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Rolling Element Bearing Fault Diagnostic System

Defining futures

COLLEGE OF
ELECTRICAL AND MECHANICAL ENGINEERING
NATIONAL UNIVERSITY OF SCIENCES AND TECHNOLOGY
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College of Electrical & Mechanical Engineering

THESIS REPORT

ROLLING ELEMENT BEARING FAULT DIAGNOSTIC SYSTEM

Submitted to the Department of Mechatronics Engineering
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ABSTRACT

Machine condition monitoring (MCM) is always been challenging and most important task for engineers / scientist. As there are some parameters from which medical doctor can easily tells about the health of human being and cure, similarly engineers / scientist are more interested in the health of machine. They want that before any major break down they can easily detect and predict the fault in machine.

There are different major components and parts of machine like gears, piston, shaft, pulley etc. and defect in one of them can create catastrophic failure and result in major down time. Among all the above component one of the most critical component is bearing. Bearings are the most common components and are used in almost every machine.

The purpose of this thesis was to create a software and methodology to easily detect the faults in rolling element bearing before complete breakdown. The software must be able to detect the fundamental faults of rolling element bearing.

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CHAPTER 1

INTRODUCTION

1 Introduction

Most common components of bearings are inner race and outer race. The bearing damage to produce a series of impact vibration when the ball surface defects are generated durring running. The belongings of all these vibrations directly effects on the storage of characteristic frequencies (BCF), which are calculated on the basis of the speed of the wave, the geometry of the bearing and the location of the fault reporting. Due to the reason of impact which is generated by a bearing fault distributes its energy over wide frequency range thus the BCF has relatively low energy, it is often overwhelmed by noise and vibrations generated from other macro fault of structural components.

To allow for, the easier detection of such faults, the Envelope Detection (ED) technique has been used .The impact vibrations are difficult to be identified in low frequency range due to their low energy and interference. The modulated amplitudes of repetitive impacts are often excited at the bearing structural resonance frequency. Hence, the amplitude demodulation provided by ED allows the detection of localized detects. Due to the inability of FFT to detect faults, which exhibit non-stationary impact signals, there is a need to seek for other alternatives.

A GUI in MATLAB is developed as diagnostic software. The main aim was to provide such a software that even a lay man who do not now about FFT, ED and side bands concept can easily detect fault. Software takes input of bearing geometry, raw data signal from accelerometer and automatically displays which fault bearing contains

1.1 Motivation for Project

Machine condition monitoring is always an interest for me for long time. This interest developed in me first time when I first worked in cement manufacturing plant as internee in my degree program. Where I observed huge induction motors, pumps, conveyers having large bearing and revolving with high speed. The first thought came in my mind that is there any methodology to observe the functionality and viability of bearings? Then after completing my graduation I worked as Assistant Manager in Pet bottle manufacturing plant, there too I observe and thought is there can be any easy method to check the condition of bearing. At that time we were using a method like stethoscope (made of SKF) and used to listen the sound of rotating bearing. This method was cumbersome and need expertise in order to detect the bearing faults. Further it was also reviewed through literature that

- A $5,000 wind turbine bearing replacement can easily turn into a $250,000 project, not to mention the cost of downtime

- In 1987, LOT Polish Airlines Flight 5055 Il-62M crashed because of failed bearings in one engine, killing all the183 people on the plane

After reading different research papers and literature which has horizon my scope that this can be made in software using vibration techniques. So I selected MATLAB as a tool for developing this software for detection of rolling element bearing fault.

1.2 Aim and Objective

While selecting and developing this project as thesis the main aim was to present a software for a lay man, who can easily detect the bearing fault having no knowledge of bearing and signal processing techniques. A person who does not know the basics of FFT, side bands, envelop detection techniques can also find bearing faults.

1.3 Apparatus and Data Sets

In order to check the performance of software data sets of reliable machine were required. This must be a vibration data signal from accelerometer of rolling element bearing rotating at different RPM. Since the facility of data acquisition was not available so I searched Website of Case Western Reserve University. Where found all the data sets of my need.

The test stand consists of 2 hp induction motor; the test bearing supported the motor shaft. The accelerometer was mounted on bearing casing vertically. The data set were collected at 12000 samples per second for drive end bearing. Figure below shows the induction motor.

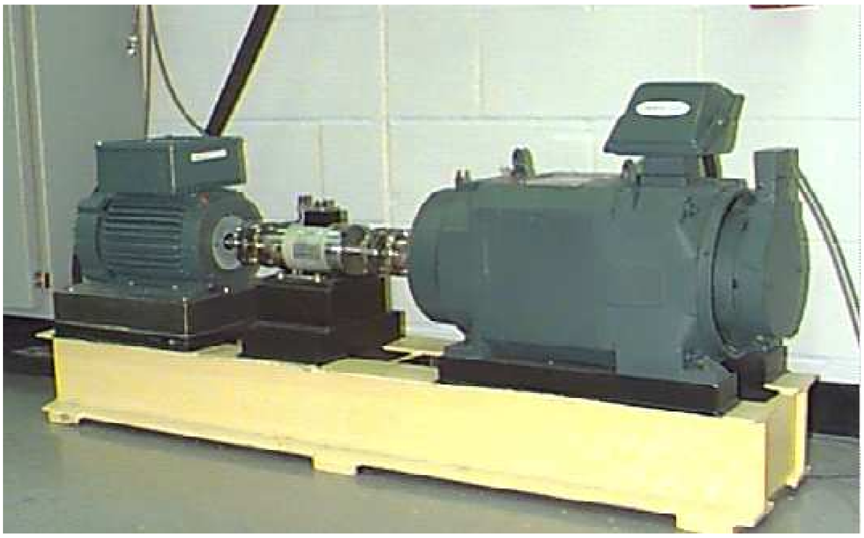


Figure 1: 2HP Induction Motor

Drive end bearing was deep grove ball bearing of type 6205-2RS JEM SKF. Defects in bearing were introduced on outer and inner race by electro discharge machining using 21 mils or 0.533 mm (1 mil = 0.001 inches).

1.4 Organization of Project

The thesis is organized as follows:

Chapter 2 describes the types of bearing and their defects.

Chapter 3 Literature mainly focuses on Data acquisition techniques.

Chapter 4 provides the details of Signal processing techniques on vibration raw data.

Chapter 5 covers the results and discussions of the software which was developed and tested.

CHAPTER 2

BEARINGS

2 Types of Bearings

There are two main types of bearings:

- Journal bearings (Sleeve bearings)
- Rolling element bearings (anti-friction bearings)

2.1 Journal Bearing

Journal or sleeve bearings have no moving parts and are normally designed to enclose the rotating shaft to provide support and stability. Rotating element are not used in journal bearings. Instead, the shaft rides on a layer of lubricating oil inside the bearing journal. A journal bearing consists of two basic parts which are journal and the bearing as illustrated in figure. The journal section is a portion of shaft which transfers the radial load to the bearing, acting as support. The journal normally rotates.

Figure 2: Journal Bearing

Fluid lubrication in radial load journal bearings depends largely on the viscosity of the lube and its adhesion to the surface of journal and the bearing. The radial clearance provided in the journal bearing allows a wedge-shaped film to form between the journal and bearing. The lubricant is dragged in to clearance space by rotation of the journal.

Journal Bearings have had a long history of application in high speed rotating machinery, such as turbines. Compared with rolling element bearings, journal bearing have the following advantages:

- High Surface speed capability
- High reliability

- Low maintenance requirements
- Simplicity of detailed configuration

Pressure fed bearing is used for heavily loaded shafts that starts and stop often. To prevent the shaft rubbing against the bearing material, lubricant is introduced under relatively high pressure in order to hydro statistically float the shaft before start-up. The is put under pressure 20 min to 1 hour before start up to insure that shaft is floating on the oil. Figure 3 illustrates the shaft rotating in the centre of journal bearing.

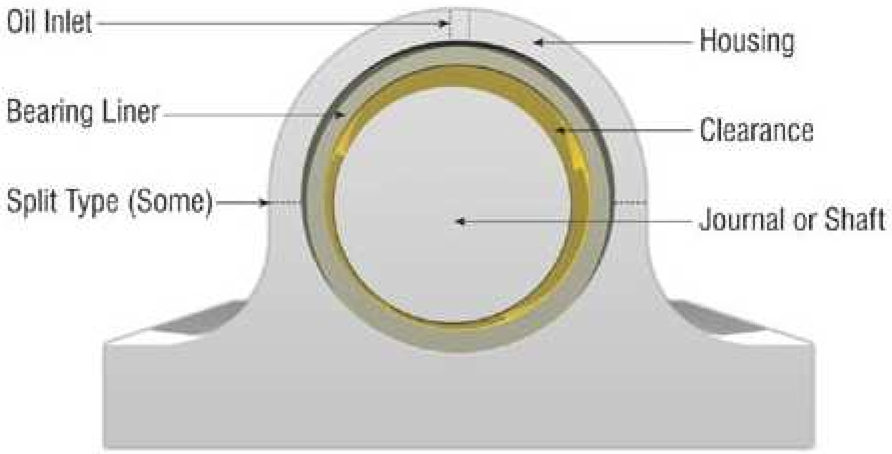


Figure 3: Shaft Riding on Lubricant in Journal Bearing

Journal bearing clearance should never be less than 2 mils, and should be at least 1 to 1.5 mils for every inch of shaft diameter. For example, a 6 inch diameter shaft should have a clearance of 6 to 9 mils. The clearance has an effect on the system's vibration, since damping increases as clearance decreases.

Journal bearing have high starting friction and low running friction. Figure 4 shows a plot of friction vs. RPM. At low RPM friction between shaft and the oil is quite high in the area called boundary layer. The friction begins to decrease as the shaft frequency passes through the mixed lubrication area. When the shaft frequency is in the fluid film area, friction is at minimum.

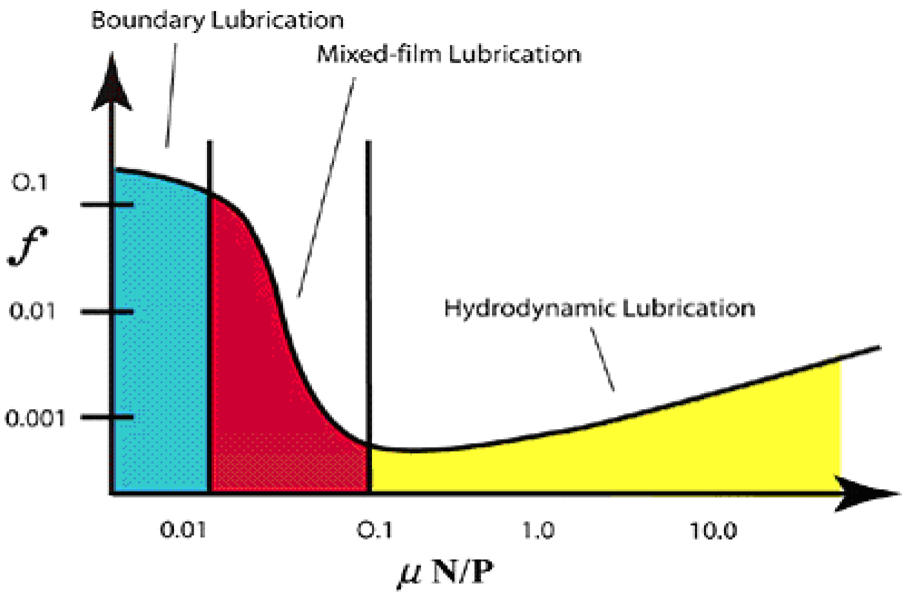


Figure 4: Friction Vs RPM for Journal Bearing

Journal bearing should always operate in the fluid film area, when the shaft is at operating conditions. Three major factors determine the threshold of the fluid film layer.

1. Frequency of shaft
2. Viscosity of lubricants
3. Load on bearing

The rotational velocity of the lubricants next to the shaft will be equalled to that of the shaft, whereas the rotational velocity of the lubricant next to the bearing surface will be zero. If the shaft were perfectly cantered in the bearing, and the friction losses were ignored, the average rotational velocity of lubricants would be 50% of shaft frequency. However, frequency runs somewhere between 35 to 49 % of the shaft frequency due to friction losses and shaft eccentricity.

2.2 Vibration from Journal Bearing

Vibration due to faulty journal bearing is generally the result of the following conditions:

- Excessive bearing clearances
- Oil whirl
- Lubrication problems

2.2.1 Excessive Bearing Clearance

A journal bearing with excessive clearance may cause some misalignment or other sources of vibration force resulting in conditions such as mechanical looseness or bearing pounding. Correct journal bearing clearances would make for less amplitude of vibration. Excessive bearing clearance will also result in poor load distribution within the bearing, decreased bearing life, and excessive shaft deflection.

Insufficient bearing clearances may produce excessive bearing and lube temperatures, increased journal and bearing wear, constricted flow of lubricant and eventual bearing failure and finally bearing seizure.

There are two types of clearance: initial clearance and running clearance, the initial clearance may come from the manufacturer and can be altered by shimming the split bearing housing, which changes the bearing diameter. The running clearance is the clearance remaining in the journal bearing during normal operation. Running clearance requirements may vary depending on load, speed, temperature, application and type of bearing material.

2.2.2 Oil Whirl

Swirl of oil is used at relatively high speeds, another common problem with bearings on machines with pressure lubrication systems operating.

Rotate as a lubricant to 50% of the rotational speed to pass through the narrow zone, which is pressed on the next shaft bearings. The average velocity of the lubricant accelerates interior and slows as it leaves away. This downward acceleration, and generates turbulence on both sides of the slot, and a vortex is formed in the region of the high-pressure lubricant.

The shaft in this fluidized oil like a surfboard and rides on the surface of the shaft. Called fluidized oil, which drives the shaft, is unstable to a little less than half the speed. The oil is proportional to the vortex frequency of the shaft and to be removed when the shaft RPM below the instability threshold.

Vortex of oil approaches the critical first tree as the tree approaches its second criticism, creating a resonance condition called whisk in oil. The frequency of vortex oil remains constant during the review of the first shaft and is removed when the frequency of the wave can be below the second critical. Both phenomena can be severe and in penetration of the lubricating film. When this happens, the impact of the tree can cause serious damage and against the pin.

One solution is to change the loading, lubricant, temperature or space, so that the tree does not cause the instability and the second way is to use tilting pad bearing.

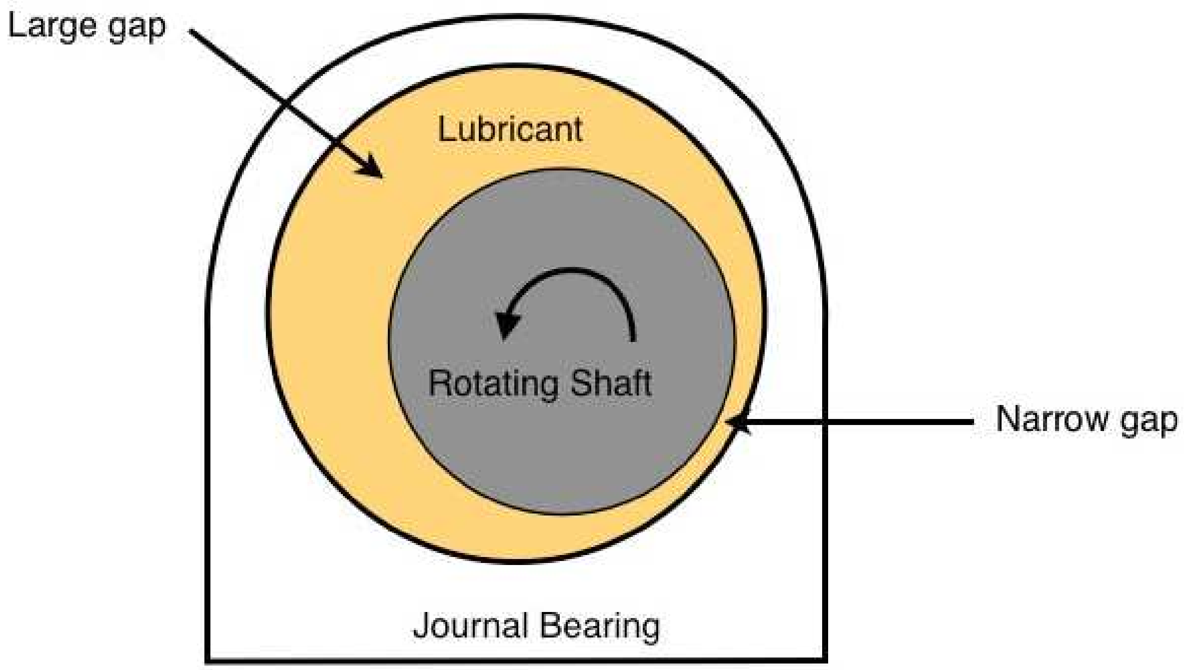


Figure 5: Shaft off-Center in Journal Bearing

2.2.3 Improper Lubrication

Improper lubrication, lack of lubrication, or the wrong type of lubrication, can cause vibration problems in journal bearing.

Improper lubrication usually promotes excessive friction between the bearing surface and the shaft journal area and the friction excites vibrations in the bearing and other part of the machines. Vibration, due to improper lubrication, where the excessive friction occurs, is called “dry whip”. Damage to the bearing and journal will occur almost immediately.

The frequency of vibration due to dry whip is usually high, often producing high pitched squealing sounds normally associated with insufficiently lubricated bearings. It is not associated with any multiples of shaft RPM. If dry whip is suspected as the cause of vibration, inspect the lubricant for quality and type, check the journal bearing clearance and compare to specifications.

2.3 Rolling Element Bearing

Rolling element bearings are used in many classes of rotational machinery where compact, high load capacity rotor support systems are required. They have low starting friction and high running friction. There is a frequency barrier which seriously limits the use of rolling element bearings in high speed applications. For conservative designs and long life, the shaft speed of grease lubricated anti friction bearing should not exceed 7200 RPM divided by shaft diameter. For example, a 6 inch diameter shaft mounted in grease lubricated anti friction bearing should not be run at a frequency higher than 1200 RPM. These numbers are conservative, but they should be followed if bearing life is critical in the production process.

There are two major classes of rolling element bearings: those with spherical balls and those with tapered or cylindrical rollers as illustrated in Figure. Tapered roller and spherical ball bearings sustain both radial and axial loads, while cylindrical roller bearing sustains only radial loads.

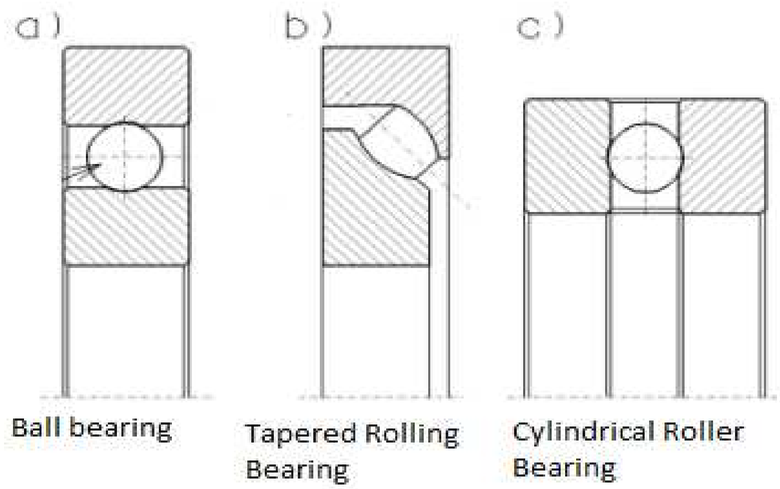


Figure 6: Rolling Element Bearings

The unique advantage of rolling element bearings is the absence of any destabilizing forces in the bearings such as those which induce oil whip in rotors supported by hydrodynamic bearings.

However, there are two major disadvantages of rolling element bearings are as follows:

1. In rolling element bearing there is always rolling contact, and there are some aspects of rubbing or sliding contacts between rollers and the stator and rotor races and the cage, with life limiting consequences. In the fluid film bearings, there is no metal to metal contact thus has no life-limiting wear.
2. Rolling element bearing have essentially no intrinsic internal damping, and where damping is necessary to good rotor dynamic operation, some damping elements must designed into apparatus. Fluid film bearing have, intrinsic to their operation. Significant internal damping that serves in good stead for limiting critical speed peak amplitude and for delaying the onset of any unstable whirling.

2.4 Vibration from faulty Rolling Element Bearing

Since no bearing is perfect, all rolling element bearings will have faults such as irregularities in the surface of race and in the roundness of the balls. When a bearing spins, these faults create periodic frequencies called fundamental defect frequencies.

There are four fundamental defect frequencies in rolling element bearings.

FTF-Fundamental Train Frequency. This is rotational speed of the bearing cage and ball/roller assembly.

BPFI-Ball Pass Frequency of the Inner Race. This is the frequency created as all the balls roll across a fault in the inner race.

BPFO-Ball Pass Frequency of the Outer Race. This is the frequency created as all the balls roll across a fault in outer race.

BSF-Ball/Roller Spin Frequency. This is the circular frequency of each ball/roller as its spins during revolution around shaft. For a roller bearing, two times the ball/roller spin frequency

2xBSF is a more appropriate parameter. This is a frequency at which a single defect on a roller contacts the inner and outer race of bearing.

Fundamental defect frequency depends on both the bearing geometry and the shaft speed. If you know what type of bearing is in the machine, you can use the manufacturer data to find the ball diameter, pitch diameter, number of rolling element, and the angle between the surfaces of each rolling element bearing.

There are situation where bearing fault frequencies that appear in the vibration spectra do not match your calculated frequency. The most common reason is that an unanticipated load in the bearing changes one of the parameters used in your calculation. Typically the parameter that changes is the contact angle.

Unanticipated thrust loads can occur from many sources. Such as magnetic centre misalignment in an electric motor. In certain type of bearings (such as zero contact angle bearings or bearing related only for pure radial load), this thrust load causes the bearing to run at a different contact angle than its normal contact angle.

A rolling element bearing can be used in two ways:

- The inner race is stationary and the outer race is rotating. This is often used in the front wheels of automobiles.
- The inner race is rotating and the outer race is stationary. This is the most common industrial application.

For the case one (inner race stationary, outer race rotating), the fundamental fault frequencies BPFI, BPFO and FTF are calculated by the following formulas:

$$BPFI = N_b / 2 * S * (1 - B_d / P_d * Cos\theta)$$

$$BPFO = N_b / 2 * S * (1 + B_d / P_d * Cos\theta)$$

$$FTF = S/2 * (1 + B_d / P_d * Cos\theta)$$

Where:

N_b: the no. of balls

B_d = ball diameter

P_d = pitch diameter

S = rotational speed

θ = contact angle

For the case two (inner race rotating, outer race stationary), the fundamental fault frequencies BPFI, BPFO and FTF are calculated by the following formulas:

$$BPFI = N_b / 2 * S * (1 + B_d / P_d * Cos\theta)$$

$$BPFO = N_b / 2 * S * (1 - B_d / P_d * Cos\theta)$$

$$FTF = S/2 * (1 - B_d / P_d * Cos\theta)$$

The formula for ball spin frequency BSF is identical for both cases:

$$BSF = P_d / 2B_d * S * [1 – (B_d / P_d * Cos\theta)^2]$$

The most common bearing problem is outer race fault in the load zone. Inner race fault are the next most common.

It should be pointed out that above equations are based on the following assumptions

1. Total numbers of rollers / balls are same in diameter.
2. Contact between balls, inner race and outer race are rolling.
3. Slipping between the shaft and the bearing are negligible.

In practice there is always some sliding and slippage especially when bearing is under load and after some wear. You can refer to the following approximate formulas:

$$BPFI = 0.55\text{-}0.6*S*N_b$$

$$BPFO = 0.45*S*N_b$$

$$BSF = 3.5*S*N_b$$

Several important points to keep in mind when analyzing for roller element bearing defects include:

- Defect frequencies are generated by discrete flaws on races and balls.

- Bearing defect frequencies are asynchronous (non synchronous). This is important for differentiating bearing defects from synchronous components such as gears.
- Bearing defect frequencies will change with variations in contact angle, caused by changes in thrust load, when sliding occurs.
- For equal size defects, the amplitude at outer race defect frequency will be larger than amplitude of inner race frequency.
- In a typical failure, the race frequencies will appear first. Ball and cage frequencies may appear later as side bands around the race defect frequencies.

The amplitude of vibration will depend largely on extend of bearing fault. It must be noted that even momentary impacts can excite natural frequency vibration. All machines have natural frequencies vibration and are excited by impact forces. When the rolling element of bearing strike a crack or pitted area in its raceway, the natural frequency of bearing, shaft or bearing support may be excited.

The natural frequency vibrations usually appear as vibration peaks in frequency ranges many times the shaft RPM.

CHAPTER 3

DATA ACQUISITION TECHNIQUES

3 Introduction

The output from transducer is an analog signal. This means that it is a continuous varying voltage that is proportional to vibration. The analog output is directed to input channel of data acquisition device (alias DAQ) or electronic data collector. The data collector stores some and all of data / information depending on its capability and the instructions from the user. Some data collector are “dumb” in that they only stores data for downloading elsewhere and do not perform any further signal manipulation. Other data collectors are combined with analyzers that can process the signal and develop meaningful information. This course addresses the combined data collector and dynamic signal analyzer.

3.1 Data Sampling and Aliasing

3.1.1 Data Sampling

As illustrated in figure 7 , “sampling” is the processing of converting a continuous analog signal into a discrete sequence of numbers or digital samples. The digitizing process(analog to digital converter) does not represent the waveform exactly but, with enough samples, the approximation improves. Therefore, spacing between samples are highly critical and important. The actual transfer of information occurs in “blocks” which consists of amount of data or samples required to satisfy one transformation. A standard block size consists of 1024 samples. Since the use of smaller block sizes reduces resolution, it is uncommon practice. The analyzer must be instructed on how to capture the data in order to satisfy the user’s ultimate requirement. Analyzer typically requests the following information in the set-up program:

- Numbers of scans (blocks).
- Frequency range.
- Number of spectral lines.
- Window type.

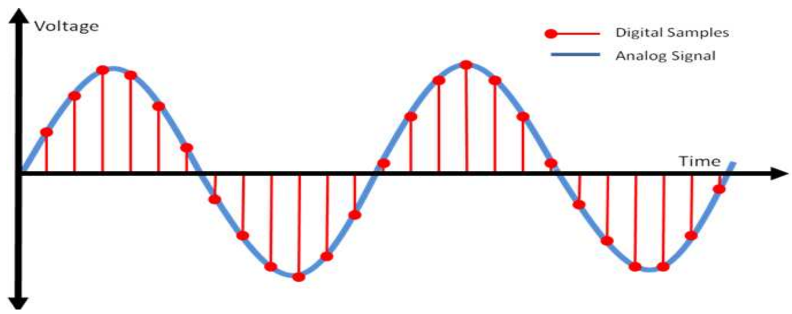


Figure 7: Converting Continuous Signal into Discrete Signal

A. Numbers of Scan (blocks)

Each scan is equivalent to one complete sampling of the data. If the system is steady state and never changes, one sample would be just fine. However, it may appear to be steady state to the naked eye, the activity within machinery tends to vary somewhat over time. As a result it is advisable to obtain a representative sample by taking several scans. Four to five scans is typical unless the variability suggests a greater number. Some analyzers average the scan is typical unless the variability suggests a greater number. Some analyzers have the ability to process each scan and / or the averaged value depending on the instructions from the user.

B. Frequency range

The analyzer must be informed as to the maximum frequency of the interest. If the user does not provide an instruction, the analyzer will spend consider able time going to the highest possible frequency on each scan. This would not only be waste of time and memory but might also corrupt the data by exceeding the capability transducer. As a result, the user must specify the maximum frequency. The frequency range is selected upon evaluating the following criteria:

- Machines and component typically generate disturbances across different frequency spans. The analyst must calculate or estimate the highest frequency of interest.
- The transducer must be linear across the range of interest. Sometimes the maximum frequency is determined by the linear range of transducer.

C. Spectral lines

The analyzer must be instructed as to the amount of clarity or resolution required. The FFT algorithm computes the amplitude versus frequency and displays it in discrete quantities. The computed data points are placed in a ‘bin’ or in on a ‘line’ created by dividing the maximum frequency into numerous equal quantities. For example, assume that maximum frequency of interest is 400Hz. To graduate amplitude/frequency data points in 1 Hz increments, it would necessary to specify 400 lines (400Hz/400 lines=1 Hz/line). The data that falls in between the increments is forced into another bin or line. This situation result in a distorted and non distinct display. To improve the accuracy of display for the purpose of analysis, the user could specify additional lines. The total line required to process an FFT is equal to the amount of time required for data collection plus computation plus display. Since current dynamic signal analyzers are very fast, the computation and display time is negligible in most instances. Therefore, the time required to capture the data is related the number of lines and maximum frequency as expressed below

$$T_s = N / F_{max}$$

Where

T_s : Sampling Time

N: number of lines

F_{max} : maximum frequency

Greater resolution requires more time. Not only that but more data must be stored. As a result, vibration analyst are frequently instructed to perform routine machinery health monitoring at lower resolution (less lines) to save time and conserve memory.. For troubleshooting, however, resolution is critical to discerning the peaks and not missing something important. The lowest resolvable frequency or the sampling rate, f_s, is related to sampling time as follows:

$$f_s = N / F_{max} = 1 / T_s$$

The number of samples is related to number of lines by the following relationship:

$$S = 2.56\ N$$

3.1.2 Aliasing

Aliasing is the creation of false disturbances during digital sampling process. Figure, illustrates the concept of aliasing caused by sufficient sampling rate. The vibration analyst could very easily be misled by the peaks, making an in correct diagnosis. Two signal "alias" if the difference in their frequencies appears in the frequency range of the interest. To avoid this situation, the Shannon or Nyquist sampling theorem states that a continuous signal can only be properly sampled when it does not contain frequency components above ½ the sampling rate. As a result it is a common practice to sample the data at minimum of twice the highest frequency of interest. A sampling rate equal to twice the input frequency is called "Nyquist Frequency" in signal processing. To further assure that stray signals do not initiate aliasing, low pass filters, called an "anti alias filters", are installed prior to performing analog to digital conversion as well as prior to performing the FFT. In most stances, the appropriate sampling rate is automatically established by the dynamic signal analyzer when set-up information is entered.

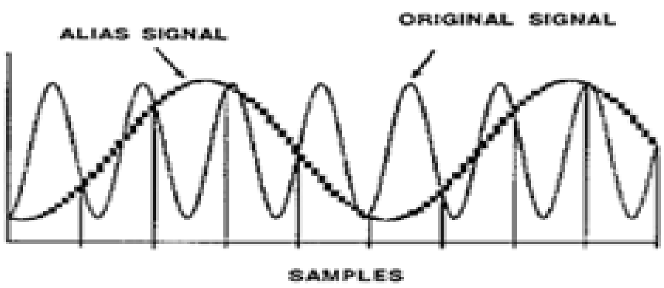


Figure 8: Illustration of Aliasing

CHAPTER 4

Signal Processing (Conditioning) Techniques

4 Introduction

The term signal processing is the process, amplification and making it easy to understand of signals. On the other hand signal conditioning is the processing of the shape or style of signal so as to make it understandable to, or well-suited with, a given device. It can be said that signal processing is a more general term and it includes signal conditioning. Signal conditioning is special term refers to signal processing related with a device.

Unrefined vibration signal always contain components which are smeared ("noise") and frequently some original components which may partially unclear to actual components which are the important part of measurements taken. There are several choices that can have the noise or other uninterested signal parts removed.

4.1 Windows

When we use FFT to find the frequency content of signal, it is inherently assumed that the data that you have is a single period of periodically repeating waveform. This is show in Figure 9 .

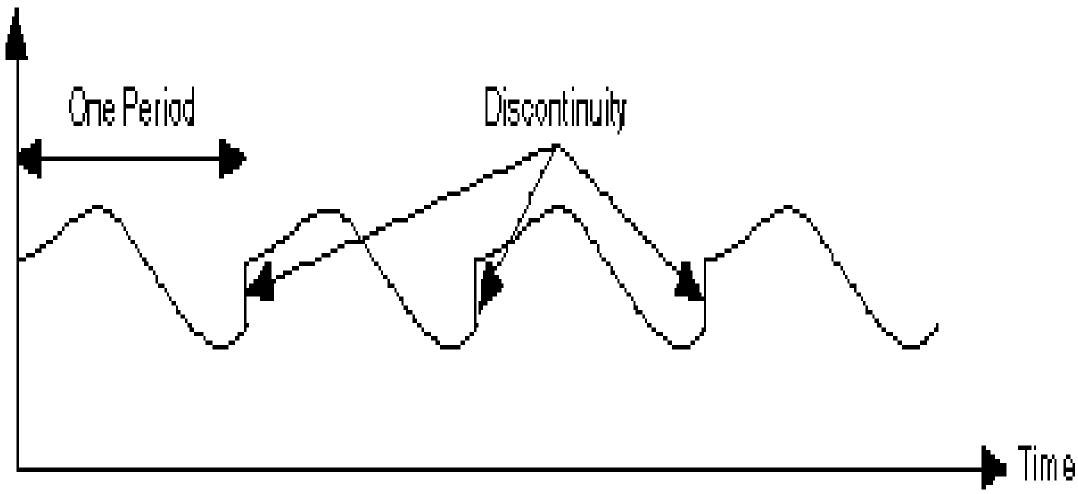


Figure 9: Periodic Extension of Sampled Data

The first period shown is the single sampled. The waveform equivalent to this period is then repeated in time to produce the periodic / constant waveform. If this condition is not achieved, an issue called "spectral leakage" occurs resulting in an incorrect spectral display. Specifically, some of the energy at a given line or bin will transfer to a higher frequency bin , resulting in a spectral peak that is not really present. When leakage occurs, the vibration analyst can be mislead by erroneous information and perhaps make a false diagnosis.

To minimize leakage, a window factor is applied to force the data block amplitude or input to zero at each end. Non periodic disturbances will still cause some leakage.

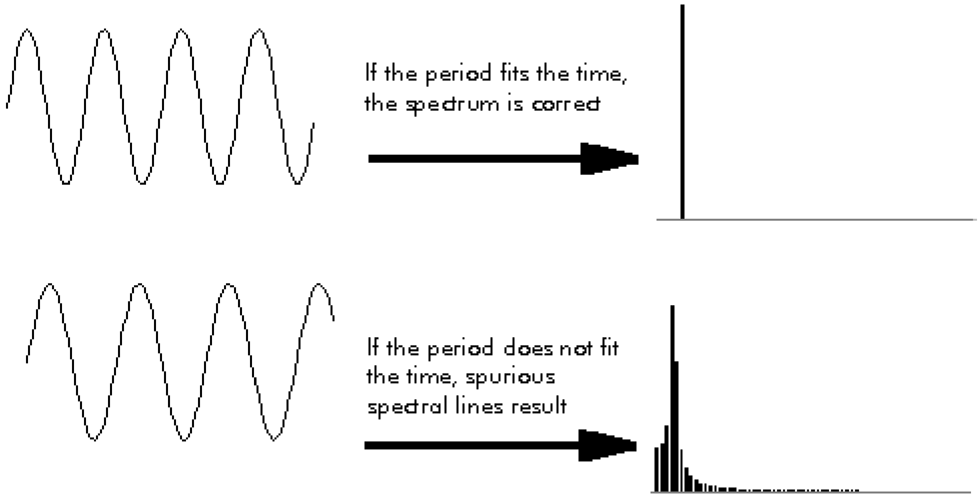


Figure 10: Illustration of Spectral Leakage

There are numerous choices of windows that effect the shape of the spectral peak in different ways. As a result, must make a choice as to the most appropriate window factor to apply under the circumstances. Several of most common windows are displayed in table below

Window Type	Application	Window Factor
Uniform or None	Bump test, transients, event occurs within time record	1
Hanning	Fault diagnosis, random noise, periodic signals	1.5
Flat top	Refined fault diagnosis	3.8

A Uniform Window is essentially no window at all and is used where periodic FFT requirements are fully met. A Hanning Window provides a clear peak for ease of resolution but results in amplitude errors. That the amplitude is not entirely correct is, for the most part, not a significant complication for machinery diagnostics. With a Flat Top Window, the amplitude is represented well at the cost of frequency resolution. Window factor effects the resolution and if not select properly may combine peaks. Line or Bin resolution is ultimately determined by the following relationship:

$$\text{Analyzer resolution} = 2\ F_{max}\ WF / N$$

Where, WF: Window factor

4.2 Averaging

The raw waveform signals of operating machinery tend to contain complex information that varies during the sampling process. Much of unwanted information is random noise. A comparison of individual waveforms taken close together reveals the extend of the variation both in amplitude and frequency. To improve the quality of information, the data is frequently averaged to provide more meaningful results, hopefully typical of steady state process. Either the waveforms or spectra can be averaged. There are several types of averaging:

a. Synchronous time averaging: To assume that the blocks of data are always captured starting at the same periodic moment; the data collector can be prompted or triggered. This type of averaging is appropriate when asynchronous background noise is interfering with identification of peaks.

b. RMS or Power averaging: This is amplitude averaging based on square root of arithmetic mean of squared spectra. This technique works well to reduce the amplitude variance, but still permits noise interference. This type of averaging is appropriate when background noise is not overpowering.

c. Difference (negative) averaging: The intend with this technique is to highlight resonance activity in the spectrum by linearly subtracting the normal operating spectrum from a spectrum containing deliberately excited resonance.

d. Peak averaging: This is the averaged amplitude of maximum peaks at particular frequency. This technique is used to capture peak amplitudes at each line during transient testing.

e. Overlap: Time records are typically scanned in sequence. When one record is complete other is started. With overlap processing the analyzer is instructed to combine old and new time records by overlaying a portion of the new onto old.

4.3 Filters

A filter is a system that is designed to remove some component or modify some characteristic of signal. Hardware or software filters are fundamental to signal conditioning. The filter limits or cleans the raw vibration signal in some predictable fashion such that a single frequency or group of frequencies may be isolated for further processing / study. They can be classified as follows:

- Low pass filters
- Band pass filters
- High pass filters

4.3.1 Low Pass Filters

This type of filters allows only low frequency signal components to pass through for further processing. In essence, all frequencies that lie within the pass band, ($f<f_0$) are accepted for processing while signals above the cut-off frequency are attenuated and rejected to extend possible. Roll-off occurs from the cut-off frequency to full rejection or stop band.

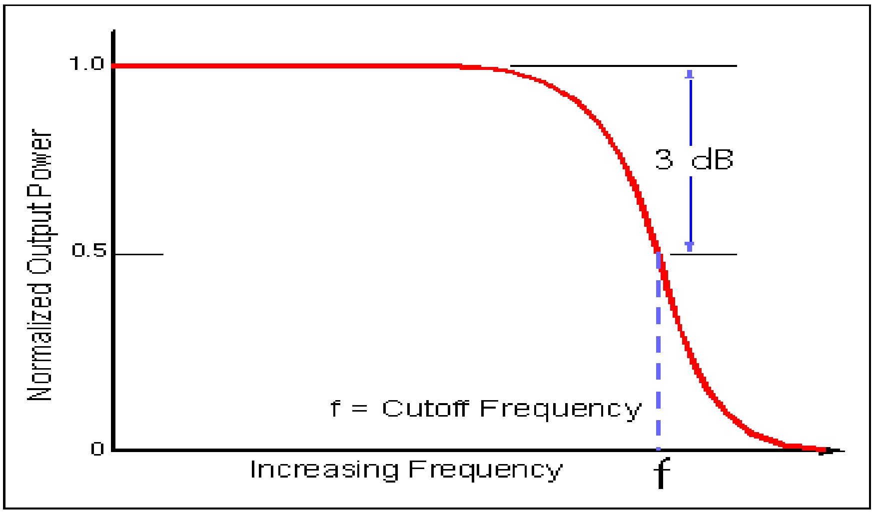


Figure 11: Low Pass Filter

4.3.2 Band Pass Filters

A band pass filter passes a particular band of frequencies for further processing while rejecting frequencies both above and below the desired pass band. A notch or band reject filter is opposite of band pass filter. In fact notch or band reject filter eliminates all frequencies which are un wanted.

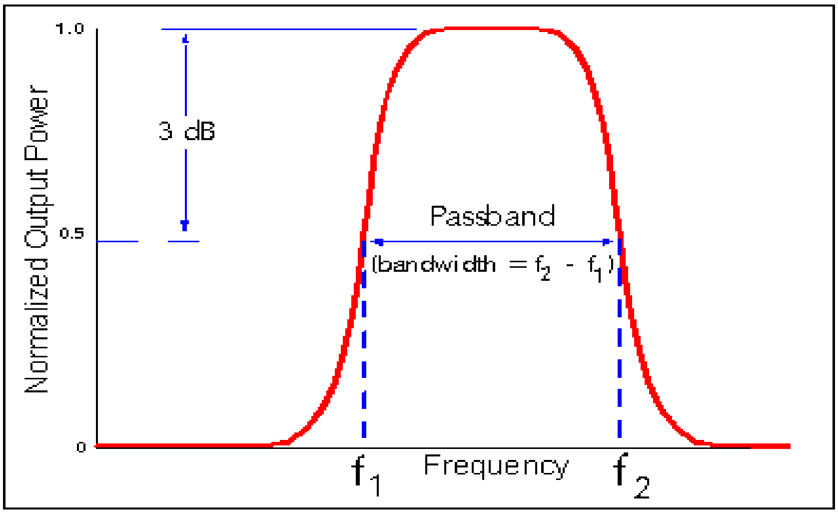


Figure 12: Band pass Filter

4.3.3 High Pass Filters

A high pass filter passes only high frequencies for further processing while rejecting frequencies both below the point of acceptance. High pass filters are usually used with accelerometer to eliminate the lower frequencies.

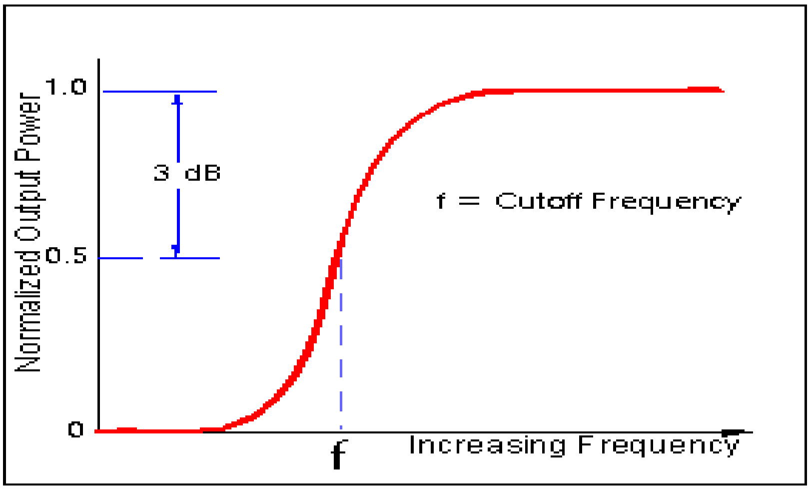


Figure 13: High Pass Filter

4.4 Dynamic Range and Logarithmic Range

In many instances, a spectrum contains peaks at location across the frequency band that varies from low to very high. Some of the peaks may be close to the noise floor whereas others may be almost off scale. When there is big disparity in the amplitudes it is very easy to oversee important low level activity as the high peaks take precedent.

If the dynamic range is not set properly, then this can lead to decreased resolution, sometime clipped data (i-e the peaks of signal cannot be read because the signal is much larger than what we thought it would be) or even introduce noise or have a poor signal to noise ratio.

To further improve visibility, a logarithmic scale is often used to enhance the dynamic range of the plots. Multiplying the log relationship by 20 converts the logs to decibels. A comparison of linear scale presentation of a large difference to a logarithmic presentation is illustrated in figure 14

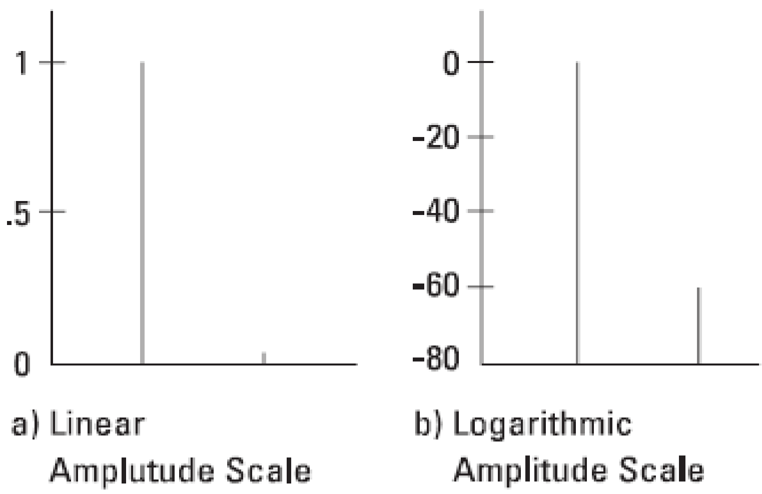


Figure 14: Linear Scale Vs Log Scale

The vibration analyst frequently encounters large range amplitudes when monitoring machinery. The initial fundamental bearing faults exhibit very low amplitudes. Issues such as misalignment, un balance, gear mesh, etc tend to overshadow the bearing signatures, making early detection of bearing failure very difficult. Applying a logarithmic scale would bring the bearing fundamentals in to view.

The dB range is computed as follows:

$$dB = 20 \log(V / V_{ref})$$

Where:

V: amplitude of the high peak

V_{ref} : amplitude of the low peak

For example, with a 5.7 mm/s gear meshing peak versus a 0.1 mm/s bearing peak, the dynamic range required would be calculated as 20 log (5.7/0.1) = 35.12 dB.

4.5 Vibration Signature Analysis

The word signature has been appointed to the trend of the signal that identifies the health of a system from which they illustrate purchased. This vibration signatures are widely used as an analytical tool for the mechanical system used.

Methods based on vibrations have been widely used for the detection and diagnosis of faults over several years. These methods are applied in the usual manner and was separated in time and frequency. An analysis in the time domain is mainly focused on the statistical properties such as peak level, asymmetry, standard deviation, kurtosis and crest factor. Frequency domain method generally used Fourier analysis, order to convert the time-domain to the frequency-domain. These techniques are generally classified into three major categories, which are given below

4.5.1 Time- domain analysis

The analysis in the time domain are passed to a display and evaluation of the vibration device according to time. The great advantage of this format is that no data is lost in the review. However, the disadvantage is that there is often too much data (also a bit of time out loud) for the diagnosis of simple and clear faults. This procedure is further characterized as: time waveform analysis, time waveform indices, time synchronous averaging, negative averaging, orbits and probability density moments.

4.5.2 Frequency- domain analysis

Frequency domain relates to a display device or the data analysis function of the oscillation frequency of the device. The main advantage of this kind of set-up is indicated recurrent clearly as peaks in the spectrum of vibration signals at frequencies at which relapse occurs but disadvantage often is a significant loss of data that falls on the transformation. This technique is further categorized as: band pass analysis, shock pulse analysis, enveloped spectrum, signature spectrum and cascades (waterfall plots).

4.5.3 Cepstrum analysis

The quefrency or cepstrum domain analysis is basically x-axis for cepstrum, it can be described as first convert time domain signal in to power spectrum then take its logarithm and agin take its spectrum. One of the main advantages is that it highlights periodicities and harmonics that occur is spectrum. This methodology is called cepstrum analysis or quefrency domain analysis, “cepstrum” is basically inverse of ‘spectrum”.

CHAPTER 5

RESULTS AND EXPERIMENTS

5 Introduction

To allow for, the easier and exact detection of inner and outer faults in Rolling Element Bearing (REB), the Envelope Detection (ED) technique has been used together with FFT. The impact vibrations are difficult to be identified in low frequency range due to their low energy and interference, the usual practice is to view these micro-structural vibrations at the bearing resonance frequency range. The modulated amplitudes of repetitive impacts are often excited at the bearing structural resonance frequency. Hence, the amplitude demodulation provided by ED allows the detection of localized detects. Due to the inability of FFT to detect faults, which exhibit non-stationary impact signals, there was a need to seek for other alternatives.

Lets have an example with bearing that has developing a crack in its outer race. Each time a ball passes over the crack, it creates a high-frequency burst of vibration, with each burst lasting for a very short time. In the simple spectra of this signal one would expect a peak at BPFO instead we get high frequency because of excitation of bearing structural resonance. The signal produced is an amplitude-modulated signal with bearing structural resonance frequency as the carrier frequency and the modulation of amplitudes

5.1 Statistical Analysis

One problem which arouse was that what will be happen if the bearing will be healthy? Because the healthy / normal REB has also vibration signal with some peaks. In order to identify only the largest peak will lead to wrong detection, so there was some need to develop more efficient technique.

KURTOSIS is statistical analysis technique used on signal as a function of time Numbers of scans (blocks).

5.1.1 Kurtosis

In mathematical theory and statistics, kurtosis (from the Greek word κυρτός, kyrtos or kurtos, meaning bulging) is any measure that whether the data is peaked or flatten. Kurtosis is descriptor of the shape of a probability distribution. It was observed that if the kurtosis value is greater than 3 then REB is considered as faulty else normal bearing.

5.2 Procedure

Raw data signal is acquired in Matlab GUI and displayed as time domain plot. On the same Matlab GUI REB characteristic frequency is calculated using bearing characteristics formulas. Raw data is converted in to frequency domain using FFT technique and band passed. ED of band passed signal is displayed using "Hilbert Transformation". Calculation of amplitudes at calculated characteristics frequency using interpolation function. Condition applied using kurtosis, if value of kurtosis is less than 3 then result displays is "normal bearing" and if the kurtosis value is greater than 3, algorithm compares amplitude of bearing characteristics frequency at inner race and outer race through enveloped spectrum signal. Finally GUI display the results automatically on 'dialogue box'.

5.3 Results

Test 1

Description	
File Name	97.mat
No of Samples	243938 samples
Kurtosis	2.78
BPFO (Hz)	107.3043 Hz
BPFI (Hz)	162.0957 Hz
RPM	1796
Result	**Normal Bearing**

In the first experiment 97.mat (Matlab file) is loaded and displayed in time domain, the bearing data file obtained from accelerometer has 243938 samples. Up to 12000 samples were displayed on left top plot (time data).In top right plot FFT is calculated in which peaks of bearing characteristics frequency are not clearly shown due to spectral leakage. In left bottom plot envelope signal is calculated and finally spectrum is plotted in bottom right plot. The bottom right plot is displayed up to 500 Hz. Above table shows all the parameters calculated for matlab file and of specific ball bearing (6205-2RS JEM SKF). As value of kurtosis is less than 3, so this bearing is normal. In last bottom plot we can still see peak, this peak is not related to bearing characteristic frequency, in fact this peak is shaft frequency which is rotating at 1796 RPM (or 30 Hz) and at 60 Hz it is 2 x RPM.

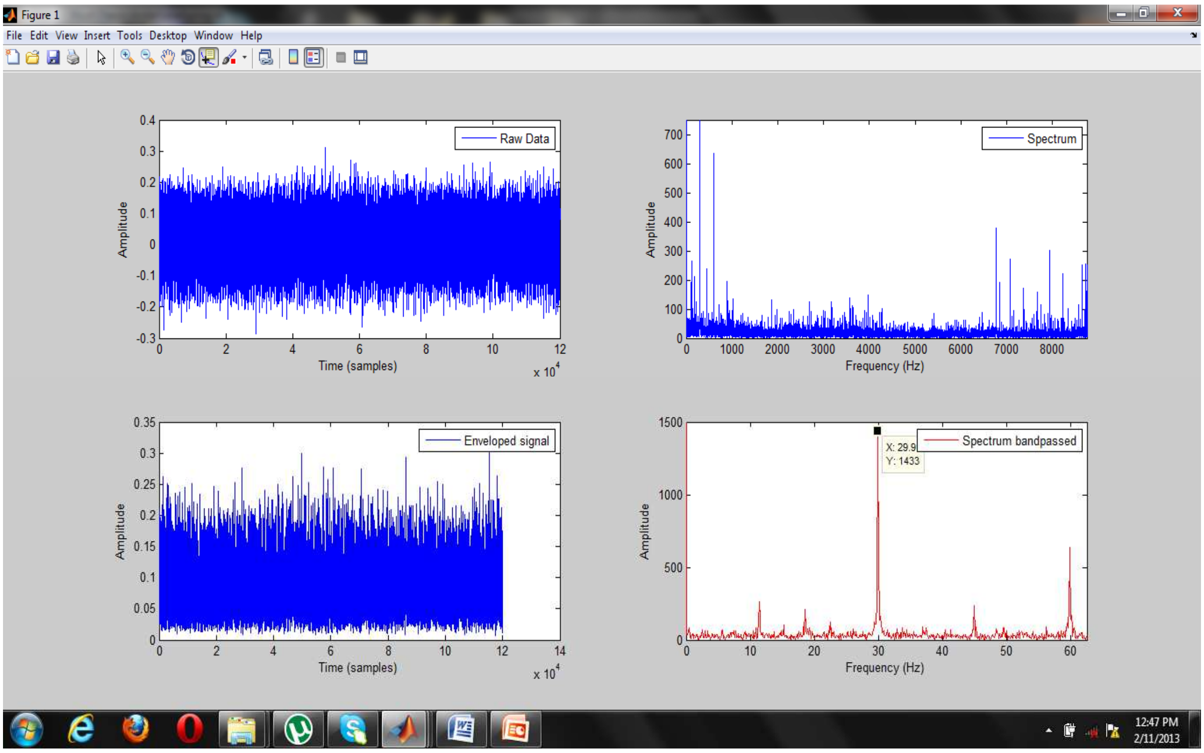


Figure 15: Matlab Plots for Normal Bearing

<u>Test 2</u>

Description	
File Name	209.mat
No of Samples	122136 samples
Kurtosis	7.46
BPFO (Hz)	107.364 Hz
BPFI (Hz)	162.186 Hz
RPM	1797
Result	**Inner Race Fault**

In the second experiment another matlab (209.mat) file is tested with similar algorithm as above. While calculating kurtosis it values shows that there is some defect as this value is greater than 3. First peak in enveloped spectrum at 29.95 Hz shows shaft frequency. Now it is clear that there is spike at 161.9 Hz (near to 162.186 Hz) of inner race frequency. However amplitude of outer race frequency (i-e 107.364 Hz) is less then inner race. As largest amplitudes displays fault hence this bearing has inner race fault. Other peaks and side bands shown in plot are harmonics of shaft frequencies and other bearing characteristics frequencies like outer race frequency, ball and cage frequencies.

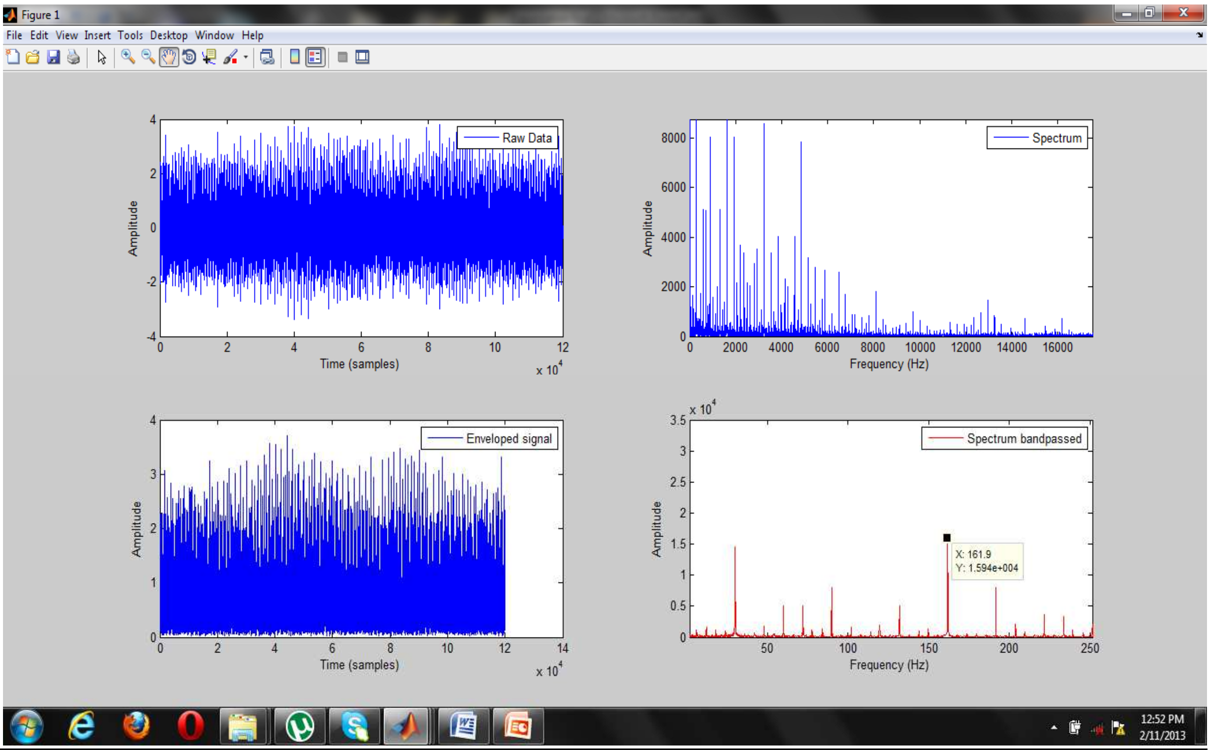


Figure 16: Matlab plots for Inner Race Fault

Test 3

Description	
File Name	210.mat
No of Samples	121556 samples
Kurtosis	7.66
BPFO (Hz)	105.989 Hz
BPFI (Hz)	160.110 Hz
RPM	1774
Result	**Inner Race Fault**

Another malab file is tested with above algorithm and procedure, again found difficulty in finding peaks due to spectral leakage and noise. Kurtosis value greater then 3 also showed that this is faulty bearing. Observing enveloped spectrum first peak at 29.56 Hz shows shaft frequency (1774 RPM). Once again clear peak is observed at 159.8 Hz near to

160.110 Hz of inner race frequency. The algorithm and matlab coding clearly proved the fault findings at different RPM easily.

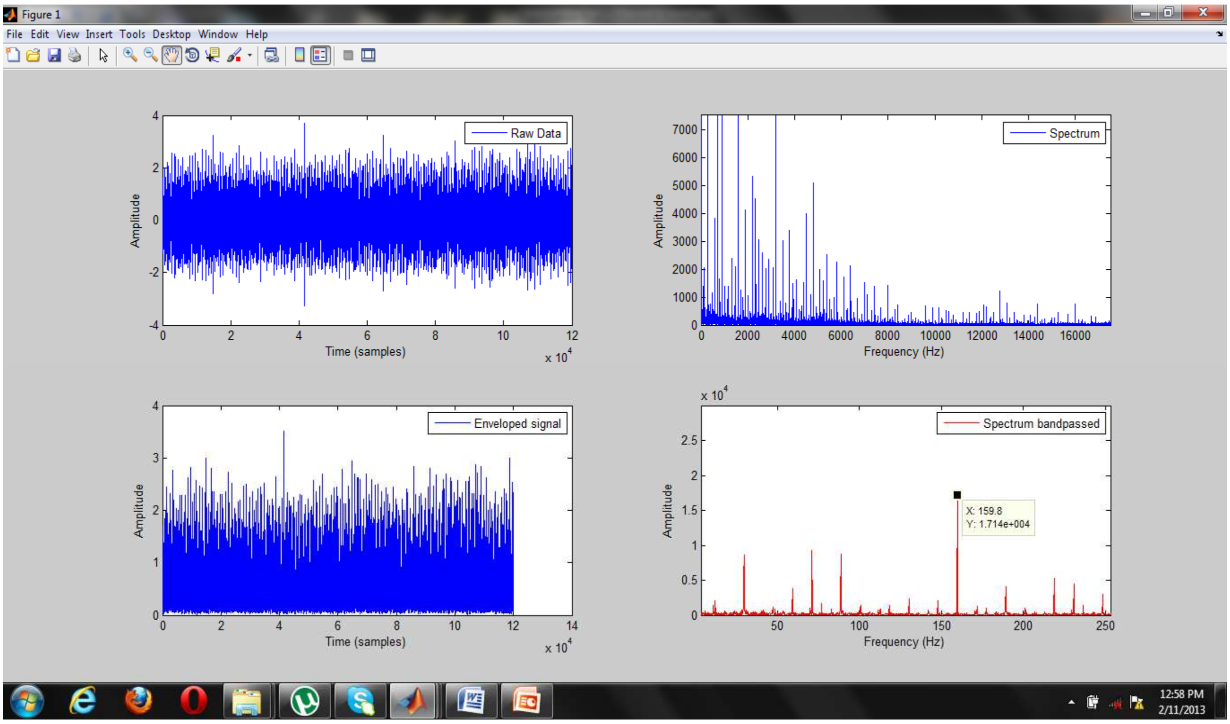


Figure 17: Matlab Plots for Inner Race Fault

Test 4

Description	
File Name	211.mat
No of Samples	121846 samples
Kurtosis	8.06
BPFO (Hz)	104.675 Hz
BPFI (Hz)	158.124 Hz
RPM	1752
Result	**Inner Race Fault**

This experiment also showed inner race fault for particular matlab file (211.mat), kurtosis value is greater than 3 tells some abruptness and peakdness. Enveloped spectrum at bottom right shows clear peak at 157.8 Hz near 158.124 Hz of BPFI. Again first peak at 29.2 Hz (or at 1752 RPM) shows shaft frequencies and rest of peaks and side bands shows outer, ball and cage frequencies and harmonics of shaft frequency.

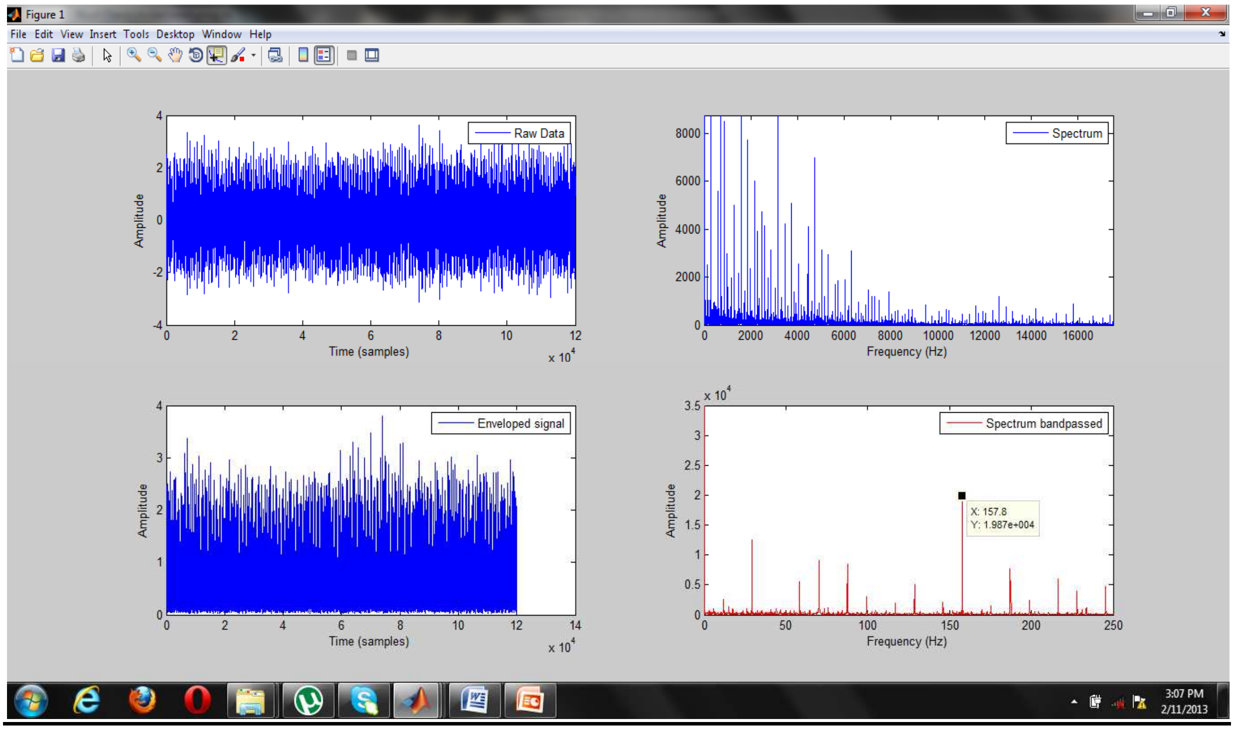


Figure 18: Matlab Plots for Inner Race Fault

Test 5

Description	
File Name	234.mat
No of Samples	122426 samples
Kurtosis	20.96
BPFO (Hz)	107.304 Hz
BPFI (Hz)	162.095 Hz
RPM	1796
Result	**Outer Race Fault**

In the test 5, another matlab file (234.mat) was tested using above algorithm. The matlab file has 122426 samples, kurtosis applied on 120000 samples, again found kurtosis value higher then 3.FFT plot showed spectral leakage and unable to find correct peak. Enveloped spectrum once again showed true picture of fault, as it can be clearly seen the peak at 107.5 Hz which is near to BPFO (i-e 107.304 Hz). Second highest peaks at 29.93 Hz shows shaft frequency (i-e 1796 RPM). Rest of peaks shows harmonics of shaft frequency and other side bands shows ball, inner race and cage frequencies respectively.

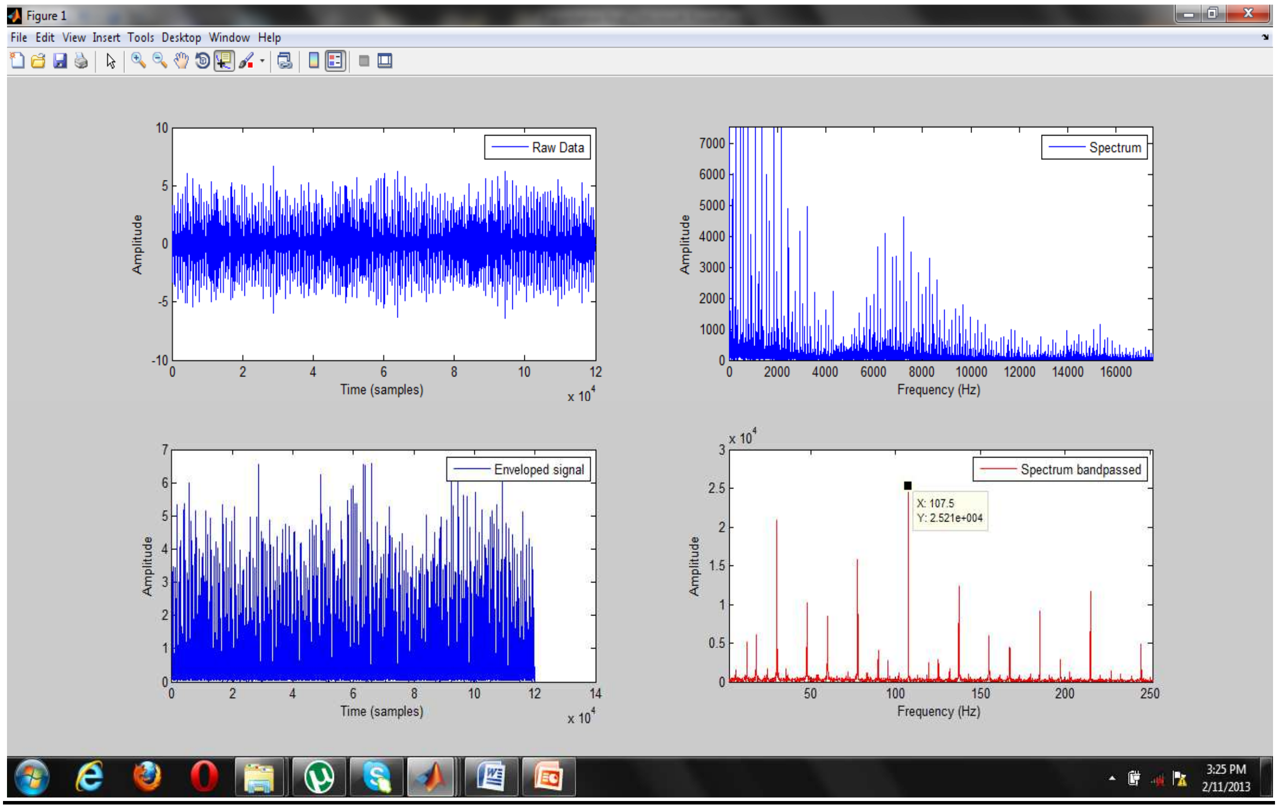


Figure 19: Matlab Plot for Outer Race Fault

Test 6

Description	
File Name	235.mat
No of Samples	121991 samples
Kurtosis	21.99
BPFO (Hz)	105.810 Hz
BPFI (Hz)	159.839 Hz
RPM	1771
Result	**Outer Race Fault**

In this matlab file again above procedure was repeated to verify the results and algorithm, kurtosis value again greater than 3 shows the faults in bearing. In order to find which bearing fault it has envelop spectrum was observed. A peak was observed at 106 Hz approx. The peak at BPFO is greater than BPFI, hence greater peak tells fault of bearing. The max peak in graph tells shaft frequency which is 29.51 Hz.

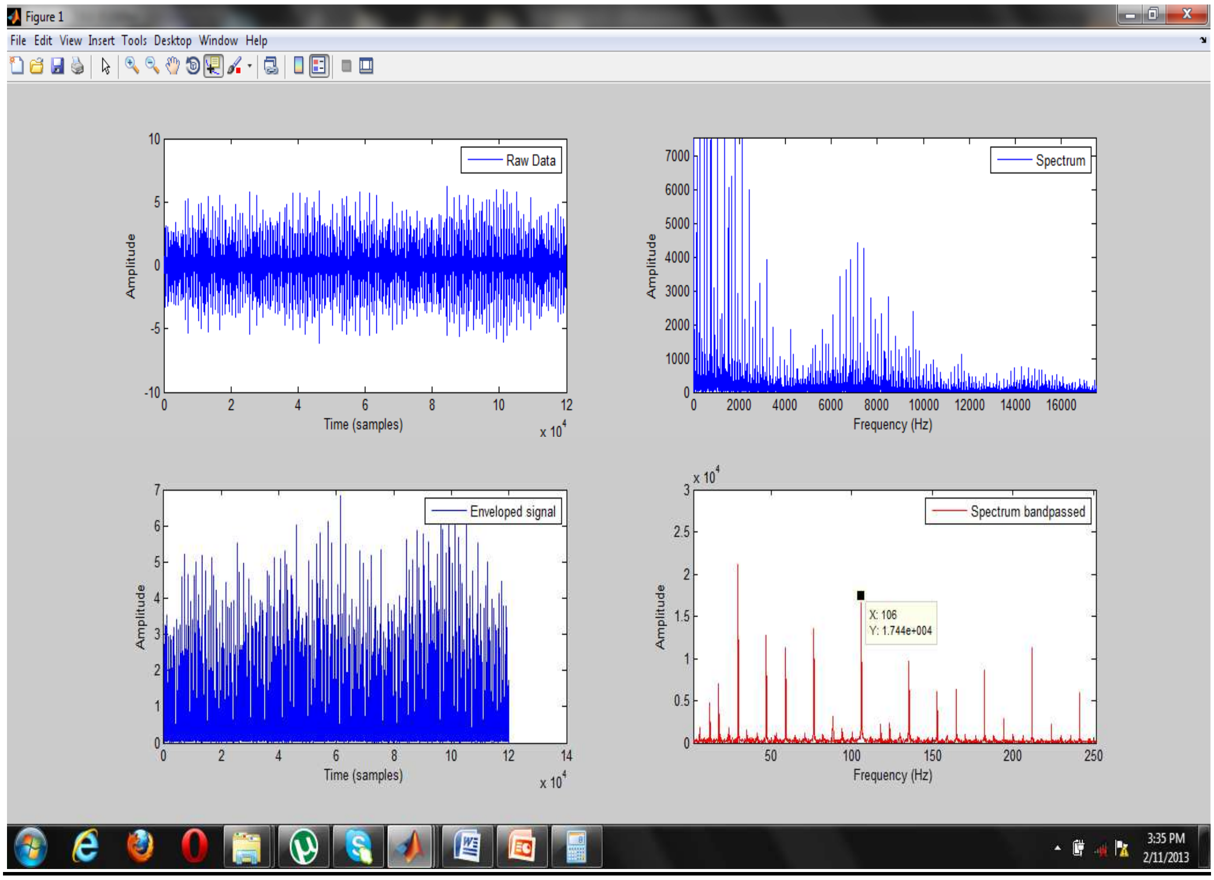


Figure 20: Matlab Plot for Outer Race Fault

Test 7

Description	
File Name	236.mat
No of Samples	122281 samples
Kurtosis	23.27
BPFO (Hz)	104.436 Hz
BPFI (Hz)	157.763 Hz
RPM	1748
Result	**Outer Race Fault**

In this matlab file (236.mat) again above procedure was repeated to verify the results and algorithm, kurtosis value again greater than 3 shows the faults in bearing. In order to find which bearing fault it has envelop spectrum was observed. A peak was observed at 104.6 Hz approx. This peak is equal to BPFO, and hence proves the fault of outer race. Second highest peaks shows shaft frequencies at 29.133 Hz. Rest of peaks and side band tells information about shaft frequency harmonics and other bearing characteristic frequencies.

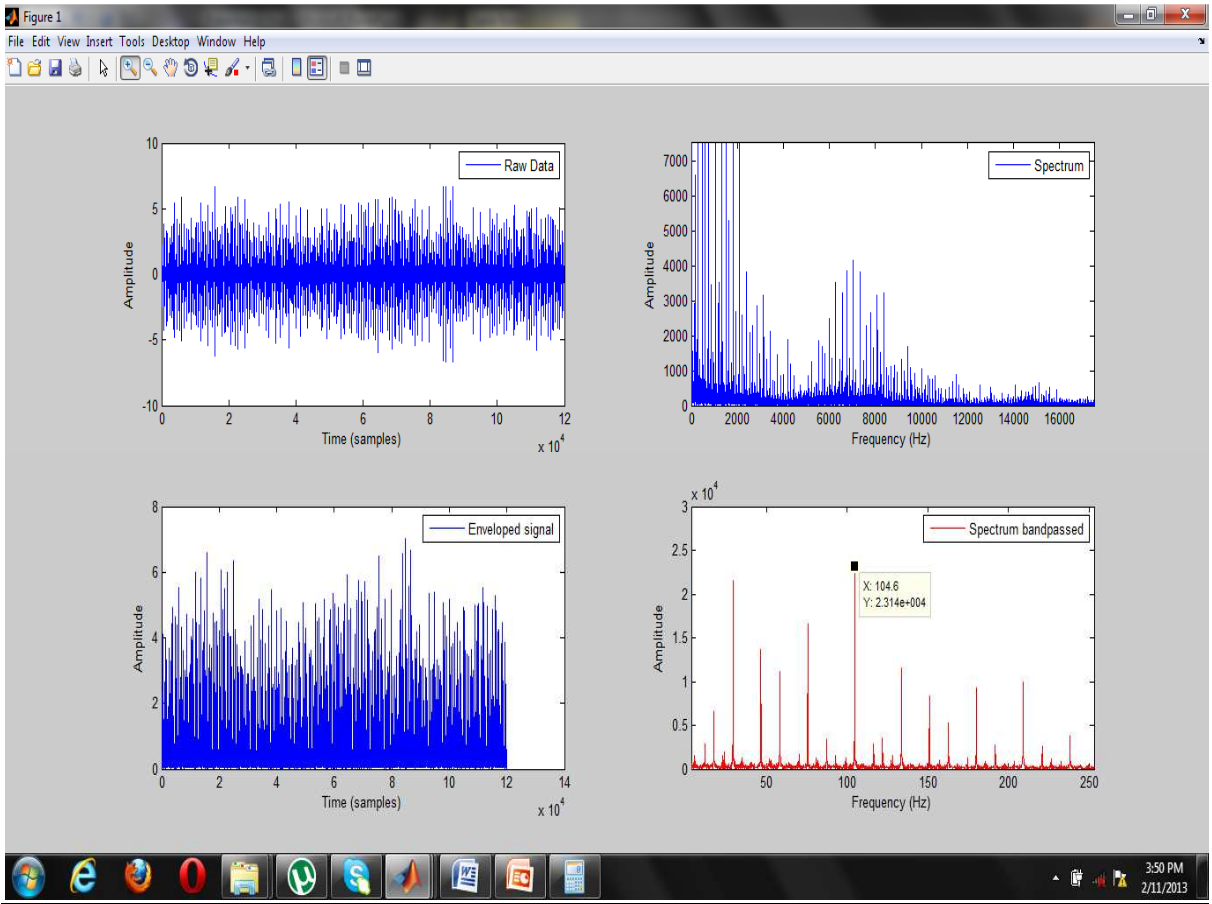


Figure 21: Matlab Plot for Outer Race Fault

5.4 Matlab GUI Display

5.4.1 Normal Bearing

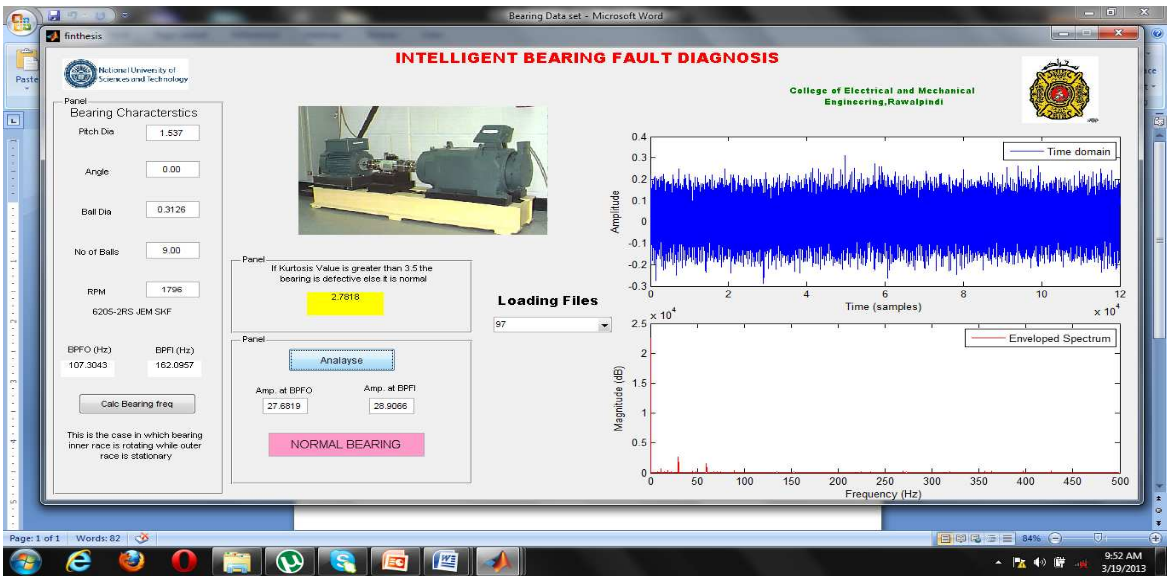


Figure 22: Matlab GUI with Normal Bearing Result

5.4.2 Inner Race Fault Bearing

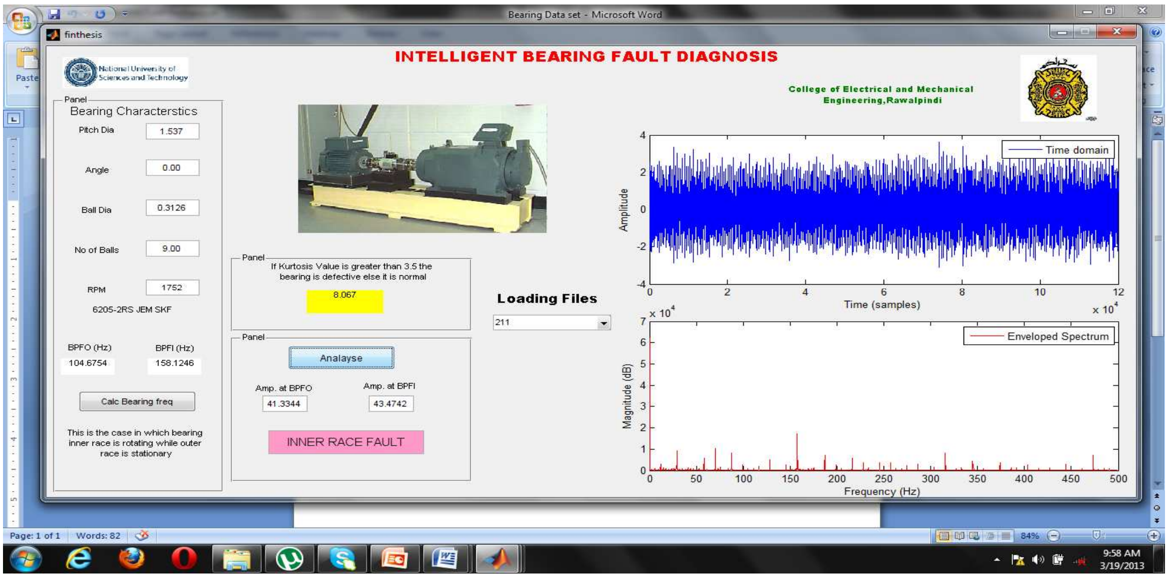


Figure 23: Matlab GUI with Inner Race Fault

5.4.3 Outer Race Fault Bearing

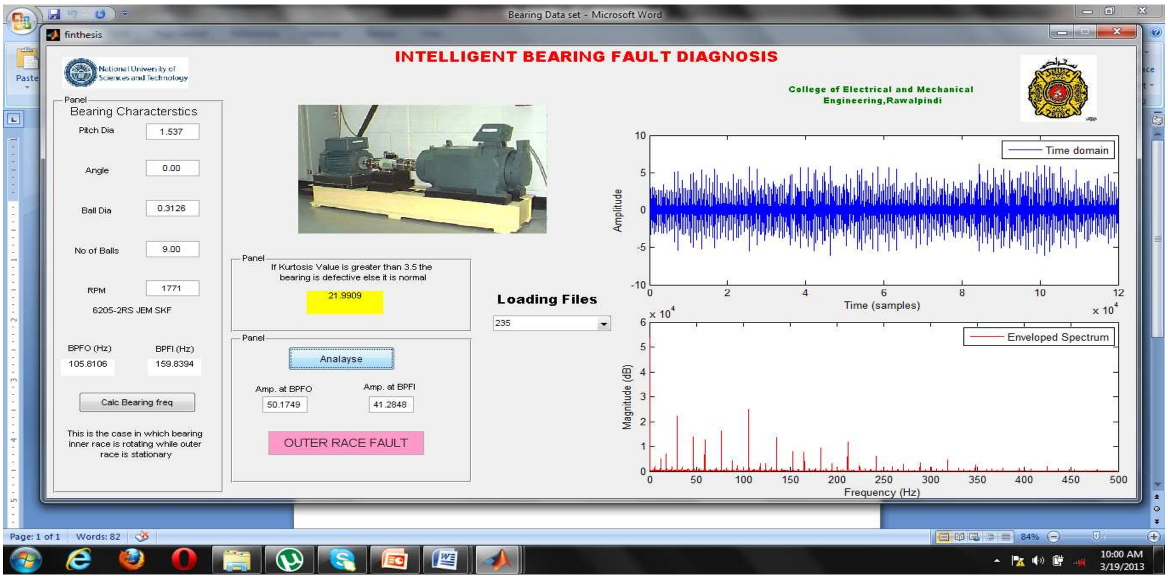


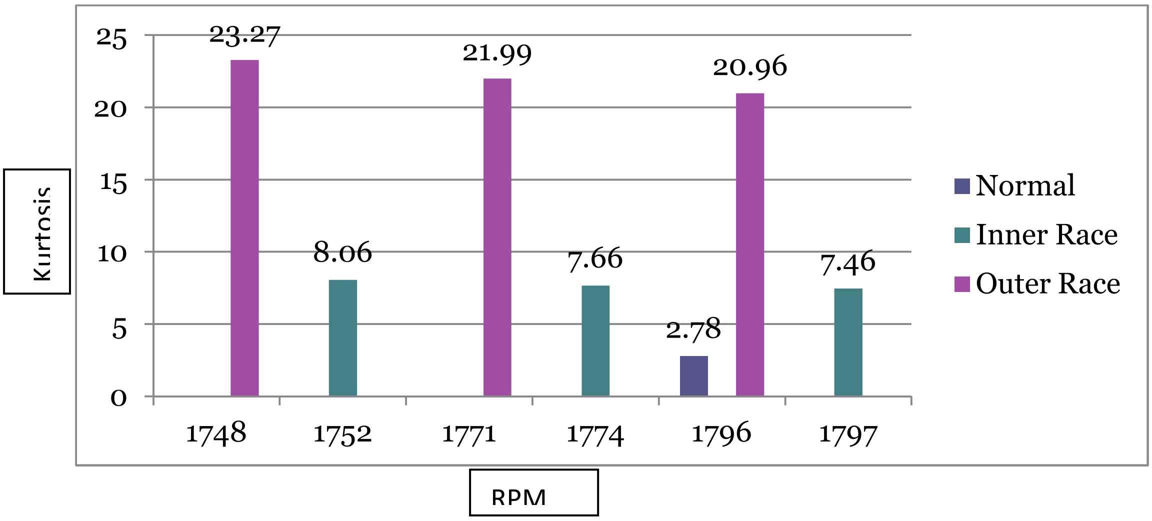
Figure 24: Matlab GUI with Outer Race Fault

5.5 Conclusions

The bearing fault diagnosis system is although not new technology. Many researchers worked and developed different methodology. This thesis provides novel idea of using minimum signal processing techniques. The results were finally calculated prior to signal processing techniques (FFT, Envelop Spectrum) through comparison of BPFO and BPFI peaks. It was also observed that kurtosis is strong tool for finding any abruptness. Kurtosis value greater than 3 shows fault in bearing. It was also observed that kurtosis value of much higher value (i-e greater than 20) shows and reveals the fault outer race and the value of kurtosis for inner race fault will always be less then outer race kurtosis value.

5.5.1 Comparison Of RPM Vs Kurtosis

The values of results also reveals another result that the Kurtosis value decreases with increase in bearing RPM. This can be shown in graph given below.



These result can also be verified from research papers [44] and [45]. The main reason is that as the bearing speed for faulty bearing increases crest factor and RMS value also increases. Since RMS is inversely proportional to Kurtosis hence its value decreases. These results also shows that bearing has been harmed with time and was started to crushed out at complex point of damage.

5.6 Future Work

The methodology described must be implemented practically. This is low cost technology and similar methodology can further be researched in the field of other analysis such as wear and tear in gears, pistons etc. Similarly other bearings like journal bearing can also be tested or some other easiest algorithm can be developed.

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