overall rock type, the two vibration conditions represent different lithologically induced events.

Both Figure 5-10 and Figure 5-11 include normalized plots. This indicates that all of the frequencies on the x-axis are normalized to the first peak to more easily view harmonics of a given frequency. For example, if there are peaks at 3, 6, 9 and 12 Hz, normalizing the plot to the 3 Hz peak will result in peaks at 1, 2, 3 and 4. This allows us to not only see the harmonics of one vibration excitation mechanism more clearly, but to also point out areas where the vibration excitation mechanism is unrelated. Again, as with Figure 5-5 and Figure 5-9, the velocity plots in Figure 5-10 and Figure 5-11 show very high levels towards the low end of the frequency spectrum. These are aberrations caused by the integration of the acceleration signal and not true values.
Figure 5.10 - Vibration Exception at 4.8 Feet with Harmonics of the 3x Rotary Speed
Figure 5-11 - Vibration Exception at 23.5 Feet at 3x and 6x Rotary Speed
5.7 Baseline Comparison

The rock type being drilled has a great impact on the drilling parameters. Different rocks have different characteristics including strength, elasticity, cleavage, density, etc. All of these factors vary from mine to mine, even from hole to hole. A full baseline analysis of vibration conditions while drilling would take into account all types of vibration scenarios from each rock type, however there is no practical way to standardize equipment from mine to mine, nor gain access to all types of rock to do such a baseline. All the drilling data presented in this thesis originate from drilling in very hard silicated chert with a large diameter drill bit, necessitating a large power input. These conditions make for some of the hardest drilling possible and should be compared to similar conditions. Vibrations can stem from any rock type as the penetration rate is inversely related to material compressive strength. As such, even soft rock can generate large vibration when being drilled quickly with an improper combination of drilling parameters. One way to characterize drilling conditions is by using specific energy.
5.7.1 Generating the Specific Energy Plot

Specific energy is the amount of energy required to excavate a given volume of rock as defined by Teale’s equation given in section 2.9. This equation applies to rotary drills only as it looks at the axial and rotational components alone in the calculation. By monitoring specific energy, three-dimensional rock strength profiles of the mine can be generated bench by bench. These additional data are useful for the geological model as well as being immediately useful for the operator to predict the drill performance on a hole to hole basis as they observe the specific energy record of the previous hole, however these data must be clear and easy to understand.

In the case of Figure 5-12 the signal is quite noisy and difficult to interpret.

The cause of this noise is a low penetration rate. At the test site used in this thesis the rock was very hard and the bits were 16 inches in diameter. This combination leads to very slow penetration rates of 0.1 – 0.3 metres/min (0.33 – 0.1 feet/min). The specific energy plot in this circumstance is shown below.

![Figure 5-12 - Specific Energy Signal with Slow Penetration Rate](image-url)
Specific energy is calculated using the calculated electrical power from the pulldown and rotary motors as well as the depth signal and/or rate of penetration signal. Since the rate of penetration component of the equation is part of the denominator, small, fluctuating rates of penetration have an adverse effect on the specific energy trace making it ‘ragged’. Also shown in Figure 5-12 is a signal called ‘Delta Volume’. This is the rate of penetration signal multiplied by the drilling area of the bit and reflects how much rock is being excavated over time. The signal roughly mirrors the specific energy plot, high when the specific energy is low and vice versa.

This ‘mirroring’ effect reflects exactly how the specific energy equation should work but also shines a light on the effect penetration rate has on the SE equation. In Figure 5-12 the depth signal is being smoothed over a fifteen second window. Figure 5-13 shows the specific energy signal without smoothing the depth signal. Without smoothing the signals, the low penetration reduces the utility of the SE calculation to nil from an operator standpoint.

![SE Plot (Electrical) with labels](image)

**Figure 5-13 - Specific Energy (Unsmoothed Depth Signal)**
It should be noted that even the depth signal in Figure 5-13 is an interpolated one; the original depth encoder signal recorded from the drill is a stepped signal that proves to be useless in generating any specific energy information. With a ‘stepped depth’ signal, the denominator is zero the majority of the time, thus precluding the possibility of calculating SE.

In order to interpolate the depth signal as accurately as possible, the hoist voltage signal is used. Hoist voltage is used as a modifier or multiplier against the depth signal because the hoist motor is what is causing the progression down the hole. Both the dead weight of the head and drill string combined with the pulldown force (analogous to Hoist motor current) supply the force, but the actual movement of the motor itself is seen in the voltage. If the motor is pulling down at full power with no movement, no voltage is seen on the motor armatures. Subsequently, to adjust for the ‘steps’ the voltage signal is summed over the step and at each point the incremental depth is calculated as follows:

\[
\Delta d = \frac{V_i \cdot \Delta D}{\sum_{i=a}^{b} V}
\]

*Equation 5-3 - Interpolated Incremental Depth*

Where:

\( \Delta d \) = Incremental depth per sample point

\( \Delta D \) = Total depth of step

\( V_i \) = Instantaneous voltage reading for sample i

\( a \) = Start time of ‘step’

\( b \) = End time of ‘step’
The application of this equation can be seen in Figure 5-14 which shows the original ‘steps’ as well as the interpolated signal by which method the SE is accurately calculated.

![Figure 5-14 - Interpolated Depth Steps](image)

The SE equation, therefore, can only be reliably calculated by using either very accurate, high resolution depth encoders or by supplying an SE value for a depth segment equal to or significantly larger than the depth of each ‘step’. Due to non-linear depth progression as a result of the action at the bit-rock interface, instantaneous SE is impracticable. In addition, depth segments near to, but not the same size as the depth steps will cause the same problem as the un-interpolated signal due to improper averaging. In larger, multi-step windows the averaging errors will get smaller as more steps are included in the window.

5.7.2 Use of the Specific Energy Data

Specific energy data are used by mines to determine the strength of the drilled rock. This information is used by the blasting department with the aim of modifying hole charges to better suit the rock type. With proper loading of the blastholes, more uniform fragmentation results
and more predictable feed is available for the crushing/grinding circuit. This reduces costs further down the line on excavation equipment as well as all the stages of the refining process. As such, accurate specific energy data are very important.

Specific energy can also give a record of the drilling conditions the drill has faced over time. By recording how much energy the drill consumes and how that consumption fluctuates, specific energy can be used as an indicator for machine health much like vibration information. Furthermore, the strength profile generated by the use of specific energy could potentially be used to alter the settings in the PLC of the drill by detecting what kind of ground the drill is in and setting the motors accordingly.

At present it is uncertain whether any drill uses specific energy in this way, however it may be worthy of future study.
Chapter 6: Vibration Correlation

6.1 Rotary Current Oscillation

Of the four main signals being recorded, the rotary current is most directly connected to the vibration signal. This relationship exists for two reasons; the rotary motor is applying ninety percent of the energy that is required to break the rock and the motor drives are speed controlled. The rotary specific energy contributes 80-90% of the energy into breaking the rock and is therefore the largest power contributor to the rock face. Since power can be represented as voltage times current (P=VI), the voltage and current signals monitored on the drill are the signals we can investigate to determine relationships with the vibration signal. Speed controlled motors try to maintain the speed as constant as possible and apply the necessary torque in order to do so. The RPM is determined by the motor voltage and the torque is determined by the motor current. The speed controller attempts to keep the motor voltage constant, however any resistance or obstruction the bit encounters will slow the bit and require an increase in current to bring the rotating bit up to speed again. As a result, the rotary motor current signal is the primary signal to investigate.

One of the more common cases of high vibrations in drilling occurs when there are fractures or hard inclusions at the bit face. Every time a cone on the bit strikes the abnormality there is a sharp increase in resistance which slows the bit momentarily. The motor drives increase the current to the motor to regain speed. Once the cone has passed over the abnormality, the current is reduced until the next cone encounters the same resistance. This repetitive action causes significant oscillation in the drill string which shows up as an increase in vibration amplitude as well as oscillations in the rotary torque signal. The torque signal shown in Figure
6-1 demonstrates how large this ripple can be; up to 10,000 Nm (7,375 ft. lbs) at three times the rotary speed, which in this case is 3.615 Hz.

![Rotary Torque Oscillation](image)

**Figure 6-1 - Current Ripple Amplitude and Frequency**

The vibration signal has been shown to correlate with the rotational speed in chapter five. The current signal also exhibits a correlation to RPM at three times and six times the rotary speed, as shown in Figure 6-2 below. Also shown is Figure 6-3 which demonstrates the relationship between current frequency (three times RPM) and vibration frequency. As in chapter five, the smooth, linear relationship (Figure 5-3) breaks down below the 60 RPM mark as background vibrations begin to dominate.
Figure 6-2 - Current Frequency Plots

Figure 6-3 - Vibration and Current Frequency Relationship
6.2 Vibration Contributors

The rotary current signal is not the only cause of vibration, the geology of the drilling face is a major factor. As different geological scenarios are encountered, the vibrations can become amplified by the drilling control system as it attempts to maintain a specific rotary speed. It is the combination of poor controller settings and difficult geology together that produce the damaging vibrations when the geology changes to where the control system set points will start to oscillate and the time constant in the system is too slow to respond. One way to mitigate vibration is to adjust the pulldown force and rotary speed such that vibration is dissipated, however the programming in current control systems is often ineffective in accomplishing this.

6.2.1 Signal Analysis

In order to show the correlation between vibration and motor signals, the voltages and currents from the hoist and rotary motors were compared visually against the vibration signal. The rotary voltage signal is not well aligned with the vibration signal as the motor is speed controlled. As the drill goes through varying rock conditions, the current signals respond with increases, decreases or oscillations to maintain motor voltage.

The hoist motor current is not as active as the rotary motor current because it is most often at its maximum limit throughout drilling. The control system is set to maximize penetration rate, but there is an upper limit on pulldown pressure or weight on bit because the tricone bit can tolerate only a certain amount of force. This ceiling is necessary but flattens the hoist current signal. Subsequently, the hoist voltage is not stable; unlike the rotary voltage, it is lower in slow drilling and higher in easier drilling. There is a maximum setting for the penetration rate as a safety mechanism for impact loading on the bit, putting another artificial ceiling in the hoist motor control loop. When this maximum is reached in softer ground, the hoist current will
begin to respond to the changes in geology. The hoist signals, together or separate, do not show an obvious connection with the vibration signal on visual inspection whereas the rotary current signal does show a more obvious correlation.

6.2.2 Signal Identification

The rotary current signal was further analyzed using a software package within Matlab called the System Identification Toolbox. This toolbox is an addition to Matlab that allows users to compare time domain and frequency domain data to search for and identify relationships between different signals. The system identification window can be seen below in Figure 6-4.

![Matlab System Identification Tool](image)

Figure 6-4 - Matlab System Identification Tool
The toolbox compares an output signal to one or more input signals using mathematical models. The default configuration uses impulse, spectral, linear parametric and state space models to compare the data automatically while manual refinement is possible. Additional operations include removing trends, means or filtering the data for low and/or high frequencies.

In analyzing the rotary current and vibration signals a state space model was the most accurate. The state space model is a mathematical model of a physical system defined by a number of ‘states’ which represent a set of first-order equations that represent a higher order equation. The variables are vectors written in matrix form where the state-space is defined by a number of axes equal to the model order. The matrix can contain free variables which describe dynamics of the system. The system identification package used in Matlab® uses final prediction error as the method to judge the best model, of which the equations used are presented below:

\[ x(t + T_s) = Ax(t) + Bu(t) + Ke(t) \]

Equation 6-1 - State Space Model Equation 1

and

\[ y(t) = Cx(t) + Du(t) + e(t) \]

Equation 6-2 - State Space Model Equation 2

The solutions for the variables are available in the appendix.

6.3 Current and Current Ripple Prediction

While rotary current and vibration are connected, it is not known for certain which causes which; the vibration could result from the oscillating current, the current could be oscillating due to the vibration or both. Due to the vibration occurring at multiples of the rotation frequency, the state of the system is the cause, however whether one signal leads the other is
unknown. The signals were tested both as inputs and as outputs in the Matlab System Identification toolbox to compare the model prediction. The outputs can be seen in Figure 6-5.

![Figure 6-5 - Current and Vibration Prediction](image)

The actual output of each relationship as shown in the figure above can be seen as the thin black line, with the model output shown as the thick red line. The model does a very good job of prediction early in the validation window, but over time the accuracy falls. This decrease occurs because the geology does not remain constant. In a short amount of time the bit moves through the rock and the vibrations change, fractures wax or wane and prediction of drill behavior becomes impossible at this level.

The system identification toolbox provides the user with a correlation percentage to help the user better determine the best model. In the two cases above, both were equally accurate at predicting each other.

Both signals in Figure 6-5 were low pass filtered at 10 Hz to show frequencies present as a result of drilling induced vibration. As previously shown in chapter five, aliased signals are present in the 75 Hz and 175 Hz range of the fast Fourier transform of the vibration signal sampled at 400
Hz but do not contribute to the frequency range of interest and were therefore filtered out. The fast Fourier transform of the vibration and rotary current signals can be seen in Figure 6-6.

![Fast Fourier Transform](image)

Figure 6-6 - Matching Frequencies in Rotary Current and Vibration

Both the rotary current and vibration signals show distinct peaks at 3.615 Hz or 3 times the rotary speed of 72 RPM. Other scenarios, such as obstructions crossing a hole can generate strong vibrations at 6 times the rotary speed. Severely fractured ground can generate vibrations at various low frequencies that are not as easily correlated to rotary speed but are still felt strongly in the operator’s cab. Regardless of the geological cause, high vibration drilling situations were accompanied by an increase in rotary current values during the field test period as can be seen in the recorded data in Figure 6-7. The vibration data have been scaled up by a factor of 100 for clarity. As shown, with increased current, overall vibration increases as well.
The above figure is insufficient and too noisy to truly show the relationship between rotary current and vibration. The current signal oscillations match well with the vibration signal and warrant further review. As a predictor for vibration, the current signal itself is insufficient, however the amplitude of the current oscillations themselves provide a good basis for vibration prediction. Figure 6-8 below shows the RMS filtered current ripple against the RMS filtered vibration signal. Both signals were low pass filtered at 10 Hz.
The above figure shows a very good match in the latter half of the data, however the current ripple in the first half is much higher than acceptable to be used as a predictor of vibration.

In order to quantify the match between the RMS Current Ripple and the RMS Vibration signal, Figure 6-9 shows the correlation between the two as well as their $R^2$ values.
This relationship looks to be rather poor, however this covers the entire hole through 40 feet (12.2 m) of changing geology. By looking at a portion of the data we can see a better correlation as shown in Figure 6-10.
In Figure 6-10 a 100 second section of the hole is presented with a much more linear relationship between the current ripple and vibration signals. This 100 second section represents approximately 3 feet (0.9 m) of drilled depth.

Figure 6-7 suggests a relationship exists between the amplitude of the vibration signal and the rotary current. By looking at both the rotary current and the rotary current oscillation amplitude together, a much better correlation can be seen in Figure 6-11.

![RMS Current Ripple and RMS Current Signal vs. RMS Vibration](image)

*Figure 6-11 - RMS Current Ripple and RMS Current Signal vs. RMS Vibration*

As with the Current Ripple alone, the correlation between this combination signal and vibration can be seen in Figure 6-12.
The relationship between RMS vibration and the combined equation of RMS current ripple and RMS current signal shows a much closer relationship with an $R^2$ value of 0.5557 instead of 0.3083. Using the same 100 second section, Figure 6-13 shows an increase in the $R^2$ value as well from 0.7608 to 0.8157.
The relative weighting of the current ripple and the current signal to vibration is not constant.

The empirical relationship describing Figure 6-11 is as follows:

\[
RMS\ Vibration \cong K (RMS\ Rotary\ Current\ Ripple^{0.6})(RMS\ Rotary\ Current^{0.4})
\]

**Equation 6-3 - Current and Vibration Relationship**

Where:

\( K \) = Geologically dependant scaling factor

The formula is not generic because it leaves out critical geological information. As previously mentioned, the breakage mechanism for one rock type is not transferable to other rock types or even adjacent holes in some mines. The formula is shown as a first step towards predicting vibration using information direct from the bit-rock interface instead of ‘second-hand’ information from the vibration sensor located on the mast, several connections away from the...
drilling face. This formula was repeated on data from other areas of the mine both from a nearby hole as shown in Figure 6-14 as well as a separate bench as shown in Figure 6-17. The application of the same formula shows a weak connection in the nearby hole and virtually no significant connection in the hole from the separate bench. Figure 6-15, Figure 6-16, Figure 6-18 and Figure 6-19 show the correlation between the variables using Equation 6-3 as in the above examples. As can be seen, the relationship degrades as the holes are further away due to geological changes.

![Current Ripple Graph](image)

*Figure 6-14 - Nearby Hole using Current Ripple Formula*
Figure 6-17 - Distant Hole using Current Ripple Formula

Figure 6-18 - Distant Hole Correlation - Entire Hole
While the initial graph shows two well matched signals for a duration of over 20 minutes, this simple equation is less applicable as the geological conditions change. While an all-encompassing equation did not present itself during the analysis, the connection between the current signal and the vibration signal is significant. By understanding the actions of the rotary motor and the subsequent recorded vibrations, a clearer understanding of vibration in the drilling process can be gleaned.
Chapter 7: Conclusions and Recommendations for Future Work

7.1 Primary Contributions and Conclusions

This thesis investigated the monitoring and analysis of signals related to the vibration of large rotary electric blasthole drills used as primary production equipment in the surface mining process. The primary research contributions and conclusions were:

1. Acquisition of a comprehensive data set for investigation of blasthole drill vibration in a truly operational context (iron ore mine in Minnesota’s Mesabi Iron Range).
2. Investigation of ambient vibration sources and vibration aliasing issues (sections 4.4 and 4.5).
3. Development of signal analysis and smoothing techniques for calculation of specific energy (section 5.7).
4. Validation of predictions in the literature that the natural frequency of the drill mast decreases with increasing bit depth, and the observation that the mounting position of the accelerometers on the mast distorts this trend (section 4.1).
5. Demonstration that the pull down force (weight on bit) has no appreciable impact on the mast’s natural frequency (section 4.1)
6. Demonstration that the pull down force (weight on bit) has no appreciable impact on the mast’s damping ratio (section 4.2).
7. Demonstration of a strong relationship between rotary speed and the dominant vibration frequency peaks at 3x and 6x rotary speed (section 5.3), and postulation of a physical explanation of the 6x vibration peak (section 5.5).
8. Selection of rotary motor current for further investigation for use either as a substitute for accelerometer feedback, or as an auxiliary signal to detect down-the-hole vibration
when it is not manifested by the mast mounted accelerometers, based on the current
signal’s frequency peaks at 3x and 6x rotation speed (section 6.1).

9. Demonstration, through use of system identification, that the dynamic relationship
between vibration and rotary current, while it can be modelled locally, varies with depth
and geology and hence is essentially a time-varying process (section 6.3). This results in
the amplitude of rotary current not being usable as a proxy for vibration amplitude.

10. Demonstration that the root-mean-square (RMS) of the low frequency current
oscillations, in a nonlinear combination with the RMS of the current signal as a whole,
may be able to serve as a proxy for the RMS of the vibration signal (section 6.3).

7.2 Recommendations for Future Work

Various avenues of investigation could not be pursued due to a combination of logistical
limitations and the finite time allocated to a Masters level thesis. Some of these merit further
research:

1. More comprehensive structural response testing with additional accelerometers
installed on the mast, as well as, if possible, on the drill string.

2. Utilization of the existing data set to group together similar drilling conditions, perhaps
on the basis of specific energy, and then use this grouped data to see if stronger,
condition-specific, correlations between vibration and current-based variables could be
established.
3. Application of pattern recognition techniques, such as artificial neural networks, to explore whether they can establish more effective mappings between the current signal characteristics and the measured vibrations.

4. Comparison of how the detection of severe vibrations based on some current-based technique compares with conventional vibration-RMS feedback controllers used in commercial drill control systems.
References


Appendices

Appendix A - DATAQ DI-718Bx Specifications

Analog Inputs

Number of Channels: DI-718B: 8 configured for signal conditioned inputs
DI-718Bx: 16 configured for signal conditioned inputs
Channel Configuration: Defined by DI-8B Module
Input Impedance: Defined by DI-8B Module
Input offset voltage: Defined by DI-8B Module
Channel-to-Channel crosstalk rejection: -75db @ 100kHz unbalance
Offset temperature coefficient: 0.25μV/°C
Digital Filtering: PC-connected models: Conditional over-sampling; peak/valley detect, last point, average, frequency, RMS
Stand-alone models: None
Measurement Range: Defined by DI-8B module on a channel-by-channel basis.

Note: Not all DI-8B amplifier modules support ± excitation, but all support ± channel inputs.

Accuracy: ±0.25%FSR
Resolution: ±1 part in 8,192
Gain (DI-718Bx only): 1, 2, 4, 8 software selectable per channel.
CJC Error: ±1.5°C plus 8B Module (spec measured at 25°C with no air circulation and Low Current Module configuration only).

A/D Characteristics

Type: Successive Approximation
Resolution: 14-bit
Monotonicity: ±2 LSB
Conversion Time: 69.4μs
**Scanning Characteristics**

Maximum sample throughput rate: PC-connected: 4,800 Hz
Stand-alone: 14,400 Hz (assumes SD memory latencies of 80ms or less)
Minimum sample throughput rate: PC-connected: 0.0034 Hz
Stand-alone: 0.0017 Hz

Maximum scan list size: DI-718B: 9 entries
DI-718Bx: 18 entries
Sample buffer size: 2kb

**Digital I/O**

Bits: DI-718B: 2 Inputs (Remote Storage and Remote Event)
DI-718Bx: 8 bi-directional (including Remote Storage/Event)

Input voltage levels: Min. required “1” 2V; Max allowed “0” 0.8V

DI–718B and DI-718Bx Series Hardware Manual

**Calibration**

Calibration Cycle: One year

**Ethernet Interface** (optional USB to Ethernet converter available — part number 101014-EA)

Type: 10/100Base-T
Connector: RJ-45
Protocol: TCP/IP
Server Type: DHCP

**USB Interface (DI-718B Models only)**

Connector: USB
Protocol: USB 2.0

Removable Memory (Stand-alone Models only)
Type: Secure Digital (SD)
Capacity: 8MB to 1GB

Real Time Clock
Type: Date, hour, minute, second
Resolution: 1 second
Accuracy: 20 ppm

Controls (Stand-alone Models only)
Single push-button: Provides manual control over Record and Standby modes.

Indicators
Stand-alone models: Three color LED indicating Record, Standby, and Error conditions.
PC-connected models: Power LED

Transfer Rate to PC
Real Time: up to 4,800 samples per second
From MMC or SD Memory (Ethernet): up to 3,000 samples per second

General
Panel Indicators: Mode LED
Panel Controls: Control push-button (Stand-alone models)
Panel slots: Accepts MMC/SD-type flash memory
DI-718B and DI-718Bx Series Hardware Manual
Input connectors: DI-718B: Two, removable sixteen position terminal blocks
DI-781BX: Four, removable sixteen position terminal blocks for signalconditioned channels.

Operating Environment: 0°C to 70°C

Enclosure: Aluminum base with steel wrap-around. Aluminum end-panels with plastic bezels. Aluminum top hatch for access to 8B backplane.

Dimensions: DI-718B: 5.4D × 4.1W × 1.5H in. (13.81D × 10.48W × 3.81H cm.)
DI-718Bx: 7.29W × 9L × 1.52H in. (18.52W × 22.86L × 3.86H cm.)

Weight: DI-718B: 14 oz. (397 grams) + DI-8B Modules
DI-718Bx: 2 lbs. 10 oz. (1.19 kg) + DI-8B Modules

Power Requirements: USB: 9 to 36 VDC, 2 watts + 8B modules
Ethernet: 9 to 36 VDC, 2.5 watts + 8B modules
ACCELEROMETER, INDUSTRIAL, ICP®

<table>
<thead>
<tr>
<th>Mode Number</th>
<th>ECU # 1434</th>
</tr>
</thead>
<tbody>
<tr>
<td>279A41</td>
<td></td>
</tr>
</tbody>
</table>

**Performance**

- **ENGLISH**
  - Sensitivity: 100 mV/g
  - ±50 g
  - Frequency Range: 20 to 6000000 Hz
  - 0.53 to 10000 Hz
  - 100 Hz: 18 V/s
  - ±1 %
  - ±1 %
  - ±5 %

- **SI**
  - Sensitivity: 2.1 mV/g
  - ±50 g
  - Frequency Range: 20 to 6000000 Hz
  - 0.53 to 10000 Hz
  - 100 Hz: 18 V/s
  - ±1 %
  - ±1 %
  - ±5 %

**Environmental**

- Overload Limit (Shock): 5000 g pk
- Temperature Range: 40°C to 70°C
- Temperature Response: See Graph
- Enclosure Rating: IP68

**Electrical**

- Setting Time (within 1%): ≤10 sec
- Discharge Time Constant: >0.5 sec
- Excitation Voltage: 10 to 20 VDC
- Constant Current Excitation: 2 to 20 mA
- Output Impedance: <1000 ohm
- Output Bias Voltage: 8 to 12 VDC
- Spectral Noise (10 Hz): 50 μV/Hz
- Spectral Noise (100 Hz): 25 μV/Hz
- Spectral Noise (1 kHz): 6 μV/Hz
- Electrical Isolation (Case/Case): >107 ohm
- Electrical Protection: RFI/EMI

**Physical**

- Size (Length x Width): 136 mm x 22.5 mm
- Weight: 3.3 oz
- Mounting Thread: 3/8-24 Female
- Mounting Torque: 2.7 to 6.6 N-m
- Sensing Element: Quartz
- Sensing Geometry: Shear
- Housing Material: Stainless Steel
- Sealing: Welded Hermetic
- Electrical Connector: 2-pin MIL-C-5015
- Electrical Connection Position: Top

**Optional Versions**

- Standard model: Model 279A41 Mounting Stud, 1/4-20 to M8 x 1 replaces Model 279A40

**Notes**

- 1) Typical
- 2) Conversion Factor: 1g = 9.81 m/s²
- 3) The high frequency tolerance is ±10% of the specified frequency.
- 4) Zero-based, least squares, straight line method.
- 5) 1/4-20 has no equivalent in S.I. units.
- 6) See PCB Declaration of Conformance PS223 for details.

**Supplied Accessories**

- 081440 Mounting Stud (1)

---

All specifications are at room temperature unless otherwise specified.

In the interest of constant product improvement, we reserve the right to change specifications without notice.

ICP® is a registered trademark of PCB group, Inc.
Appendix C - Drill Specifications
**P&H 120A**

**ROTARY BLASTHOLE DRILL**

**OPERATING SPECIFICATIONS**

**WORKING RANGES**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
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<tbody>
<tr>
<td>Drill Hole Diameter Range</td>
<td>To 22 in.</td>
</tr>
<tr>
<td></td>
<td>559 mm</td>
</tr>
<tr>
<td>Maximum Single Pass</td>
<td>65 ft. 0 in.</td>
</tr>
<tr>
<td>Standard Hole Depth</td>
<td>19.81 m</td>
</tr>
<tr>
<td>Maximum Bit Loading</td>
<td>150,000 lbs.</td>
</tr>
<tr>
<td></td>
<td>68,038 kg</td>
</tr>
</tbody>
</table>
### MAST

<table>
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<tr>
<th>Construction:</th>
<th>Lattice type using alloy steel structural shapes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Single Pass:</td>
<td>65 ft. 0 in. (19.81 m), 70 ft. (21.3 m) optional</td>
</tr>
<tr>
<td>Drilling Depth:</td>
<td>Two hydraulic cylinders, 10.5 in. (267 mm) diameter (each)</td>
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### HOIST / PULLDOWN

<table>
<thead>
<tr>
<th>Design:</th>
<th>DC electric motor driven chainless rack and pinion design</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bit Loading (maximum):</td>
<td>150,000 lbs. (68,038 kg)</td>
</tr>
<tr>
<td>Feed Rate:</td>
<td>To 80 fpm (24.4 m/min)</td>
</tr>
<tr>
<td>Hoist Rate:</td>
<td>To 80 fpm (24.4 m/min)</td>
</tr>
<tr>
<td>Auxiliary Winch:</td>
<td>Optional, 10,000 lbs. (4,536 kg) capacity</td>
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### PIPE HANDLING

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<thead>
<tr>
<th>Type:</th>
<th>Parallelogram style pipe rack, one rack is standard</th>
</tr>
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<tbody>
<tr>
<td>Pipe Size:</td>
<td>5 5/8 in. - 16 in. (219 mm - 407 mm) diameter</td>
</tr>
<tr>
<td>Options:</td>
<td>Additional pipe racks up to four total</td>
</tr>
<tr>
<td>Auxiliary Equipment:</td>
<td>Standard deck wrench</td>
</tr>
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</table>

### ROTARY MACHINERY

<table>
<thead>
<tr>
<th>Design:</th>
<th>Dual DC electric motor drive</th>
</tr>
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<tbody>
<tr>
<td>Rotation Speed:</td>
<td>0-119 RPM standard, 0-101 RPM and 0-138 RPM optional</td>
</tr>
<tr>
<td>Maximum Torque:</td>
<td>Up to 25,000 ft-lbs. (33,895 NM) for standard gear ratio, up to 22,000 ft-lbs. (29,830 NM) and 30,000 ft-lbs. (40,875 NM) for optional gear ratios</td>
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### ELECTRICAL CONTROL SYSTEMS

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<tr>
<th>Control System:</th>
<th>Allen Bradley PLC 5/40 - SLC Remote I/O Ladder Logic-Based</th>
</tr>
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<tbody>
<tr>
<td>GUI (Graphical User Interface):</td>
<td>15&quot; Touch screen including operating parameters</td>
</tr>
<tr>
<td>Standard:</td>
<td>15&quot; Advanced Diagnostics/Troubleshooting</td>
</tr>
<tr>
<td>Optional:</td>
<td>Standard and optional operator error protection systems available (over-temperature shut-downs, over-tilt protection, pipe rack interlock protection, pipe protection software, etc.)</td>
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### GENERAL DIMENSIONS

<table>
<thead>
<tr>
<th></th>
<th>Description</th>
<th>Value</th>
<th>Unit</th>
</tr>
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<tbody>
<tr>
<td>A</td>
<td>Width, overall</td>
<td>21 ft. 10 in.</td>
<td>6.66 m</td>
</tr>
<tr>
<td>B</td>
<td>Height, mast up</td>
<td>93 ft. 0 in.</td>
<td>28.35 m</td>
</tr>
<tr>
<td>C</td>
<td>Height, mast down</td>
<td>20 ft. 0 in.</td>
<td>6.10 m</td>
</tr>
<tr>
<td>D</td>
<td>Length, mast up</td>
<td>44 ft. 0 in.</td>
<td>13.41 m</td>
</tr>
<tr>
<td>E</td>
<td>Length, mast down</td>
<td>94 ft. 7 in.</td>
<td>28.82 m</td>
</tr>
<tr>
<td>F</td>
<td>Overall width of crawlers</td>
<td>19 ft. 6 in.</td>
<td>5.94 m</td>
</tr>
<tr>
<td>G</td>
<td>Overall length of crawlers</td>
<td>24 ft. 2 in.</td>
<td>7.36 m</td>
</tr>
<tr>
<td>H</td>
<td>Width of jacks</td>
<td>14 ft. 6 in.</td>
<td>4.42 m</td>
</tr>
<tr>
<td>I</td>
<td>Length between jacks</td>
<td>33 ft. 9 in.</td>
<td>10.29 m</td>
</tr>
<tr>
<td>J</td>
<td>Height to top of op. cab</td>
<td>14 ft. 0 in.</td>
<td>4.27 m</td>
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### ELECTRICAL SYSTEM

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<tr>
<th>Description</th>
<th>Specification</th>
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<tbody>
<tr>
<td>Incoming Power Supply Voltage</td>
<td>To 7200 V, 3 phase, 60 Hz or 6,600 V, 3 phase 50 Hz</td>
</tr>
<tr>
<td>Recommended Supply Transformer</td>
<td>1 MVA Continuous, 5 MVA Short Circuit</td>
</tr>
<tr>
<td>High Voltage Switchgear</td>
<td>External disconnect at trail cable junction box, high voltage load break disconnect and high voltage vacuum contactor for main drive motor and drives</td>
</tr>
<tr>
<td>Main Drive Motor</td>
<td>700 HP (522 kw)</td>
</tr>
<tr>
<td>Compressor Oil Cooler Fan</td>
<td>30 HP (22.4 kw)</td>
</tr>
<tr>
<td>Machinery House Fans</td>
<td>2 x 7.5 HP (5.6 kw)</td>
</tr>
<tr>
<td>Hoist and Rotary Blowers</td>
<td>2 x 1 HP (.75 kw)</td>
</tr>
<tr>
<td>Auxiliary Hydraulic Pump</td>
<td>1.5 HP (1.1 kw)</td>
</tr>
<tr>
<td>Oil Circulation Pump</td>
<td>10 HP (7.5 kw)</td>
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### LOWER WORKS

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<th>Description</th>
<th>Specification</th>
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<tbody>
<tr>
<td>Crawler Type</td>
<td>Heavy-duty lug and tumbler drive system, P&amp;H design</td>
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<tr>
<td>Shoe Width</td>
<td>36 in. (914 mm)</td>
</tr>
<tr>
<td>Standard</td>
<td>44 in. (1,118 mm), 54 in. (1,372 mm)</td>
</tr>
<tr>
<td>Optional</td>
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<tr>
<td>Propel Machinery</td>
<td>Dual hydrostatic planetary drive with spring set, hydraulic release brake, 310 HP (231 kw)</td>
</tr>
<tr>
<td>Propel Speed (maximum)</td>
<td>1.0 mph (1.61 kph) High</td>
</tr>
<tr>
<td></td>
<td>.6 mph (.97 kph) Low</td>
</tr>
<tr>
<td>Gradeability</td>
<td>60%</td>
</tr>
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<td>Take-Up Adjustment</td>
<td>Jack and shim</td>
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### WEIGHTS - Approximate

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<th>Description</th>
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<tr>
<td>Operating Weight (maximum)</td>
<td>365,000 lbs. (165,564 kg)</td>
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<tr>
<td>Shipping Weight with Mast</td>
<td>359,400 lbs. (153,849 kg)</td>
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<tr>
<td>Ground Bearing Pressure for Track Pads:</td>
<td>21.0 psi (36” shoes), 17.2 psi (44” shoes), 14.0 psi (36” shoes)</td>
</tr>
<tr>
<td>Ground Bearing Pressure for Jacks:</td>
<td>129 psi standard size, 46 psi optional size</td>
</tr>
</tbody>
</table>

### LIGHTING

<table>
<thead>
<tr>
<th>Description</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Standard</td>
<td>Quartz halogen floodlights for work area supplemented by incandescent and fluorescent area and interior lighting</td>
</tr>
<tr>
<td>Optional</td>
<td>High pressure sodium floodlights</td>
</tr>
</tbody>
</table>
# AIR SYSTEM

<table>
<thead>
<tr>
<th>Compressor:</th>
<th>Gardner-Denver SSY Series oil-flooded rotary screw type</th>
</tr>
</thead>
<tbody>
<tr>
<td>Standard Volume:</td>
<td>3,000 cfm (85 cm³/min)</td>
</tr>
<tr>
<td>Optional Volume:</td>
<td>2,500 cfm (85 cm³/min), 3,600 cfm (102 cm³/min)</td>
</tr>
<tr>
<td>Pressure:</td>
<td>65 psi (448 kpa)</td>
</tr>
<tr>
<td>Air Filters:</td>
<td>Dual Donaldson 2-stage, dry type</td>
</tr>
</tbody>
</table>

# HYDRAULIC SYSTEM

<table>
<thead>
<tr>
<th>Main System:</th>
<th>Closed loop design utilizing two variable-displacement piston pumps for propel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Auxiliary System:</td>
<td>Open loop design utilizing vane pumps for mast raising, machine leveling, and pipe handling</td>
</tr>
<tr>
<td>Control Valves:</td>
<td>Manifold mounted electro-hydraulic, PLC controlled - PLC 5/40 Discrete 110V DC Solenoid</td>
</tr>
<tr>
<td>Hydraulic Lines:</td>
<td>Extensive use of high pressure steel tubing</td>
</tr>
<tr>
<td>Filtration:</td>
<td>3-micron return filters, 3-micron charge filters, suction strainers</td>
</tr>
</tbody>
</table>

# LEVELING JACKS

<table>
<thead>
<tr>
<th>Cylinders:</th>
<th>Four (4) 9 in. (229 mm) diameter x 66 in. (1,676 mm) stroke</th>
</tr>
</thead>
<tbody>
<tr>
<td>Jack Pads:</td>
<td>Two sizes available</td>
</tr>
<tr>
<td>Standard:</td>
<td>30 in. (762 mm) diameter</td>
</tr>
<tr>
<td>Optional:</td>
<td>50 in. (1,270 mm) diameter</td>
</tr>
<tr>
<td>Auto Level:</td>
<td>Optional feature</td>
</tr>
</tbody>
</table>

# OPERATOR'S CAB

<table>
<thead>
<tr>
<th>Type:</th>
<th>Rear mounted with vibration and noise suppression</th>
</tr>
</thead>
<tbody>
<tr>
<td>Controls:</td>
<td>PLC controlled, backlit for night operation</td>
</tr>
<tr>
<td>Glazing:</td>
<td>Tinted shatter-resistant, laminated glass on all sides, roof window with guard</td>
</tr>
<tr>
<td>Climate Control:</td>
<td>Mine Air Systems or Sigma HVAC unit available, provide pressurization and filtration</td>
</tr>
</tbody>
</table>
MACHINERY DECK PLAN

ADDITIONAL OPTIONAL EQUIPMENT

- 70 ft. (21.3 m) Single Pass
- Dust Suppression (dry or water)
- Dust Curtains
- Cold Weather Package
- Sure Wrench® Breakout Wrench
- Centralized Lubrication
- Remote Propel Controls
- Wiggins Remote 'Fastlift' System

- GPS Interface
- Shock Subadapter
- Deck Bushing (roller type)
- Mast Ladder with Saf-T-Climb®
- Fire Suppression
- Bit Lube
- Cab Window Shades
- Cable Reel

NOTE: All designs, specifications and components of equipment described above are subject to change at manufacturer's sole discretion at any time without advance notice. Data published herein is informational in nature and shall not be construed to warrant suitability of machine for any particular purpose as performance may vary with conditions encountered. The only warranty applicable is our standard written warranty for this machine.

P&H MINING EQUIPMENT
A Joy Global Inc. Company

P&H Mining Equipment • Box 310 • Milwaukee, Wisconsin 53201 USA

XS-1299-3
.SAG-006
Appendix D – Matlab Code for RMS Filtering

% RMS filter with d.c. bias averaged over a 20 point window
% iteratively recalculated at each sample time
% assumes variable x contains the raw vibration signal

window= 400;
% number of points in window for calculation of d.c. bias

len=n;
% this first bit just dimensions and zeros the vectors used to store the
% calculated bias and RMS signal

bb=A(:,15); % this is the vibration channel we're interested in.
ff=zeros(len,1);
for i=1:len
    bb(i)=0;
    ff(i)=0;
end

initial_bias=mean(A(1:window,15));
% wait one second and then calculate first bias

for i=1:window
    bb(i)=initial_bias;
    ff(window)= ff(window)+(A(i,15)-bb(i))^2;
% This is building up the average which is divided below
end

ff(window)= sqrt( ff(window)/window );
% The average signal is divided and ready for the bulk of the data.

for nn=(window+1):len;
% calculate it for the total number of samples in the test data set
% these next two lines are what needs to run in Control Builder at
% every sample time

    bb(nn) = bb(nn-1) + ( (A(nn,15) - A((nn-window),15) )/window);
    ff(nn) = sqrt( (window*(ff(nn-1)^2)) -((A((nn-window),15) - bb(nn-window))^2) + ((A(nn,15) - bb(nn))^2) )/window );

% as you can see it's not very computationally heavy
% just two additions, three subtractions, two divisions, three
% squares,
% and one square-root you could implement it with a buffer of length
% (window+1)
end

A(:,15)=ff; % This puts the value back into the correct channel after
D.C. bias removal and moving average window RMS calculation.
Appendix E – System Identification Formulae

Current Predicting Vibration

State-space model:

\[ x(t+T_s) = A x(t) + B u(t) + K e(t) \]

\[ y(t) = C x(t) + D u(t) + e(t) \]

\[ A = \begin{bmatrix}
  x1 & x2 & x3 & x4 & x5 & x6 & x7 \\
  x1 & 1.005100 & 0.092049 & 0.003962 & 0.000325 & -0.000126 & -0.000071 & -0.000070 \\
  x2 & -0.098432 & 0.989060 & 0.089310 & 0.007775 & -0.000893 & 0.000696 & 0.002631 \\
  x3 & 0.015384 & -0.090642 & 0.974370 & 0.154700 & 0.094227 & -0.056812 & -0.074420 \\
  x4 & -0.037973 & -0.006701 & -0.173750 & 0.906790 & 0.142400 & 0.025247 & 0.257040 \\
  x5 & 0.019324 & -0.018876 & -0.012811 & -0.314430 & 0.145880 & -0.674820 & 0.103050 \\
  x6 & -0.014209 & -0.013889 & -0.008063 & -0.045026 & 0.179120 & 0.718120 & 0.882890 \\
  x7 & -0.002486 & -0.001646 & -0.010295 & 0.047881 & -0.138800 & -0.384030 & 0.882500 \\
\end{bmatrix} \]

\[ B = \begin{bmatrix}
  u1 \\
  x1 & 0.21711 \\
  x2 & -2.59720 \\
  x3 & 7.67880 \\
  x4 & 2.36770 \\
  x5 & 89.52100 \\
  x6 & 26.47100 \\
  x7 & 60.69300 \\
\end{bmatrix} \]

\[ C = \begin{bmatrix}
  x1 & x2 & x3 & x4 & x5 & x6 & x7 \\
  y1 & -1.143700 & -0.046481 & -0.002026 & -0.000171 & -0.000040 & 0.000021 & 0.000008 \\
\end{bmatrix} \]

\[ D = \begin{bmatrix}
  u1 \\
  y1 & 0 \\
\end{bmatrix} \]
$K = \begin{bmatrix}
  \text{y1} \\
  x1 & -1.709 \\
  x2 & -26.668 \\
  x3 & -267.010 \\
  x4 & -643.650 \\
  x5 & 269.320 \\
  x6 & 72.303 \\
  x7 & 121.160 
\end{bmatrix}$

$x(0) = \begin{bmatrix}
  x1 & -12.25 \\
  x2 & 432.18 \\
  x3 & -2714.30 \\
  x4 & -8571.00 \\
  x5 & 6955.30 \\
  x6 & -22949.00 \\
  x7 & -5052.60 
\end{bmatrix}$

Estimated using N4SID from data set CurrPredVib

Loss function 0.00117064 and FPE 0.00125545

Sampling interval: 0.0025

**Vibration Predicting Current**

State-space model:

$x(t+Ts) = A \, x(t) + B \, u(t) + K \, e(t)$

$y(t) = C \, x(t) + D \, u(t) + e(t)$

$A = \begin{bmatrix}
  x1 & x2 & x3 & x4 & x5 \\
  x1 & 1.001200 & -0.011495 & -0.006284 & 0.025056 & -0.001089 \\
  x2 & 0.019933 & 1.001700 & 0.071944 & 0.012680 & -0.000550 \\
  x3 & -0.003470 & -0.056972 & 0.945660 & 0.082116 & -0.003299 \\
  x4 & -0.009830 & -0.004934 & -0.030697 & 0.983140 & -0.103440 \\
  x5 & -0.010154 & 0.023771 & -0.069943 & 0.161060 & 0.931120 
\end{bmatrix}$

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\[ B = \begin{array}{c|c}
  & u1 \\
\hline
 x1 & -0.002773 \\
 x2 & 0.008136 \\
 x3 & -0.014079 \\
 x4 & 0.031257 \\
 x5 & 0.014565 \\
\end{array} \]

\[ C = \begin{array}{c|ccccc}
  & x1 & x2 & x3 & x4 & x5 \\
\hline
 y1 & 3787.70000 & 488.06000 & 1079.60000 & 2.75000 & 0.07811 \\
\end{array} \]

\[ D = \begin{array}{c|c}
  & u1 \\
\hline
 y1 & 0 \\
\end{array} \]

\[ K = \begin{array}{c|c}
  & y1 \\
\hline
 x1 & -0.14132 \\
 x2 & -0.11888 \\
 x3 & -0.43512 \\
 x4 & -2.07030 \\
 x5 & 3.95360 \\
\end{array} \]

\[ x(0) = \begin{array}{c|c}
  & x1 \\
\hline
 x1 & -0.00400 \\
 x2 & -0.00118 \\
 x3 & -0.01416 \\
 x4 & 0.20913 \\
 x5 & 1.77470 \\
\end{array} \]

Estimated using N4SID from data set mydatafe
Loss function 8.47814e-006 and FPE 8.91236e-006
Sampling interval: 0.0025