# FAULT DIAGONOSIS BY VIBRATION ANALYSIS OF SYNCHRONOUS GENERATORS

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# Abstract

All machinery with moving parts generates mechanical forces during their normal operation As mechanical condition of machine changes because of wear, changes in operating environment, load variations etc., so do these forces. Generator has one or more machine elements that turn with the shaft - e.g. bearings, rotors. In a perfectly balanced machine, all rotors run on their true centre-line and forces are equal.

Rotor imbalance will generally be present due to uneven weight distribution or due to the imbalance between generated lift and gravity. Combination of these forces with stiffness of rotor-support system will determine the vibration level. Vibration profile that results from motion is the result of a force imbalance - there is always some imbalance in real-world applications. All mechanical equipment in motion generates a vibration profile, or signature, that reflect sits operating condition.

Many vibrations are normal for rotating or moving machinery, e.g. normal rotation of shafts and other rotors, contact with bearings etc.

Synchronous Generator faults such as mechanical misalignments, rotor imbalance, loose bolts, bearing faults and incipient metal fatigues cause to generate abnormal identifiable vibrations.

In the research,
First, the relationship between the generator mechanical faults and the vibration harmonic magnitudes are studied for particular machine problems occur previously and for that analysis the critical frequencies in the frequency spectrum of the synchronous generators are identified by using the Micrologue instrument CMVA 60, which is a property of the Mahaweli Complex of Ceylon Electricity Board. Then, the magnitude ratios of the harmonics at critical frequencies to the fundamental component of the vibration profiles are determined from Experiment with using

single phase motor with load coupled which had been used with originated mechanical faults, is used for the vibration analysis.

Next, the results obtained from the case studies and the experiment are compared with the standards that have been evolved from the past studies and the researches in order to determine the feasibility of setting defective levels or standard on the vibrations harmonics at critical frequencies of generator faults. Finally, the possibility of developing a condition monitoring system to identify mechanical faults is investigated for synchronous generators in Ceylon Electricity Board. The results can then be extended to indicate the faults at early stage to minimize the unwanted long outages, minimize costly rotating failures & reduce maintenance inventory cost.

Thus it helps to provide the necessary lead-time to schedule maintenance to suit the needs of the plant management.

# Declaration

I hereby declare that the work presented in this report is my own work and not has been submitted earlier or concurrently for any other degree.

***UOM Verified Signature***

Signature . ......................

Name : S. H. Ediriweera

Date : 08th February 2010

I certify that this work was supervised by me and the above declaration is true.

***UOM Verified Signature***

Signature : ..............................

Name : Dr. J. P. Karunadasa

Date : 08th February 2010

# Preface



Since this condition monitoring system is not much longer used in the CEB and there is little number of literature found in the University Library, it was little difficult for me to decide where to start at the beginning. The condition-monitoring tool purchased from the SKF Condition Monitoring Inc. was helpful in this concept and I could gather more details from the Internet.

I express my sincere gratitude to Dr. J P Karunadasa for all the encouragement, guidance and support given throughout my Engineering Carrier to make this task a success and directing towards the research towards the realization of the ultimate goal.

I sincerely thank Mr. A.K.Samarasinghe, DGM (AMHE), Thilakasiri Vijayananda EE (Controls & Instrumentation) and the technical staff at Canyon Power station for providing me with the details of the vibration monitoring systems, other required literature, and support for builds a machine model for the experiment and for their comments on some difficulties encounter during the project.

Finally A big thanks go to my wife Iroshini and my parents for finding me free time and free mind taking my responsibilities to do the research.

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# Chapter 1
# INTRODUCTION

## 1.1 General

Condition monitoring of synchronous generators can be effective in minimizing overall maintenance costs and downtime associated with the operation of these machines. Rotor shaft misalignment and Imbalance of the synchronous generator are two of the most common failure modes associated with the synchronous generator.

The statistical data of failures among generators indicates that over half of all machinery problems are caused by misalignment and almost half of all machinery problems are caused by imbalance.[1]

The other machine faults include mechanical looseness or improper fit between component parts, electrical faults measured as vibration, bearing cocked on a shaft and bearing defects, etc.



Condition Monitoring Systems are available to assess the condition of the synchronous generators and to identify indications of generator problems and hence to determine the most effective time to schedule the maintenance.

## 1.2 Condition Monitoring Tools

The principle of a Condition Monitoring System is straightforward:

- The goal is to identify changes in the condition of a machine that will indicate some potential failure.
- Physical characteristics are identified that collectively indicate the present condition of the machine.
- Each of these characteristics is measured, analyzed, and recorded so that trends can be recognized.

Among the Condition Monitoring Tools, which are used in the industry followings are at the top: [4]

- *Visual Inspection* - The most preliminary form of condition monitoring is visual inspection by experienced operators and maintainers. Failure modes such as cracking, leaking, corrosion, etc can often be detected by visual inspection before failure is likely. This form of condition monitoring is generally the cheapest. Consequently, other forms of condition monitoring should generally augment, rather than replace, visual inspection.

- *Vibration Analysis* – the analysis of the unique patterns of vibration created by specific components of an electro mechanical system. These unique patterns can provide early indications of problems such as machinery imbalance, misalignment, bearing wear, worn gears, etc.



- *Temperature Analysis* – identifying abnormal temperature variations across the machinery surface can be discovered with non-destructive testing with thermographs. That could result from problems such as bearing wear, lack of lubrication, degrading electrical contacts and terminations, etc.

- *Oil Analysis* – analyzing the physical and chemical properties of lubrication to identify contamination, wears particles, as well as changes in viscosity and other chemical properties. Controlling these critical parameters is the key to preventing premature failure of mechanical systems such as bearings, gears, etc. For example high silicon content indicates contamination of grit etc, and high iron levels indicate wearing components.

- *Performance analysis* - where the physical efficiency, performance, or condition is found by comparing actual parameters against an ideal model. Deterioration is typically the cause of difference in the readings. Generators and motors are arguably the most common machines.

### 1.3 Vibration Analysis

In electric machinery vibration analysis can determine misalignment, unbalance, mechanical looseness, eccentric rotors, bearing wear, loose rotor bars, and poor end turn connections. Vibration analysis can be applied to all rotating equipment, (from less than 1 rpm to 10000 rpm and above).

Many vibrations are normal for rotating or moving machinery, e.g. normal rotation of shafts and other rotors, contact with bearings etc. Specific problems with machinery generate abnormal identifiable vibrations, e.g., misaligned shafts, Imbalance, loose bolts, worn bearings, leaks and incipient metal fatigue.



Key to using vibration signature analysis for predictive maintenance is ability to differentiate between normal and abnormal vibration profiles. Predictive maintenance using vibration signature analysis is based on the following:

- All common machinery problems and failure modes have distinct vibration frequency components that can be isolated and identified.
- Frequency-domain signature is generally used because it contains discrete peaks, each representing specific vibration source.
- There is a cause for each frequency component.
- When the machine signature is compared over time, it will repeat until some event changes the vibration pattern.

To monitor the dynamic operating conditions of these SGs (in addition to the maintenance procedures performed in routine) a vibration monitoring system is being used.

**1.4 Thesis Objective & Outline**

In electric machinery vibration analysis can determine lot of electrical and mechanical faults. A bearing that has a small developing defect will cause a telltale change in the machine vibration level. Vibration analysis, properly applied, allows detecting small developing mechanical defects long before they become a threat to the integrity of the machine.

The main objective of this is to identify and verify the vibration analysis results to detect the SG faults at small developing mechanical defects long before they become a threat to the integrity of the machine. This analysis results helps to identify misalignment, imbalance, mechanical looseness, eccentric rotors, bearing wear, loose coupling, and rotor bars of Generators.



Thus it helps to provide the necessary lead-time to schedule maintenance to suit the needs of the plant management. Numerous studies, such as those conducted by the Electric Power Research Institute (EPRI), have shown that on average, the cost to industry for maintenance will be reduced by more than 50% if a predictive maintenance program is used instead of run-to-failure.

Procedure:

Following guidelines are used to achieve the final goal,

a) Generator fault identification through mechanical vibration monitoring.
   - Analysis of the velocity vibration FFT spectrum.
   - Deriving a relationship between critical vibration frequencies and the mechanical faults.

b) Synchronous Generator fault identification and verified by using an experiment from machine model with originated faults to identify FFT patterns at these faults.

c) Compare obtained results from case study and experiment with the standards of vibration analysis.

d) Formulation of a fault detecting scheme that indicate SG faults at early stage.

## 1.5 Advantages of vibration Monitoring System

**a) Avoiding expensive failures**

It can realize substantial savings if equipment is shut down for repairs before a major failure occurs. Repair costs can very well triple if, for example, a generator is permitted to run to failure - versus a planned replacement of bearings found to be worn prior to that same failure.



Since vibration analysis addresses all moving parts of any type of rotating equipment, it can identify not only specific machine faults, but can also simplify repairs by identifying the root cause of the problem. Most importantly, vibration analysis is capable of identifying problems long before they become noticeable. This early warning of emerging equipment faults allows the engineering staff to plan for repairs, to order parts, and to conduct those repairs at convenience, rather than under emergency conditions.

Such a predictive and pro-active style of maintenance not only avoids the unnecessary high costs involved with emergency repairs, but prevents potentially disruptive failures in critical plant operations.

**b) Minimize spare parts Inventory**

The proper implementation of a vibration analysis system allows a reduction in spare part inventory, and to establish "*Just-In-Time*" inventories without adding risk. Parts can then be ordered when problems first arise - instead of maintaining an expensive and redundant parts inventory, and without the risk of installing new replacement parts whose shelf life may have been exceeded.

**c) Energy savings**

The loss due to wastage attributed to the misalignment and imbalance of rotating machines can be saving with energy being converted into heat rather than output power.

**d) Other significant benefits**

A thorough vibration analysis program will also provide the following benefits:

- ***Transition from "Run to Failure" to pro-active style maintenance***
- ***Increased operating reliability***
- ***Increased mean time between failure (MTBF)***
- ***Elimination or lessen unscheduled equipment downtime***
- ***Improve maintenance management***
- ***Lower utility bills***

# Chapter 2
# PROJECT IMPLEMENTATION

## 2.1 Monitoring System Outline

A complete block diagram of the proposed monitoring system is shown in figure 2.1. At the baseline measurement stage, the Synchronous Generator is equipped with an accelerometer for the vibration measurements.

As described in the next section, instrumentation, above task is done with the help of the Microlog CMVA60 instrument, which gives its output as frequency spectra. The vibration spectrum is then evaluated to determine critical frequencies.

The critical frequencies ($f_{v,k}$) ; k=1, 2, 3 .... are defined as those frequencies which lie significantly above the noise floor.



Let, $V(f_{v,k})$ be the magnitude of the vibration at frequency $f_{v,k}$.

The set of ratio constants, $m_k$, is then calculated for each critical frequency, as the ratio of present magnitude to the previous magnitude of the vibration.

$$m_k = V(f_{v,k})_{current} / V(f_{v,k})_{previous} \quad (2.1)$$

Vibration harmonics are scaled by frequency to account for the fact that vibration is measured as a velocity.

The condition monitoring of the generator then involves sampling the vibration, calculating its spectrum and then determining the ratio constants, $m_k$ of critical frequency components of the spectrum.

The values of $m_k$ can be used as an indication for the generator health condition. The experimental results and the existing standards can then be used as a starting point in defining the limit on $m_k$.

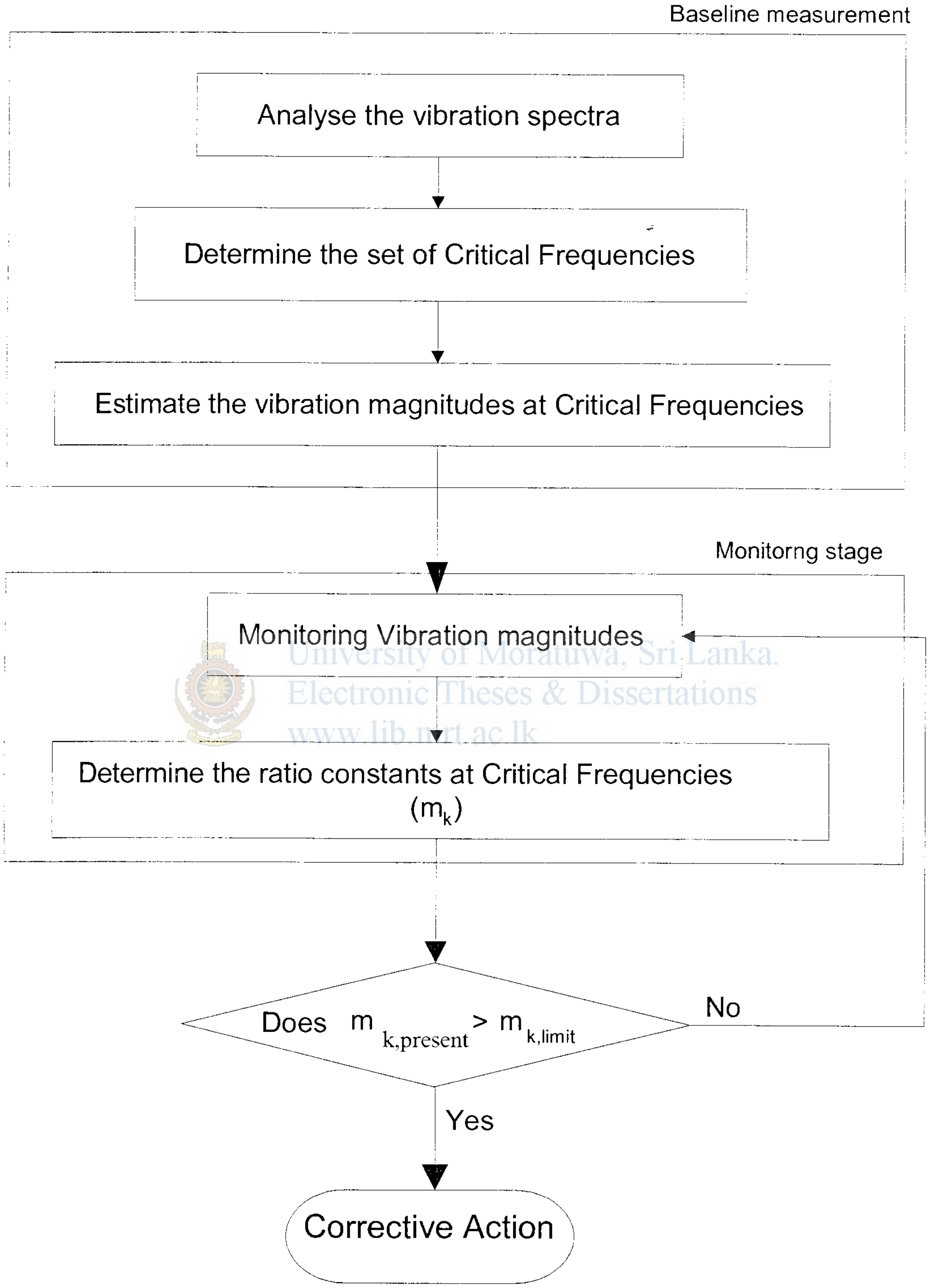



Figure 2.1: Block diagram of the proposed system

## 2.2 Instrumentation

The equipment setup for measuring the Synchronous Generator vibration is shown in Figure 2.3. The data collected by the accelerometer is fed to the instrument called Microlog CMVA60. Then, by using the Software called Prism 4, accompanied by the Microlog instrument is used for analyzing the data.

### 2.2.1 The Accelerometer SKF CMSS2200- Data acquisition equipment:

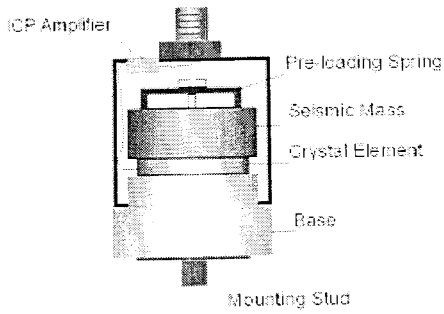



Figure 2.2: Accelerometer



The piezo-electric accelerometer can be considered the standard vibration transducer for machine vibration measurement. The compression-type accelerometer, diagrammed in Figure 2.2, was the first type to be developed. The principle of operation of this type of accelerometer is illustrated as follows.

The seismic mass is clamped to the base by an axial bolt bearing down on a circular spring. The piezo-electric element is squeezed between the mass and the base. When a piezo-electric material experiences a force, it generates an electric charge between its surfaces.

When the accelerometer is moved in the up and down direction, the force required to move the seismic mass is born by the active element. According to Newton's second law, this force is proportional to the acceleration of the mass. The force on the crystal produces the output signal, which is therefore proportional to the acceleration of the transducer.

The frequency range of the accelerometer is very wide, extending from very low frequencies in some units to several tens of kilohertz.
Refer Data Sheet of Accelerometer SKF CMSS2200 in Appendix 1.

### 2.2.2 Microlog CMVA 60 and Prism 4 Database

The vibration monitoring system for the condition monitoring of the synchronous generator is governed by the Prism 4 Pro software and the data, collected by the instrument, SKF Microlog CMVA 60. Advanced signal processing techniques in the Micrologextract the modulating frequency and clearly represent the amplitude relationship of modulating frequency to the line frequency.

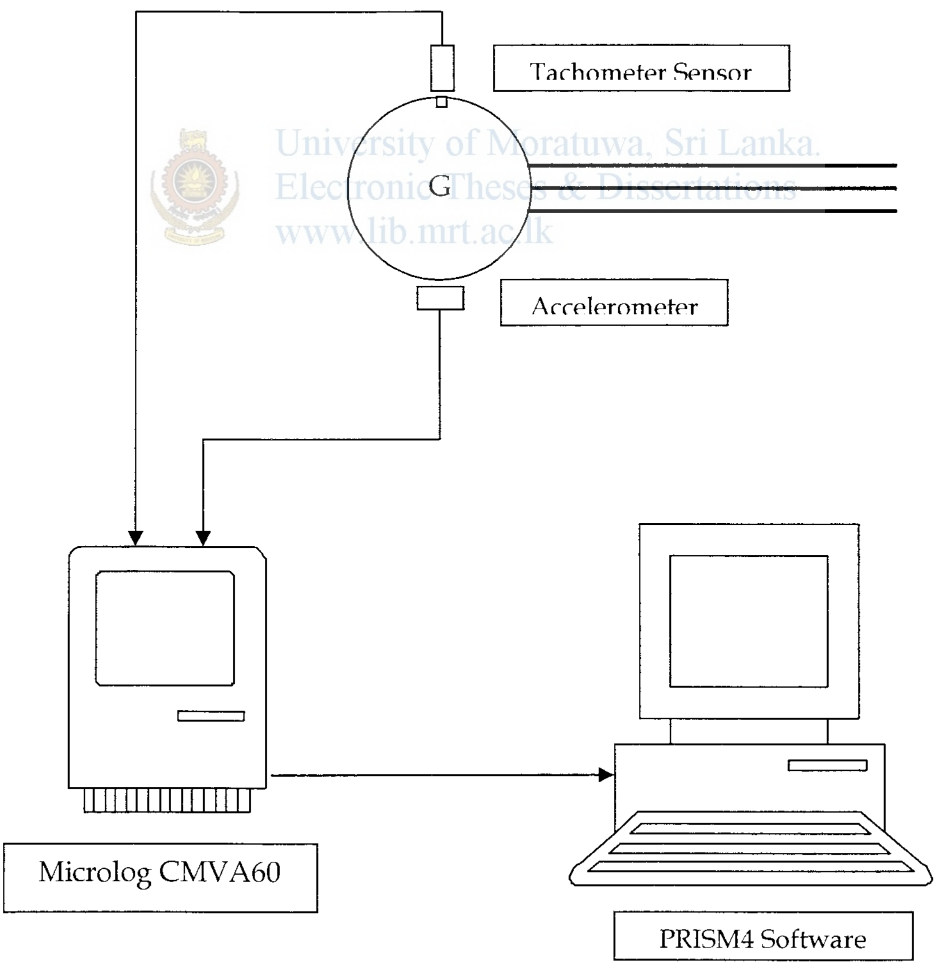





Figure 2.3: Data Acquisition System

Use of the Microlog CMVA60 and the software for monitoring has the following advantages:

- The Microlog, without interrupting the system can perform testing on-line.
- Prism 4 combines mechanical and electrical measurements to give the edge to root cause analysis.
- The analysis needs no mathematical computations since the vibration spectrum of displacement, velocity or acceleration is a direct output of the prism 4-software where FFT is used.

Refer Data Sheets of used Vibration monitoring instrument SKF Microlog CMVA60 and accessories in Appendix 2 and Data Sheet of Prism 4 Software in Appendix 3.

# Chapter 3
# SYNCHRONOUS GENERATOR VIBRATION MONITORING SYSTEM

## 3.1 Vibration monitoring system

Vibration is simply the movement of a machine or machine part back and forth from its position of rest. Further, it is the response of a system to some internal or external force applied to the system.

Analysis of system and equipment vibration levels is one of the most commonly used Condition Monitoring techniques. Vibration monitoring helps determine the condition of rotating equipment and structural stability in a system. It also helps identify noise sources.



Mechanical vibration is considered the best operating parameter to judge Synchronous Generator (SG) dynamic conditions such as, [1]

- Imbalance
- Misalignment
- Mechanical looseness
- Bearing faults
- Bent shafts

Vibration monitoring instrumentation typically uses piezoelectric accelerometer as a transducer/sensor, which produces a voltage proportional to the force to which it is subjected and it should be permanently affixed to the generator being monitored.

Preferred locations for measuring (structure borne) noise levels on installed machinery have evolved over a period of approx. 30 years. In concept for horizontally mounted machinery, measurement should be taken in the horizontal and vertical planes.

To assist in the determination of machine problems, it is very helpful to have vibration data from each measurement point in three directions. These directions are called Axial, Radial, and Tangential. Axial is the direction parallel to the shaft in question, radial is the direction from the transducer to the center of the shaft, and tangential is 90 degrees from radial, tangent to the shaft.
Vibration sampling points are selected as shown in Appendix 4.

The vibration monitoring system for condition monitoring of the Synchronous Generators which is being implemented at Mahaweli and Laxapana Complexes power stations will be governed by the Prism 4 Pro software and the data collected by the instrument, SKF Microlog CMVA 60. The accelerometer, SKF CMSS2200 is used to measure the machine vibration. A typical arrangement of the instrument for measuring the vibration is shown in Figure 2.3.



Vibration data obtained by mounting transducers on the Generator at various locations, typically Generator housing and bearing caps is shown in figure 3.1.

Figure 3.1: Vibration monitoring process

The portable type data-gathering device, Microlog SKF Condition Monitoring Instrument, which is capable of acquire vibration data, convert to FFT (fast Fourier Transform) and store and software - PRISM4 for Windows 1.35.1 for analysis data in any form of displacement, velocity or acceleration is shown in Figure 3.2.




Figure 3.2: Vibration monitoring Instruments

The used accelerometer SKF CMSS2200, which responds to the acceleration of the vibration source, is shown in Figure 3.3.

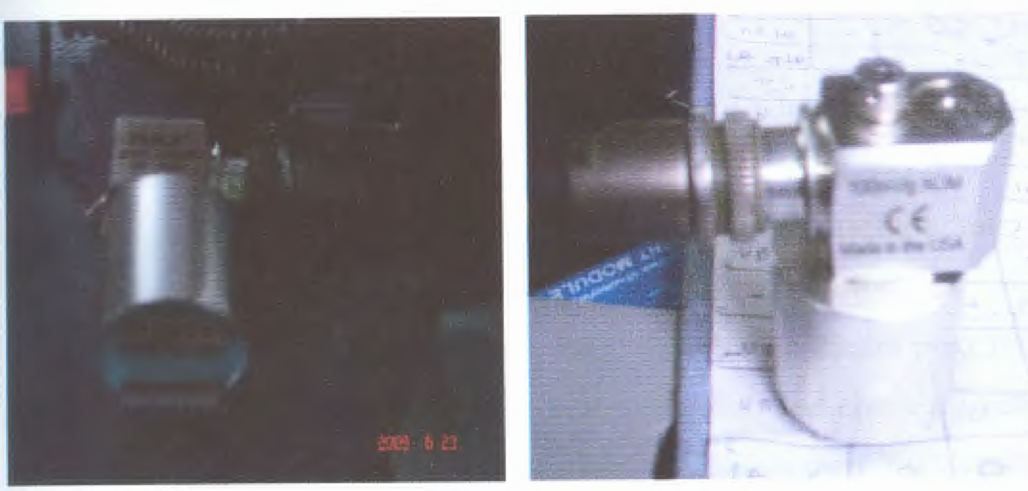



Figure 3.3: SKF CMSS2200 Accelerometer

## 3.2 Collect useful information

Correct diagnosis of rotating machinery mechanical faults depends on having complete information about the vibration spectral data. Therefore following facts should be considered for obtaining accurate and useful data for analysis.

### 3.2.1 Identify all components of the machine that could generate vibration.

Before a spectrum can be analyzed, the components that cause vibration within the machine must be identified. That means first we must check what are the possibilities as follows.

- If bearings are present, know their bearing default frequencies.
- Is the machine operating in the same vicinity as another machine, if so, know the running speed of the adjacent machine. Vibration from one machine can travel through the foundation or structure and affect vibration levels on an adjacent machine.
- Is the machine mounted horizontally or vertically?
- Is the machine overhung, or connected to anything that is overhung?



### 3.2.2 Identify the Machine's Running Speed

Knowing the machine's running speed is critical when analyzing an FFT spectrum. There are several ways of determining running speed.

Read the speed from instrumentation at the machine or from instrumentation in the control room monitoring the machine.

An FFT's running speed peak is typically the first significant peak reading the spectrum from left to right. Look for this peak and check for peaks at two times, three times, four times, etc. The suspected running speed frequency (2X, 3X, 4X). Harmonics usually cause vibrations at multiples of the running speed frequency.

### 3.2.3 Identify what type of Measurement produced the FFT spectrum

Determine the type of measurement that produced the spectrum such as displacement, velocity, acceleration, enveloping, etc.

### 3.2.4 Selection of test point locations

In general, it is desirable to locate the test transducer as close as possible to the bearing with solid metal between the bearing and the sensor. Avoid bearing caps, which are of thin metal and are thus poor conductors of vibration energy. If possible, pick test point locations so that there is no metal-to-metal joint between the bearing and the sensor.

### 3.2.5 Obtain any historical Machinery Data

Find out whether there are previously recorded values, FFTs or overall trend plots available of the machine and check whether was a baseline recorded previous occasions of monitoring.



## 3.3 Analysis of the Vibration Spectrum for Generator Fault Identification

Once the above information is known, we can proceed to analyze the spectrum. Analysis usually follows a process of elimination. Eliminate what is not on the spectrum and what is left is the problems.

### 3.3.1 Once Running Speed is Determined, Identify the Spectrum's Frequency Ranges

- Identify any harmonics of running speed (1X, 2X, 3X, etc.).
- Identify bearing fault frequencies.
- Identify adjacent machinery vibration, if applicable.

### 3.3.2 Verify Suspected Fault Frequencies

The spectra may produce peaks at identified fault frequencies. These peaks may or may not represent the indicated fault. Observe for harmonic to determine if the identified frequencies were generated from the indicated fault.

- If peak appears at the fundamental fault frequency and another peak appears at two times the fundamental fault frequency, it is a very strong indication that the fault is real.
- If no peak appears at the fundamental fault frequency but peaks are present at two, three, and may be four times the fundamental fault frequency, then this also represents a strong indication that the indicated fault is valid.

### 3.3.3 Determine the Severity of the Fault

- One way to determine the fault's severity is to compare its amplitude with past readings taken under consistent conditions.
- Another way is to compare the amplitude to other readings obtained by similar machines running under the same conditions. A higher than normal reading indicates a problem.



## 3.4 Determine the Generator mechanical faults

The vibration signal is analyzed in the frequency domain. That is, the amplitude of the signal's frequency components is plotted against the respective frequencies.

The first method for analysis is comparison of the RMS value of the vibration, given as a vibration signal with a vibration standard such as the international ISO 2372, the German VDI 2056 or the British BS 4675. These recommended running vibration standards have been developed using the extensive statistical base on machinery failures and are used to give an indication of overall health.

Refer Vibration standards VDI 2056 in Appendix 5 and ISO 2372 in Appendix 6.

Then it was used a machine model with developed faults to identify FFT patterns at various mechanical faults.

After that several case studies were carried out to identify several bearing faults of the CEB generators by using vibration database of machine condition monitoring.

### 3.4.1 Misalignment

Misalignment is created when shaft, couplings and bearings are not properly aligned along their centerlines. About 50% of the machine problems are due to misalignment. There are two types of misalignments:

**Angular misalignment**

This occurs when two shafts are joined at a coupling in such a way as to induce a bending force on the shaft.

- Angular misalignment causes axial vibration at fundamental frequency (1x). Figure 3.4 is shown the axial direction vibration frequency as same as in the 1x running frequency.



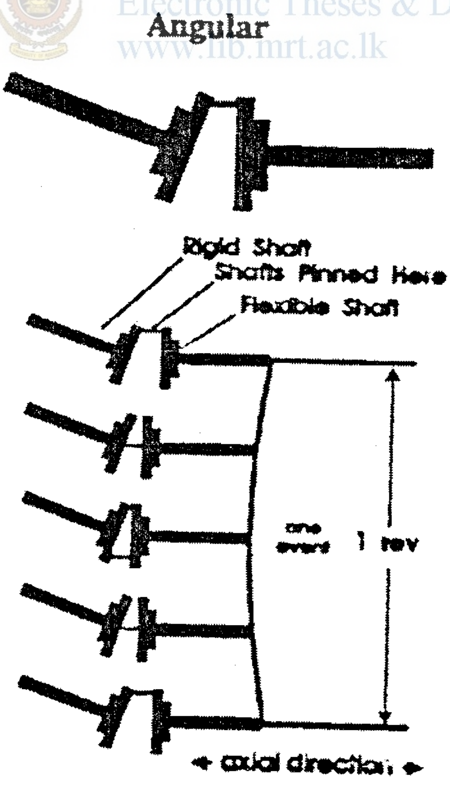



Figure 3.4: Angular Misalignment

**Parallel misalignment**

This occurs when the shaft centerlines are parallel but displaced from one another.

- Parallel misalignment causes Radial vibration at double the fundamental frequency (2x). This was illustrated in Figure 3.5 as double revelations on radial direction per one revolution of the shaft.

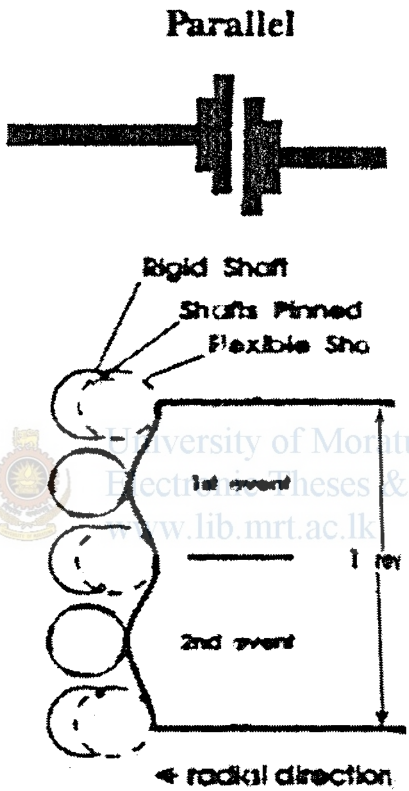





Figure 3.5: Parallel Misalignment

In parallel misalignment the vibration amplitudes at double the fundamental frequency ($A_{2X}$) can vary from 30% to 200% of that of the fundamental frequency ($A_{1X}$),

| | | |
|---|---|---|
| If | $(A_{2X})/(A_{1X}) < 50\%$ | : considered as acceptable |
| If | $50\% < (A_{2X})/(A_{1X}) < 150\%$ | : a coupling damage is probable |
| If | $150\% < (A_{2X})/(A_{1X})$ | : problem should be fixed as soon as possible [1] |

**Causes for misalignments**

- Thermal expansion due to a process working with heat. Since most machines are aligned cold, then as they operate and heat up, thermal growth causes them to grow misaligned.
- Machines directly coupled not properly aligned.
- Forces transmitted to the machine by piping and supports.
- Foundation uneven, shifting or settling.

Misalignment usually causes the bearing to carrying load than its design specification, which in turn causing failure due to fatigue. Fatigue is the result of stresses applied immediately below the load carrying surfaces and is observed as spalling of surface metal.

### 3.4.2 Imbalance



Imbalance occurs when the shaft's mass centerline does not coincide with its geometric centerline. Almost half of all machinery problems are caused by imbalance. There are three types of imbalance

Static Imbalance – Only one force is involved

Couple Imbalance – Two equal forces are $180^0$ from each other

Dynamic Imbalance – Combination of above two

Imbalance can be caused by a number of factors, including improper manufacture, an uneven build up of debris on the rotors, or the addition of shaft fittings without an appropriate counter balancing procedure. Key characteristics of vibration caused by imbalance are;

- Typical imbalance shows abnormally high vibration amplitude in the radial (vertical and horizontal) direction compared to that of the axial direction. [2]

- It is sinusoidal, occurring at a frequency of once per revolution (1X).
- The spectrum generally does not contain harmonics of 1X running speed, unless severe.
- Amplitude increases with speed up to the first critical speed of the machine.

The axial imbalance frequency, that is in line with the machine running frequency of 1x as shown in Figure 3.6.

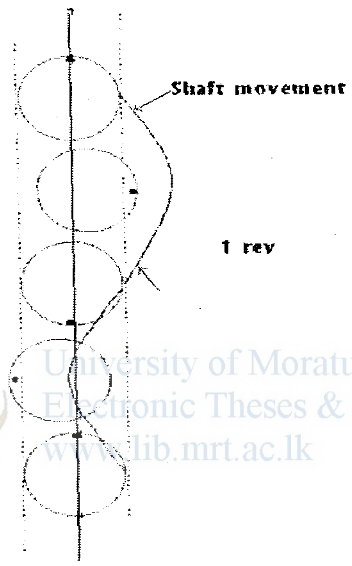





Figure 3.6: Imbalance

**Phase Analysis**

- Sensor shows, a phase shift of $90^0$ between the horizontal and vertical positions.

**Effects of Imbalance**

Imbalance usually causes the bearing to carry a higher dynamic load than its design specification, which in turn causes the bearing to fail due to fatigue. Fatigue is the result of stresses applied immediately below the load carrying surfaces and is observed as spalling away of surface metal.

### 3.4.3 Mechanical looseness

Mechanical looseness or the improper fit between component parts, is generally characterized by a long string of rotating frequency harmonics and half fundamental frequency (1/2 x) harmonics at abnormally high amplitudes.

i.e. 2x, 3x, 4x, …… or 3.5x, 4.5x, 5.5x, …….

- Range is from 2x to 10x
- Their magnitudes are greater than 20% of that of the fundamental

i.e. $A_{2x}$ , $A_{3x}$ , …… > 20% * $A_{1x}$

**Causes for looseness**

The major reasons for the mechanical looseness as follows:

- ❖ Machine has come loose from its mounting
- ❖ A machine component has come loose.
- ❖ The bearing has developed a fault which has worn down the bearing elements or the bearing seat.



**Effects of looseness**

If the looseness is bearing related, the effects are the same as imbalance, only more severe. If looseness is generated from a component, there is a possibility the part will become detached, causing secondary damage.

### 3.4.4 Bearing defects

Often bearing defect is not the source of the problem. It may be due to another fault like misalignment or imbalance.

Bearing defects occur at much higher frequencies with much lower amplitudes. The bearing defect frequencies should be calculated & over-laid on the vibration spectra. If those frequencies align with the peak amplitudes in the vibration spectrum, there is probably a bearing defect.

### 3.4.5 Bent Shafts

With overall vibration and spectral analysis, a bent shaft problem usually appears identical to misalignment problem. Phase measurements are needed to distinguish between the two.

A bent shaft would produce indications of motor imbalance and angular misalignment. Obviously, this test could be followed up by measuring the phase angles of the 1x vibration in each axis, measuring the shaft for runout and checking the alignment at the coupling.

**Causes for bent shaft**

- Cold Bow- As a result of gravity, a shaft with a high length to width ratio can, at rest, develop a bend.
- Improper handling during transportation.
- High torque.



As with imbalance, a bent shaft usually causes the bearing to carry a higher dynamic load than its design specification, which in turn causes the bearing to fail due to fatigue.

**Phase Analysis**

- Radial phase measurements typically appear "in phase".
- Axial phase measurements are typically $180^0$ out of phase.

***Note:***

It may sometimes be difficult to quantify the degree of severity of the fault based on the vibration spectrum at a sight because of the presence of resonance, unbalance and other unknown conditions and hence, the vibration measurements can be trended to determine the rate of rise of the degradation, which is a good indicator of generator's condition.

# Chapter 4
# MACHINE MODEL TO VERIFY VIBRATION SPECTRUMS OF THE GENERATOR FAULTS

## 4.1 Implementation of Machine Model

As shown in theory of the vibration spectrums when a misalignment , imbalance, mechanical looseness is present in a shaft or coupling of a generator, the frequency spectrum of the vibration will consist of specific frequency components with specified magnitudes.

The case studies of bearing faults also based on the standards and guidelines and there is no real observation for verify that results. Therefore, to identify and to verify the relationship of machine faults with vibration spectrum it is required to carry out an experiment by using machine model with developed faults.



In the case studies in Chapter 5 it was observed that this relationship of vibration spectra with generator faults. Then it is required to verify that relationship, constructed in theory was shown to be true with the experimental results.

It was used a single phase motor with 500rpm to verify these results. This motor is initially at healthy condition with coupling and balanced load wheel as shown in the Figure 4.1. Then it was tested for misalignment, imbalance and mechanical looseness with developed faults. Then vibration spectrums at each of these fault condition was analyzed.

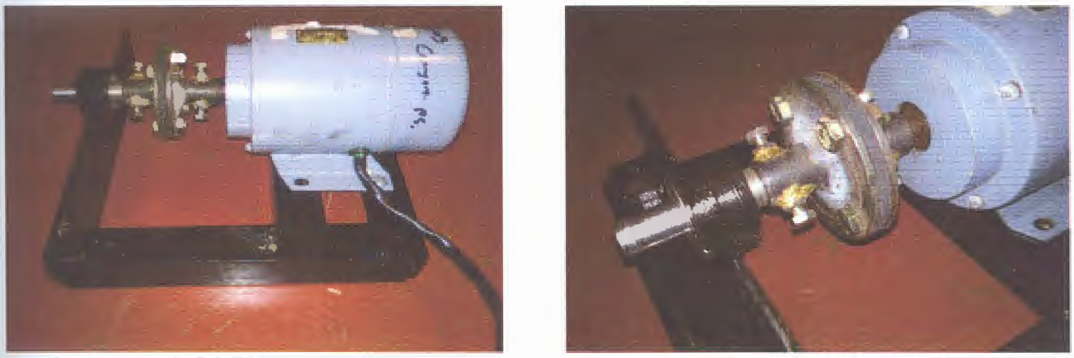


Figure 4.1: The experimental Machine Model

Initially, the machine model was tested at healthy condition to obtain velocity vibration spectrum of the bearing housing. The obtained vibration spectrum is shown in the Figure 4.2.

In this spectrum it was observed some harmonics that may be due to inherent problems or looseness of components of the model. That will not significantly effect for testing due to their lower magnitudes.

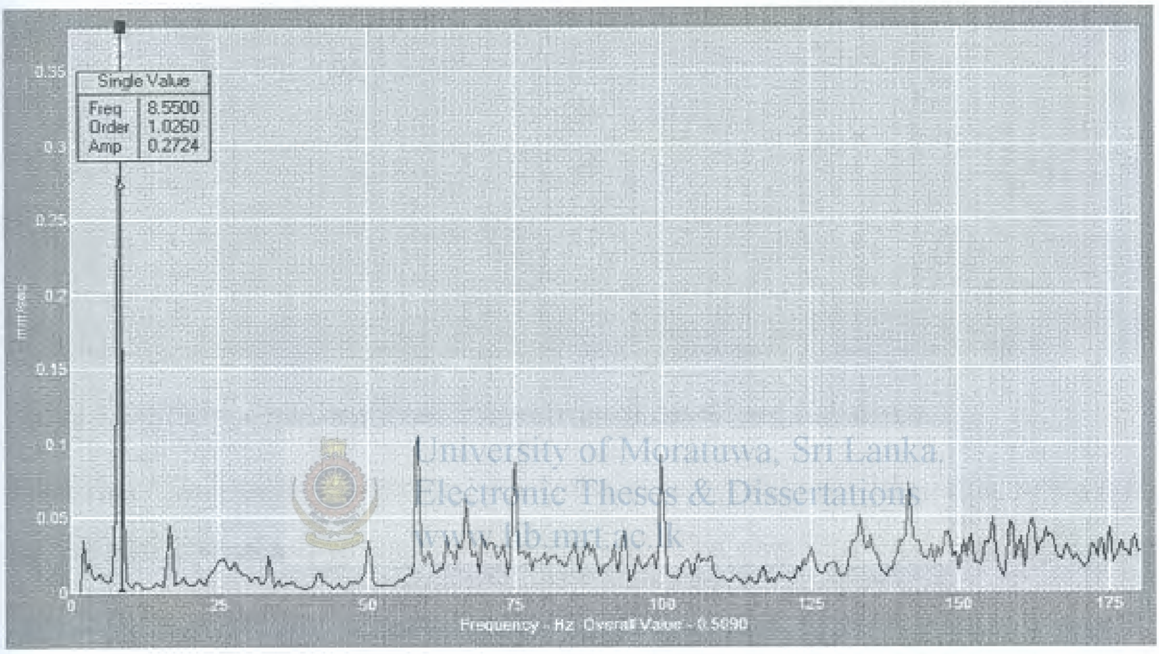




Figure 4.2: Machine model Vibration Spectrum at Healthy Condition

## 4.2 Imbalance Condition

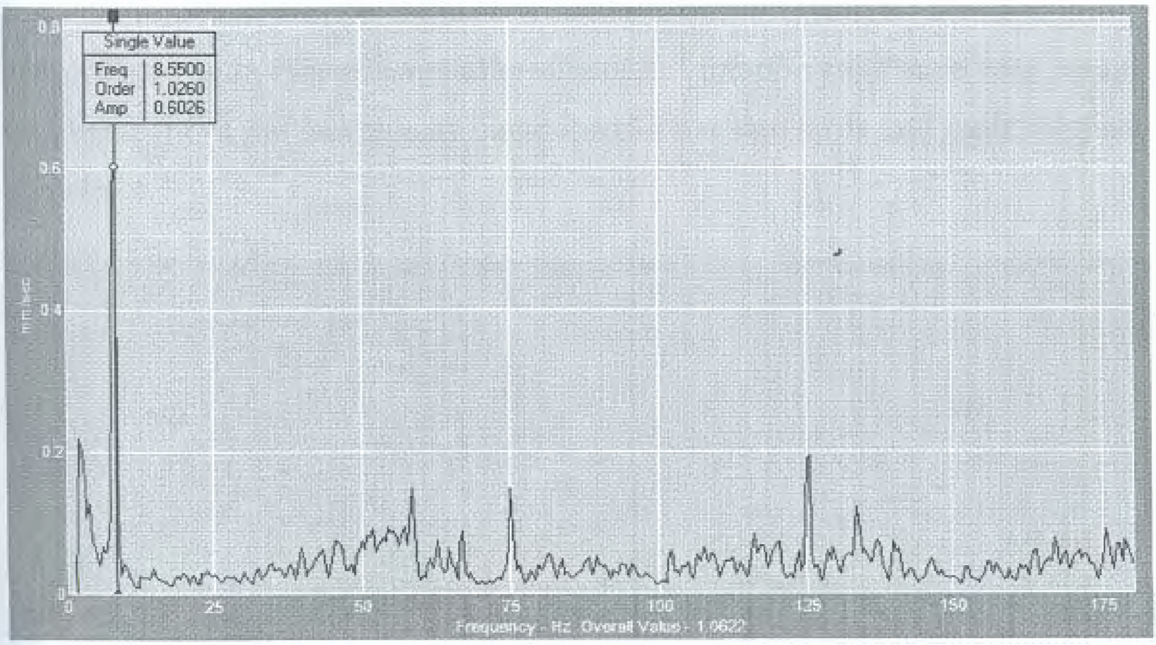



Figure 4.3: Machine model Vibration Spectrum at developed Imbalance



Then machine model was tested for Imbalance condition with using unbalance load wheel connected to the motor shaft end. The obtained vibration analysis is shown in the Figure 4.3.

When analyzing and compare these vibration spectrums at healthy condition and Imbalance condition it can observe the significant increase of 1x amplitude at Imbalance condition than healthy condition. This will prove that there is a strong relationship of 1x vibration amplitude with Imbalance as in the standards and guidelines of fault diagnosis.

### 4.3 Parallel Misalignment

Then machine model was tested for Parallel misalignment by coupling and mounting adjustments and vibration spectrum was obtained is shown in Figure 4.4 and compare with Figure 4.2 which is at healthy condition. When these two instances of spectrums results was analyzed and observed abnormally high 2x frequency component (137% of Fundamental frequency) at the instance of parallel misalignment.

This will prove that there is a relationship of 2x vibration amplitude with parallel misalignment of the machine as mentioned in the standards and fault diagnosis guidelines.

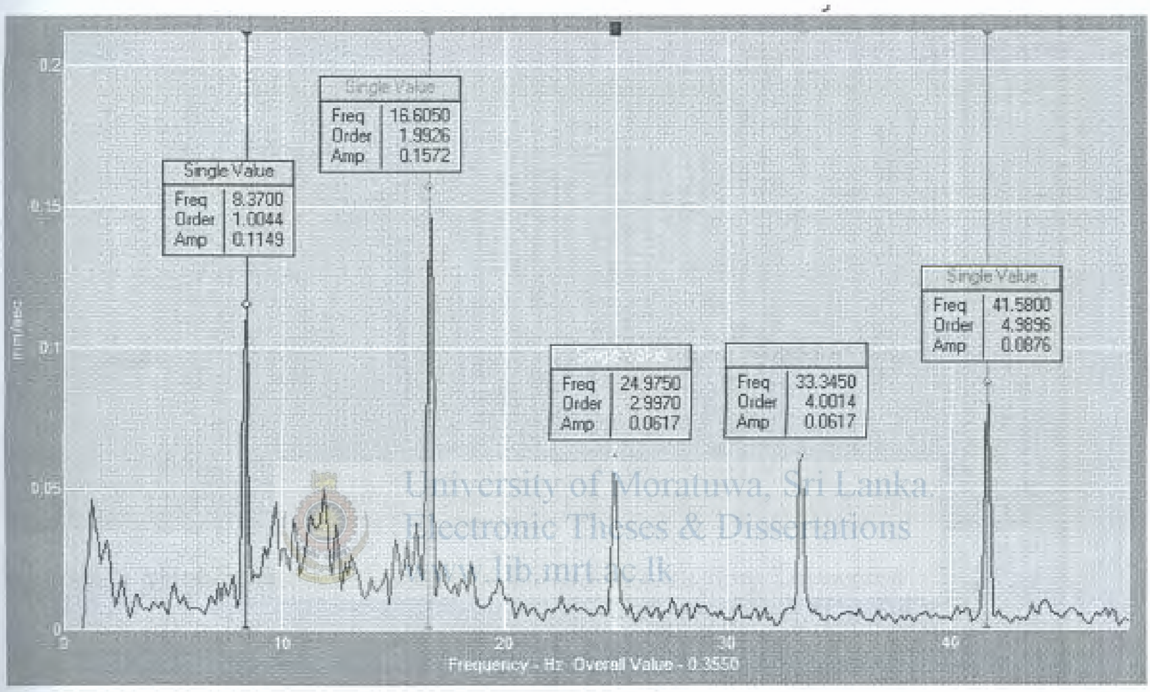





Figure 4.4: Machine model Vibration Spectrum at developed Parallel Misalignment

### 4.4 Coupling and Mounting Looseness

Then machine model was tested for coupling and mounting looseness by loosing coupling bolts and mounting bolts and vibration spectrum was obtained is shown in Figure 4.5, Figure 4.6 and Figure 4.7 for different looseness conditions of coupling and mountings of the model. When these instances of spectrums results was analyzed and observed there are a series of several synchronous multiples of running speed (range 2x to 10x) and their magnitudes are greater than 20% of the 1x in spectrum.

This will prove that there is a relationship of vibration amplitude patterns with mechanical looseness which was observed in case studies and mentioned in Standards and fault diagnosis guidelines.

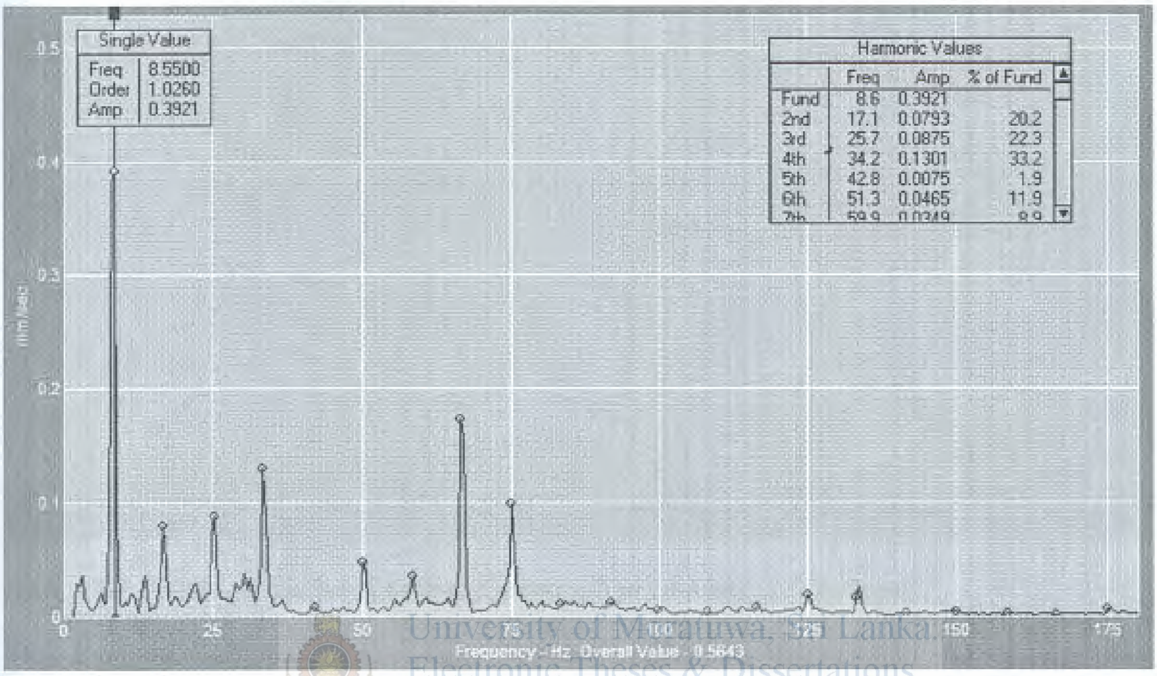





Figure 4.5: Machine model Vibration Spectrum at Coupling Looseness

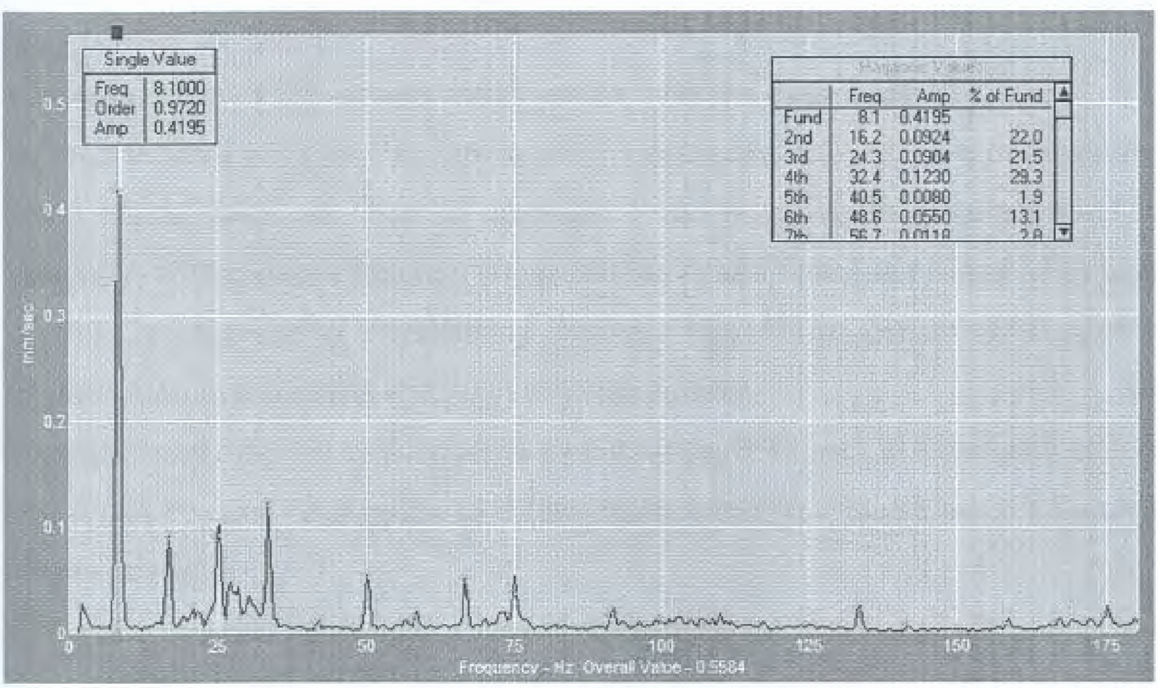



Figure 4.6: Machine model Vibration Spectrum at Coupling Looseness

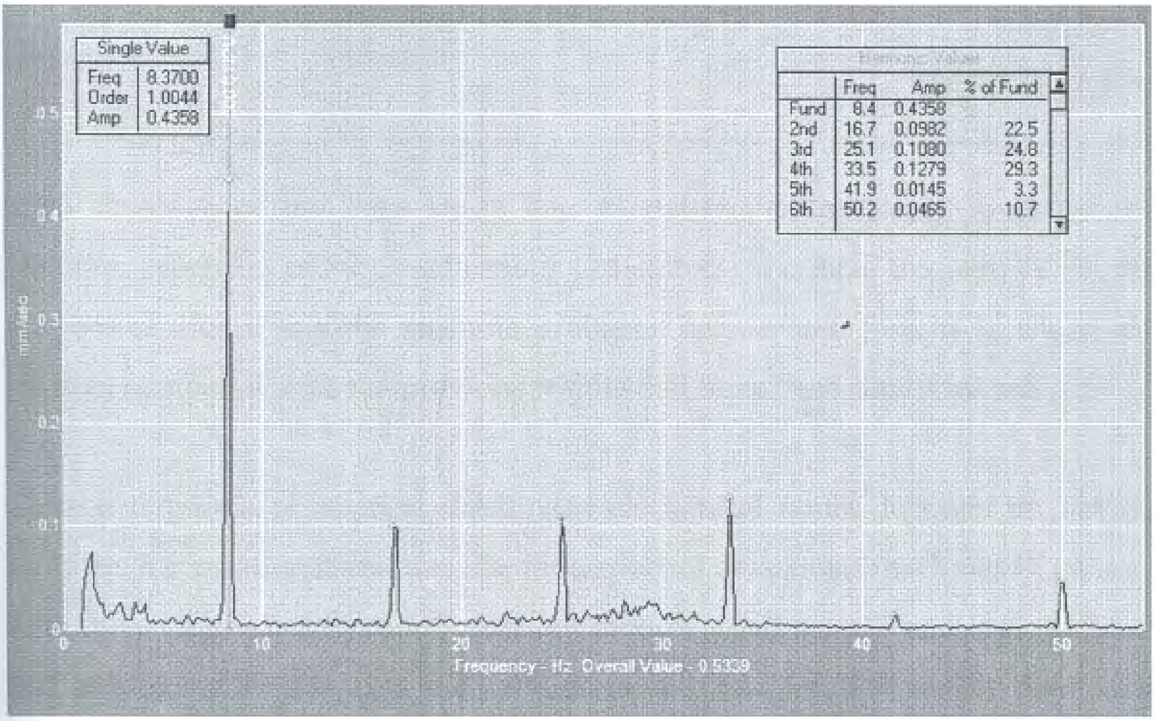



Figure 4.7: Machine model Vibration Spectrum at Mounting Looseness



These results can then be compared with the standard values to distinguish between a healthy generator and a generator with faults.

### 4.5 Comparison of the experimental results with the Standards

When compared with the VDI 2056 severity level charts & ISO 2372 & in appendix 5 and 6 and diagnosis chart in appendix 7 the faulty machine categorized is compared with the experimental results of above and which is proved to be true since it is observed a relationship between the faulty generator diagnosis identification in standards with experimental results.

Since the experimental results seem to follow the ISO and VDI severity charts and it can be used as a guide to differentiate between a faulty machine and a healthy machine.

Therefore, we shall conveniently differentiate the severity level charts and guide lines in diagnostic tables as the reference for assessing the generator condition.

# Chapter 5
# CASE STUDY

Main objective of this case study was to determine the safety limits of the vibration spectrum of the Synchronous Generators, at critical frequencies for the purpose of identifying the machine problems beforehand. Results of the study are then compared with the previous results that have been standardized.

Correct diagnosis of rotating machinery mechanical faults depends on having complete information about the vibration spectral data. Since machines in general have three degrees of freedom of lateral motion, good science and logic suggest that data from all three axes will provide more information, if we can analyze it properly.



The three orthogonal axes are designated axial, radial and tangential. As indicated by the term, the axial is the direction in line with, or parallel to, the shaft. Radial and tangential are the two perpendicular axes in the plane of rotation. Generally, radial is vertical for a horizontal machine while tangential is in the transverse direction. For a vertical machine, radial and tangential are both horizontal axes such that radial is toward the center of the shaft and tangential is as the name implies.

The used software can accommodate vibration displacement, velocity or acceleration in any unit of measurement; we generally measure machinery velocity vibration in V.

The spectral amplitudes of vibration are compared to average baseline data accumulated for each spectral peak from selected past test data for the specific machine type.

## 5.1 Measurements & Observations

### 5.1.1 Angular Misalignment

**Case 1: Randenigala Unit no 02 Shaft Vibration Analysis**

On 07th May 2003 Randenigala Unit 02 was tested due to abnormally high vibration of the shaft. The vibration spectrums were obtained from both axial and radial directions. The measured axial velocity vibration spectrum is shown in the Figure 5.1.

After examine the above spectrum details it was suspected that diagnosis could be angular misalignment, since 1x axial is abnormally high. It was decided that this couldn't be rotor imbalance, since 1x radial is normal and much lower than 1x axial. The axial vibration could be due to the rocking motion. The rocking motion can cause 1x axial vibration to be abnormally high as long as the axial amplitudes exceed of average value. This information was sufficient to diagnose the fault as an angular misalignment.



Having studied the above information coupling was suspected. Because it was carried out some repairs of the coupling by the maintenance people recently before the fault occur.

Then on 11th June and shaft coupling alignment was checked with a laser tool and alignment was properly adjusted. Then the axial velocity vibration spectrum was measured again after the repair and that is shown in Figure 5.2. After that it was observed shaft vibration is significantly reduced and reduction in axial vibration amplitude.

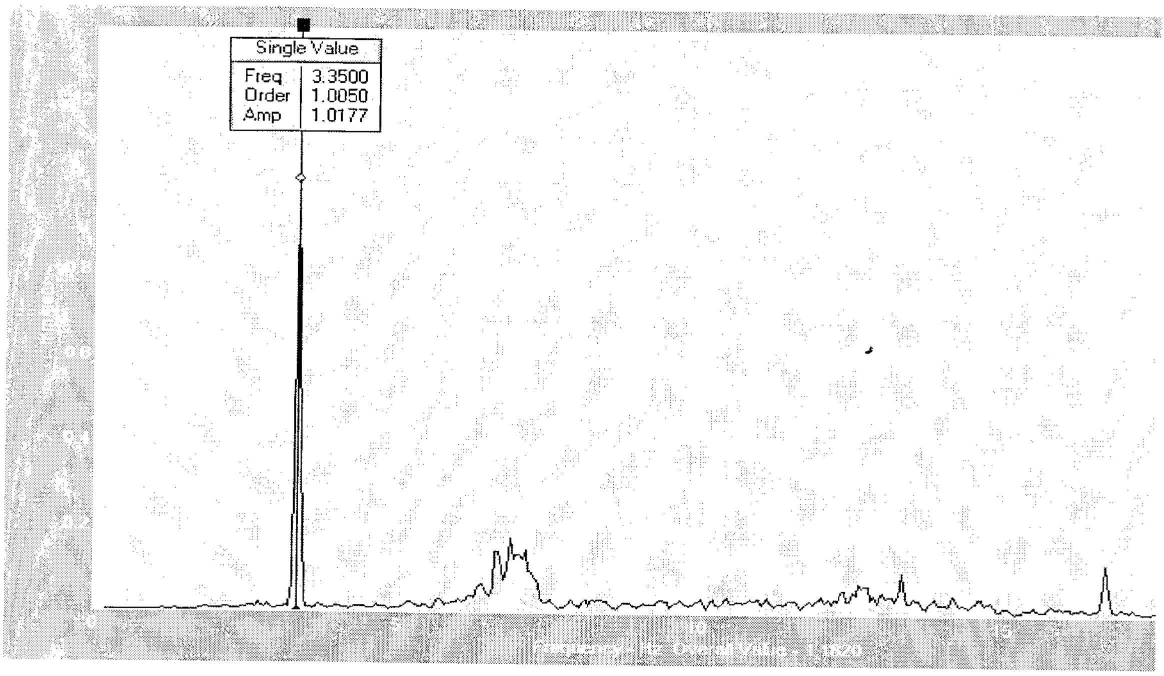



Figure 5.1: Axial Vibration Spectrum of Randenigala Unit 02 Upper Guide Bearing on 2003 May 07 (Before Repair)



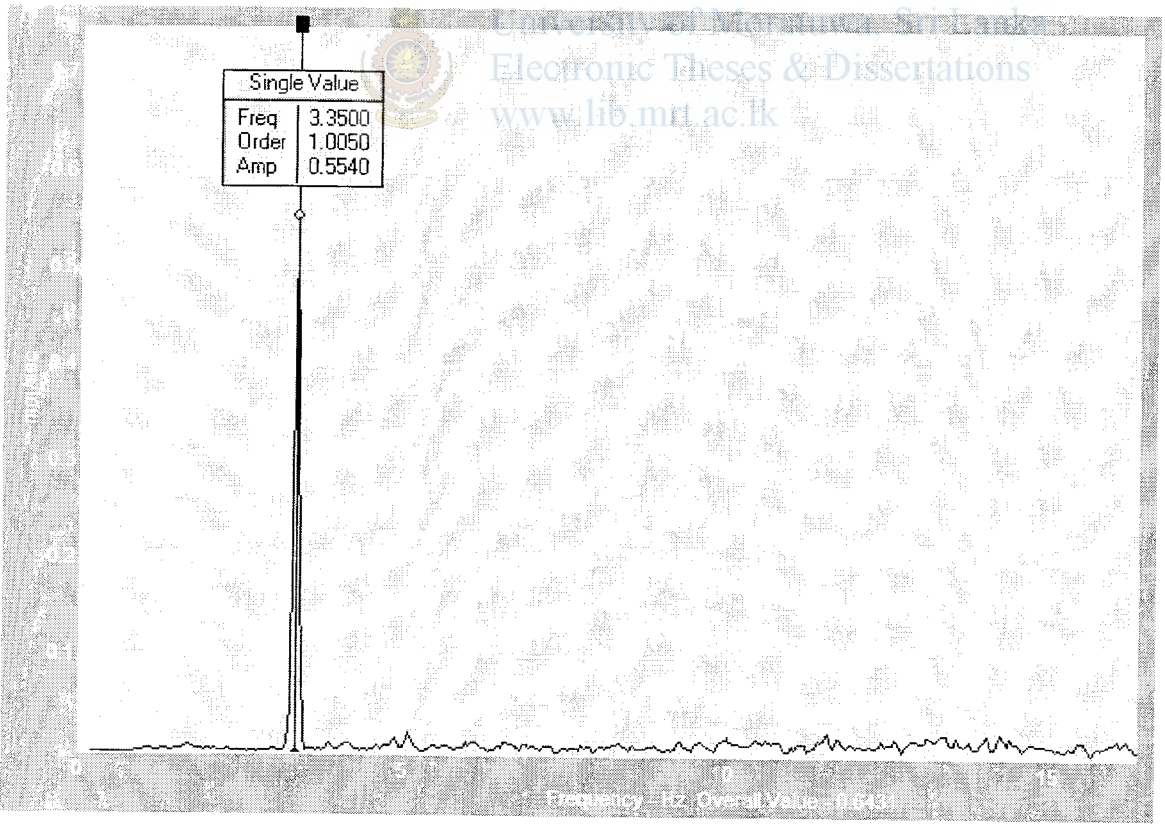



Figure 5.2: Axial Vibration Spectrum of Randenigala Unit 02 Upper Guide Bearing on 2003 June 11 (After Repair)

### 5.1.2 Parallel Misalignment

**Case 1: New Laxapana Unit 01 Vibration Analysis**

New Laxapana unit no 01 was exhibited high shaft vibration on 1st May 2005 and it was tested for analyze the vibration spectrums. The radial velocity vibration spectrums were measured from two directions from downstream side and Loading bay. These vibration spectrums are shown in the Figure 5.3 and 5.4.

| | Harmonic Values | | |
|---|---|---|---|
| | **Freq** | **Amplitude** | **% of Fund.** |
| **Fund** | 7.125 | 0.6153 | |
| **2nd** | 14.25 | 0.8882 | 144.4 |
| **3rd** | 21.375 | 0.1920 | 31.2 |

Table 5.1: New Laxapana Unit 01 Radial Vibration- Downstream



| | Harmonic Values | | |
|---|---|---|---|
| | **Freq** | **Amplitude** | **% of Fund.** |
| **Fund** | 7.125 | 0.6469 | |
| **2nd** | 14.25 | 0.5914 | 91.4 |
| **3rd** | 21.375 | 0.1308 | 20.2 |

Table 5.2: New Laxapana Unit 01Radial Vibration- Loading Bay

The harmonic analysis for vibration spectrum is shown in the Table 5.1 and 5.2. By examine these spectrum details it was evident that the data indicates the diagnosis could be parallel misalignment, since 2x radial frequencies are abnormally high (144% and 91.4%) in both measured directions and also present of high 3x.

Due to above fault symptoms shaft coupling was suspected. Coupling bolts were tightened after proper aligned by using a Laser Tool for rectification of the problem. Then the Radial velocity vibration spectrum was measured again after the repair and that is shown in Figure 5.5.

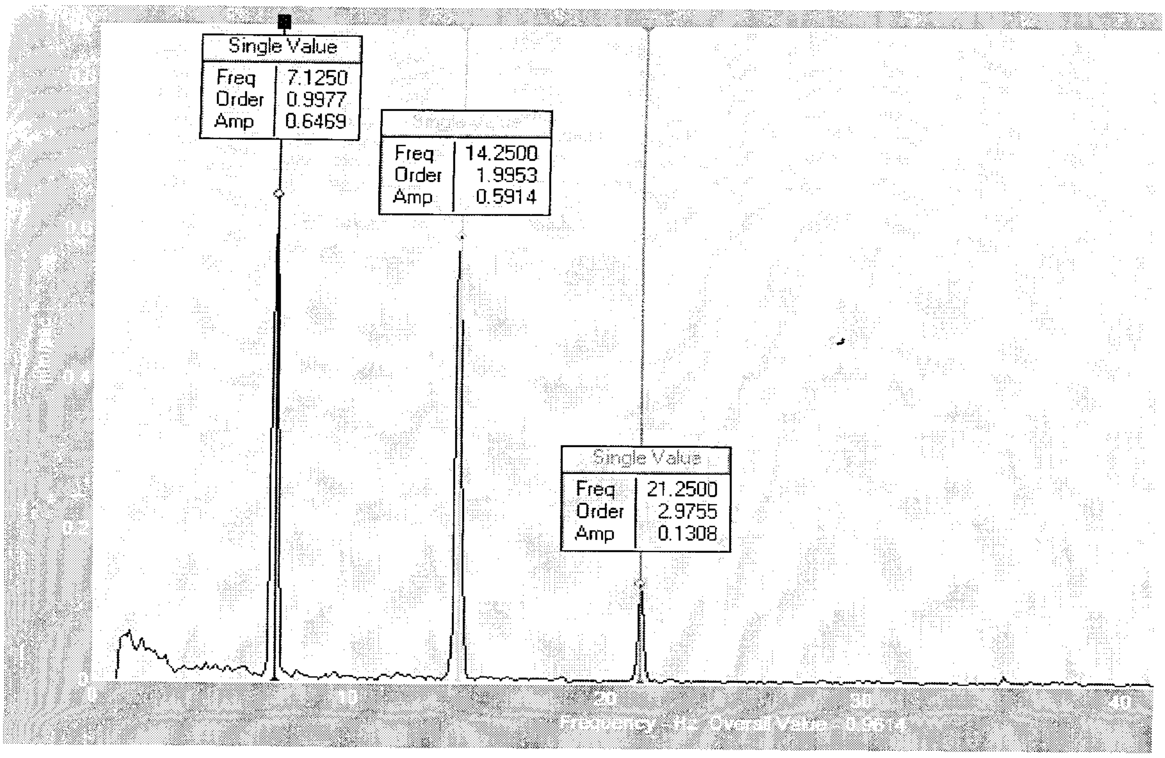



Figure 5.3: Radial Vibration Spectrum of New Laxapana Unit 01 Generator Upper Bearing Downstream side on 2005 May 01 (Before Repair)



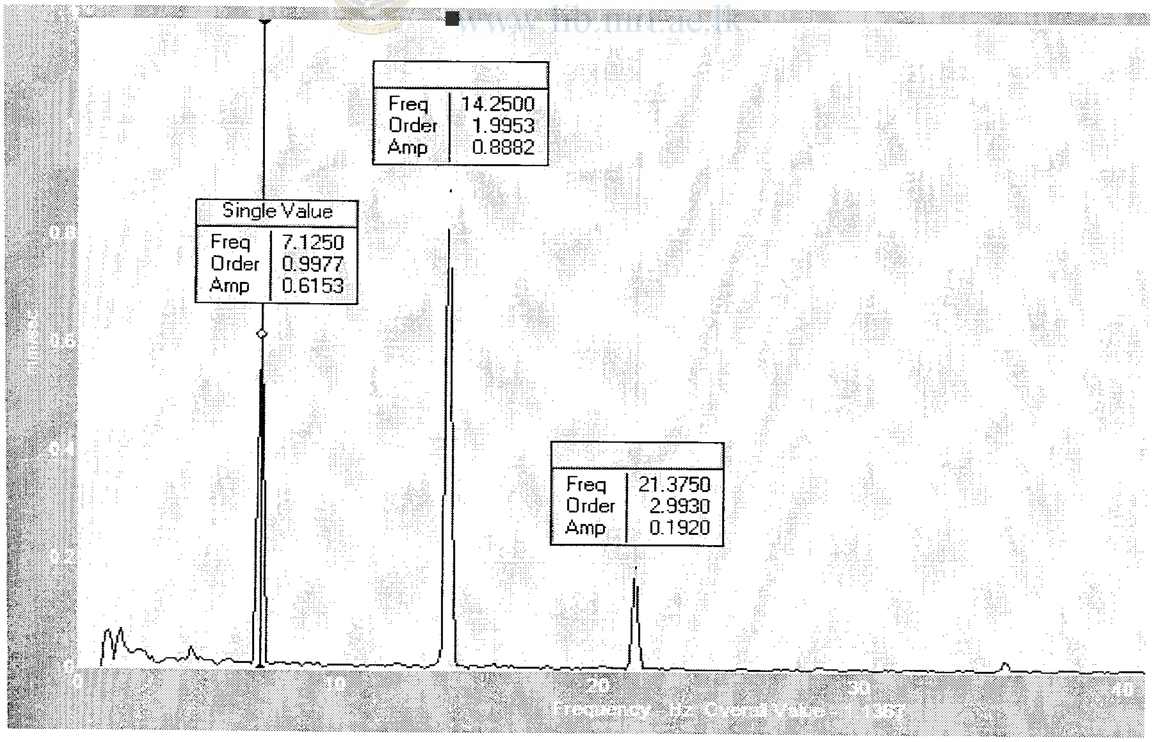



Figure 5.4: Radial Vibration Spectrum of New Laxapana Unit 01 Generator Upper Bearing Loading bay side on 2005 May 01 (Before Repair)

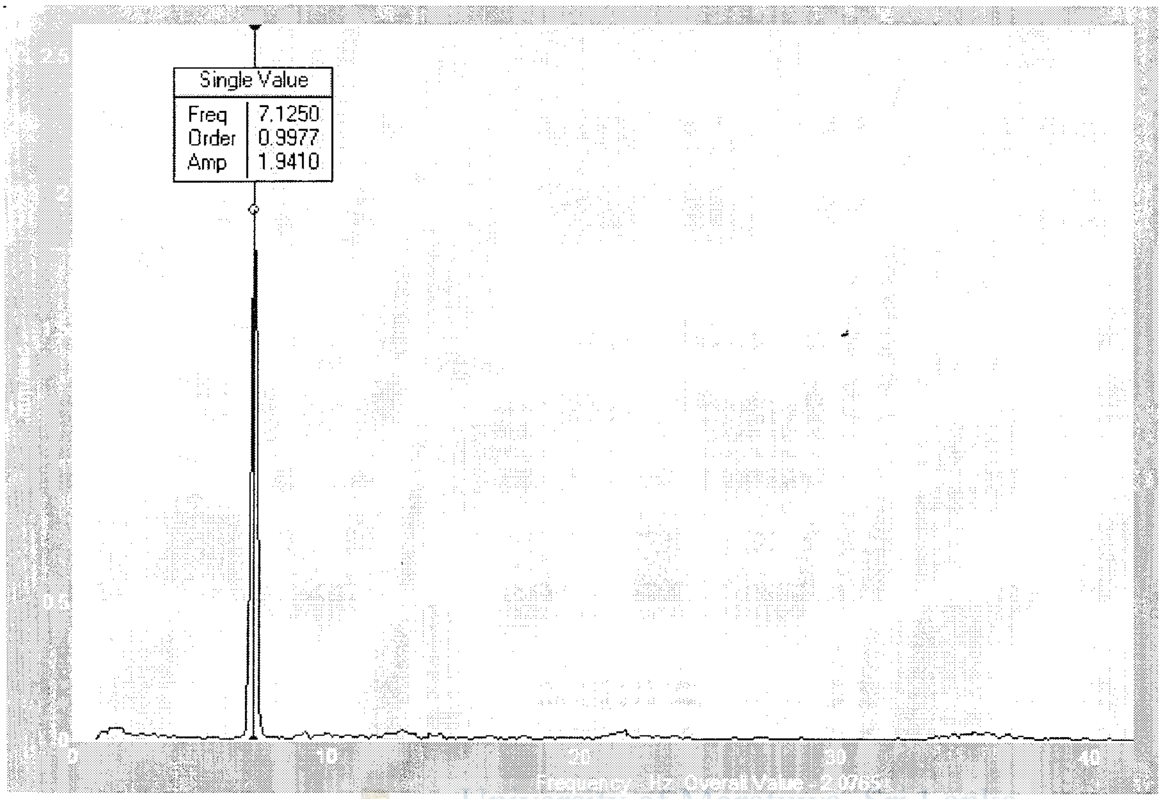





Figure 5.5: Radial Vibration Spectrum of New Laxapana Unit 01 Generator Upper Bearing Downstream side (After Repair)

**Case 2: Nilambe unit 2 Bearing on 2003 June 11**

Nilambe Unit 02 was exhibited high vibration on 11th June 2003, on Generator bearings & Thrust bearing when load increases from 1 to 1.6 MW.

So machine vibration spectrum was tested from both axial and radial directions and suspected locations of Coupling, Thrust & Guide bearings vibration were measured.

The radial velocity vibration spectrums were measured from two directions from downstream side and Loading bay for Guide bearings and Trust bearing. These vibration spectrums are shown in the Figure 5.6, 5.7 and 5.8.

The harmonic analysis for vibration spectrum is shown in the Table 5.3. 5.4 and 5.5 respectively.

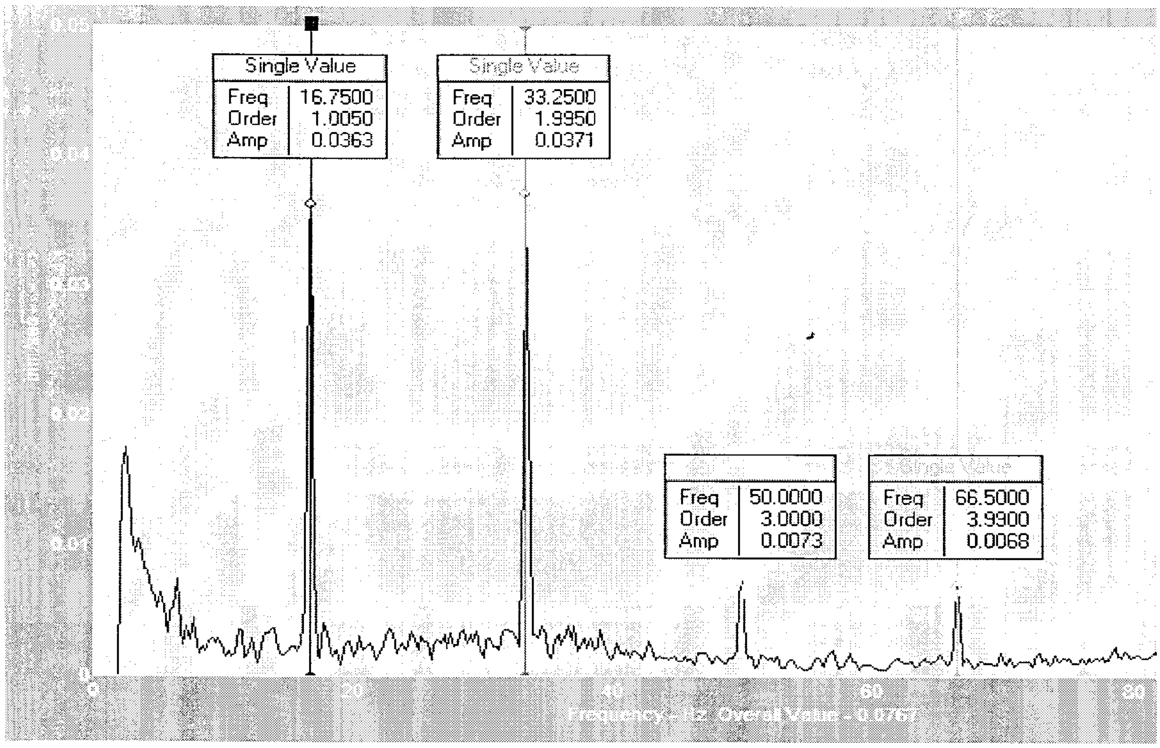



Figure 5.6: Radial Vibration Spectrum of Nilambe Unit 02 Bearing 01 Down stream side on 2003 June 11 (Before Repair)



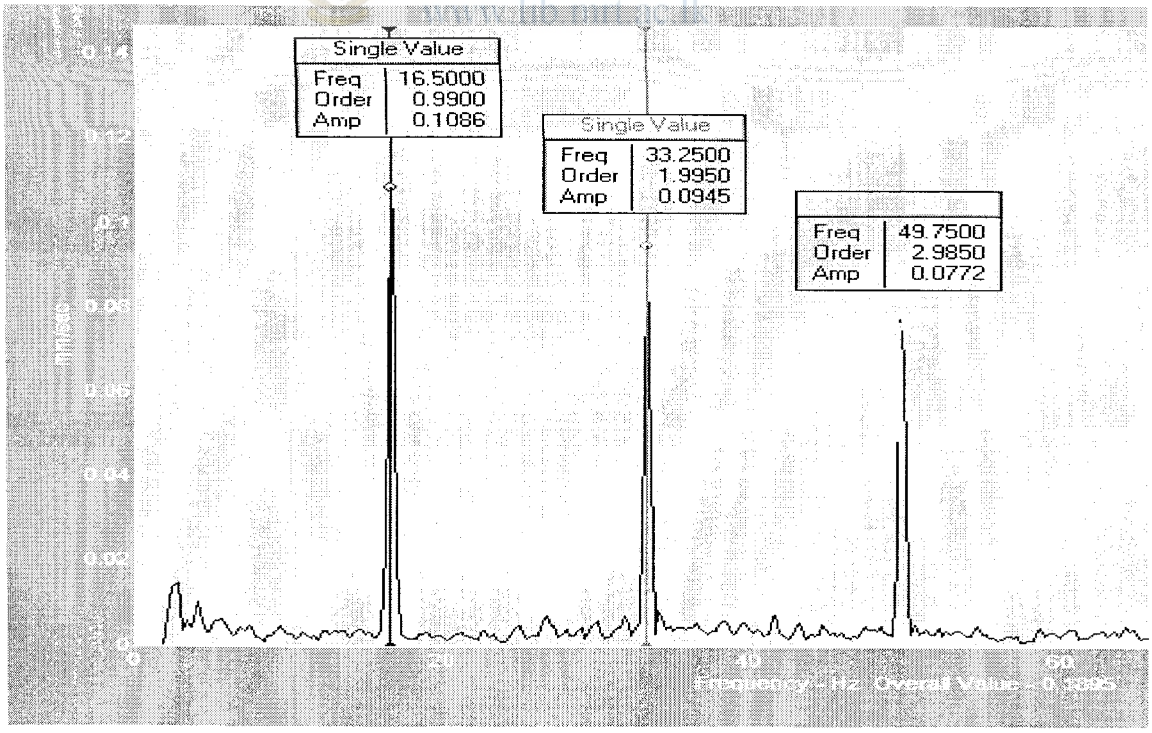



Figure 5.7: Radial Vibration Spectrum of Nilambe Unit 02 Bearing 01 Loading Bay side on 2003 June 11 (Before Repair)

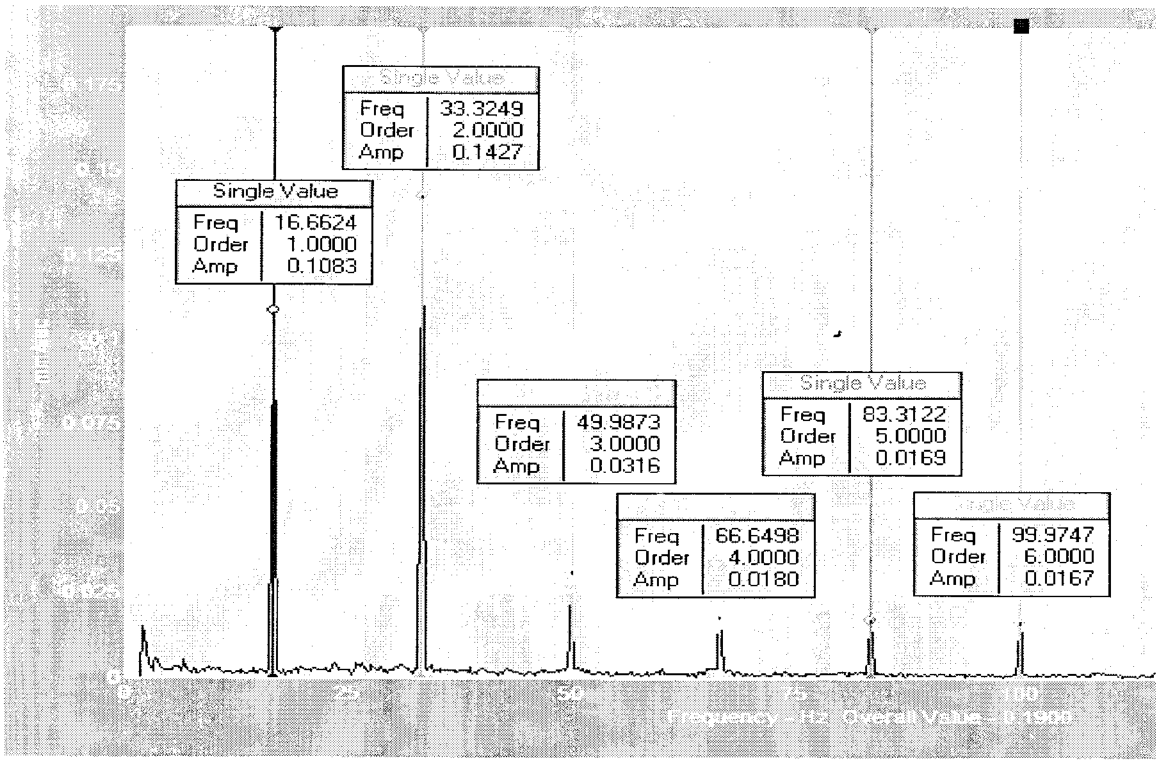



Figure 5.8: Radial Vibration Spectrum of Nilambe Unit 02 Trust Bearing Down stream side on 2003 June 11 (Before Repair)



| | Harmonic Values | | |
|---|---|---|---|
| | **Freq** | **Amplitude** | **% of Fund.** |
| **Fund** | 16.75 | 0.0363 | |
| **2nd** | 33.25 | 0.0371 | 102.2 |
| **3rd** | 50 | 0.0073 | 20.1 |
| **4th** | 66.5 | 0.0068 | 18.7 |

Table 5.3: Nilambe Unit 02 Bearing 01 Radial Vibration- Downstream side

| | Harmonic Values | | |
|---|---|---|---|
| | **Freq** | **Amplitude** | **% of Fund.** |
| **Fund** | 16.75 | 0.1086 | |
| **2nd** | 33.25 | 0.0945 | 87.0 |
| **3rd** | 50 | 0.0772 | 71.1 |

Table 5.4: Nilambe Unit 02 Bearing 01 Radial Vibration- Loading Bay side

| | Harmonic Values | | |
|---|---|---|---|
| | **Freq** | **Amplitude** | **% of Fund.** |
| **Fund** | 16.75 | 0.1083 | |
| **2nd** | 33.25 | 0.1427 | 131.8 |
| **3rd** | 50 | 0.0316 | 29.2 |
| **4th** | 66.5 | 0.0180 | 16.6 |
| **5th** | 83.31 | 0.0169 | 15.6 |

Table 5.5: Nilambe Unit 02 Trust Bearing Radial Vibration- Downstream side

After examine the above spectrum details it was decided that diagnosis could be parallel misalignment, since 2x radial is abnormally high and also present of high 3x. It was also suspected there may be mechanical looseness of coupling due to present of 3x, 4x, 5x and higher in the radial spectrum.

After that it was properly aligned the guide and thrust bearing by using laser tool and tightened the coupling bolts to rectify looseness fault. After the repair Radial velocity vibration spectrum was measured that is shown in Figure 5.9.



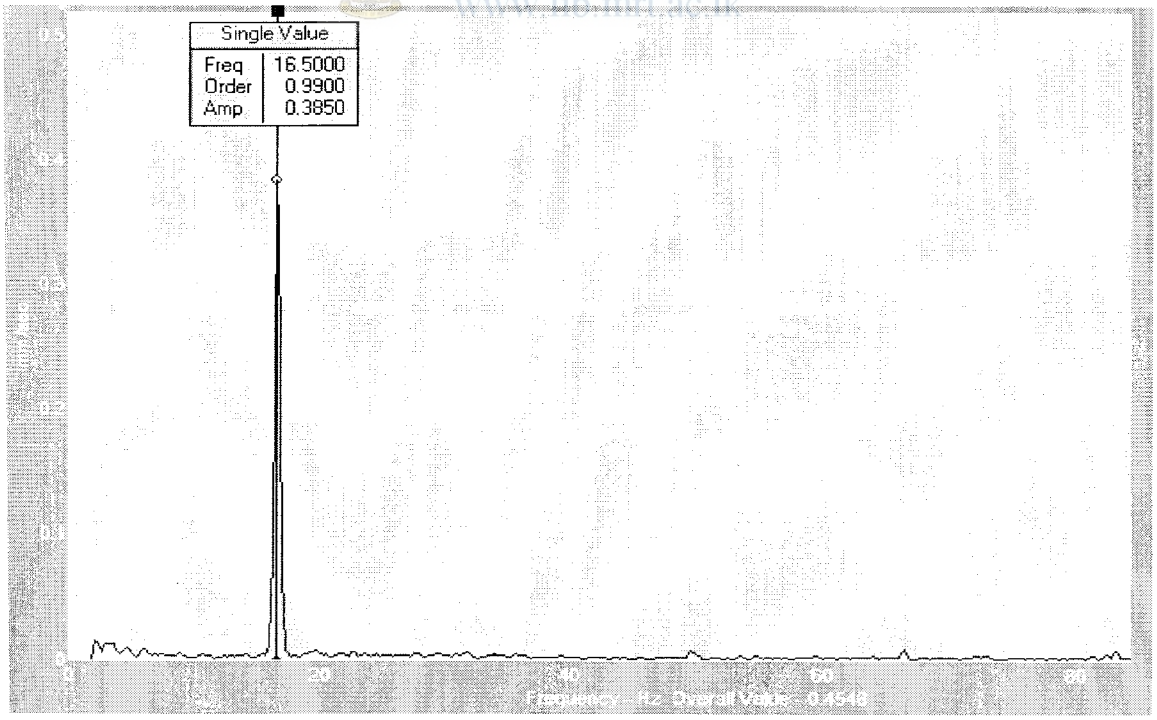



Figure 5.9: Radial Vibration Spectrum of Nilambe Unit 02 Trust Bearing Down stream side (After Repair)

**Case 3: Victoria Unit 2 Vibration analysis 2003 Sep 24**

Victoria Unit 2 exhibited high shaft vibration on 24th September 2003, on Generator bearings.

So machine vibration spectrum was tested for suspected Guide bearings and coupling from both axial and radial directions and radial vibration spectrum which was obtained for analyze the fault is shown in the Figure 5.10.

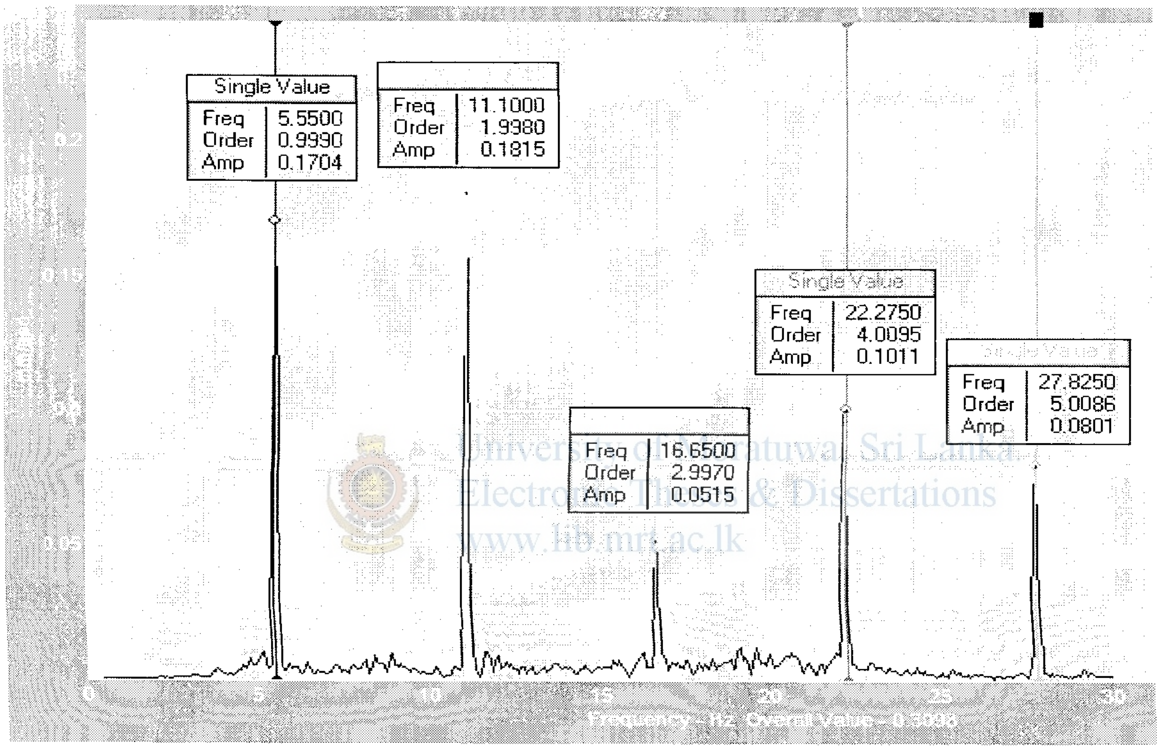




Figure 5.10: Radial Vibration Spectrum of Victoria Unit 02 Lower Guide Bearing on 2003 Sep 24 (Before Repair)

| | Harmonic Values | | |
|---|---|---|---|
| | **Freq** | **Amplitude** | **% of Fund.** |
| **Fund** | 5.55 | 0.1704 | |
| **2nd** | 11.1 | 0.1815 | 106.5 |
| **3rd** | 16.65 | 0.0515 | 30.2 |
| **4th** | 22.275 | 0.1011 | 59.3 |
| **5th** | 27.825 | 0.0801 | 47.0 |

Table 5.6: Victoria Unit 02 LG Bearing Radial Vibration

The harmonic analysis for vibration spectrum is shown in the Table 5.6.

After examine the above spectrum details as same logic for the previous example it was decided that diagnosis could be parallel misalignment, since 2x radial is abnormally high and also present of high 3x. It was also suspected there may be mechanical looseness of coupling due to present of 3x, 4x and 5x in the radial vibration spectrum.

After that it was properly aligned the guide and thrust bearing by using laser tool and tightened the coupling bolts to rectify looseness fault. Then the Radial velocity vibration spectrum was measured again after the repair and one of the vibration spectrums of that is shown in Figure 5.11.



Figure 5.11: Radial Vibration Spectrum of Victoria Unit 02 Lower Guide Bearing (After Repair)

### 5.1.3 Imbalance

**Case 1: Nilambe Unit 01 Vibration on 08th August 2000**

Nilambe Unit 01 exhibited high vibration on 08th August 2000, on Generator bearings & Thrust bearing when load increases from 1 to 5MW.

So machine vibration spectrum was tested from both axial and radial directions and having studied these data Coupling, Thrust & Guide bearings were suspected.

The radial velocity vibration spectrums were measured on 08th August 2000 was compared with previous velocity spectrum of the same machine which was obtained in January 2000. These vibration spectrums are shown in the Figure 5.12 and 5.13.

1x amplitude on 2000 Aug 08 = 0.5614

1x amplitude on 2000 Jan = 0.3681

Exceedance on 1x amplitude = 0.5614 - 0.3681

= 0.1933

% Exceedance than previous = (0.1933/0.3681) x 100

= **52.5%**



After inspected the vibration spectrums amplitudes with previous obtained values it was observed that 1x amplitude is abnormally higher than previous values and all harmonics are less than 15% of the 1x. Also it was observed that 1x amplitude increases with speed up of the machine.

After examine the above spectrum details it was decided that diagnosis could be imbalance of rotor mass, since 1x radial is abnormally high.

It was decided to inform this rotor imbalance to the OEM for checking and continuously monitor the condition of the machine.


Figure 5.12: Radial Vibration Spectrum of Nilambe Unit 01 Bearing on 2000 Jan 24 (Before Fault)



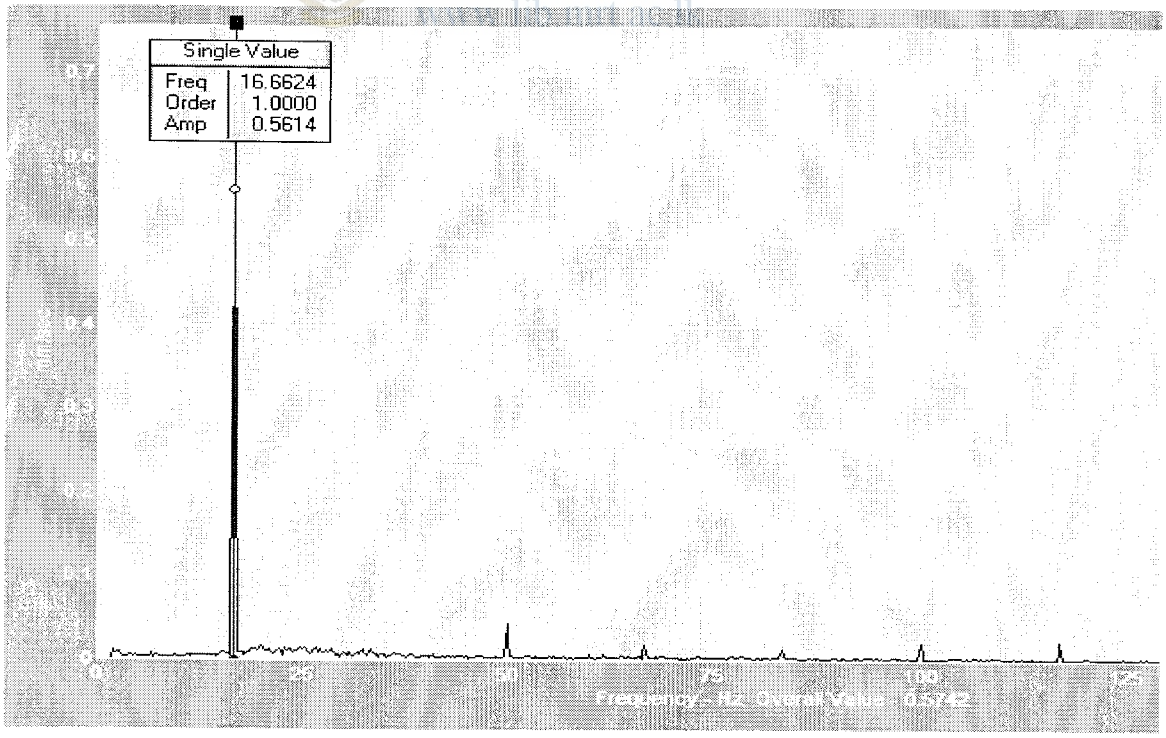



Figure 5.13: Radial Vibration Spectrum of Nilambe Unit 01 Bearing on 2000 Aug 08 (After Fault)

**Case 2: Victoria Unit 01 Vibration analysis on 30th May 2005**

Victoria Unit 01 Lower Guide Bearing exhibited high vibration on 30th May 2005, on Generator bearings & Thrust bearing when the load is increased.

So machine vibration spectrum was tested from both axial and radial directions and having studied these data Coupling, Thrust & Guide bearings were suspected.

The radial velocity vibration spectrums were measured on 30th May 2005 was compared with previous velocity spectrum of the same machine which was obtained in March 2001. These vibration spectrums are shown in the Figure 5.14 and 5.15.

1x amplitude on 2001 March = 25.5957

1x amplitude on 2005 May 30 = 29.1463

Exceedance on 1x amplitude = 29.1463 – 25.5957

= 3.5506

% Exceedance than previous = (3.5506/25.5957) x 100

= **13.9%**



After inspected the vibration spectrums amplitudes with previous obtained values it was observed that 1x amplitude is higher than previous value by around 14% and all harmonics are less than 15% of the 1x. Also it was observed that 1x amplitude increases with speed up of the machine.

By examine the above information it was suggested this diagnosis could be unbalance of rotor mass and decided to closely monitor the machine condition in future until obtain corrective action.

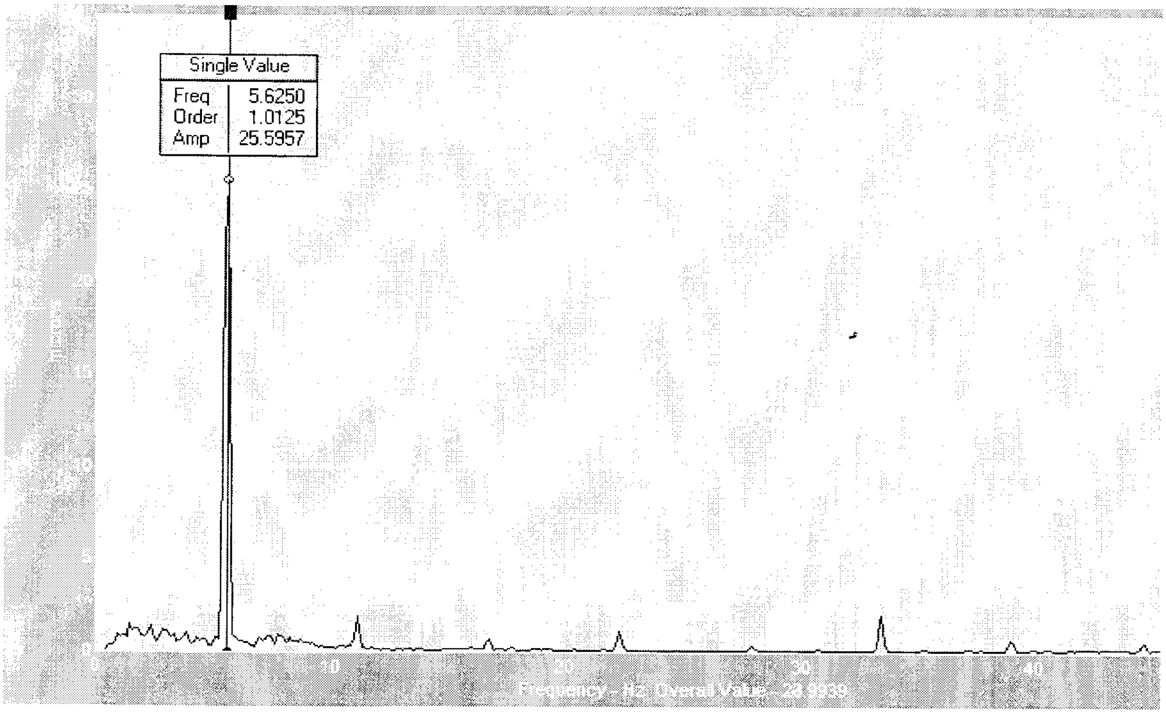



Figure 5.14: Radial Vibration Spectrum of Victoria Unit 01 Lower Guide Bearing on 2001 March (Before Fault)



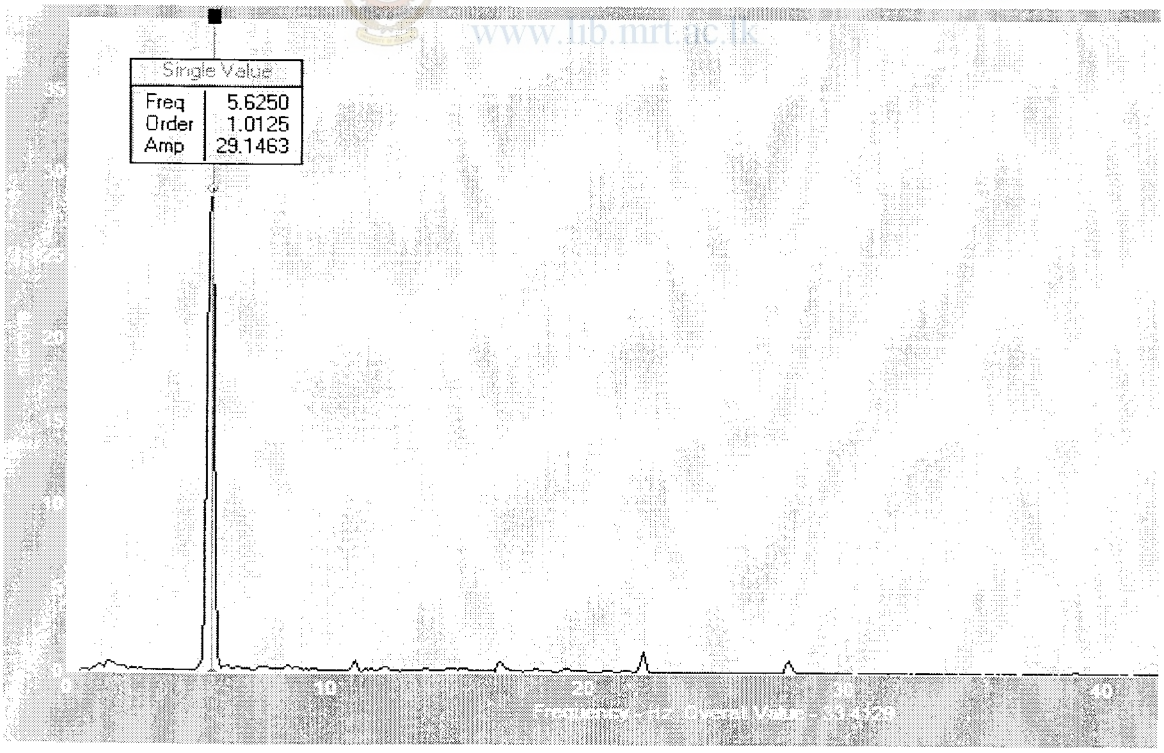



Figure 5.15: Radial Vibration Spectrum of Victoria Unit 01 Upper Guide Bearing on 2005 May 30 (After Fault)

### 5.1.4 Mechanical looseness

The mechanical looseness occurs in machines that has come loose from its mounting or machine component or if the fault has developed in the bearing that may be worn down the bearing elements or the bearing seat.

The following case studies were carried out of the generator faults at the with abnormally high machine vibration situations to find out the reason for that vibration. Most of the instances the vibration spectrums indicate abnormally high running speed amplitude followed by multiples or ½ multiples. Also it can be observed these harmonic peaks may decrease in amplitude as they increase in frequency.

These faults can be rectified by proper tightened of the coupling, machine mounting or other machine components or replaced the worn down bearing elements, seats.



**Case 1: Nilambe Unit 01 vibration analysis**

Nilambe Unit no 01 was exhibited high vibration on 080th August 2000 and it was tested for analyze the vibration spectrums. The velocity vibration spectrums for bearing 2 and bearing 1 are shown in the Figure 5.16 and 5.17 respectively.

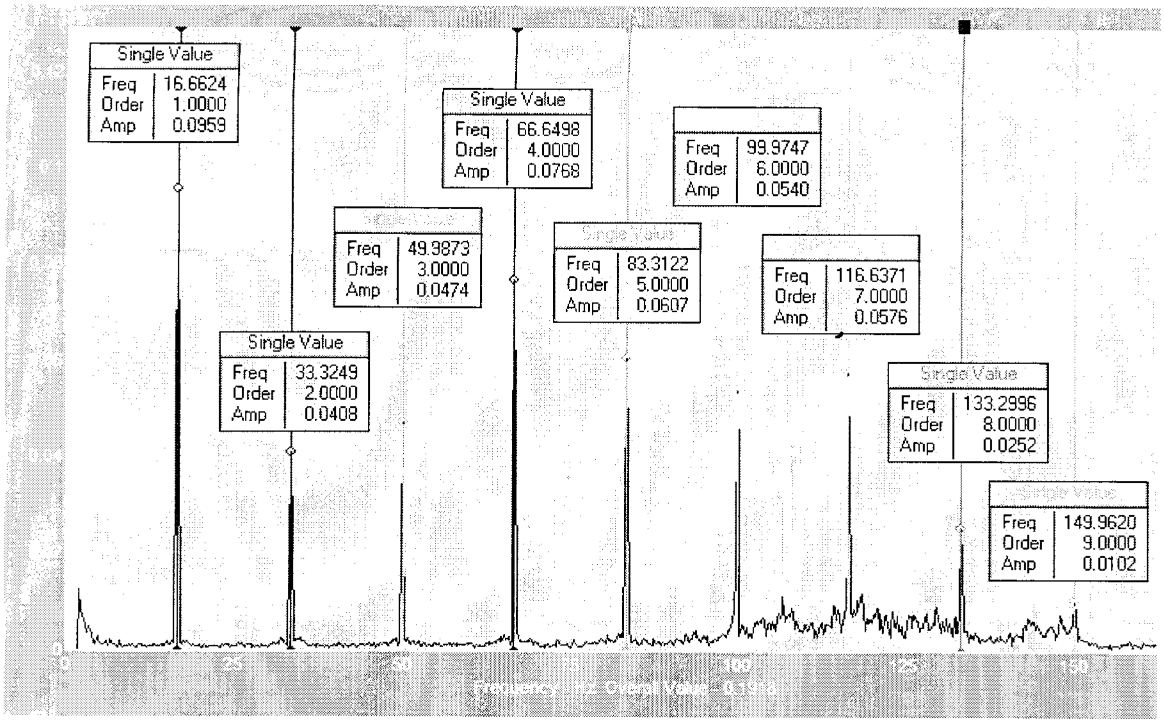



Figure 5.16: Vibration Spectrum of Nilambe Unit 01 Bearing 02 on 2000 Aug 08 (Before Repair)



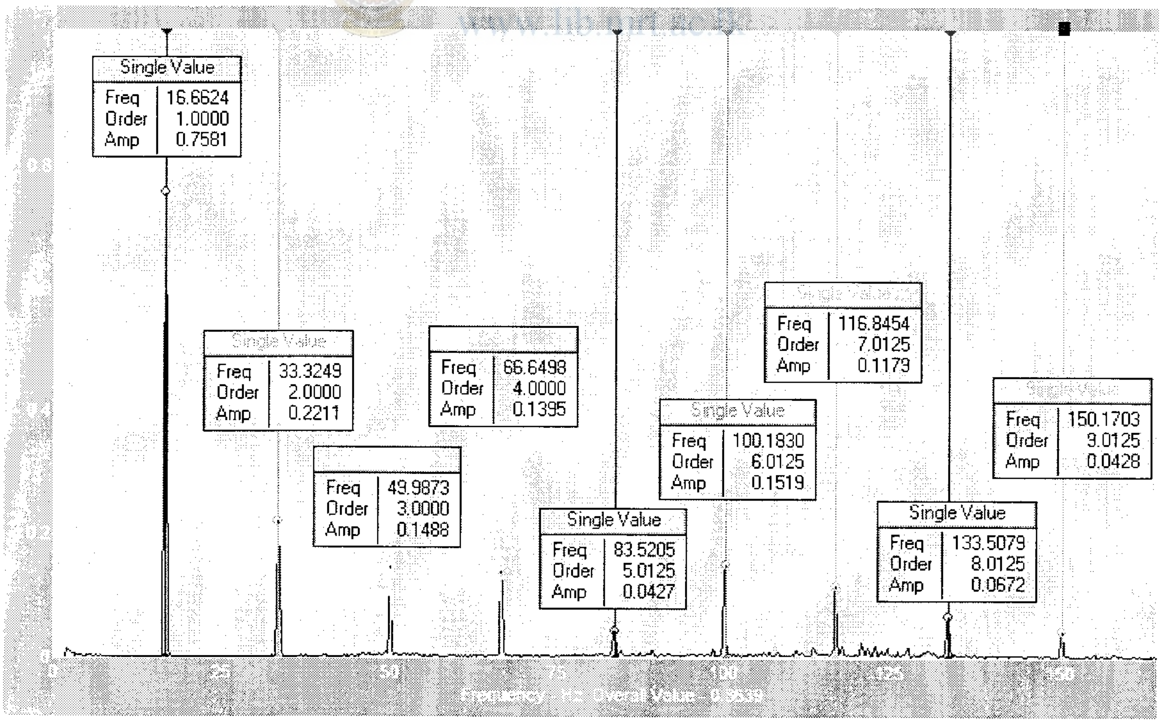



Figure 5.17: Vibration Spectrum of Nilambe Unit 01 Bearing 01 on 2000 Aug 08 (Before Repair)

| | Harmonic Values | | |
|---|---|---|---|
| | Freq | Ampli. | % of Fund. |
| Fund | 16.66 | 0.0959 | |
| 2nd | 33.32 | 0.0408 | 42.5 |
| 3rd | 49.98 | 0.0474 | 49.4 |
| 4th | 66.65 | 0.0768 | 80.1 |
| 5th | 83.31 | 0.0607 | 63.3 |
| 6th | 99.97 | 0.0540 | 56.3 |
| 7th | 116.64 | 0.0576 | 60.1 |
| 8th | 133.30 | 0.0252 | 26.3 |
| 9th | 149.96 | 0.0102 | 10.6 |

Table 5.7: Nilambe Unit 01 Bearing 02 vibration Harmonic analysis

| | Harmonic Values | | |
|---|---|---|---|
| | Freq | Ampli. | % of Fund. |
| Fund | 16.66 | 0.7581 | |
| 2nd | 33.32 | 0.2211 | 29.2 |
| 3rd | 49.98 | 0.1488 | 19.6 |
| 4th | 66.65 | 0.1395 | 18.4 |
| 5th | 83.31 | 0.0427 | 5.6 |
| 6th | 99.97 | 0.1519 | 20.0 |
| 7th | 116.64 | 0.1179 | 15.6 |
| 8th | 133.30 | 0.0672 | 8.9 |
| 9th | 149.96 | 0.0428 | 5.6 |

Table 5.8: Nilambe Unit 01 Bearing 01 vibration harmonic analysis



The vibration harmonic amplitude analysis of Bearing 2 and Bearing 1 are shown on the Table 5.7 and Table 5.8 respectively. According to these data it can be observed there are a series of eight synchronous multiples of running speed (range 2x to 9x) and their magnitudes are greater than 20% of the 1x in spectrum of bearing 2.

Also in vibration spectrum of bearing 1 indicates that majority (range 2x, 3x,4x & 6x) synchronous multiples of running speeds magnitudes are greater than or nearly 20% of the 1x.

Thus, this diagnosis is clear indication of mechanical looseness of the machine.

Then it was decided to check and tighten the couplings and bearing mounts to rectification of this fault. Then the velocity vibration spectrum was measured again after the repair and Bearing No 02 vibration spectrum is shown in Figure 5.18.

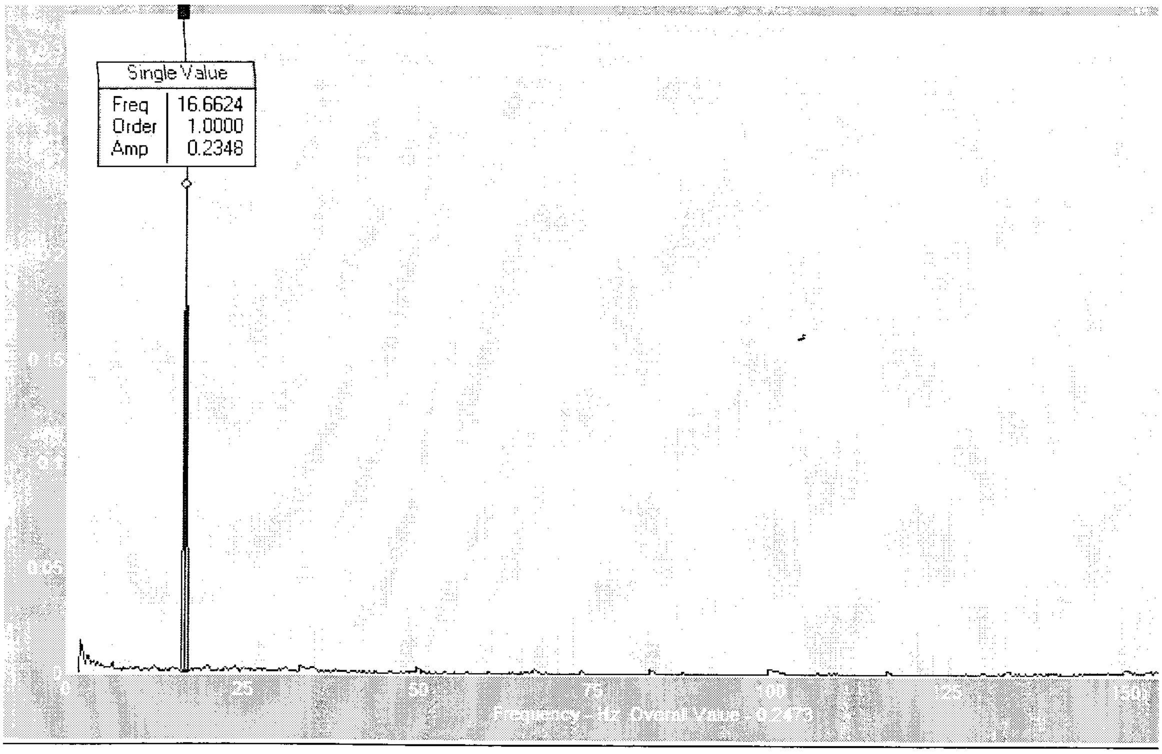



Figure 5.18: Vibration Spectrum of Nilambe Unit 01 Bearing 02 (After Repair)



**Case 2: Victoria Unit 01 vibration analysis**

Victoria Unit no 01 was exhibited high vibration on 16th June 2003 and it was tested for analyze the vibration spectrums. The displacement and velocity vibration spectrums for LG bearing are shown in the Figure 5.19 and 5.20 respectively.

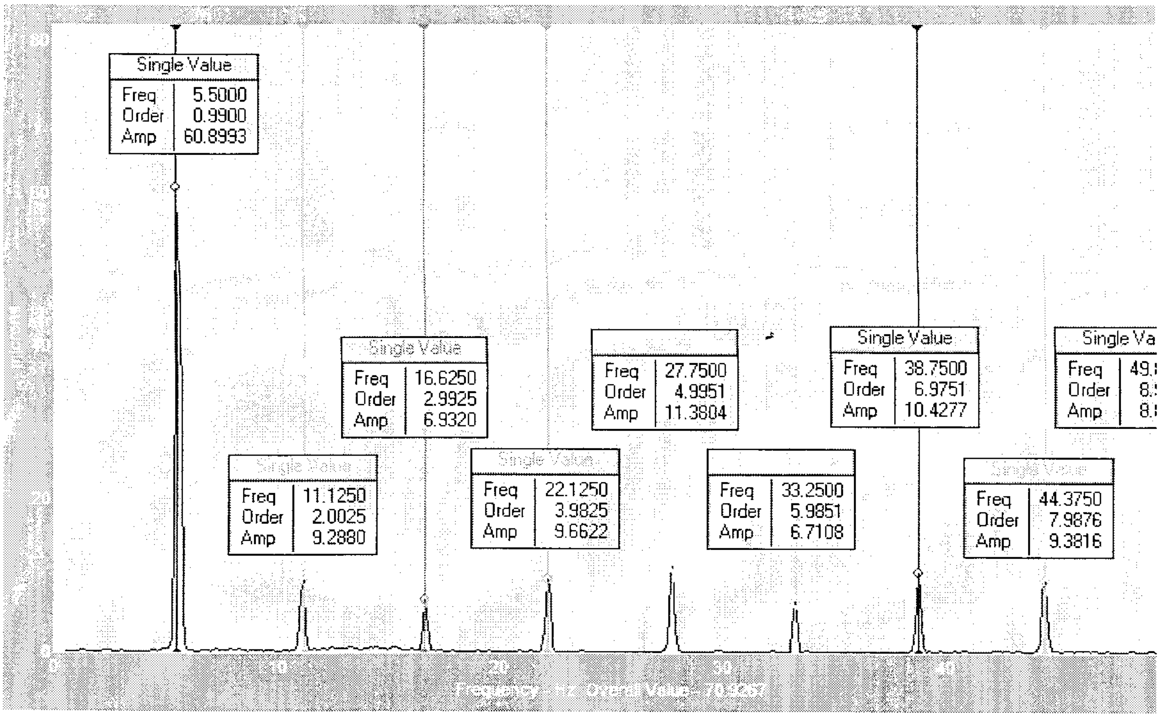



Figure 5.19: Displacement Vibration Spectrum of Victoria Unit 01 Lower Guide Bearing on 2003 June 16 (Before Repair)



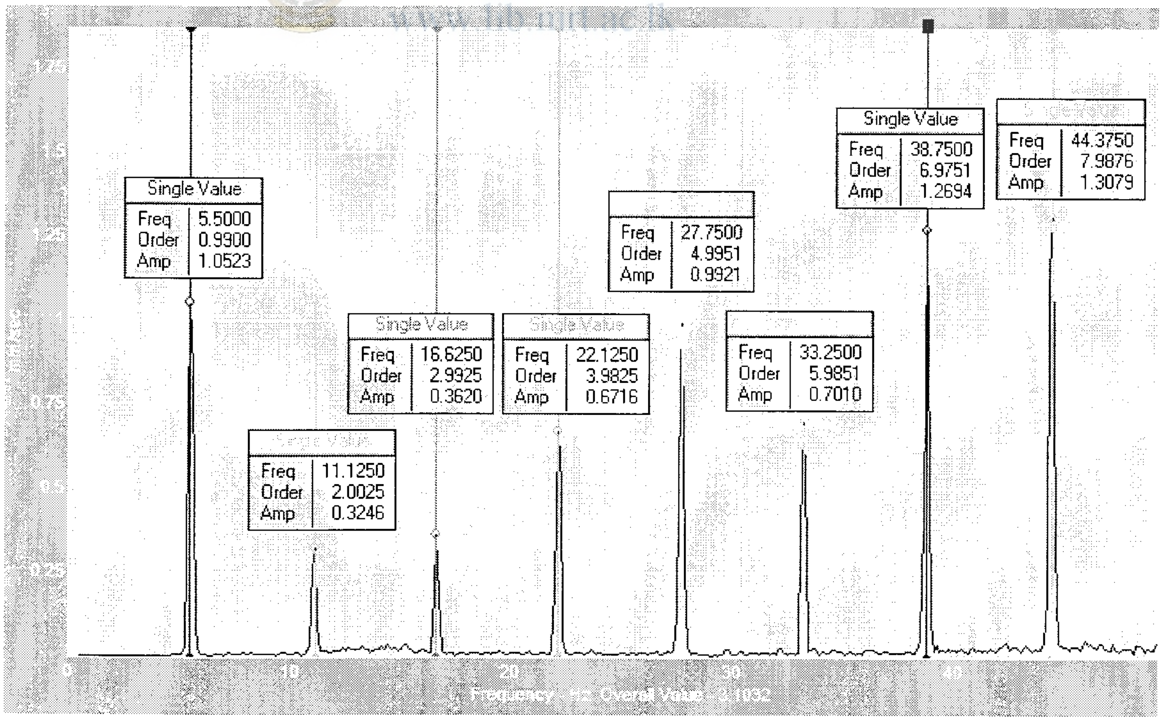



Figure 5.20: Velocity Vibration Spectrum of Victoria Unit 01 Lower Guide Bearing on 2003 June 16 (Before Repair)

| | Harmonic Values | | |
|---|---|---|---|
| | Freq | Ampli. | % of Fund. |
| Fund | 5.5 | 1.0523 | |
| 2nd | 11.125 | 0.3246 | 30.8 |
| 3rd | 16.625 | 0.3620 | 34.4 |
| 4th | 22.125 | 0.6716 | 63.8 |
| 5th | 27.75 | 0.9921 | 94.3 |
| 6th | 33.25 | 0.7010 | 66.6 |
| 7th | 38.75 | 1.2694 | 120.6 |
| 8th | 44.375 | 1.3079 | 124.3 |

Table 5.9: Victoria Unit 01 LG Bearing velocity Vibration harmonic analysis

| | Harmonic Values | | |
|---|---|---|---|
| | Freq | Ampli. | % of Fund. |
| Fund | 5.5 | 60.8993 | |
| 2nd | 11.125 | 9.288 | 15.3 |
| 3rd | 16.625 | 6.932 | 11.4 |
| 4th | 22.125 | 9.6622 | 15.9 |
| 5th | 27.75 | 11.3804 | 18.7 |
| 6th | 33.25 | 6.7108 | 11.0 |
| 7th | 38.75 | 10.4277 | 17.1 |
| 8th | 44.375 | 9.3816 | 15.4 |

Table 5.10: Victoria Unit 01 LG Bearing displacement vibration harmonic analysis



The vibration harmonic amplitude analyses of Lower Guide Bearing (LGB) velocity and displacement vibration are shown in the Table 5.9 and Table 5.10 respectively. According to these data it can be observed in LGB velocity vibration spectrum there are a series of seven synchronous multiples of running speed (range 2x to 8x) and their magnitudes are greater than 20% of the 1x in spectrum of bearing 2.

Also in displacement vibration spectrum of LGB indicates that majority (range 2x, 3x,4x & 6x) synchronous multiples of running speeds magnitudes are nearly 20% of the 1x.

So that is clear indication of mechanical looseness of the generator.

Then it was decided to check and tighten the couplings and bearing mounts to rectification of this fault. Then the velocity vibration spectrum was measured again after the repair and Lower Guide Bearing vibration spectrum is shown in Figure 5.21.

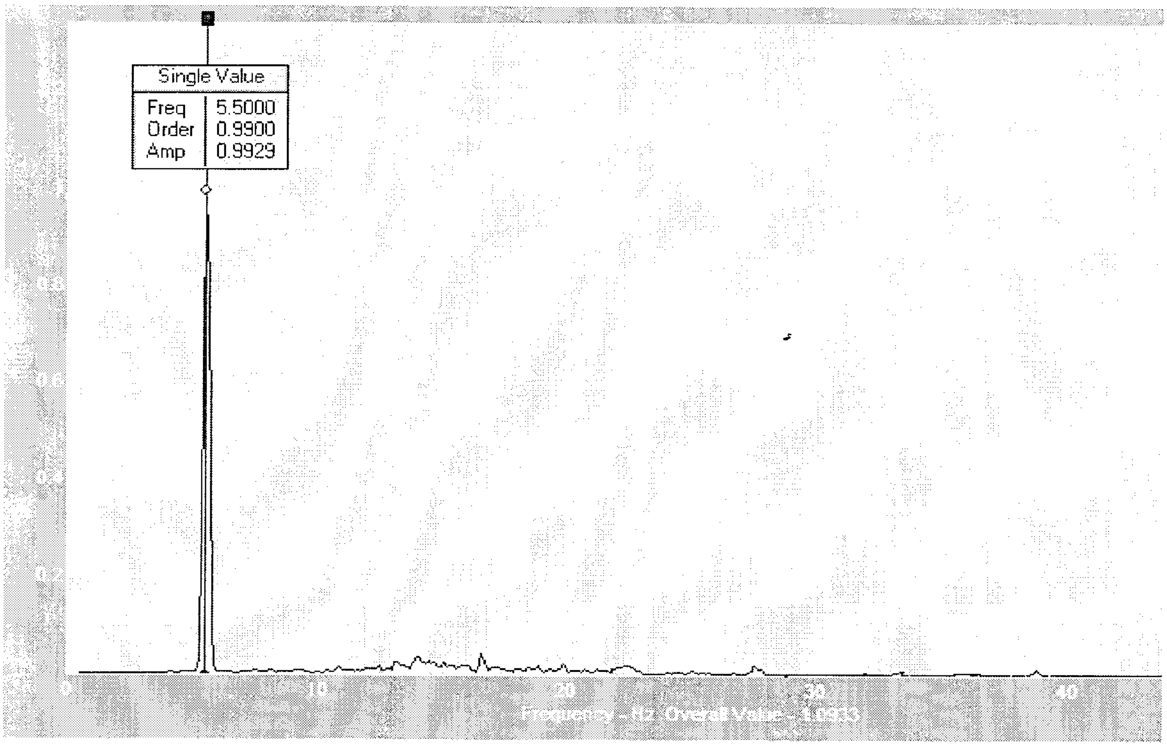



Figure 5.21: Velocity Vibration Spectrum of Victoria Unit 01 Lower Guide Bearing (After Repair)



**Case 3: Randenigala Unit 02 vibration analysis**

Randenigala Unit no 02 was exhibited high vibration on 07th May 2003 and it was tested for analyze the vibration spectrums is shown in the Figure 5.22. According to the spectral data it was found that the cause is the mechanical looseness of the machine coupling and it was properly tightened and aligned.

Then the velocity vibration spectrum was measured again after the repair and Trust Guide Bearing vibration spectrum is shown in Figure 5.23.

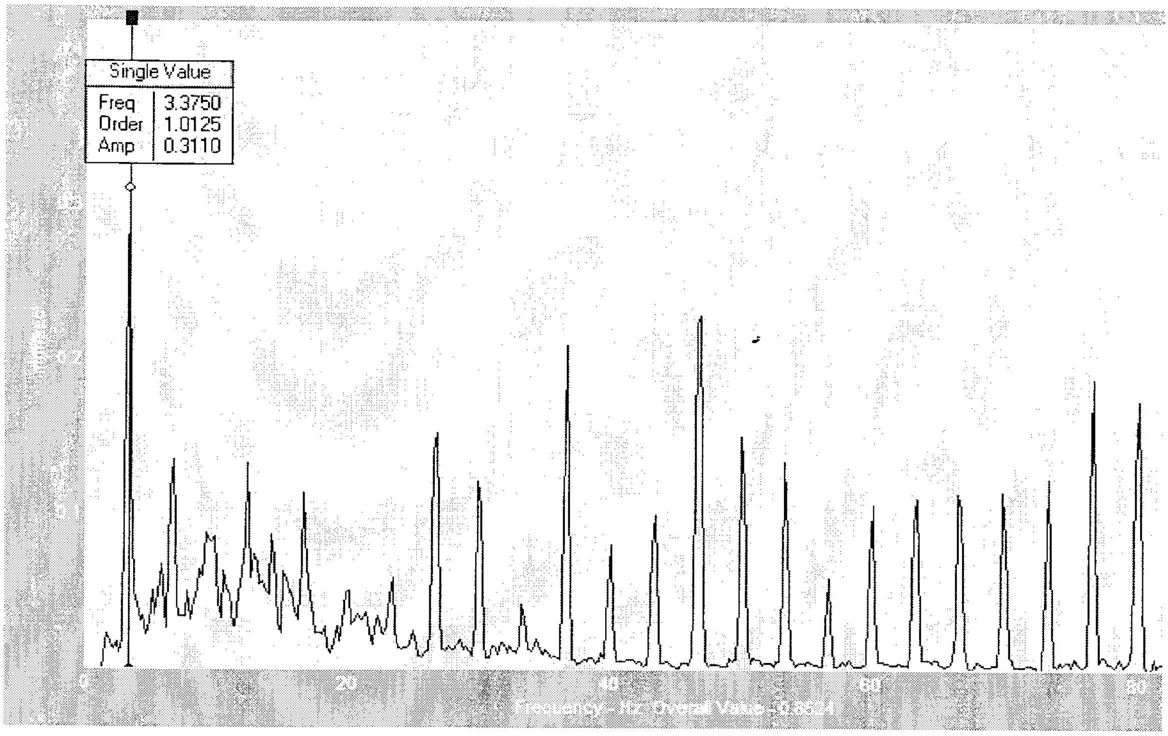



Figure 5.22: Vibration Spectrum of Randenigala Unit 02 Trust Guide Bearing @ 45 MW on 2003 May 07 (Before Repair)



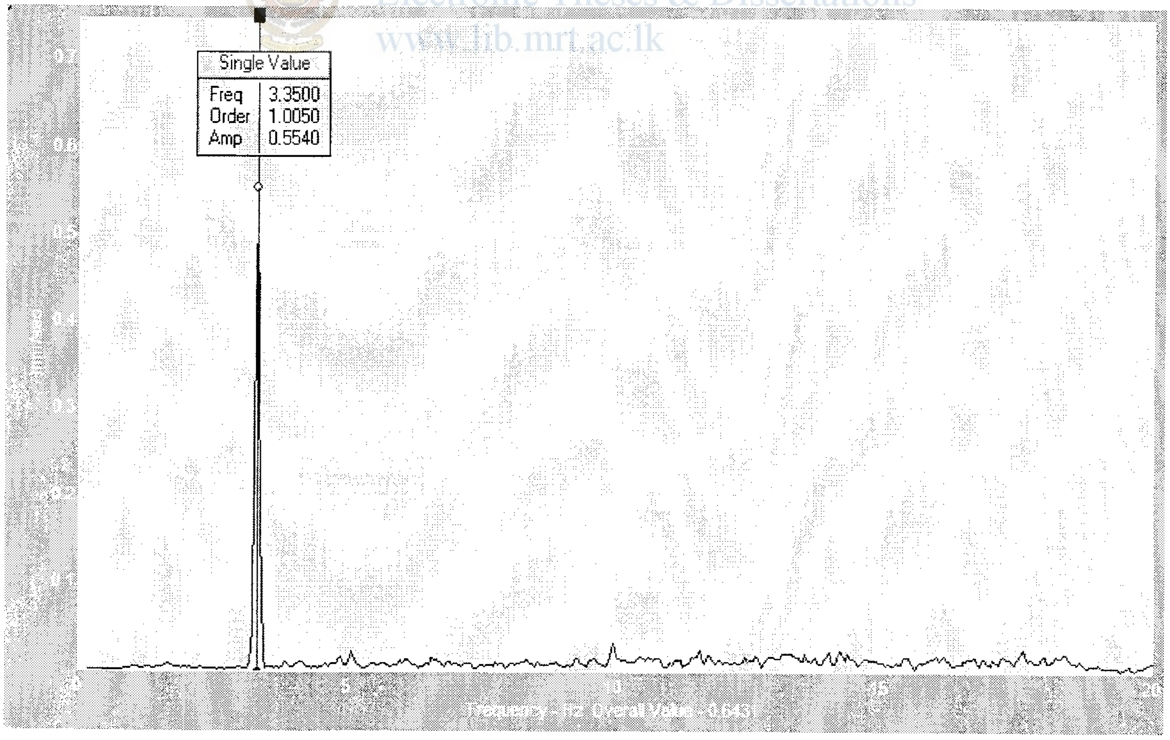



Figure 5.23: Vibration Spectrum of Randenigala Unit 02 Trust Guide Bearing @ 45 MW (After Repair)

### 5.1.5 Bearing Defects

There are many machinery problems that can contribute to bearing failure. The most prevalent are excessive load caused by misalignment and/or imbalance and lubrication problems such as lack or improper lubrication, excessive or contaminated lubrication.

The velocity vibration measurements are typically very useful for detecting and analyzing low frequency rotational problems such as imbalance, misalignment, looseness, etc.

But bearing defects occurs at much higher frequencies and much lower amplitudes bearing defects may not be detectable on its early stages on these vibration spectrums.

Therefore to assist in determining if a machine's problem include a faulty bearing, the defect frequencies of the bearing should be calculated and can be overlaid on the vibration spectra.



Then these defect frequencies are aligned with peak amplitudes in the vibration spectrum, there is possibility to the bearing defect.

**Case 1: Randenigala Unit 02 Vibration analysis on 28th August 2001**

Randenigala Unit 02 exhibited high vibration on August 2001, on Generator bearings & Thrust bearing when load increase.

So machine vibration spectrum was tested from both axial and radial directions and having studied these data Coupling, Thrust & Guide bearings were suspected. These vibration spectrums which were obtained at various loads are shown in the Figure 5.24, 5.25 and 5.26.

When analyzing the vibration spectra it was observed much higher frequencies with much lower amplitudes and suspected there may be bearing defects.

Also in vibration spectrum of upper guide bearing indicates that synchronous multiples of running speeds magnitudes (2x and 3x) which are greater than or nearly 20% of the 1x.

So that is indication of mechanical looseness of the generator coupling or bearings.

Then it was decided to check and tighten the couplings and bearing mounts to rectification of this fault.

After that upper bearing oil sump was inspected and some metal particles were found in the thrust & guide bearing sump. This will cause may be a bearing seems to have damaged.

On 2001 September a shear pin failure in the machine has been detected. When inspecting this fault maintenance staff have found a piece of wood (4 inch long 1.5 inch diameter approximately cylindrical shape, app 150gm) entrapped in the Runner that may caused bearing defects due to coupling misalignment or imbalance forces on the shaft.

Then shaft coupling alignment was checked with a Laser Tool and alignment was found perfect and upper bearing was replaced with spare one. The measured coupling velocity vibration spectrum after the repair is shown in Figure 5.27.



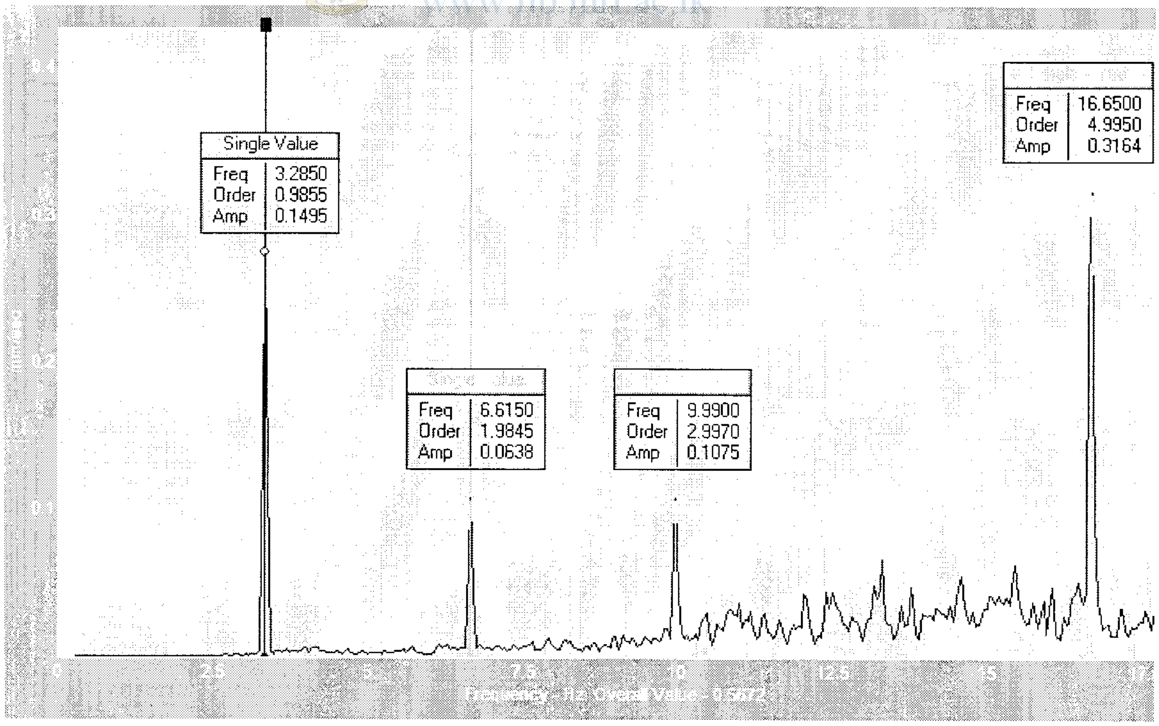



Figure 5.24: Vibration Spectrum of Randenigala Unit 02 Upper Guide Bearing @ 32.5 MW on 2001 Aug 28 (Before Bearing Replacement)

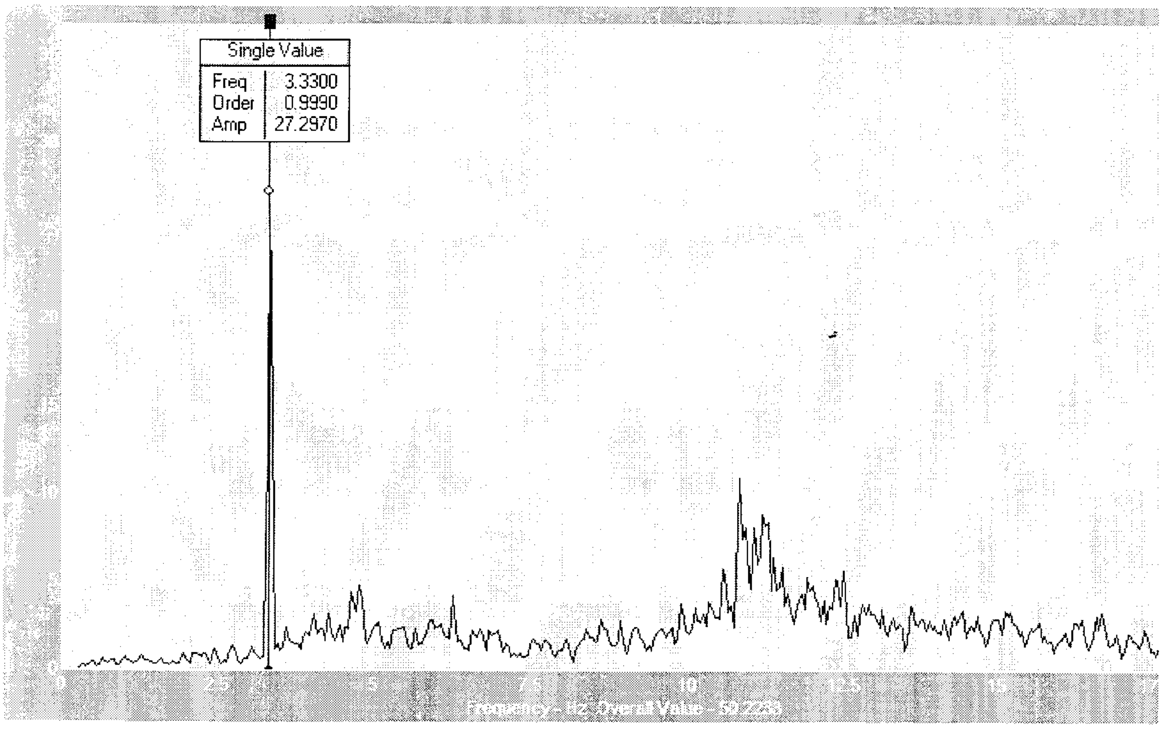



Figure 5.25: Displacement Vibration Spectrum of Randenigala Unit 02 Coupling @ 34 MW on 2001 Aug 28 (Before Bearing Replacement)



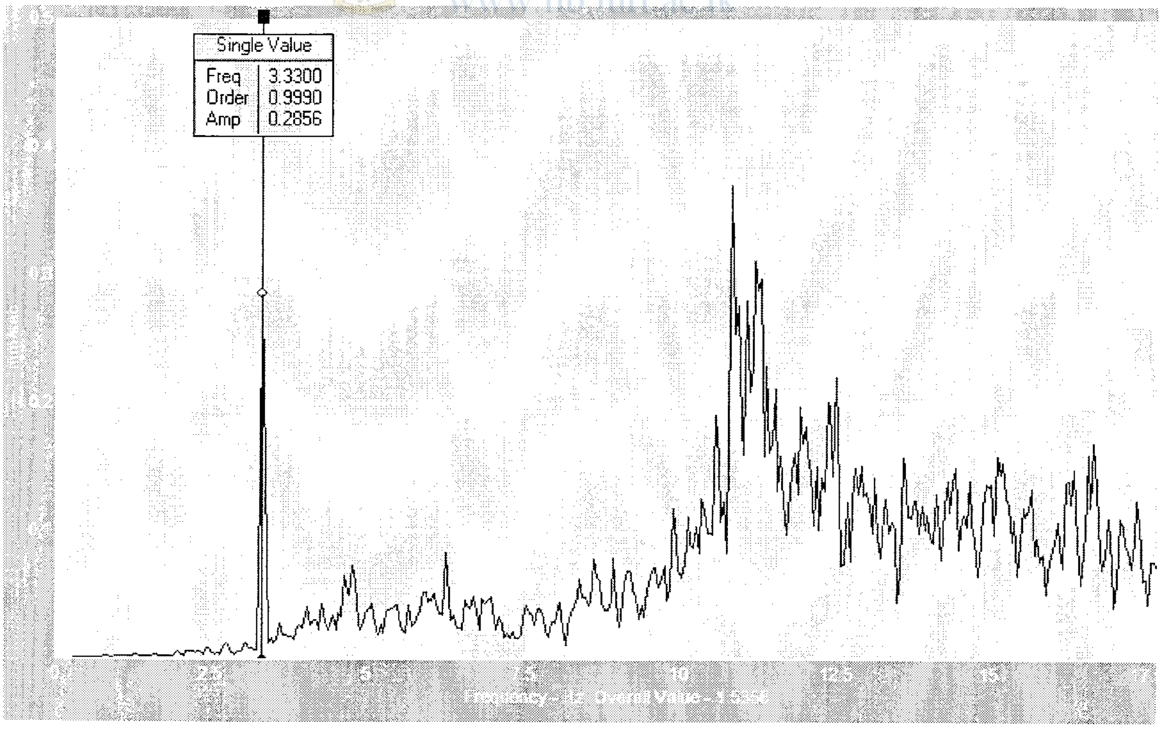



Figure 5.26: Velocity Vibration Spectrum of Randenigala Unit 02 Coupling @ 34 MW on 2001 Aug 28 (Before Bearing Replacement)

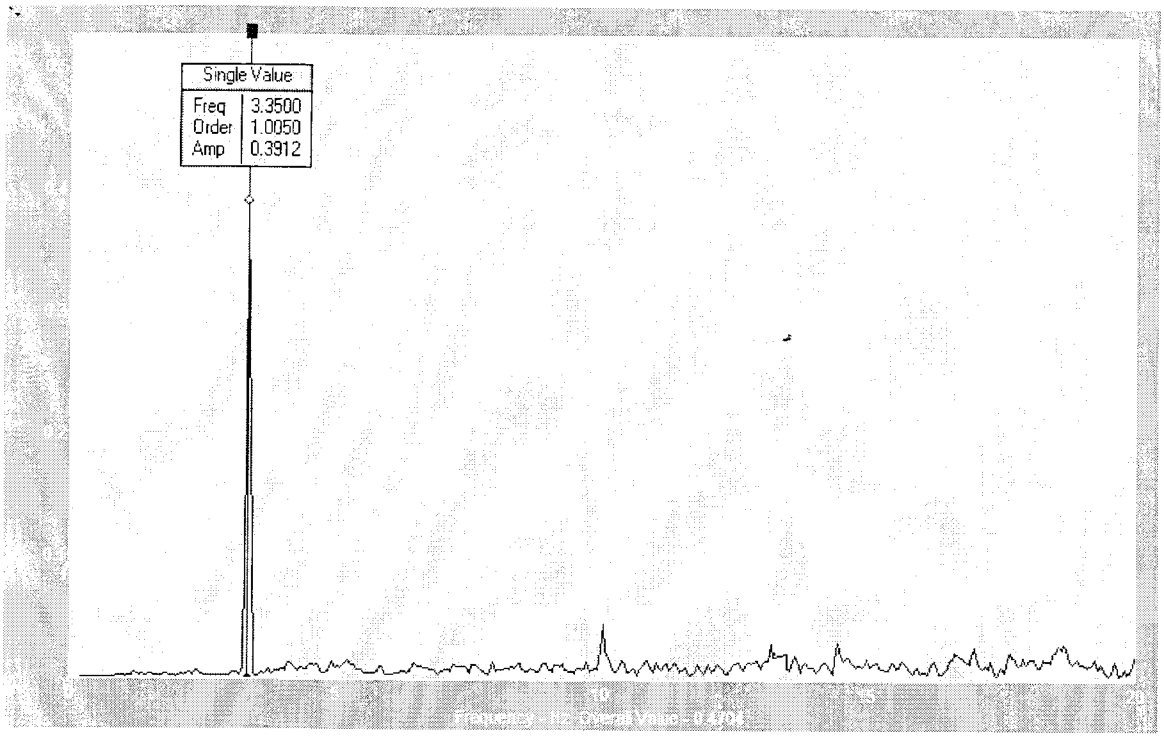



Figure 5.27: Velocity Vibration Spectrum of Randenigala Unit 02 Coupling @ 34 MW (After Bearing Replacement)



**Case 2: Ukuwela Unit 01 Vibration analysis on October 2004**

Ukuwela Unit 01 exhibited high vibration on October 2004, on Turbine bearing when load increase.

So machine vibration spectrum was tested from and having studied these data coupling and Turbine bearing were suspected. The one of vibration spectrum which were obtained is shown in the Figure 5.28.

After that Turbine bearing oil sump was inspected and some metal particles were found in the bearing sump. This will cause may be a bearing seems to have damaged.

So it was decided to replace the bearing after further analysis of the machine condition. After the replacement of the Bearing then it was obtained the velocity vibration spectrum and Bearing vibration spectrum is shown in Figure 5.29.

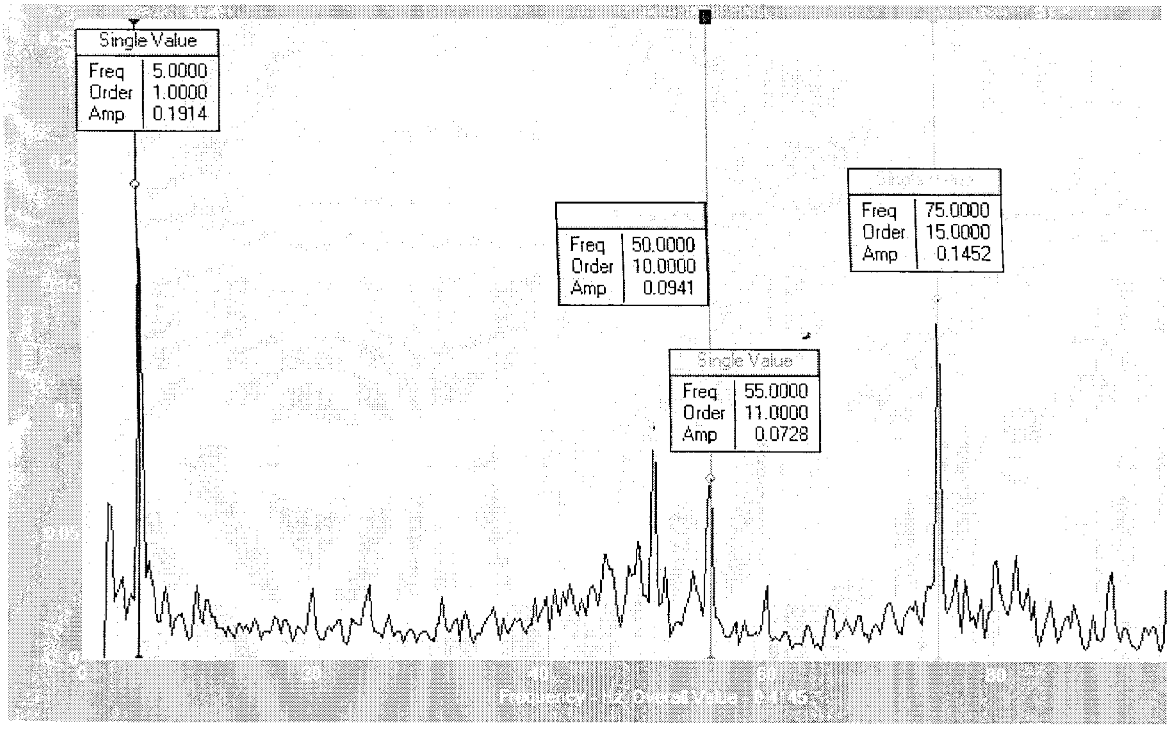



Figure 5.28: Vibration Spectrum of Ukuwela Unit 01 Turbine Bearing Bracket on 2004 Oct (Before Bearing Replacement)



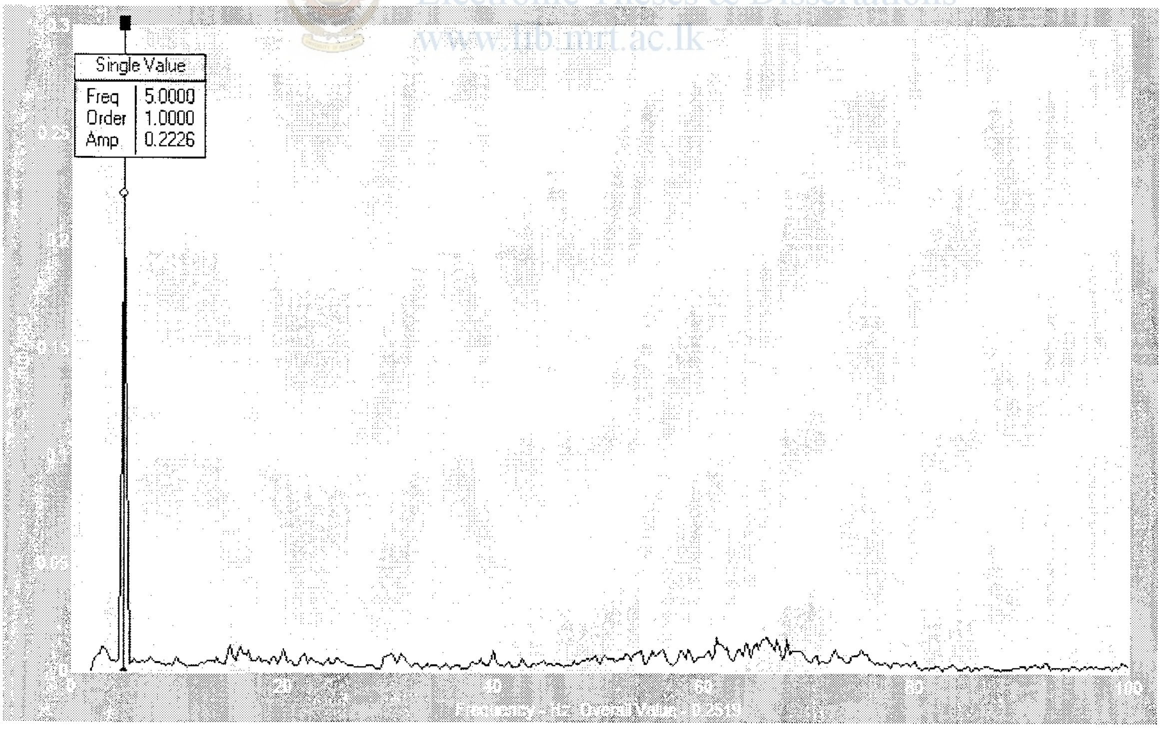



Figure 5.29: Vibration Spectrum of Ukuwela Unit 01 Turbine Bearing Bracket (After Bearing Replacement)

# Chapter 6
# DISCUSSION

The good healthy operating condition of the generator is very much important for assuring a sound operating system in a power station.

Machine vibration analysis is a good method in synchronous generator fault diagnosis especially when the other monitoring systems such as temperature analysis are difficult to implement.

The experiment showed that the generator vibration spectrum analysis could be effectively used to detect major mechanical faults such as misalignment, imbalance etc. In the experiment, this result was verified with the theory.

The results of the case studies of previous works, experiment and the standards derived could be used to determine the fault level of the condition monitoring system which in turn would help to identify and verify the generator problem with a minimum interruption to the system operation.



The effect of distinguish the specific fault frequencies in the vibration spectra of the SG is a major issue in identifying the faults. The standards for detecting scheme and the results of the experiment help to build and confirm the relationships between the vibration spectra and faulty condition of machine.

**Advantages of the Vibration monitoring system**

- No alteration in the machine circuit or shut down is required, as the vibration measurement is done at online.

- Analysis is convenient since a particular problem gives similar results irrespective of the machine capacity: Standardization gives rise to easy Fault recognition.

- Both, monitoring and analysis were simple with Microlog instrument, which released the burden of calculating FFT vibration components at critical frequencies separately. Thus, it ensured the accuracy and increased the efficiency of the system.

- Very convenient to analysis data since the Prism 4 software also have capability to convert the same spectra in forms of displacement, velocity or acceleration.

**Disadvantages**

- Vibration harmonics can result from several different sources in addition to the sources that we are interested here such as other machine operating in the same vicinity of the tested machine



- Therefore, these harmonics can be overlooked as machine problems and the results sometimes can be misleading. So that, a sudden conclusion cannot be made by looking at the vibration spectrum and the method of 'Trending' will have to be used instead: testing over several weeks or months will confirm if the machine is stable and not changing/degrading.

- Cannot be used for testing in environment with high temperatures or external noise levels.

- May sometimes not be suitable for detecting some machine faults such as the Bearing fault, since the high frequency components appear in the vibration spectrum only when the fault is severe and the bearing failure, which most of the time can be detected by its noise level.

# Chapter 7
# CONCLUSION

Generator vibration acts as an excellent transducer for detecting mechanical faults in the Synchronous generator. Spectrum analysis of the generator's vibration signal can hence be used to detect a faulty generator without disturbing its operation.

The procedure includes,

a) Acquiring the generator vibration signals
b) Performing signal conditioning
c) Analyzing the derived signals to identify the fault.

Among the faults detectable from this technique are Synchronous generator's angular and parallel misalignments, Imbalance, mechanical looseness, bearing defects, etc.



In this project it shows there is strong co-relation between specific vibration spectrum patterns with developing faults of the model. The case study done with the various synchronous generators also shows this co-relation of vibration spectrums with their faults.

This technique becomes more useful for trending generator indications, where it can be used to find the information required for predictive maintenance.

Therefore, this highly versatile technology of condition monitoring and fault analysis of generator solves the biggest hurdle of any maintenance engineer, i.e. to obtain a shut down for testing the machine and `How long until I need to replace or repair the generator?'

# REFERENCE


[1] Vibration diagnostic guide: CM5003-EN, SKF Condition Monitoring, VDG Revision 1.01, 1995. pp 21-37.

[2] Guidelines for Implementing a Portable Condition Monitoring Program Manual part no. 31641400 - 1995, SKF Condition Monitoring Inc. pp 3-6.

[3] Electric Motor Monitoring with your Microlog, Application note CM3005, SKF Condition Monitoring, 1997.

[4] Adtya Koebe, `On-line condition monitoring of motors using electrical signature analysis': a paper presented at the 'Recent advances in condition-based plant maintenance', seminar organized by the Indian Institute of Plant Engineers, May 2002, Mumbai.

[5] Caryn M. Riley, K. Lin, Thmas Habetler, 'A Method for online Vibration Monitoring of Induction Motors' IEEE transactions, 1997, pp 201-205.

[6] Bert Lundgaard DLI Engineering, 'Using Transducers for Machinery Fault Detection', Sensors magazine, November, 1988.

[7] Web sites in The Internet : www.reliability.com
www.equipmenthealth.com
www.galaxyscientific.com
www.electricityforum.com
www.reliabilityweb.com

# ISS 2200/ ISS 2200-M8

## Profile Standard Acceleration Sensor ...

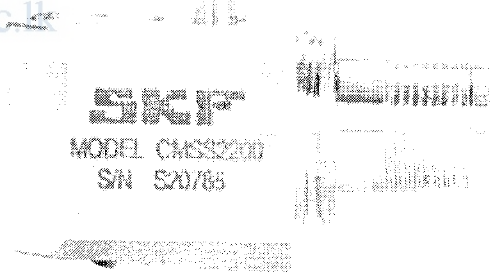
*... Incorporating the latest in technology to meet the demanding CE, EMC and low level noise requirements.*

Using detailed knowledge acquired from many years of supplying high quality sensors to a broad spectrum of industry users, SKF Condition Monitoring offers rugged accelerometers designed for the Pulp and Paper, Petrochemical, Steel, Mining and Construction, Metal Working, and Machine Tool Industries.

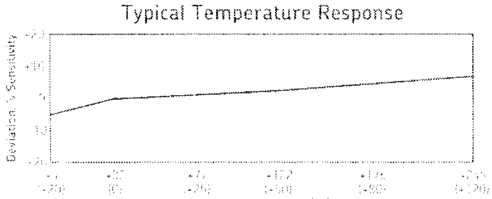
## Features

- *For use with the SKF Local Monitoring Unit (LMU's) On-Line Systems, Protection Systems and the ever versatile Portable Data Collection Instruments.*
- *Rugged, economical, and all around general purpose sensor.*
- *Sensitivity, 100 mV/g to optimize use in multiple applications.*
- *Exceptional bias voltage (BV) stability at elevated temperatures.*
- *Designed for exceptional low noise level for low frequencies at elevated temperatures.*
- *Meets stringent CE, EMC requirements.*
- *Low profile for side industrial accelerometer with captive mounting bolts (1/4–28, M6 x 1.00 or M8 x 1.25) provided.*
- *Corrosion resistant and hermetically sealed.*
- *Reverse polarity wiring protection.*

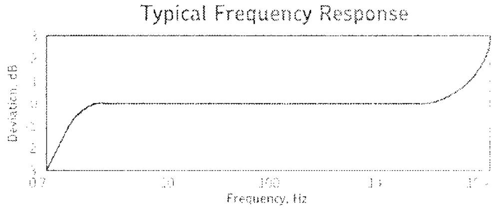





Typical Temperature Response




Typical Frequency Response



…00/
2200-M8
Profile
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or

## Specifications

Specifications conform to ISA-RP-37.2 (1-64) and are typical values referenced at +75°F (+24°C), 24 Vdc supply, 4 mA constant current and 100 Hz.

*DYNAMIC*

**Sensitivity:** 100 mV/g

**Sensitivity Precision:** ± 10% at +77 °F (+25 °C)

**Acceleration Range:** 80 g peak

**Amplitude Linearity:** 1%

**Frequency Range:** ± 10%: 1.0 Hz to 5,000 Hz
± 3 dB: 0.7 Hz to 10,000 Hz

**Resonance Frequency:** Mounted, minimum 22 kHz

**Transverse Sensitivity:** ≤ 5% of axial

*ELECTRICAL*

**Power Requirements:**

Voltage Source: +18 Vdc to +30 Vdc

Constant Current Diode: 2 to 10 mA dc, recommended 4 mA

**Electrical Noise:** 2.0 Hz: 20 μg/√Hz

**Output Impedance:** < 100 Ohms

**Voltage Bias:** 12 Vdc

**Grounding:** Case isolated, internal shielding

*ENVIRONMENTAL*

**Temperature Range:** -58 °F to +248 °F (-50 °C to +120 °C) operating temperature

**Vibration Limits:** 500 g peak

**Shock Limit:** 5,000 g peak

**Electromagnetic Sensitivity, Equivalent g:** 70 μg/Gauss

**Sealing:** Hermetic

**Base Strain Sensitivity:** 200 μg/microstrain

**CE:** According to the Generic immunity standard for Industrial Environment EN50082-2.

Acceptance Criteria: The generated "false equivalent g level" under the above test conditions should be less than 2 milli g peak to peak.

*PHYSICAL*

**Dimensions:** See drawing

**Weight:** ≈ 5.10 oz. (145 gms)

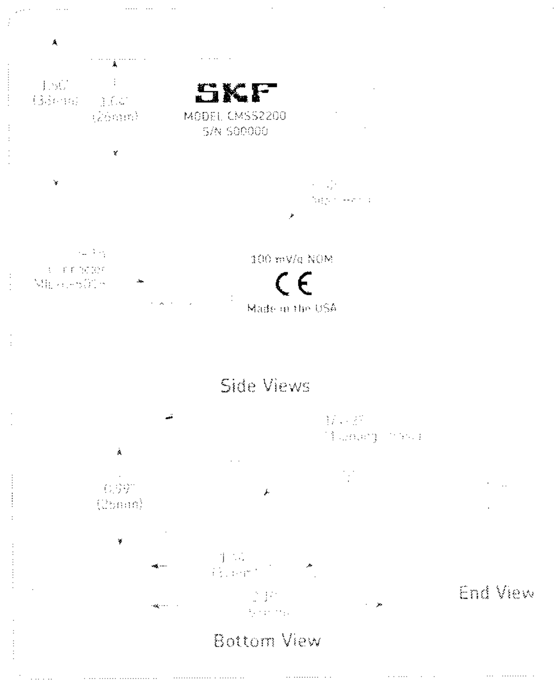
**Case Material:** 316L stainless steel

**Mounting:** Captive mounting bolts provided. One (1) 1/4-28 English thread and one (1) M6 x 1.00 Metric thread.

**Mounting Torque:** 30 in-lbs, 3.4 N-m

**Connector:** Pin A: Signal/Power (White Wire)
Pin B: Common (Black Wire)

## Diagrams




**Mating Connector:** CMSS 3106F-10SL-4S or equivalent

**Recommended Cable:** Two conductor shielded, teflon jacket, 30 pF/ft (100 pF/m)

## Ordering Information

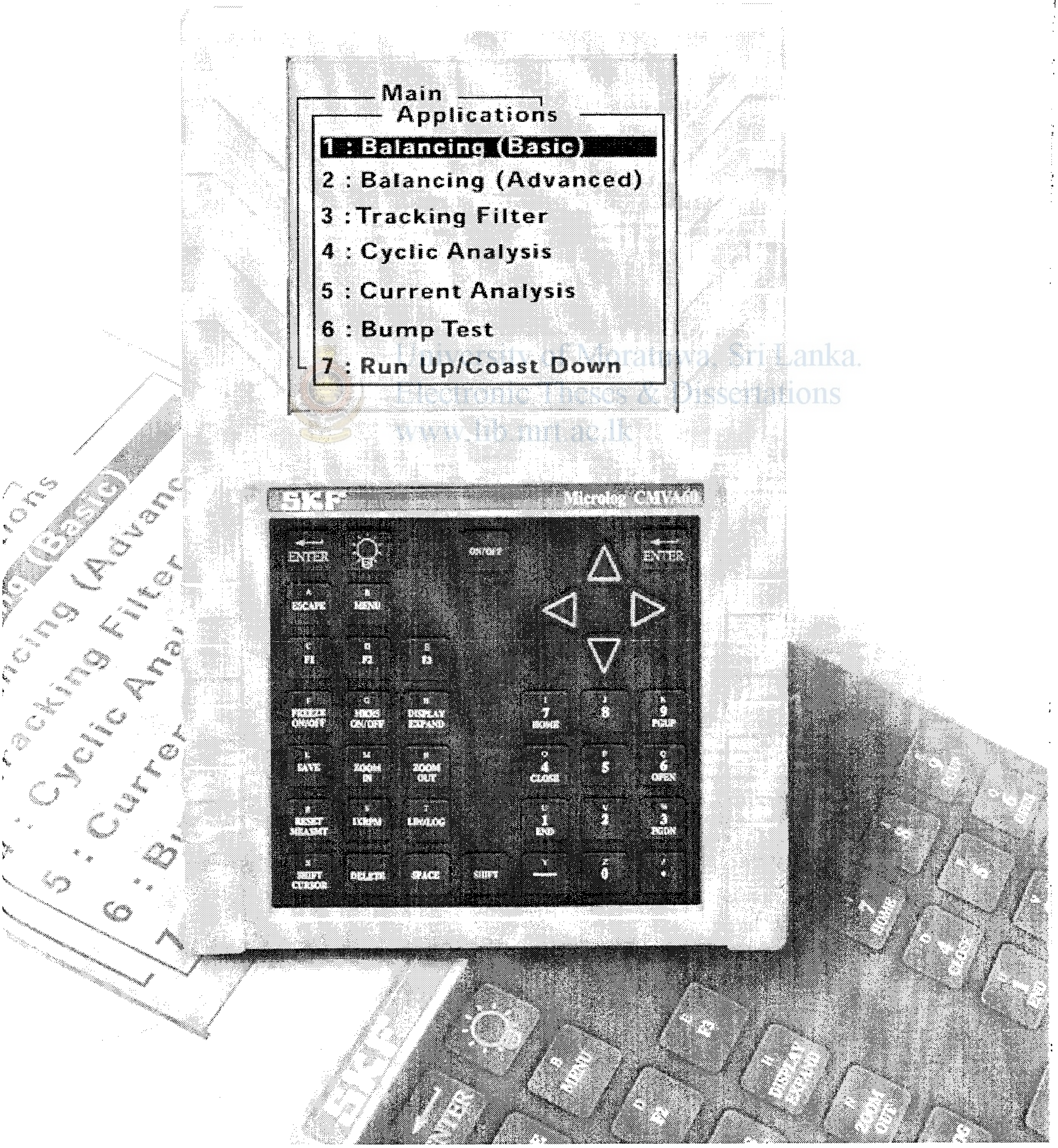
- **CMSS 2200** Low Profile Acceleration Sensor with side exit C5015 two (2) pin connector.
  1/4-28 and M6 mounting studs provided. Calibration sensitivity is provided on each accelerometer package with nominal sensitivity etched on each unit.

- **CMSS 2200-M8** Low Profile Acceleration Sensor with side exit C5015 two (2) pin connector.
  M8 mounting stud provided. Calibration sensitivity is provided on each accelerometer package with nominal sensitivity etched on each unit.


R… …ility Systems
… Court • San Diego, California 92123 USA
… 858-496-3400 • FAX: +1 858-496-3531
…ww.skf.com/cm

Appendix 2

# he "*World's Smartest*" Data Collector
# ust Got Smarter ...



# THE MICROLOG CMVA60™

### *Adds Power, Efficiency and the Expertise of SKF to Your Reliability Efforts*

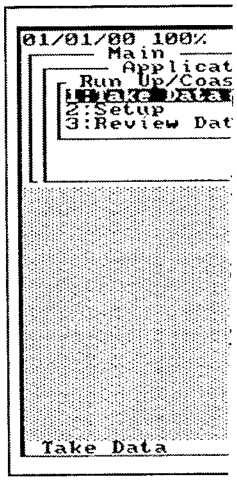
t's  analyst's dream. Imagine an instrument that not onl ollects data, and performs standard analysis fun ns, but also incorporates on-board intelligence to fac the detection, analysis and correction of machine rob ms. With a range of new features and functions, he MVA60 offers the greatest value yet for your eli lity investment. Whether you are new on the job, r a asoned veteran, the CMVA60 is the perfect ana s partner.

## HOT NEW TECHNOLOGY

t ma look the same on the outside, but get a load of the ot w technology inside. While digital signal process-ng ontinues to provide for clarity and reliability in data olle on and processing, a new digital board with dv ced programmable logic brings you the utmost in stat f the art technology. What this means to you is dra tically enhanced processing speed, unbeatable per mance, data capacity and analysis flexibility.

nd the CMVA60 is designed with the future in min setting the stage for ongoing enhancements and additional application capabilities. Upgrades to the unit are easily downloaded to the onboard FLASH memory. Simply connect the Microlog to your PC, and download the electronic files provided, or sign on to the Internet, and download your upgrade right from our web site!

SKF is serious about utilizing the best available technology to power the most advanced Data Collector/Analyzer on the market- right down to the Intel processor inside of the Microlog.




## A COMPREHENSIVE SOLUTION

Virtually all you need to perform like a maintenance superstar is at your finger-tips with the CMVA60. While competi-tors provide "teaser" capabilities that require the purchase of additional mod-ules or products to actually utilize the function, the CMVA60 provides it all in a single package. What does this mean to you? The power to dramatically improve your plant's reliability efforts– well within your budget.



*Application Wizards™ make in the field analysis and correction easy and convenient.*

## INCREASING YOUR EFFICIENCY

Enhanced analysis features make in the field analysis easy and convenient. Improved multi-parameter auto-mation enables the collection of multiple data types for the same point with a single key stroke, then lets you view the first spectrum and band alarms to assess the quality of the data collected. A new battery has more capacity than ever, enabling you to complete your route without interruption. And more than twice the data storage and largest standard internal memory in the industry are just the start.

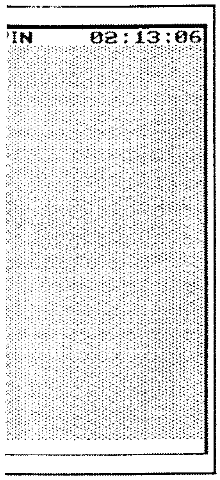
A new Triax Accelerometer, in combination with the SKF Stud Adapter and MARLIN Quick Connect™ Stud, is SKF's fast system setup of positive, prelocated, prepositioned mounting surfaces for more accurate axis alignment and triaxial data measurement.

## NEW APPLICATION WIZARDS!

The CMVA60 carries on the tradition established by the CMVA55 with "Wizards"- embedded intelligence which

I\ ..5 with "Wizards"- embedded intelligence ich facilitate critical analysis and correction with nimal setup effort. Intuitive menus and preset aults ensure that all of the commands you need for a cific application are conveniently located and cessible with a single keystroke. Wizards make it y for novices and experts alike to detect, analyze and rect machine problems.

A new Configuration Wizard is a great time saving feature allowing storage of up to six user-defined application configurations. Unique names can be assigned to identify each configuration.

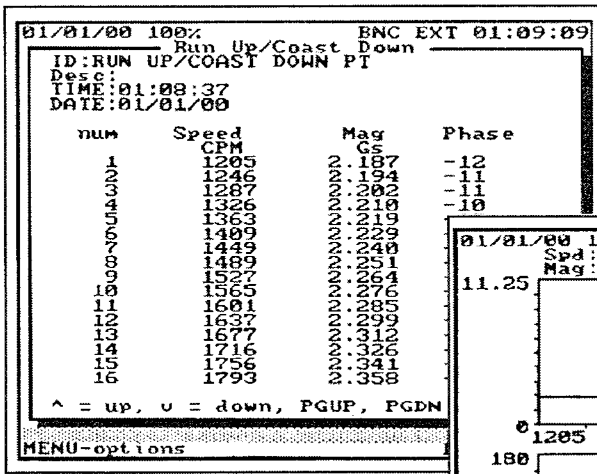


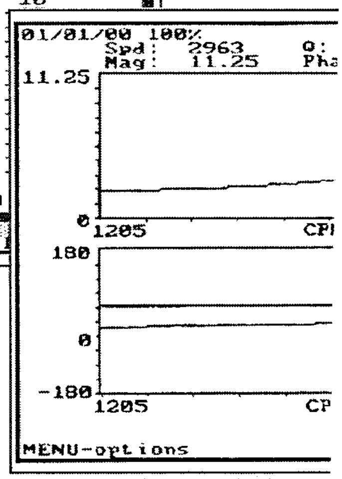

New Run-Up/Coast Down and Bump Test Wizards have been added to the CMVA60 to further enhance your analysis power. Measuring shaft criticals, and resonance testing have never been easier.

An enhanced Field Balancing Wizard enables a beginner to balance a machine with minimal training. It also ovides for greater control and flexibility when performing single plane and dynamic balancing of machine. Permanent storage makes future balancing easier d faster by enabling a user to retrieve information om previous balance jobs, while built in reporting pability makes it easy to document results.

Motor Current and Cyclic Analysis wizards also add wer to the CMVA60 package. Easily analyze and onitor problems like high resistance joints, broken tor bars, cracked end rings and blowholes in die-cast otors. A rotor bar condition assessment report diagnoses the state of the rotor bar. Or use the cyclic alysis wizard to analyze a range of cyclic time domain ents in reciprocating engines, compressors, stamping esses, and more.

A new Operator ID feature akes the MVA60 smart ough to tag the llection of every ta value with a er ID and icrolog serial umber. Important r facilitating aceability for O, insurance, d plantwide

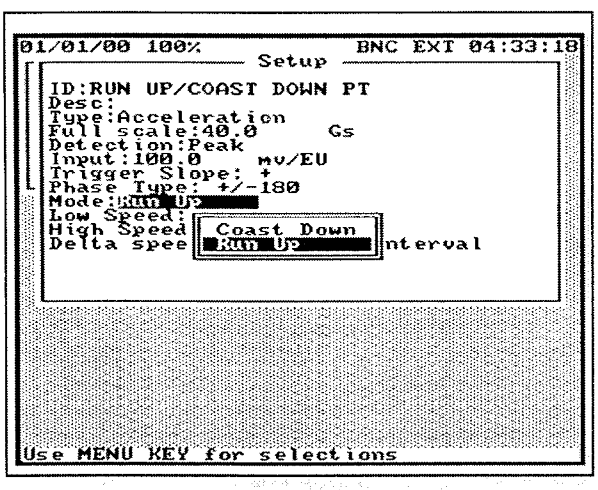


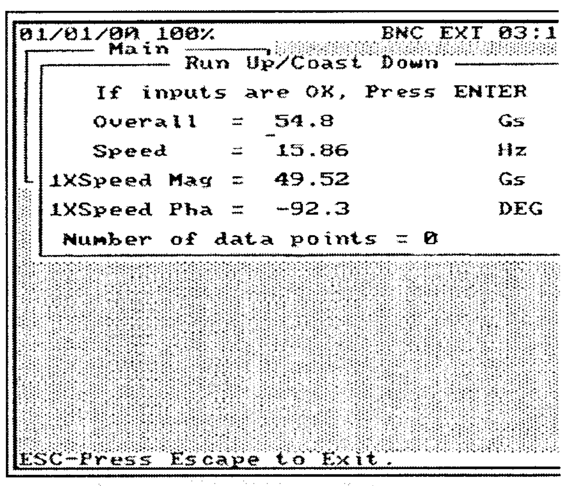




quality programs. A sensor and cable signal checking feature alerts you to cable or sensor problems before data collection begins. The Microlog's smart screen technology automatically accomodates for changes in lighting and provides for better viewing in direct sunlight. And to keep your Microlog running optimally, the CMVA60 even reminds you when calibration is due. Another benefit to ensure adherence to ISO requirements. Does a data collector get any smarter than this?



*...a downloadable Frequency Analysis Module™ automatically overlays not only bearing defect frequencies on a collected spectra, but any defect frequency including unbalance, misalignment, looseness and more!*

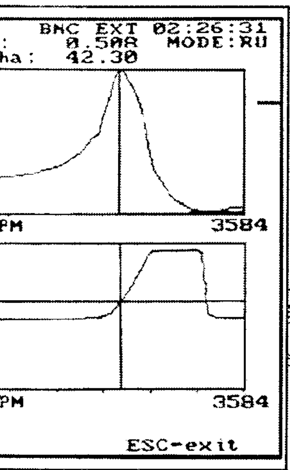


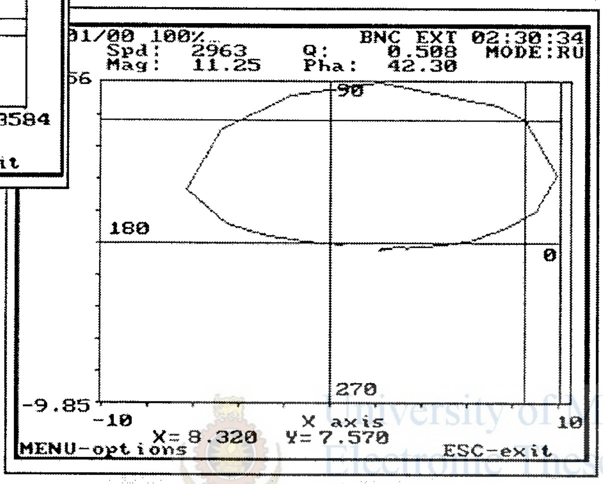
# THE MICROLOG CMVA60 THE ULTIMATE ANALYSIS PARTNER

## ULTIMATE ANALYSIS

Also new in the CMVA60 is a downloadable Frequency Analysis Module which automatically overlays not only bearing defect frequencies on a collected spectra, but any defect frequency including unbalance, misalignment, looseness and more! This makes identification of bearing and other mechanical/structural problems a snap.




And only the CMVA60 enables




effective slow speed bearing monitoring and analysis as low as 0.5 rpm.

## DATA TO DECISION MAKING

PRISM[4] for Windows™ software works with the Microlog CMVA60 to provide an added dimension to your analysis repertoire. Use PRISM[4] for Windows to store, manage and perform more extensive analysis on collected data. Or link to a range of exciting PRISM[4] Solutions™ software programs to perform operating deflection shape analysis, lube-oil analysis, remote monitoring and more. PRISM[4] for Factory Suite™ even provides a condition monitoring interface to enable data to be accessed and displayed in a control room environment.


## SUPPORT AND COMMITMENT

More than ever, the new Microlog CMVA60 is backed by SKF's solid reputation for service and support. With offices in over 130 countries around the globe, SKF provides the products and expertise to help you achieve significant results from your program- right from the start.

Contact your local SKF representative to find out more about how the CMVA60 can work for you!

## ACCELERATION ENVELOPING

Early, reliable and accurate detection of rolling element bearing faults

## APPLICATION WIZARDS:

Embedded intelligence provide step-by-step instructions for performing critical analysis functions.

- **Basic Balancing**
- **Advanced Balancing**
- **Cyclic Analysis**
- **Run-up/Coast-Down**
- **Bump Test**
- **Tracking Filter**
- **Motor Current Analysis**
- **Configuration**

## Frequency Analysis Module:

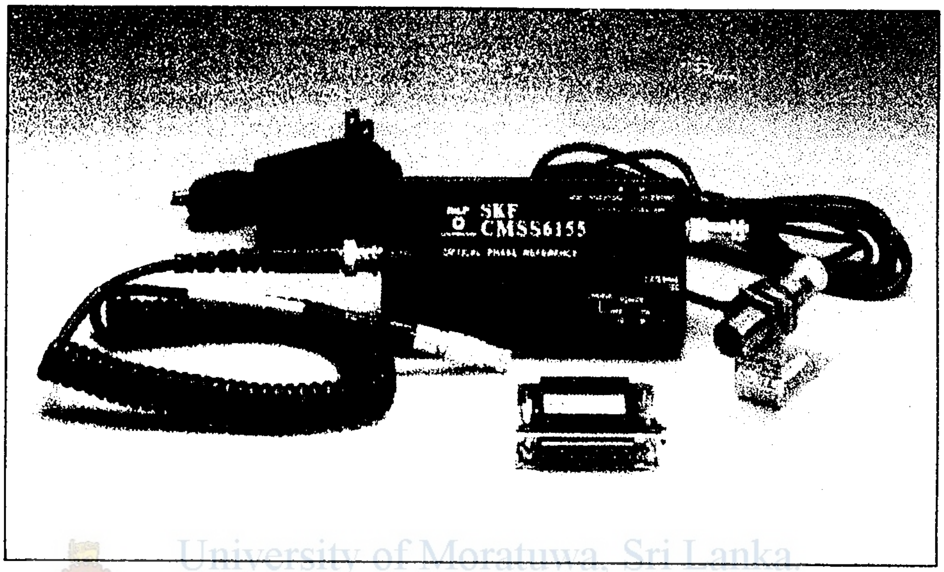
Overlay defect frequencies on collected spectra to detect:

- **Bearing defects**
- **Gearmesh**
- **Misalignment**
- **Unbalance**
- **Looseness**

SKF Condition Monitoring

# Optical Phase Reference Kit
## CMSS6155

CE




### Features

- *Single pulse per revolution TTL trigger source*
- *Positive and negative pulse outputs*
- *Self-contained*
- *Rechargeable Batteries*
- *Green LED on Optical Sensor indicates "ON" target operation*



*The Optical Phase Reference Kit (CMSS6155) is the essential tool for the proactive machinery analyst, and is one of several accessories that SKF Condition Monitoring recommends for use with the popular Microlog Portable Data Collector/FFT Analyzer.*

The Optical Phase Reference Kit extends the functionality of the Microlog, enabling the analyst to perform a range of crucial analysis functions including order tracking and analysis, synchronous time averaging, and field balancing. A TTL compatible pulse train output derived from its single lens retroreflective optical source enables the most effective and highly accurate acquisition of phase data. The instrument also facilitates the analysis of very slow speed machinery by covering a wider usable RPM range.

The Optical Phase Reference Kit consists of a Remote Optical Sensor equipped with a steady state DC light source and photodetector, an interface module, on-off switch, BNC connectors, internal rechargeable batteries and a recharger input. A separate 115 or 230 Vac plug-in charger module is also supplied. Standard camera tripods, user supplied mounting brackets or the optional Magnetic Holder with Movable Arm (CMSS6156) may be used for mounting.

For more information on how the Optical Phase Reference Kit complements your predictive maintenance program, please contact your local SKF Condition Monitoring Representative.

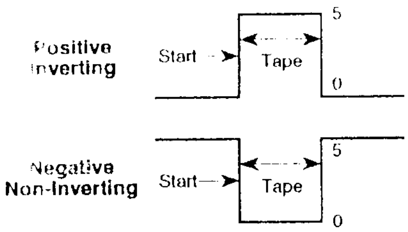
CERTIFIED
ISO 9001

# Optical Phase Reference Kit CMSS6155

## Specifications

**RPM Range:** 1 RPM to 150,000 RPM.

**Output Signals:** TTL positive inverting pulse, 0 to 5 volts. TTL negative non-inverting pulse, 5 to 0 volts.




**Output Impedance:** Less than 50 ohms.

**Output Connectors:** BNC Connector

**Pulse Width:** Determined by size of reflective marker and rotational speed of equipment being monitored.

**Power Source:** Four (4) rechargeable NiCad "AA" batteries provide up to eight (8) hours of continuous operation. Complete recharge requires fifteen (15) hours.

**Cable Length:** 8 feet (2.4 meters) cable from sensor to interface module.

**Dimensions:** Remote sensor, 1.00" (25.4mm) diameter x 2.80" (71.2mm) long with 8 feet (2.4 meters) cable and mounting clamp.

## Ordering Information

**CMSS6155-0-CE**
Self-Powered Sensor complete with Remote Optical Sensor, Mounting Clamp, Interface Module, four (4) "AA" NiCad Batteries, 115V Recharger/Power Supply, and Reflective Tape

**CMSS6155-1-CE**
Self-Powered Sensor complete with Remote Optical Sensor, Mounting Clamp, Interface Module, four (4) "AA" NiCad Batteries, 230V Recharger/Power Supply, and Reflective Tape

### *CMVA55 Micrologs*

**CMSS6155-0-W** Includes:
CMSS6155-0, CMSSR2-J5A-10, and CMSS6155M

**CMSS6155-1-W** Includes:
CMSS6155-1, CMSSR2-J5A-10, and CMSS6155M

**CMSSR2-J5A-10**
Interface Cable

**CMSS6155M**
User Manual

### *CMVA10, CMVA30* and CMVA40 Micrologs*

**CMSS6155K-0-CE** Includes:
CMSS6155-0, CMSS6135E, CMSS50189, and CMSS6155M

**CMSS6155K-1-CE** Includes:
CMSS6155-1, CMSS6135E, CMSS50189, and CMSS6155M

**CMSS6135E**
Microlog Phase Adaptor

**CMSS50189**
CMSS6155/CMSS6135E Interface Cable

## Accessories

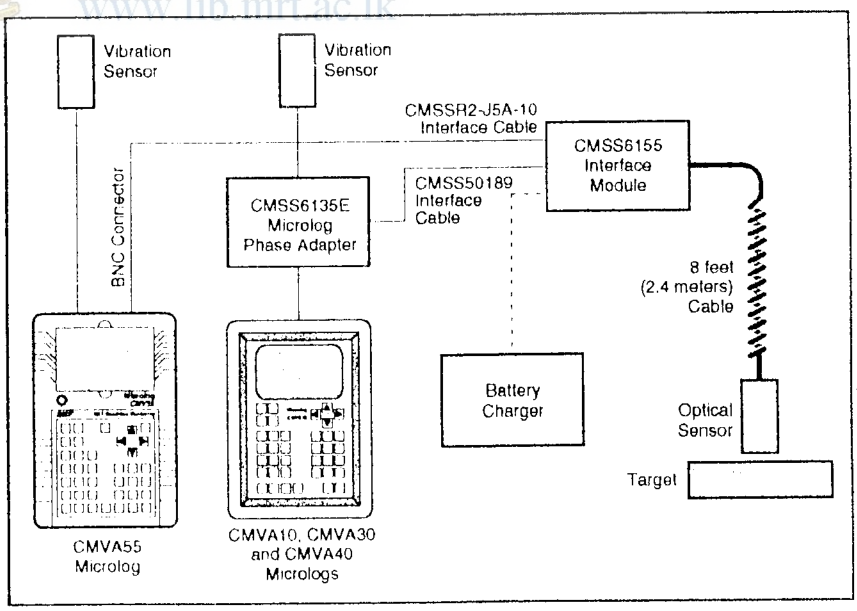
**CMSS T-5**
Reflective Tape for use with Remote Optical Sensor, 5 feet (1.5 meters) Roll x 1/2" (12.7mm) Wide.

**CMSS50401**
25 feet (7.5 meters) CMSS6155 Interface Module to Optical Sensor Extension Cable.

**CMSS6156**
Magnetic Holder with Movable Arm and Remote Optical Sensor Attachment Hardware.

** Not approved for use in hazardous areas.*



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**Your SKF Condition Monitoring Representative:**

# PRISM⁴ for Windows™ Data Management and Analysis Software

## Version 1.32

## Description

### Database Filtering and Querying

Database Filtering allows PRISM⁴ users to "filter" their PRISM⁴ databases for specific POINTs that meet user selected filtering criteria. Over 40 filter criteria are available. Examples of filter criteria are POINT Type, Last Measurement Value, ID and Description text strings, POINT setup parameters such Full Scale, Maximum Frequency and others. Filtering criteria can be combined together for very specific queries. A filtered list of POINTs are the output of this process and this new list of POINTs allows very efficient data analysis, reporting and maintenance. Database filter configurations can be stored for later reuse. All modes of database listing; hierarchy (standard) and the new "Active" ROUTE and Workspace listing modes can be filtered.

### "Active" ROUTEs and Workspaces

"Active" ROUTEs allow full access to data displays, POINT Setups and Reporting from a ROUTE. Workspaces are a new database listing mode that allows users to drag and drop database elements (Sets, Machines or POINTs) from a ROUTE or the Plant Hierarchy to a "Workspace" window. Users can name and store Workspaces for later reuse. Workspaces provide a convenient way to analyze, report and maintain small numbers of Machines or POINTs of special interest.

Spectral Band Alarms can now be assigned to one or more POINTs at one time. This greatly enhances the user's ability to use spectral band alarms as a general defect detection method. Spectral Band Alarms can also be cleared from a group of user specified POINTs with one user command.

PRISM⁴ now provides a fully ASCII data import/export feature as well as a full MIMOSA CRIS import/export functionality. The MIMOSA Export feature is a special user option.

PRISM⁴ is now Year 2000 compliant.

PRISM⁴ fully supports the new MARLIN™ data collection system. New MARLIN™ POINTs can be created, stored and downloaded into the new MARLIN™ data collector. Process and overall vibration data from the MARLIN™ can be uploaded, stored, displayed and reported on from PRISM⁴.

Other important improvements to PRISM⁴ include; a new Modify by Attribute screen, the ability to delete all or a group of FAM frequencies for one or more POINTs, improved screen size and location defaults as well as new tiling options.

PRISM⁴ for Windows Version 1.32 is part of the SKF Condition Monitoring's family of Windows based condition monitoring products. This family includes the PRISM⁴ On-line System and the PRISM⁴ Pro Knowledge Based System.

## Detailed Features

### Global Settings

- System Settings
  - English or Metric
  - CPM or Hz
- Spectrum
  - Across Machine Data Traversal
  - Speed Tagging
  - Cursor Micro-manipulation (Harmonic, Sideband, Time Interval)
  - Color Coded Cursor Information Boxes
  - Linear or Log X and Y Scaling
  - Auto Scaling, Grid and Order Normalized Frequency Axis (On or Off)
  - Amplitude Threshold Percentage
  - Waterfall Display Slant Angle Setting
  - Palogram Display Slant Angle Setting
  - Date Range Setting
  - User Selectable Color Settings
- Windows Printer Setup

### Database

- Unlimited hierarchical data structure
- Unlimited data storage per Point, data storage depth defined by user
- Database supports portable data collection as well as on-line monitoring system
- On-line database structure has three areas;
  - Scheduled data
  - Unscheduled data (data in alarm)
  - Archived data (long term data storage)
- Machine templates available for rapid database setup
- Data Import/Export functionality
  - Full ASCII (Setups and Data) Import/Export
  - MIMOSA CRIS Import/Export, Export is a special option
  - PRISM⁴ Binary Format Import/Export
- Network database functionality

### Measurement Types

- Acceleration
- Velocity
- Displacement
- Enveloping (Acc or Vel)
- *SEE*™ (Spectral Emitted Energy)
- HFD (High Frequency Detection)
- Orbit (Filtered and Unfiltered)
- Shaft Centerline
- Smax
- Volts (AC or DC)
- Temperature
- Pressure
- Flow
- RPM
- Current (AC or DC)
- Logic Level
- Linear Displacement
- Counts and Count Rate
- User Defined Units (Wild Card)
- MARLIN™ MCD POINT (Vel, Env, Temp)
- MARLIN™ Inspection POINT (Single, Multiple)
- MARLIN™ Operating Hours

### Alarming

- Unique visual alarm status indicators
- Four (4) alarm levels per Point
- User definable spectral fault frequencies including rolling element bearing frequencies via the Frequency Analysis Module functionality

# PRISM⁴ for Windows™ Data Management and Analysis Software Version 1.32

**ALARMING (CONTINUED)**

- Overall Level Alarms
  - Danger High
  - Alert High
  - Alert Low
  - Danger Low
- Overall Forecast Alarm
- Overall Statistical Alarm
- Overall Percent Change Alarm
- Spectral Band Alarms (unlimited)
- Spectral Envelope Alarm
- Phase Angle Alarm

**COMMUNICATION**

- MARLIN™ data collector
- Microlog (CMVA55, CMVA10, CMVA30, CMVA40)
- Picolog (CMVL10)
- Download and Download Functions by ROUTE
- FAM Frequencies Downloadable to CMVA55 (firmware 3.80)
- Downloadable Spectral Band Alarms (up to 12 bands per POINT, Microlog firmware 3.22+)
- Upload Reporting per Route (Standard or User Defined Content including data plots)
- Manual data entry
- Full modem support with selectable baud rate (1200–115200) and COM ports

**DATA DISPLAYS**

- Overall Trend Displays (Single/Overlay)
- Spectrum Displays (Single/Overlay/ Waterfall/Palogram)
- Phase Plot (Polar Vector, Mag/Phase Trend)
- Time Domain Displays (Single, Overlay)
- Shaft Orbit (Filtered, Unfiltered, Single Orbit, Multiple Orbit Overlay, Mag/Phase Trend)
- Shaft Centerline (Clearance plot, Distance Trend plot)
- Smax (Polar, Mag/Phase Trend)
- Event Log

*Display features include:*

- Trend Curve Fit
- Trend Percent of Full Scale Display
- Extract Spectra/Time from Trend
- Spectrum Envelope Editing Capability
- Baseline Spectrum Storage
- User definable Machinery and Bearing Fault Frequency Markers (Frequency Analysis Module – FAM)
- Spectrum Post-Processing (integration/ differentiation)
- Extract Trend or Spectrum from Waterfall
- Extract Spectral Band Trend Display from Waterfall

**REPORTS**

- Last Measurement
- Exception
- Overdue
- Upload
- Set Statistics
- Database Setup
- Route Statistics
- History Reports
- Quick Reports
  - On Screen Report with access to data plots, such as trend, spectra and time domain
  - Last Measurement, Exception, Overdue
- Send reports to screen or printer
- Manipulate reports in a word-processor via Windows Clipboard
- Customizable report formats
- Data Displays can be included in reports

**PRISM⁴ PRO CONNECTION**

- Interface with PRISM⁴ Pro Knowledge Based Diagnostic system
- PRISM⁴ Pro capable to go "On-Line" for continuous data collection and data analysis

**NETWORK OPTION**

- Users can access the on-line data via PRISM⁴ 's network licenses (five, ten and fifteen user licenses are available)
- Compatibility with Novell Netware 3.12, Microsoft NT 4.0 Server, Windows 95 Peer to Peer



**GENERAL**

- Microsoft Windows 95 and NT 4.0 graphical-user interface with pull-down menus, tool bars and icons
- Multi-tasking environment
- Edit, delete, copy and move commands via Mouse
- On-line context sensitive Help
- Supports all Microsoft Windows 95 or NT 4.0 compatible printers
- Complete User Manual
- One year Support and Maintenance Warranty
- On-site Product Training and System Setup Available


CERTIFIED
ISO 9001

## Hardware Requirements

**RECOMMENDED HARDWARE**

- 166 MHz Pentium Processor or better
- Microsoft Windows 95, DOS 7.0
- 32 Mbytes RAM
- 3 Gbyte Hard Disk Drive
- 12X CD-ROM
- High density 3.5 inch floppy disk drive
- Super VGA graphics adapter with 256 colors, 1 Mbyte RAM
- High Speed Serial/Parallel Ports
- HP LaserJet 5 or HP DeskJet Color
- Backup System:
  *Tape:* HPT1000
  *Zip Drive:* Iomega Zip or Jazz Drive

**MINIMUM HARDWARE**

- 100 MHz Pentium Processor
- Microsoft Windows 95 and DOS 7.0 or greater
- 16 Mbytes RAM
- 1 Gbyte Hard Disk Drive
- High density 3.5 inch floppy disk drive
- Super VGA graphics adapter with 256 colors, 1 Mbyte RAM
- High Speed Serial/Parallel Ports
- HP DeskJet
- Backup System:
  *Tape:* HPT1000
  *Zip Drive:* Iomega Zip or Jazz Drive

## Ordering Information

**CMS100** PRISM⁴ for Windows

**CMS200-x** PRISM⁴ for Windows Network Authorization

**CMS150** LMU/MIM Driver

**CMS101** PRISM⁴ Pro

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**Appendix 4** Vibration Sampling Points

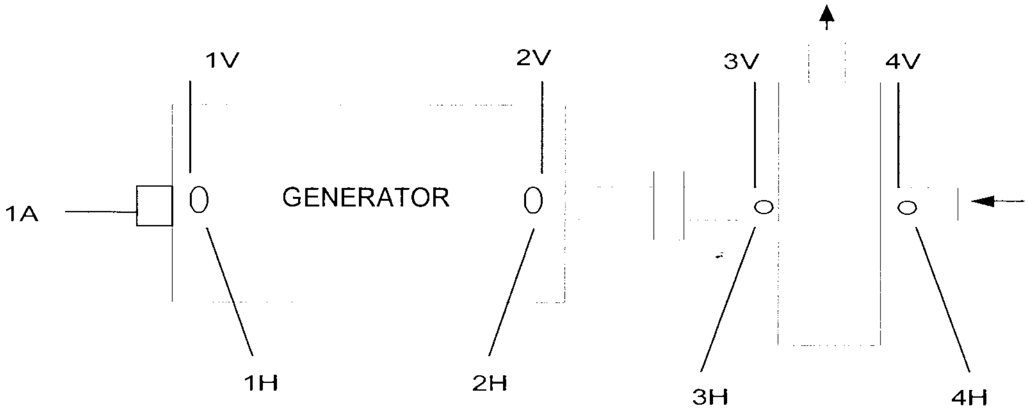


Figure A : Horizontal pump sampling points

1 – Generator non-driven end
2 – Generator Driven End
3 – Turbine driven End
4 – Turbine Non-driven End
H – Horizontal
V - Vertical
R – Radial [2]



For a vertically mounted machinery sample positions are taken radially.

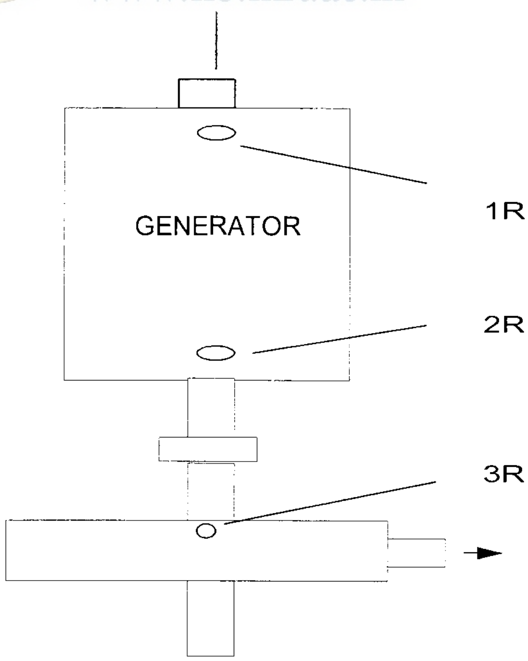


Figure B : Vertical pump measurement locations

## Appendix 5 Vibration Standard VDI 2056

| | | | RMS Velocity (mm/s) |
|---|---|---|---|
| Not Permissible | Not Permissible | Not Permissible | 45<br>11.2 |
| Just Tolerable | Just Tolerable | Just Tolerable | 4.5 |
| Allowable | Allowable | Allowable | 2.8<br>1.8 |
| Good<br>Small machines up to 15kW | Good<br>Medium machines 15-75kW or up to 300kW on special foundations | Good<br>Large machines with rigid and heavy foundations whose natural frequency exceeds machine speed | 1.12<br>0.71<br>0.45<br>0.18 |
| **Group K** | **Group M** | **Group G** | |

<u>Table: Vibration levels</u>

Appendix 6

# ISO 2372 Vibration Diagnostic Table (Horizontal Shaft)

| | Excessive **Horizontal** Vibration Indicates: | Excessive **Vertical** Vibration Indicates: | Excessive **Axial** Vibration Indicates: | Excessive **Structural** Vibration Indicates: | Notes |
|---|---|---|---|---|---|
| Imbalance | YES | NO | NO | NO | Horizontal > Axial |
| Misalignment | NO | YES | YES | NO | Axial > Horizontal |
| Looseness | YES | YES | NO | YES | Vert. ≥ Horizontal |
| Electrical Faults Measured as Vibration | | | | | To detect an electrical problem: Turn off machine power and monitor vibration. If the vibration immediately drops, the problem is electrical. |

Note:

On an overhung machine, imbalance and misalignment may display similar characteristics. Use phase measurements to differentiate between the two.

Note:

YES = ISO 2372 Unsatisfactory - Unacceptable Levels

NO = ISO 2372 Good - Satisfactory Levels.

# ISO 2372 Vibration Diagnostic Table (Vertical Shaft)

| | Excessive **Radial 1** Vibration Indicates: | Excessive **Radial 2** Vibration Indicates: | Excessive **Axial** Vibration Indicates: | Excessive **Structural** Vibration Indicates: | **Notes** |
|---|---|---|---|---|---|
| **Imbalance** | YES | NO | NO | NO | Radial > Axial |
| **Misalignment** | YES | NO | YES | NO | Axial > Radial |
| **Looseness** | YES | NO | NO | YES | |
| **Electrical Faults Measured as Vibration** | | | | | To detect an electrical problem: Turn off machine power and monitor vibration. If the vibration immediately drops, the problem is electrical. |

Note:

Radial 1 and Radial 2 positions differ by 90 degrees.

Note:

YES = ISO 2372 Unsatisfactory - Unacceptable Levels

NO = ISO 2372 Good - Satisfactory Levels.

# ISO 2372 Vibration Diagnostic Table (Overhung - Horizontal Shaft)

| | Excessive Horizontal Vibration Indicates: | Excessive Vertical Vibration Indicates: | Excessive Axial Vibration Indicates: | Excessive Structural Vibration Indicates: | Notes |
|---|---|---|---|---|---|
| Imbalance | YES | NO | YES | NO | Hor. & Axial > Vert. |
| Misalignment | YES | NO | YES | NO | Hor. & Axial > Vert. |
| Looseness | YES | YES | NO | YES | Vert. ≥ Horizontal |
| Electrical Faults Measured as Vibration | | | | | To detect an electrical problem:<br><br>Turn off machine power and monitor vibration. If the vibration immediately drops, the problem is electrical. |

Note:

On an overhung machine, imbalance and misalignment may display similar characteristics. Use phase measurements to differentiate between the two.

Note:

YES = ISO 2372 Unsatisfactory - Unacceptable Levels

NO = ISO 2372 Good - Satisfactory Levels.

Appendix 4

# Spectrum Analysis Table

| | Primary Plane | Detection Units | Dominant Freq(s) | Phase Relationship | Comments |
|---|---|---|---|---|---|
| **Imbalance** | | | | • Note: Phase references are accurate within ±30 degrees. | |
| Mass | radial | Acc/Vel/Disp | 1X | 90 degree phase shift as sensor is moved from horiz. to vertical position. No radial phase shift across the machine or coupling. | |
| Overhung Mass | axial & radial | Acc/Vel/Disp | 1X | Axial reading will be in phase. | Account for change in sensor orientation when making axial measurements. |
| Bent Shaft | axial & radial | Acc/Vel/Disp | 1X | 180 degree° phase shift in the axial direction across the machine with no phase shift in the radial direction. | |
| **Misalignment** | | | | | |
| Angular | axial | Acc/Vel/Disp | 1X,2X | A phase shift of 180 degrees in the axial direction will exist across the coupling. | With severe misalignment, the spectrum may contain multiple harmonics from 3X to 10X. If vibration amplitude in the horizontal plane is increased 2 or 3 times, then misalignment is again indicated. (Account for change in sensor orientation when making axial measurements.) |
| Parallel | radial | Acc/Vel/Disp | 1X,2X | A phase shift of 180° degrees in the radial direction will exist across the coupling. Sensor will show 0° or 180° degrees phase shift as it is moved from horizontal to vertical position on the same bearing. | |
| Combination of Ang. & Parallel | axial & radial | Acc/Vel/Disp | 1X,2X | A phase shift of 180° degrees in the radial and axial direction will exist across the coupling. | |
| **Mechanical Looseness** | | | | | |
| Structural | radial | Acc/Vel/Disp | 1X | Phase shifts of 180° degrees will exist between the machine's feet, baseplate, and/or foundation if the machine is rocking. | Usually caused when the machine's foundation degrades to such an extent that it is no longer stiff, causing the machine to 'rock'. |
| Soft Foot | radial | Acc/Vel/Disp | 1X,2X,... | Phase will shift when the machine foot is tightened. | Result of the machine footing coming loose from the foundation. |
| Wear/Fitting | axial & radial | Acc/Vel/Disp | 1X,2X,...10X | Phase reading will be unstable from one reading to the next. | Vibration amplitudes may vary significantly as the sensor is placed at different locations around the bearing. (Account for change in sensor orientation when making axial measurements.) |

# Spectrum Analysis Table (Continued)

| | Primary Plane | Detection Units | Dominant Freq(s) | Phase Relationship | Comments |
|---|---|---|---|---|---|
| **Local Bearing Defects** | | | | • Note: Phase references are accurate within ±30 degrees. | |
| Race Defect | radial | Acc/Env/*SEE* | 4X...15X | No correlation | With Acceleration measurements, bearing defect frequencies appear as a wide "bump" in the spectrum. Bearing defect frequencies are non-integer multiples of running speed (i.e. 4.32 X Running Speed). |
| | | | | | |
| **Gear Defect** | | | | | |
| Gear Mesh | radial | Acc/Env/*SEE* | 20X...100X | No correlation | The exact frequency relates to the number of teeth each gear has times the shaft rotation speed (running speed). |
| | | | | | |
| **Electrically Induced** | | | | | |
| AC Motors | radial | Acc/Vel/Disp | Line Freq. (100 or 120 Hz) | No correlation | Defect Frequencies can be seen at exactly twice the line frequency. |
| DC Motors | radial | Acc/Vel/Disp | SCR Freq. | No correlation | DC motor problems due to broken field windings, bad SCR's or loose connections are reflected as higher amplitudes of the SCR frequencies (6x$F_L$) |