DEVELOPMENT OF A PREDICTIVE ELECTRICAL MOTOR AND PUMP MAINTENANCE SYSTEM USING VIBRATION ANALYSIS

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DECLARATION OF INDEPENDENT WORK

I, ERWIN SMITH, hereby declare that the research project which has been submitted to Technikon Free State by me for the attainment of the degree MAGISTER TECHNOLOGIAE: ENGINEERING: MECHANICAL, is my independent work and has not been submitted by me or anyone else previously in view of attaining any qualification.

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SYNOPSIS

This dissertation covers the development and implementation of a predictive maintenance monitoring programme for the Water Supply Directorate of the Department of Water Affairs, Namibia. The maintenance policy in the Directorate was based on a combination of breakdown maintenance and preventative maintenance. Thus maintenance was carried out when a specific type of equipment was forced out of production. The cost of the replacement and repair of equipment increased substantially and a condition-based maintenance system was investigated and implemented. The purpose of condition monitoring maintenance is to find a convenient time for maintenance to be carried out.

Different types of condition monitoring technologies exist. After the different types of technologies have been investigated, vibration-based predictive maintenance was chosen. The project includes results from a number of field case studies and proves that vibration analysis can be used to determine the mechanical condition of electrical motors and pumps. The monitoring programme covers a total of 80 pump sets comprising mainly of electrical motors and pumps ranging from 45 to 2 400 kilowatt. In general, the programme is based on the determination of suitable monitoring parameters by taking measurements at regular intervals of the vibration characteristics of a machine.

The generalised approach to vibration analysis in a predictive maintenance programme of machinery requires a sound understanding of fundamental theoretical concepts associated with machine element dynamics and the nature of the dynamic forces and instabilities which excite vibration in electric motors and centrifugal pumps, together with the ability to plan concise experiments to obtain practical data regarding the cause of failure.

Machine faults will cause a change in the shape of the vibration frequency spectrum. The cause of the fault can be diagnosed by determining which frequency components have increased and to match them with the different characteristics of vibration. Basically, all machines vibrate at the same characteristic level depending upon the machine's design and operation. As a machine begins to age and deteriorate, vibration increases sporadically or gradually and each machine, regardless of its mechanical design, creates its own unique vibration. A vibration problem can be analysed by reviewing its component frequencies and determining at what frequency the vibration occurs. Using a vibration analyzer, it is possible to measure the frequency and corresponding amplitude of each component.

It was found that the greatest vibration normally occurs at the running speed of the machine. It can be concluded that unbalance could be a major cause of this. Misalignment was normally identified at two or three times running speed. Rolling element bearings produce their own high frequency with low amplitude vibration. Defects in rolling element bearings can be separated from the vibration produced by other mechanical components. On sleeve bearings, excessive clearances were found to be the main cause of vibration, producing many harmonic-related frequencies.

Another problem which may arise, is mechanical looseness, of which the amplitude is normally dependent on the amount of looseness and the mechanical design of the machine. This was characterised at twice the running speed with higher than usual harmonics. Resonance is another problem that could cause excessive vibration. Each part of a machine, as well as the machine itself, has a natural frequency and this frequency, relative to a machine's running speed, is of great importance since no machine should be operated in a resonant condition.

By utilising a predictive maintenance programme such as vibration monitoring, the condition of vital machinery can be determined effectively. This monitoring system can give early warnings of impending failures, determine the cause of fault and can be used to schedule repairs. Such a system can therefore prevent catastrophic failure, lengthen the life of machinery and reduce maintenance costs. Since installation of the programme, the number of unexpected failures on monitored machines has been greatly reduced and the savings gained from the programme (savings associated with maintenance costs) enabled a pay-back on investments within 18 months of installation.

UITTREKSEL

Die verhandeling bespreek die ontwikkeling en implementering van 'n voorkomende instandhouding-moniteringsprogram vir die Waterleweringsdirektoraat van die Departement van Waterwese in Namibia. Voorheen is daar slegs aandag aan 'n masjien gegee wanneer die bedryfsbehoeftes dit genoodsaak het. Die koste van die vervanging en herstel van toerusting het sodanig toegeneem dat 'n ondersoek na 'n toestandgebaseerde instandhoudingstelsel geloods is. Die doel van toestandgebaseerde instandhouding is om die mees geleë tyd vir die uitvoering van herstelwerk te bepaal.

Verskeie toestandmoniteringstegnieke bestaan. Nadat verskeie tegnieke ondersoek is, is daar op vibrasie-gebaseerde, voorkomende instandhouding besluit. Die projek sluit resultate in met verwysiging na 'n aantal gevallestudies, wat daarop dui dat vibrasie-monitering effektief gebruik kan word om die meganiese toestand van elektriese motors en pompe te bepaal. Die moniteringsprogram dek 80 pompstelle met 'n drywing van 45 tot 2 400 kilowatt. Die doel van die program was om geskikte moniteringsparameters te bepaal en dan periodieke metings van die eienskappe van die masjien met gereelde tussenposes te neem.

Die algemene benadering van vibrasie-analise in 'n voorkomende instandhoudings-program benodig 'n deeglike begrip van die fundamentele teoretiese konsepte wat verband hou met masjien-elementdinamika en die oorsake van die dinamiese kragte en onstabiele toestande wat vibrasie op elektriese motors en sentrifugale pompe het. Dit sluit ook die vermoë in om eksperimente op te stel en velddata in te samel om die oorsake van onklaarheid te bepaal. Die projek demonstreer dat vibrasie-monitering gebruik kan word om die meganiese toestand van pompe en elektriese motors te bepaal.

Masjienprobleme sal 'n verandering in die patroon van 'n frekwensiespektrum veroorsaak. Die oorsaak van die fout kan dan gediagnoseer word deur te bepaal watter frekwensiekomponent verander het en deur die frekwensie te vergelyk met die verskillende karakteristieke eienskappe van vibrasie. Basies toon alle masjiene ooreenkomste in hul karakteristieke vibrasievlakke, afhangende van elke masjien se ontwerp en bedryf. Met die

veroudering en gepaardgaande agteruitgang van masjiene, sal die vibrasie begin toeneem en elke masjien sal, ongeag die ontwerp, sy eie unieke vibrasiepatroon ontwikkel. 'n Vibrasieprobleem kan geanaliseer word deur te kyk na die komponentfrekwensies en te bepaal by watter frekwensie die vibrasie plaasvind. Deur gebruik te maak van 'n vibrasie-analiseerder is dit moontlik om elke komponent se frekwensie te meet met sy ooreenstemmende amplitude.

Daar is gevind dat die mees dominerende vibrasie gewoonlik plaasvind by die bedryfspoed van 'n masjien. Dit kan afgelei word dat wanbalans die grootste oorsaak van vibrasie is. Wanbelyning is gewoonlik gevind by die tweede en derde harmoniese frekwensie van die bedryfspoed. Rollaers genereer hulle eie hoë frekwensie met lae amplitude in vibrasie. Defekte in rollaers kan onderskei word van vibrasie wat opgewek word deur ander meganiese oorsake. Op astaplaers is gevind dat groot speling tussen die as en laer, losheid veroorsaak met harmoniese frekwensies as gevolg daarvan. Meganiese losheid hang van die mate van losheid af, asook die meganiese ontwerp, of word gekarakteriseer deur die tweede harmoniese frekwensie van die bedryfspoed en waarvan die harmoniese frekwensies hoër as gewoonlik is. Resonansie is 'n ander probleem wat uitermatige vibrasie veroorsaak. Elke komponent van 'n masjien, asook die masjien as geheel, het sy eie natuurlike frekwensie. Die frekwensie relatief tot die masjien bedryfspoed is baie belangrik, aangesien 'n masjien onder geen omstandighede by 'n resonante toestand bedryf mag word nie.

Deur vibrasie-analise in 'n voorkomende instandhoudingsprogram te gebruik, kan die toestand van kritieke masjiene effektief bepaal word. Hierdie moniteringsstelsel kan 'n vroeër waarskuwing van moontlike masjienfalings gee, die oorsaak van die probleem kan dan bepaal word en instandhouding kan geskeduleer word. So 'n stelsel kan katastrofiese falings voorkom, die leeftyd van masjinerie verleng en instandhoudingskostes verlaag. Sedert die implementering van die program is onverwagte falings op die gemoniteerde masjiene verlaag. Die besparing op instandhoudingskoste het meegebring dat die koste van die moniteringstoerusting oor 'n tydperk van agtien maande verhaal is.

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CHAPTER 1

INTRODUCTION

1.1 Introduction and Background

The function of bulk water supply in Namibia is to supply water to towns, mines and communities. Substantial amounts of money were spent over the years and nowadays on the repair and maintenance of equipment.

It is estimated in recent surveys around the world that one third of all maintenance costs is wasted as a result of unnecessary or improperly carried-out maintenance [1]. Because the Department has a standby for each installed machine it is possible to use statistical information, and machines were allowed to run to failure. The dominant reason for this ineffective maintenance activity is the lack of factual data that quantifies the actual need of repair or maintenance of equipment. In the past several efforts have been launched in the Department to start a condition-monitoring programme, but without much success.

The purpose of this thesis is to describe the implementation of a condition-monitoring system to extract information from machines to indicate their condition, their rate of change and to operate and maintain the machines safely and economically.

One major problem is the fact that the equipment is distributed all over the country. Long distances must be travelled to monitor equipment. Therefore the maintenance system must be efficient and provision must be made for other methods to be incorporated, for example on-line monitoring by using modems. Another problem which was faced was the total lack of equipment history and the fact that most of the equipment was approximately 5 years and longer in operation.

During recent years, development, research and implementation have grown in the industry of condition-monitoring techniques and have been applied to monitor the operating condition of equipment to provide the means to manage the maintenance operation. Therefore the researcher has committed himself to the project by setting up and carrying out a condition-monitoring system based on the detection and analysis of machinery developing mechanical and electrical defects. Condition-monitoring techniques will be investigated with more emphasis placed on vibration monitoring as a predictive maintenance tool. Using the vibration characteristics of a machine as an indication of its state of health can be used to predict failures and to help with the planning of maintenance [2]. The system should provide the ability to automate data acquisition, data management, trending, report generation, and diagnostics of incipient problems.

1.2 Maintenance Strategies

1.2.1 General

To select the most suitable maintenance strategy, the following different standard maintenance strategies should be considered.

1.2.2 Breakdown Maintenance (Fix it when it breaks)

Breakdown is a maintenance technique that is based on demand - the person responsible for maintenance waits for machine or equipment failure before any maintenance action is taken. In truth it is a "no maintenance approach to management". It can be the most expensive method of maintenance management. When this strategy is used, management does not spend any money on maintenance unless a machine fails to operate. High expenses are also associated with this type of maintenance because of high spare parts inventory costs, high overtime labour costs and high machine downtime [3]. Analysis of maintenance cost indicates that a repair performed in this strategy mode will average about three times higher than the same repair made within a scheduled or preventative mode [1]. Scheduling the repair provides the ability to minimize the repair time and associated labour costs.

1.2.3 Preventative Maintenance (Regularly scheduled inspections and overhauls)

Preventative maintenance is time-driven. This involves scheduling periodic downtime for the visual inspection of parts or to replace equipment, regardless of its condition. Maintenance tasks are based on elapsed time or hours of operation[4]. Figure 1-1 illustrates the Hazard rate curve or "bathtub" of the statistical life of a machine-train [1][3]. The curve relates the failure rate and time in accordance with the stage (1) early failures (2) random failure and (3) wear-out failure. It indicates that a new machine has a high probability of failure during the first few weeks of operation. Following this initial period, the probability of failure is relatively low for an extended period of time. After this normal machine life period the probability of failure increases sharply with elapsed time. In a preventive maintenance policy machine repairs or rebuilds are scheduled on the mean time to failure statistic (MTTF) [1][4]. This is the expectation of time to the first failure.

The choice of the best maintenance interval presents a difficult problem, with two opposite extremes to be avoided [2]: Too frequent maintenance wastes production time and increases the risk of trouble arising from human errors in re-assembly. Too long an interval results in an unacceptable number of unexpected machine failures during operation. A compromise between these two extremes can be established by experience, but machine failures will continue to occur. This is also illustrated in Figure 1-2. This method can be applied if many similar machines are used, which have the same operating conditions. The probability of similar problems is higher and with some experience the best time for scheduled inspections or repairs can be obtained.

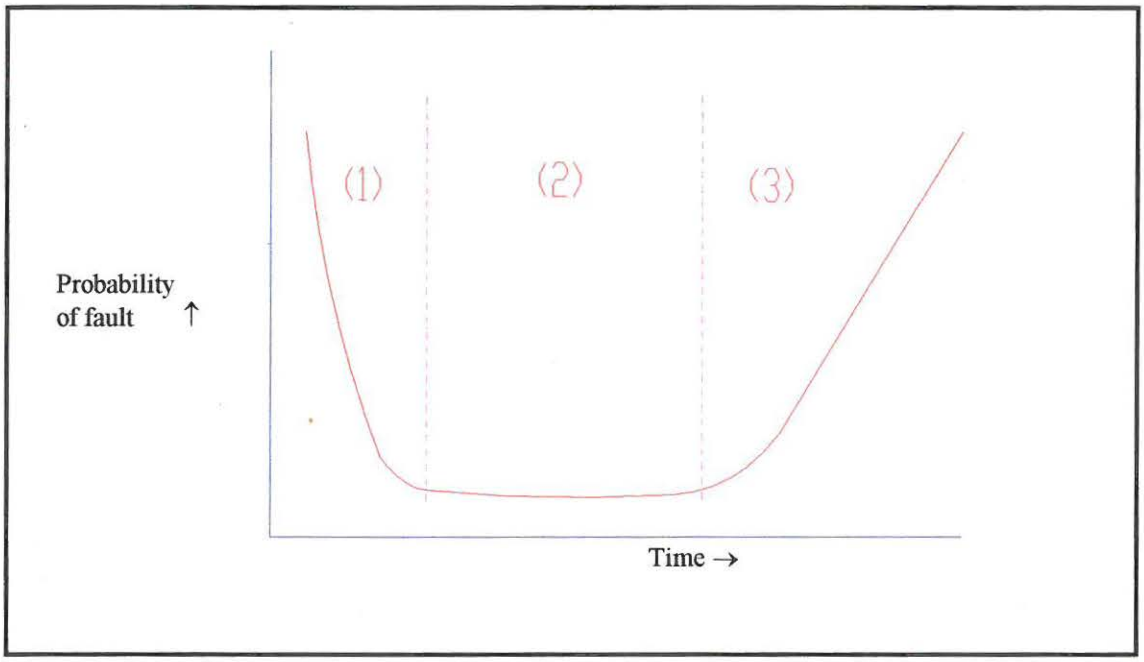


Figure 1-1: Bathtub curve illustrates the life cycle of a specific classification of machinery

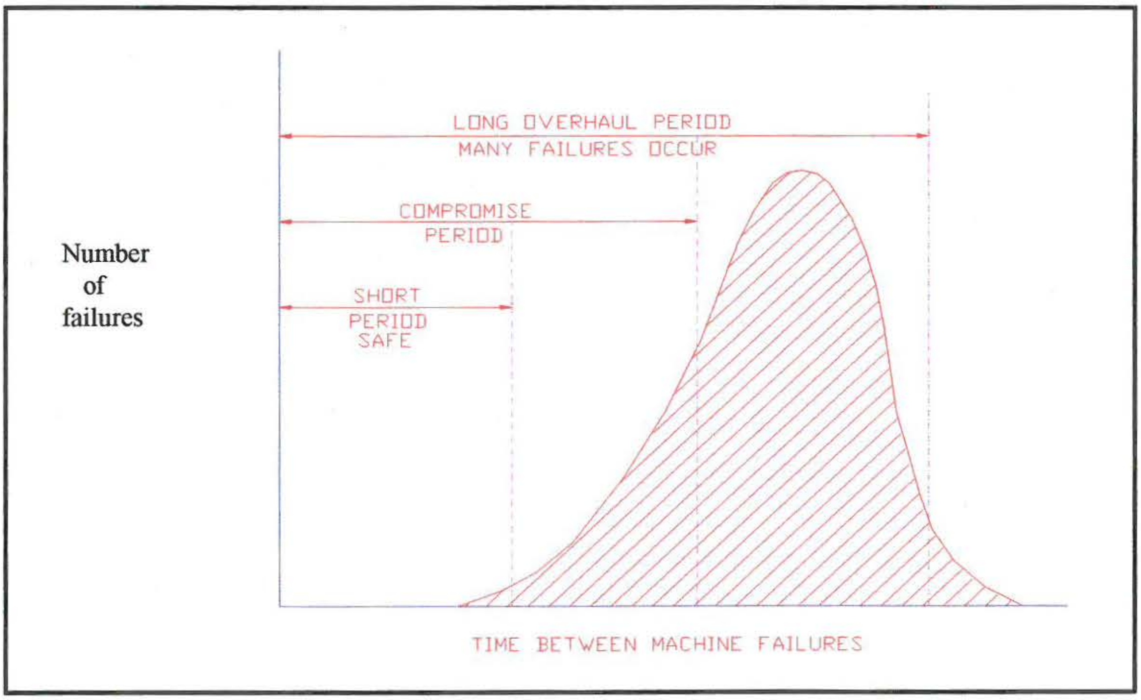


Figure 1-2: Time between machine failures

1.2.4 Predictive Maintenance (Monitor condition and service when needed)

Predictive maintenance assesses a machine's condition to decide when repairs should be performed and was developed to get the maximum life out of a machine without allowing the machine to fail [5]. The common premise of this strategy is that regular monitoring (Figure 1-3) of the actual mechanical condition and operating efficiency will provide the data required to ensure the maximum interval between repairs[2].

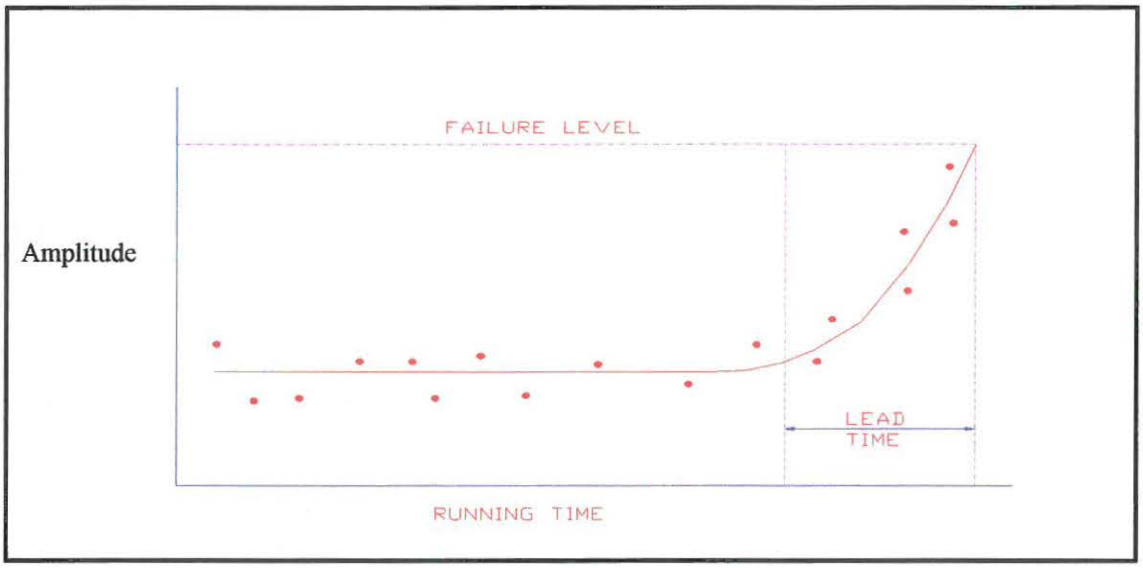


Figure 1-3: Trending a suitable parameter for measuring deterioration

Predictive maintenance can use a wide variety of measurement types to monitor a machine's condition, and they must be practical and cost-effective to obtain on a regular basis [1]. Time is also a significant factor. The first few sets of measurements may help with the diagnosis of pre-existing faults, but an effective predictive maintenance programme can only be based on consistent monitoring of trends in those measurements. Results must be repeatable to avoid confusion between variations in machine and measurement conditions.

For predictive maintenance, the condition is monitored, such as vibration and other parameters on a machine over a period of time. Changes in these parameters indicate specific faults, and one can predict the time at which they will become unacceptable. It is necessary in a Predictive maintenance programme to find a suitable technique or combination of techniques to detect the fault developing in time to take preventive action [6]. This must be done in such a manner that the total cost of repair and monitoring is less than when leaving the machine to run-to-failure.

The long-term objectives of a predictive maintenance program are to [2]:

a) Avoid unexpected catastrophic breakdowns with expensive or dangerous consequences to follow.

b) Reduce the amount of overhauls on machines to a minimum and thereby reduce maintenance costs.

c) Eliminate unnecessary interventions and risks of introducing faults on smoothly operating machines.

d) Allow spare parts to be ordered in time and thus eliminate costly stocks.

e) Reduce the intervention time and thereby minimize production loss as the fault to be repaired in machines is known in advance, and the overhauls can be scheduled when most convenient.

1.3 Selection of a Maintenance Strategy

Before a selection can be made, the following situation existing at the Department of Water Affairs must be considered:

a) There are large quantities of similar machines and equipment situated far apart.

b) Virtually all failures are due to time-related degradation and deterioration.

c) Insufficient qualified and experienced personnel

d) Centralized high-level technical support is available.

e) The Department supplies an absolutely essential service.

The above maintenance strategies are applicable to time-related deterioration. The selection of the most appropriate strategy will depend on the type of machine and importance to production. Because equipment is far apart, it would be impractical to attempt to include every machine in the programme. Only machines which are critical to operation, and those with a record of high maintenance costs, should be included.

Breakdown maintenance is preferred on smaller equipment as repair costs are normally much lower than monitoring costs. Preventative maintenance can give an indication of machine efficiency as well as some indication of impending trouble, but the shortcoming is that many serious component defects become quite advanced before they produce a measurable effect on machine performance.

For detecting incipient machine failure, the condition of a machine must be evaluated, using various predictive maintenance techniques [1]. Therefore all machines that are critical for water supply must be included in a predictive maintenance strategy. The end result will be that a combination of strategies are applied, but for critical equipment a condition-based maintenance strategy should be followed.

1.4 Predictive Maintenance Techniques

There are a variety of condition-monitoring techniques (Appendix A), but the most widespread approach is to monitor a broad range of machinery faults through the measurement of mechanical vibrations [2] [7]. For example, vibration is simple to measure and is very informative on both overall machinery condition and specific defects.

Other common monitoring and diagnostic techniques are: thermography, tribology, process parameters, visual inspections and ultrasonic monitoring. The exact combination will depend on the specific requirements [1].

1.4.1 Vibration Monitoring

This technique uses the vibration created by mechanical equipment to determine their actual condition. Monitoring the vibration can provide direct correlation between the mechanical condition and recorded vibration data. Any degradation of the mechanical condition can be detected. Used properly, vibration analysis can identify specific degrading machine components before serious damage occurs.

Using vibration as a predictive maintenance tool utilizing vibration signature analysis is based on two basic facts [3]:

a) All common failure modes have distinct vibration frequency components that can be isolated and identified.

b) The amplitude of each distinct vibration component will remain constant unless there is a change in the operating dynamics of the machine-train. Machinery distress very often manifests itself in vibration or a change in vibration pattern.

1.4.2 Shock Pulse Monitoring

The purpose of Shock Pulse monitoring is to be able to detect rolling-element bearing deterioration at an early stage [2][8]. When initial bearing damage occurs, high frequency shock waves are set up each time an element hits a defect. The Shock Pulse Meter is tuned to the resonant frequency of the accelerometer, so that when this resonant frequency is excited by the high frequency shock waves, the meter records the maximum value of the shock experienced [9].

To be successful with this technique the transducer should be located close to the load region of the bearing, and in a position favourable to the transmission of shock waves. This technique provides a relatively cheap and simple monitoring tool for rolling element bearings. The disadvantage of this technique is that it is fairly insensitive to changes in operating conditions. It shows significant increases in meter reading for defects such as a lack of lubricant and loss of bearing clearances [10]. High meter readings, indicative of a bearing fault, have also been observed for defects such as pump cavitation, or a bearing which is loose in its housing. These factors should be considered before taking any corrective action.

1.4.3 Thermography

It uses instrumentation designed to monitor the emission of infra-red energy (heat) to determine the operating condition. By detecting thermal anomalies (areas that are hotter or colder than what they should be) it is possible to locate and define incipient problems on machines, structures and systems [1]. The three general types of infra-red instruments are Infra-red Thermometers, Line Scanners and Infra-red Imaging.

Infra-red thermometers are relatively inexpensive. Interpretation of infra-red data requires extensive training and experience.

1.4.4 Tribology

Tribology is the general term that refers to the design and operating dynamics of the bearing-lubrication-rotor support structure of machinery. Several tribology techniques can be used for predictive maintenance: lubricating oil analysis and wear particle analysis [11]. There are three major limitations when using tribology analysis in a predictive maintenance programme: equipment costs, acquisition of accurate oil samples and interpretation of data.

1.4.4.1 Lubricating Oil Analysis

Lubricating Oil Analysis is an analysis technique that determines the condition of lubricating oils in mechanical and electrical equipment. It is not a tool to directly determine the operating condition of machinery.

1.4.4.2 Wear Particle Analysis

It is related to oil analysis only in that the particles to be studied are collected by drawing a sample of lubricating oil. Whereas lubricating oil analysis determines the actual condition of the oil sample, wear particle analysis provides direct information about the wearing condition of the machine-train [11]. Particles in the lubricant of a machine can provide significant information about the condition of the machine.

1.4.5 Process Parameters

Vibration-based predictive maintenance will provide the mechanical condition of a pump whereas infra-red imaging will provide the condition of the electric motor and bearings [1]. Neither technique would provide any indication of the operating efficiency of the pump. Therefore, the pump could be operating at less than 50% efficiency, and the existing predictive maintenance programme would not detect the problem. Process parameters should be included: suction and discharge pressures, current, etc. Using these parameters, the operating efficiency of the pump can be determined.

1.4.6 Visual Inspection

Regular visual inspection of the machinery and systems is a necessary part of the predictive maintenance programme. Routine visual inspection will augment the other techniques and help insure that potential problems are detected before serious damage can occur [2]. It can be incorporated in the programme of recording visual observation as part of the routine data-acquisition process. The incremental costs of these visual observations are small.

1.4.7 Ultrasonic Monitoring

This technique uses principles similar to those of vibration analysis. Unlike vibration monitoring, ultrasonic monitoring monitors the higher frequencies generated by the machine [9]. The normal monitoring range for vibration analysis is from less than 1 Hz to 20 kHz. Ultrasonic techniques monitor the frequency range between 20 kHz and 100 kHz. The principle application of ultrasonic monitoring is in leak detection. This technique is ideal for detecting leaks in valves, steam traps, piping and other process systems.

However, care should be exercised when applying this technique in the programme for bearing condition. Even though the natural frequencies of rolling element bearings will fall within the bandwidth of ultrasonic instruments, this is not a valid technique for determining the condition of rolling element bearings. Cavitation and other machine dynamics will also create energy or noise that cannot be separated from the bearing frequencies.

1.5 Setting up a Condition -Based Maintenance Programme

Implementing a condition-based maintenance programme is expensive. In addition to the initial capital costs of instrumentation and systems, there are substantial annual labour costs required to maintain the programme. To be successful, the programme must be able to quantify the benefits generated by the programme. This can be achieved if the programme is properly established, and uses proper condition-monitoring techniques. Therefore a pilot study must be done to establish a Maintenance Plan for clearly defined goals, objectives and benefits of such a programme.

1.5.1 Selecting Critical Plant

It is unlikely that Condition-Based Maintenance will be applied to all plants. The cost of monitoring would become prohibitive. Monitoring should only be applied to failure modes in machinery where cost benefits can be obtained.

1.5.1.1 Identify critical units

The availability of critical machinery is assessed by examination of history records. They are then ranked in order of criticality and availability.

1.5.1.2 Identifying critical items and Modes of Failure

An analysis of breakdowns must be studied on the different types of equipment, and the use of history records will be very helpful in this regard. This is done in order to determine whether failure is mostly time-related or occurs at random. If failures mostly occur at random, this strategy will not be effective.

1.5.1.3 Match the Maintenance Strategy to the Mode of Failure

For each observed mode of failure, a strategy of maintenance must be selected. Here the failure characteristics and mean-time-to-failure are important.

1.5.2 Condition-Based Maintenance

The flow chart in Figure 1-4 shows the steps in a Condition-Based Maintenance programme.

1.5.2.1 Selecting a technique to detect each mode of failure

Neale [2] presents a summary of techniques that have proved to be effective in detecting the more common modes of failure.

A part of the technique selection is the specification of measurement location, method and frequency. The frequency of measurements is dependent on the lead-time-to-failure. This is the time between the first detection of deterioration and the machine becoming unserviceable. If many measurements are made in the mean-time-to-failure, the monitoring frequency will be sufficient to pick up most of the deterioration in time to prevent the failure (Figure 1-3).

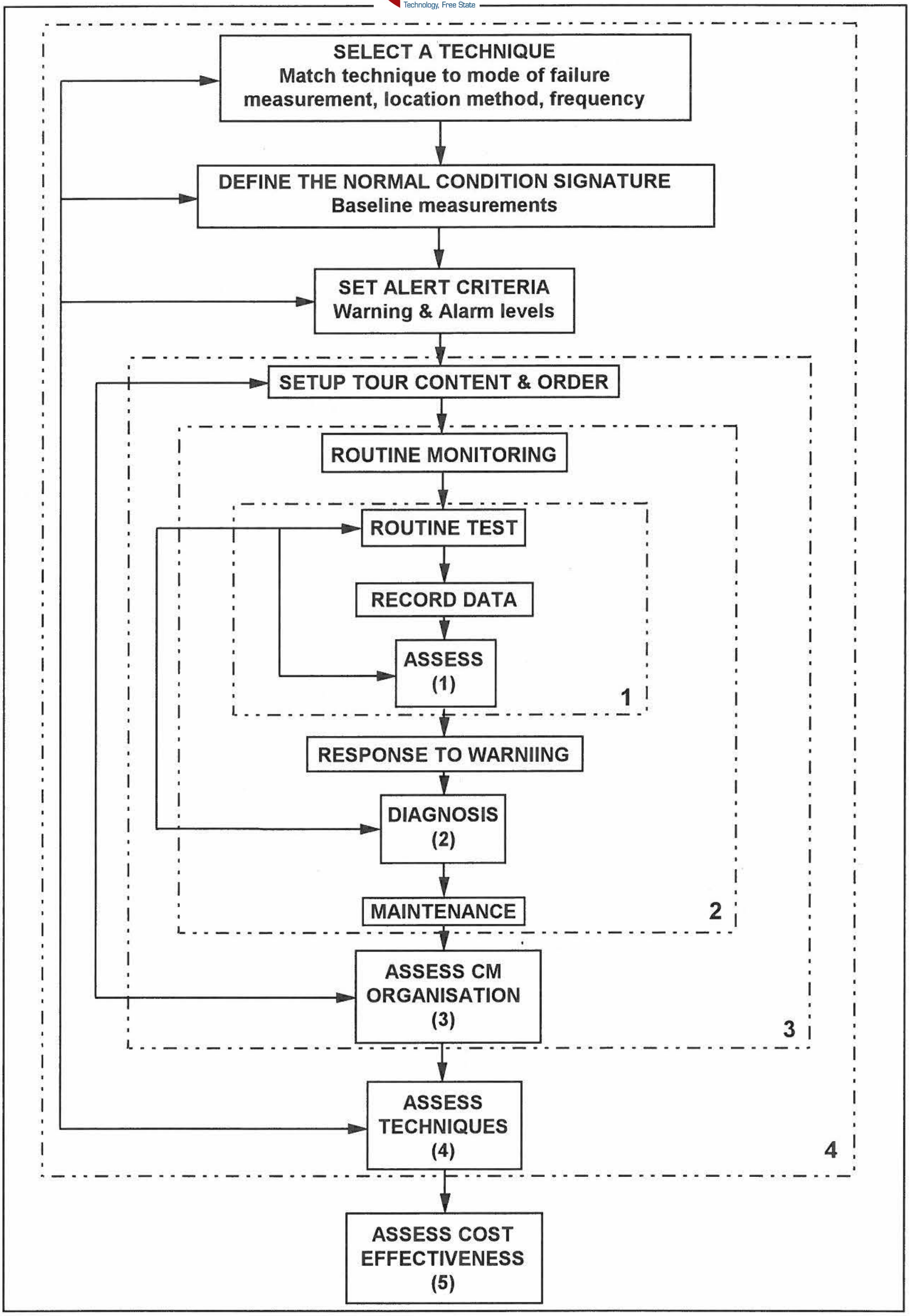


Figure 1-4: Flow chart of Condition-Based Maintenance

1.5.2.2 Define the normal condition signature

The first step in defining the normal condition of a machine is to take its baseline signatures at the defined measurement points. The baseline signature is made on the assumption that the machine is in an adequate condition. It is normally suggested that three or more baseline signatures are taken during the first weeks to ensure that a realistic baseline is established.

1.5.2.3 Setting the alert criteria

There are two important alert criteria: the Warning and the Alarm.

The warning level is the lowest level at which you can be sure that the parameter is changing from its normal range. A system that highlights any change will give maximum warning of incipient damage.

The alarm level is that level of parameter measurement that is considered to be unacceptable. The alarm level is normally set according to engineering criteria or practical experience.

1.5.2.4 Setting the tour content and order

The organization of the data collection is a purely administrative activity.

Collecting measurements using hand-held meters and paper lists is good enough for machinery with long lead-times-to-failure, but incurs low initial costs. This procedure is labour-intensive with the time spent on taking the measurements and an equal length of time spent in the office, transferring the data to the trend sheets and checking for warning.

Data collector systems, using hand-held devices, which work according to instructions, download a condition-monitoring database and then upload the measurements to the condition database, are cost-effective for the bulk of applications. They eliminate the data transfer and assessment time.

1.5.2.5 The routine Condition Monitoring

Routine monitoring is the major repetitive activity in the Condition-Based Maintenance strategy. It is normally the detection stage. It is important to select techniques which reduce the content of this activity to a minimum. Overall and normal spectra give an overview of the condition of the machine.

1.5.2.6 Action in response to a warning

When the result of a routine test of one or more parameters on a machine exceeds the warning, it is important to react quickly to diagnose the problem. An effective procedure is:

a) Examine the trend plots for all the parameters on the machine so as to highlight correlation which will help by determining the seriousness and deterioration of the fault. Using fault descriptions for the parameters contributes to the procedure.

b) Examine the recent history of maintenance actions on the machine.

c) Issue a Work Request with description of task. This stage must conform to the general work control procedures.

d) This stage is basically the verification of corrections done. Enter a history record for the machine. If work has been requested, the work feedback procedures should ensure the creation of a history record. If no work was requested, a special entry would record the reason.

1.5.2.7 Assess the Condition-Based Maintenance Strategy

The adoption of the Condition-Based Maintenance strategy incurs effort and cost. Therefore, it is essential to assess the effectiveness of its application. This is done at three levels:

a) Assess the tour organization and routine monitoring execution to improve the administration.

b) Assess the effectiveness of the techniques.

c) Assess the cost-effectiveness of the strategy.

1.6 Trending and Analysis Techniques

In the foregoing sections the procedure for constructing a Maintenance Plan has been described with the emphasis on Condition-Based Maintenance. Machines have a finite number of failure modes. If these failure modes and dynamics of the specific machines can be understood, the condition-monitoring techniques can be isolated in the specific failure mode. The term "trending" is used for the procedure of comparing a reading with previous readings and extrapolating changes to predict future situations. Therefore certain trending and analysis techniques must be incorporated into the programme to enable the obtaining of maximum benefits.

1.6.1 Trending Techniques

The method required to monitor the operating condition of equipment is trending of their relative condition over a period of time. Monitoring the trends prevents possible premature machine failure [12]. These trends are similar to the "bathtub" curve used to schedule preventative maintenance (Figure 1-1) [1]. The difference between the preventative and predictive "bathtub" curve is that the latter is based on the actual condition of the equipment, and not on a statistical average. The disadvantage of relying on trending only to maintain a predictive maintenance programme is that the reason why a machine is degrading, is unknown. Examples of trending include: vibration data, process parameters, bearing cap-temperatures, lubricating oil analysis and thermal imaging.

Trending of vibration signatures is another form of comparative analysis [12]. This method visually compares the relative change of a machine's vibration over a period of time. It identifies the specific frequency components that are creating the change in the vibration amplitude. Trending alone does not identify the specific problem or failure mode of a machine, but provides a simple way of determining if a problem exists.

1.6.2 Failure-Mode Analysis

Whereas the other techniques determine if a problem exists within the machine, failure-mode analysis tends to identify the specific problem in the machine. The premise is that there are established failure modes for a machine. Failure mode assumes that the vibration pattern for each failure mode is identifiable [3]. There are a number of failure-mode charts available to assist in analysis, and they provide an approximation of the problems in a machine. Failure-mode analysis can reduce the number of probable causes of a machine problem to a workable number. Using this technique, probable causes of machine failure can be identified, but must be verified by other techniques.

1.6.3 Root-Cause Failure Analysis

The real power of predictive maintenance lies not in the detection of problems at an early stage (although it is of great value), but the ability to pinpoint the cause of the failure, thereby preventing it from happening again [3]. It is important to match the physical symptoms with the vibration and other condition-monitoring techniques. It is an accurate method of isolating the specific machine component that is degrading and the reason for the degradation. In order to solve the problem, the reason for premature failure must be found and corrected.

CHAPTER 2

PREDICTIVE MAINTENANCE THEORY

2.1 Vibration Theory

2.1.1 Simple Harmonic Motion

This is the simplest form of vibratory motion. A mass suspended upon a spring and disturbed from its at-rest position will oscillate about this position, resulting in simple harmonic motion [10][13]. That means that the motion will repeat itself after a certain interval of time. This harmonic motion is illustrated in Figure 2-1. Until a force is applied to the weight to cause it to move, there is no vibration. By applying an upward force, the weight would move upward, compressing the spring. If the weight is released, it would drop below its neutral position to some bottom limit of travel where the spring would stop the weight. The weight would then travel upward through the neutral position to the top limit of motion and then back again through the neutral position. This motion will continue in exactly the same manner as long as the force is reapplied. When the instantaneous displacement of the mass is plotted against time, the motion takes on a sinusoidal form as shown.

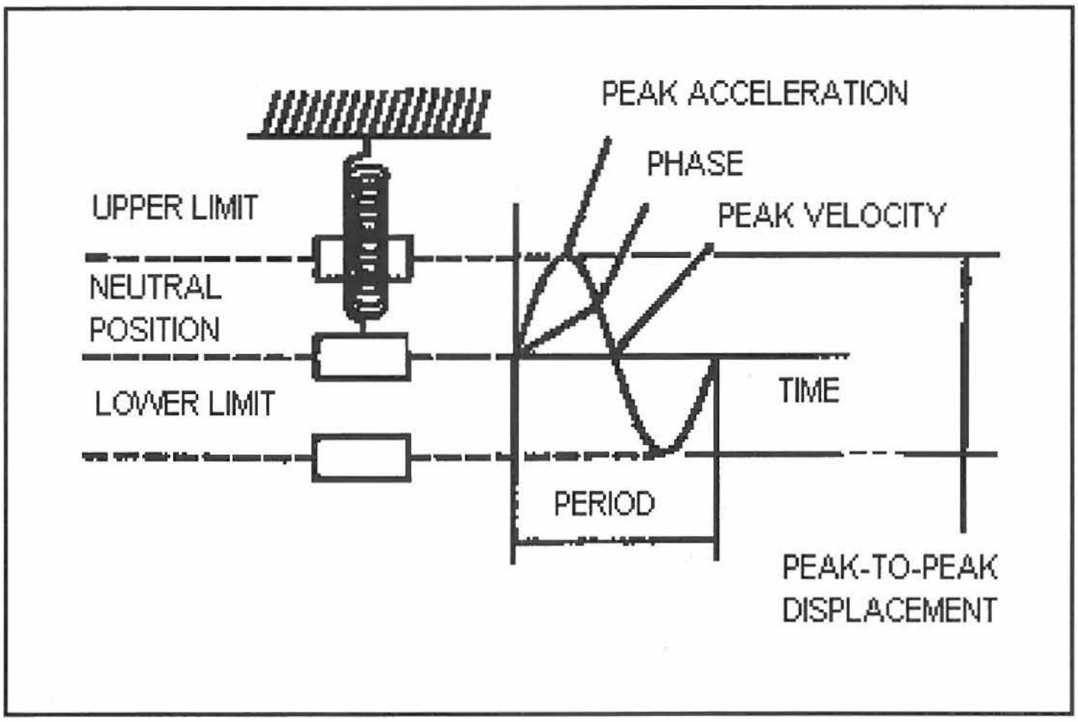


Figure 2-1: Vibration of a simple spring-mass system

2.1.2 Definitions of terms

2.1.2.1 Period and Frequency

Frequency is the number of times an event or condition repeats itself in a specific period. The units of frequency is Hertz [14][15].
Therefore: Hertz (Hz) = CPM/60 CPM: Cycles Per Minute

The period (t_p) is the time required to complete one cycle of the vibration. The period in seconds is the inverse of the frequency in Hz. For example, if a shaft is rotating at 1500 RPM (RPM and CPM are the same for rotating parts), which is equivalent to 25 Hertz (1500 RPM/60), the period of the vibration would be:

$t_p = 1/f = 1/25$ cycles per second = 0.04 seconds

This means that the vibration occurs every 0.04 seconds.

2.1.2.2 Amplitude

This measure of the displacement of a vibrating object is expressed as either root mean square (RMS), peak (P), or peak-to-peak (P-P). Figure 2-2 explains the relationship between the different amplitude measurements and is an example of a sine wave generated by the vibration of the spring system in Figure 2-1. The peak-to-peak displacement is the vertical distance measured from P to -P. The peak value is the vertical distance measured from O to P.

2.1.2.3 Fundamental frequency (f)

The fundamental frequency is the primary rotating speed of the machine or shaft being monitored and is usually referred to as the running speed of the piece of equipment. The fundamental frequency is referred to as 1 X RPM. For, example, the fundamental frequency of a 1500 RPM pump is 1500 CPM or 25 Hertz.

Vibration parameters can be converted from one input to another by using the following formulas for rotating parts:

$v = \pi \times f \times D$	$a = 0.0162 \times v \times f$
$v = 61.44 \times a/f$	$D = 0.3183 \times v/f$
$a = 0.0511 \times D \times f^2$	$D = 19.57 \times a \times f^2$

<u>where:</u>

a = Acceleration (mm/sec²)

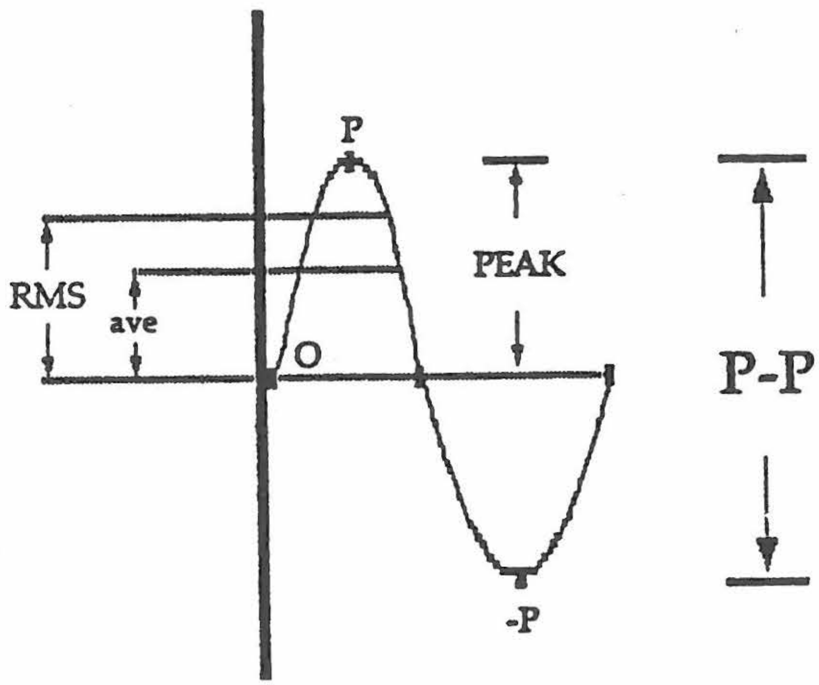
v = Velocity in mm/sec

D = Displacement in mm

π = 3.1415

g = Gravitational constant 980 mm/sec²

f = Frequency in Hertz



Peak = O to P

RMS = 0.707 peak

Average = 0.637 peak

Figure 2-2: Relationship between the different amplitude measurements

2.1.2.4 Vibration parameters

a) Displacement

Displacement is the actual physical movement of the vibrating surface and, according to standards, it is measured in meter. Displacement is defined as the change in position relative to its initial position which is the total distance from the upper limit to the lower limit (Figure 2-2) [15].

b) Velocity

Velocity is the speed at which displacement occurs and is measured in m/s. Mathematically, velocity is expressed by the equation:

$$v = dD/dt$$

where: v is velocity,
dD is the change in displacement, and
dt is the change in time.

c) Acceleration

Acceleration is the rate of change of velocity and is measured in m/s^2. Acceleration is expressed by the equation:

$$a = dV/dt$$

where: a is Acceleration
dV is the change in velocity,
dt is the change in time.

2.1.2.5 Harmonics

These are the vibration signals having frequencies that are exact multiples of the fundamental frequency (i.e., 1x f, 2x f, 3x f, etc.). Harmonics are also expressed as multiples of the running speed of a shaft, e.g., 1x, 2x and 3x RPM.

2.1.2.6 Phase

Phase is that part of a cycle (0 to 360°) through which one part travels relatively to another part of a machine or a fixed reference point [16]. Phase measurements are expressed in degrees, and one complete cycle of vibration is equal to 360°. To determine phase, a shaft is marked, or a prominent feature, such as a keyway, is observed. Either a strobe light or a key phaser is used to measure the phase. When the strobe flash rate matches the frequency of the vibration, the shaft or part will appear frozen. Phase is determined by observing the position of the mark in the frozen position. Using a strobe light, the phase is measured clockwise from 0 to 360° with top dead centre being 0°. The movement of any part on the machine can now be related to the reference mark. The strobe will provide a readout showing the speed of the shaft.

A key phaser will provide a DC voltage pulse output with one pulse for each shaft rotation. These units have a light sensitive device that pulses each time the shaft mark or keyway passes. These units should be mounted by using a magnetic base to give the most reliable data. The output pulse of these units can be monitored by several different types of instruments to provide phase and speed information. Figure 2-3 explains phase relationships. In these figures, the phase relationship is shown graphically, by using circles to indicate a shaft with a keyway or shaft mark [16]. In the circular shaft motion, top dead centre is 0°. In all figures point A is the reference point, while the four diagrams show the four main phase relationships through one cycle. Diagram 1 shows the cycle just beginning and an in-phase relationship. Diagram 2 shows 90° out of phase, diagram 3 shows 180° out of phase, and diagram four shows 270° out of phase.

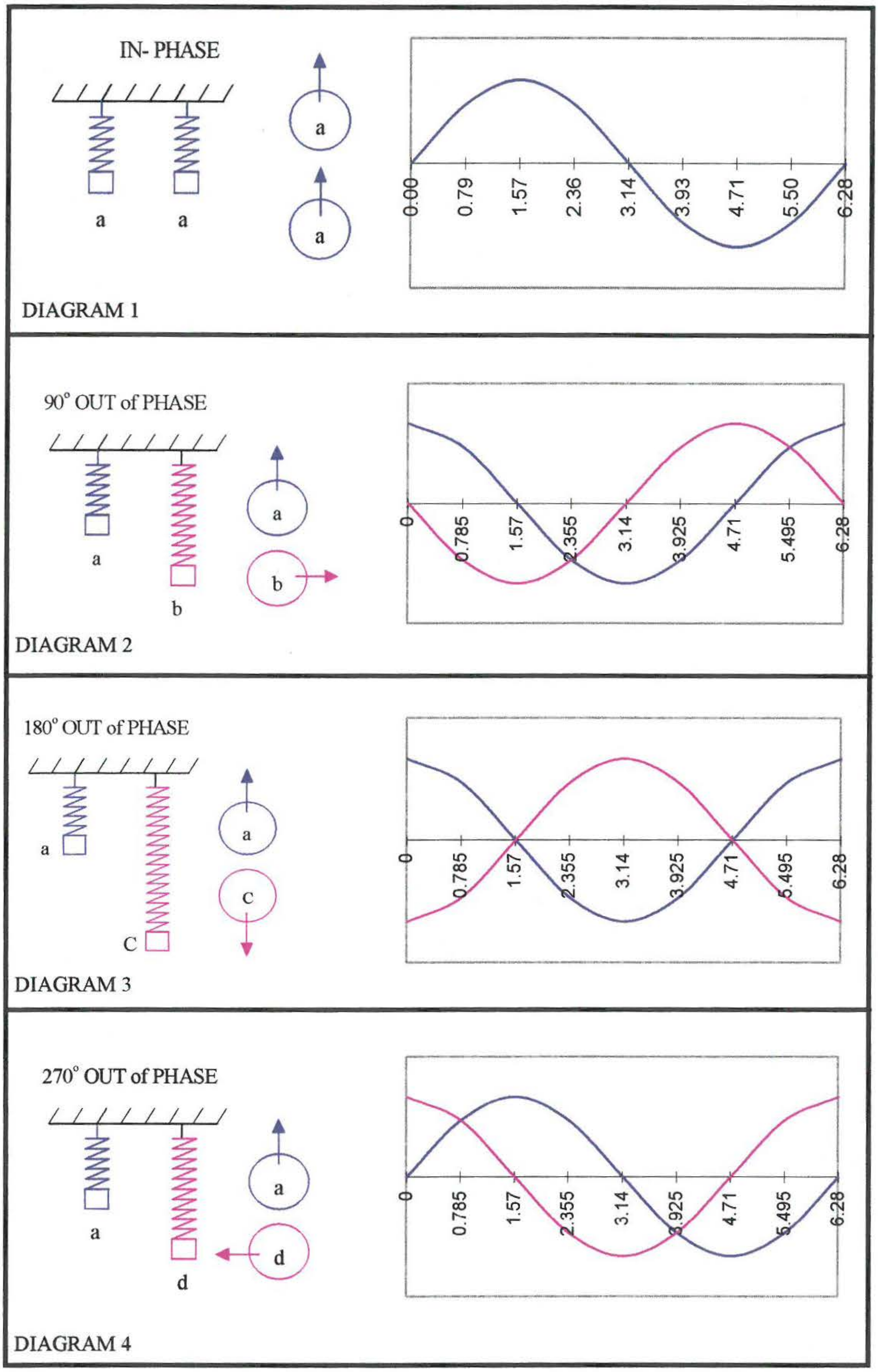


Figure 2-3: Phase relationship

2.2 Data Acquisition and Interpretation

2.2.1 General

More than 85% of mechanical problems occurring in rotating machinery can be detected and analyzed by displaying the vibration amplitude versus frequency data (Figure 2-4) [16]. Machines exhibit certain vibration characteristics which are caused by various components in the machine and it is therefore possible to identify the influence of these components by analyzing the vibration spectrum (plot of amplitude vs. frequency).

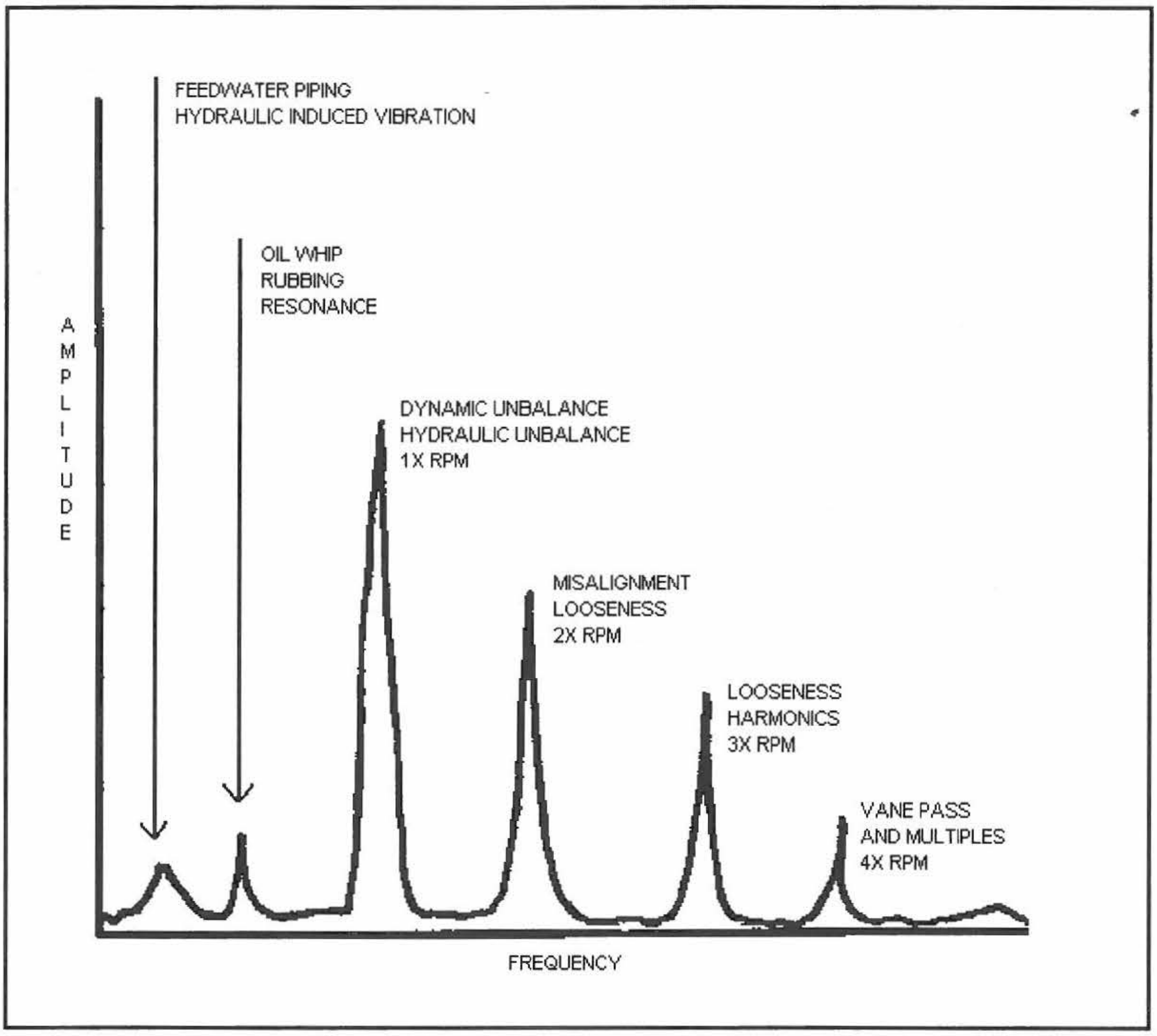


Figure 2-4 Vibration Spectrum

2.2.2 Unbalance

Unbalance results when the axis of rotation and the centre of gravity of a body are displaced from one another. The unbalance force is proportional to the speed of the rotation squared [14].

Formulating this gives:

$$F = u\,R\,\omega^2 = gm$$

where: u = unbalance mass (g)
R = radius at which unbalance weight is (m)
ω = rotational speed

Unbalance can be caused by static or couple unbalance. In operation these two normally combine and result in dynamic unbalance [16]. The 1x RPM frequency will always be present and normally dominates the spectrum (Figure 2-5). For static unbalance the vibration generated at each bearing end would have the same phase relationship. Couple unbalance will tend towards 180° out-of-phase in the same direction measured at the two bearing caps. Instead of pure static or couple unbalance, dynamic unbalance will occur, resulting in phases approximately 90° in relation to each other [17].

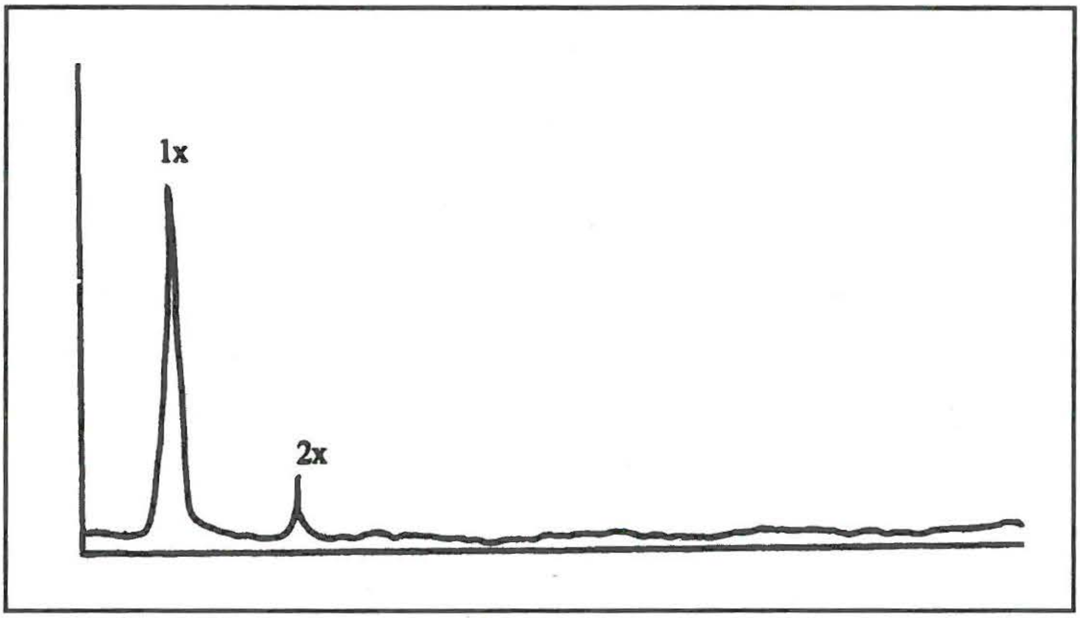


Figure 2-5: Vibration spectrum showing unbalance

Unbalance has the following amplitude and phase characteristics [14][16]:

a) It will always have a dominant frequency equal to running speed (1x RPM).
b) There will be a 90° phase shift between the vertical and horizontal at the same bearing.
c) The amplitude readings end-to-end in horizontal will have the same ratio as the vertical.
d) The amplitudes in the horizontal direction will have the same ratio as the vertical.
e) The axial amplitude will be smaller than one third of the radial amplitudes.
f) The axial phase readings will match the radial readings and occur at a dominant frequency equal to 1 x RPM.

2.2.3 Misalignment

2.2.3.1 General

The significant characteristics of misalignment is that vibration will be noted in both radial and axial directions [15][17]. The axial vibration is the best indicator of misalignment and whenever the amplitude of axial vibration is greater than one-half (50%) of the highest radial vibration (horizontal or vertical), then misalignment or a bent shaft should be suspected. A bent shaft acts very much like angular misalignment, so its vibration characteristics are included here. The dominant vibration is normally at running speed (1x) if the bend is near the shaft centre, but will be at twice running speed (2x) if the bend is near the coupling. Figure 2-6 shows the three types of misalignment [13]:

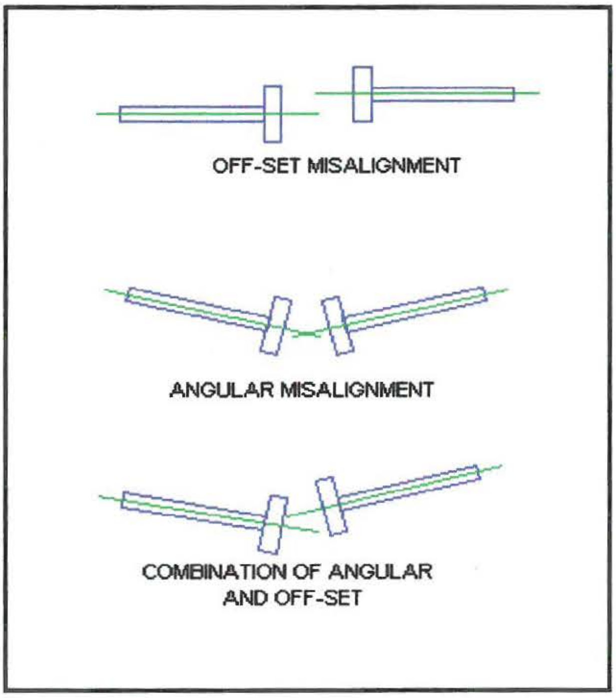


Figure 2-6: Three types of misalignment

Misalignment has the following amplitude and phase characteristics [16]:

a) The phase will be 0° or 180° measured at the two bearing caps during 75% of the time.
b) The vibration will be very directional.
d) High axial vibration.
e) The vibration will be 0° or 180° vertical to horizontal at the same bearing.

When either angular or off-set misalignment becomes severe, either high amplitude peaks at much higher harmonics (4x-8x) or even a whole series of high frequency harmonics similar in appearance to mechanical looseness can be observed. Coupling construction will often greatly influence the shape of the spectrum when misalignment is severe. Normally a combination of angular and off-set misalignment will occur [15]. Figure 2-7 shows a typical vibration spectrum of misalignment.

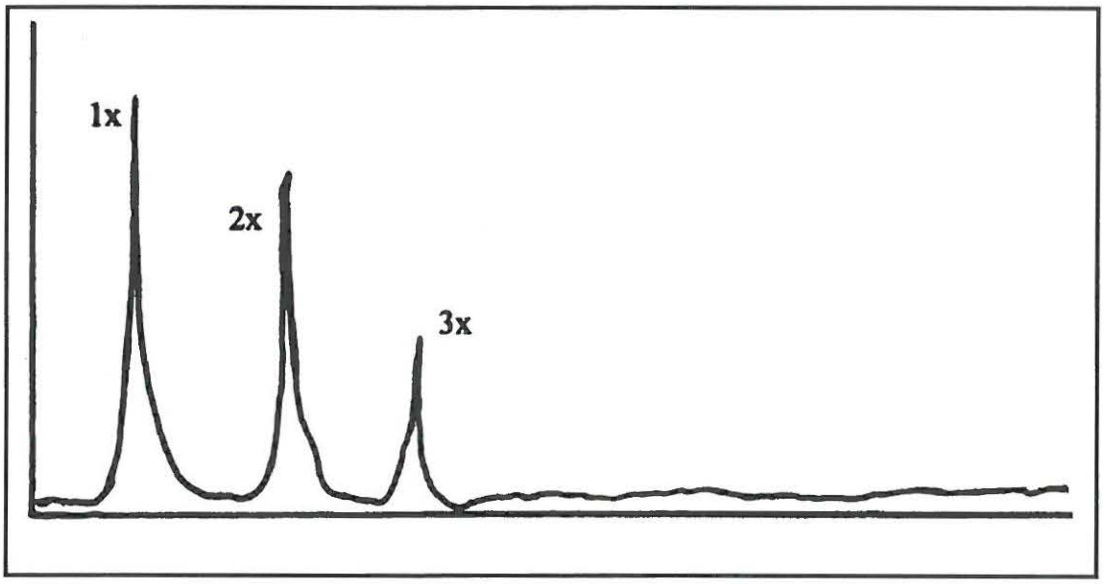


Figure 2-7: Typical vibration spectrum of misalignment

2.2.3.2 Angular Misalignment

Angular misalignment occurs when the centre lines of the two shafts meet at an angle (Figure 2-6). It is characterized by high axial vibration and the phase is 180° out-of-phase across the coupling. This form of misalignment can duplicate high axial vibration with both 1x, 2x and 3x RPM. These symptoms may also indicate coupling problems [14].

2.2.3.3 Off-Set Misalignment

Off-set misalignment is present when two shafts are parallel to each other, but have axes which do not coincide (Figure 2-6) [18]. This type of misalignment generates a radial vibration which approaches 180° out-of-phase across coupling. The vibration amplitude at 2x is often larger than 1x, but its height relative to 1x is often dictated by the coupling type and construction.

2.2.4 Misaligned Bearing On a Shaft

A misaligned bearing (Figure 2-8) will generate considerable axial vibration [13]. It will cause a twisting motion with an approximate 180° phase shift from top to bottom and/or side-to-side as measured in the axial direction of the same bearing housing. Attempts to align the coupling or balance the rotor will not alleviate the problem. The bearing must be removed and installed correctly.

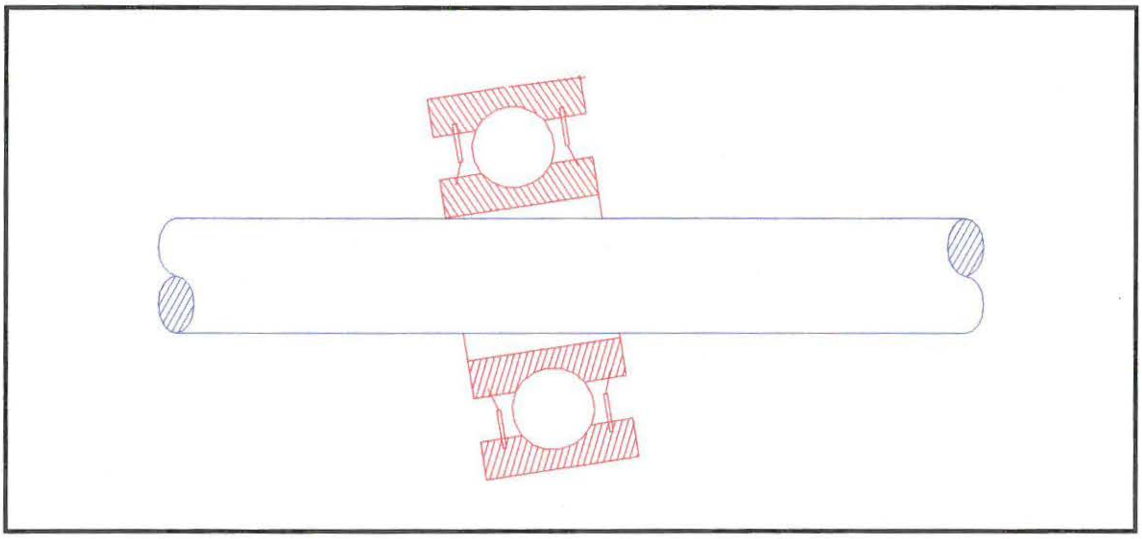


Figure 2-8: Misaligned bearing on a shaft

2.2.5 Resonance

Resonance occurs when a forcing frequency coincides with a system's natural frequency (Figure 2-9) and can cause dramatic amplitude amplification. This can result in premature or even catastrophic failure [16].

The natural frequency of a system is [10]:

$$\omega_n = \sqrt{k/m}$$

where:

k = stiffness factor
m = mass

Thus: An increase in stiffness will increase the natural frequency and an increase in mass will lower the natural frequency. The operating zone of equipment should be outside $\pm\sqrt{2N}$ if N is the operating speed corresponding to the natural frequency.

Resonance magnifies the amplitude of vibrations in relatively undampened systems anywhere from 5 to 10 over and above that of non-resonant vibrations [15][17]. Machine vibration that is magnified by resonance is almost always due to a non-rotating part such as a segment of a steel base, pedestal, support beam, brace or span of pipe .

Resonance decreases the bearing and seal life, as well as increases the possibility of creating fatigue cracks in foundations. It is estimated that at least 20% of all running machines have some magnification due to partial or full resonance of such parts as pipe lengths, pedestals, portions of bases, valves, beams and bearing housings. Whenever cracking is reported, resonance should be suspected.

To detect resonance problems, one of the following can be done[16]:

a) Resonance Bump Test

This test is based on the principle that when a "spring system" or part is bumped or deflected in some way, it will vibrate for several cycles at its natural frequency. Its amplitude decreases with each cycle, but its frequency remains the same. The low frequency of the repeated bumps is not measured, only the frequency that is generated between bumps. Resonance data can be taken with the vibration instrument using the "peak hold" function. The spectrums obtained from the bump test look similar to the regular spectra from vibrating running machines, except that the various peaks will not necessarily be related to running speed. Instead, the peaks will relate to the resonant frequencies of various machine parts that were replaced by the bumping action. The largest peak usually represents the resonant frequency. Once the bump test indicates that some part is resonant to a source vibration, determining the exact part is possible while the machine is running.

When the machine is running and the suspect part is resonating, the point-to-point method of obtaining and plotting amplitudes to obtain the part's mode shape should help to identify which parts are resonant and which are not. If a piece of pipe or structure is suspected to be in resonance, the part can be divided roughly into 10 equal measuring points. Using this spacing for a suspect part will result in a similarly shaped resonance curve for a resonant part, whether it is relatively small or large. The amplitude is plotted on the vertical scale and the positions where the measurements were taken are plotted on the horizontal scale. Resonant parts will exhibit much more pronounced curling than non-resonant parts.

b) Phase Relative to Resonance

If the vibration amplitude changes somewhat, but the phase remains approximately the same (within a few degrees), this indicates that the part is not resonant [16]. Appreciable phase changes (anywhere from 15°, 90° and up to about 150°) indicate that the part is partially to fully resonant (Figure 2-9).

To solve resonance problems, one of the following can be done:

a) Remove the exciting force

b) Change the frequency of the exciting force.

c) Change the natural frequency of the machine part or support. In order to change the natural frequency, the stiffness and/or mass of the part must be changed.

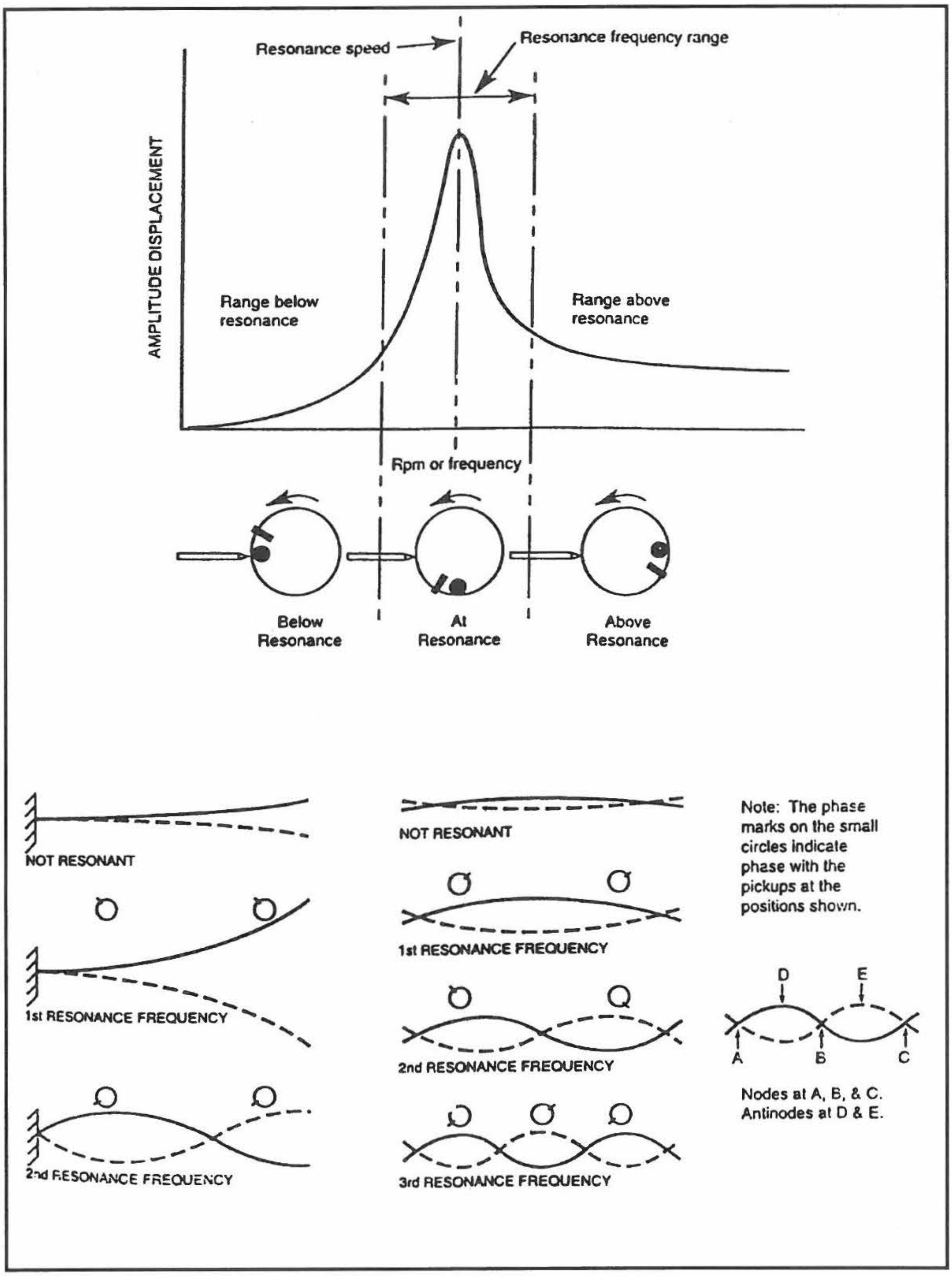


Figure 2-9: Vibration Phase Relative to Resonance Frequency

2.2.5 Mechanical Looseness

2.2.5.1 General

Looseness is caused by structural weakness of machine feet, baseplates or foundation; also by deteriorating grouting, loose hold-down bolts (soft foot) at the base or distortion of the frame or base. It results in a vibration at a frequency of twice the rotating speed (2x rpm), but may also result in higher-order frequencies such as 3,4,5 or 6x rpm [15][17].

The nature of mechanical looseness and the reason for the characteristic of vibration at 2x rpm or at higher frequencies can be explained by referring to the sequence in Figure 2-10 [13]. Ilustrated is an unbalanced rotor mounted in a bearing with loose mounting bolts. In Figure 2-10A, the heavy spot of unbalance has rotated 180°, where the unbalance force is directed downward. This tends to force the bearing down against the pedestal. In Figure 2-10B, the heavy spot has rotated 360°, and the resultant unbalance force is now in the upward direction. This upward force tends to lift the bearing off the pedestal. Figure 2-10C, the heavy spot has rotated 90°, and in this position the upward lifting force of unbalance is zero. Therefore the bearing will simply drop against the pedestal. This action produces two applied forces for each revolution of the shaft. One force is applied by the rotating unbalance, and a second force is applied when the bearing drops against the pedestal. Therefore, the vibration frequency resulting from this looseness will be 2x rpm.

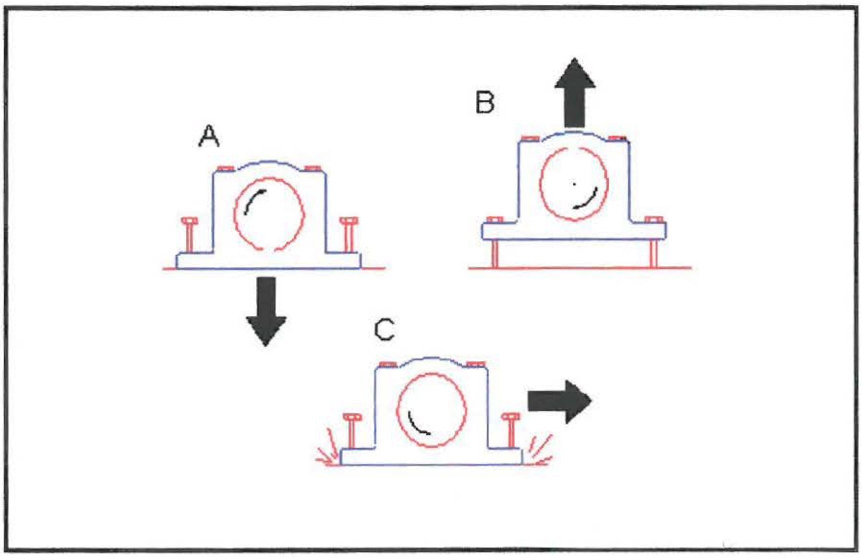


Figure 2-10: Mechanical looseness resulting in harmonically related vibration frequencies

2.2.5.2 Machine base or structural looseness

The amplitude of vibration resulting from mechanical looseness will often be unsteady, and observing the rotor with a stroboscopic light will generally reveal an erratic image. Comparing the amplitude and phase of vibration at various points on the machine or structure as illustrated in Figure 2-11 can often help detect the source of mechanical looseness. Any significant difference in amplitude or phase indicates relative motion and a source of looseness [14]. Phase analysis may reveal approximately 180° phase difference between vertical measurements on the machine foot, baseplate and base itself. It can also be caused by cracks in the frame structure or bearing pedestal.

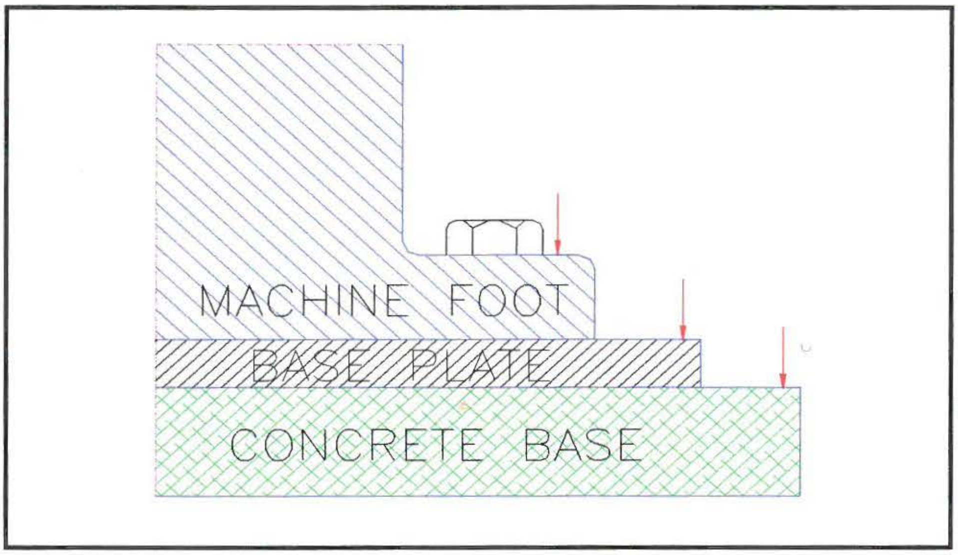


Figure 2-11: Comparative phase readings taken at the three measurement points illustrated may help in locating the source of looseness

2.2.5.3 Improper clearances or bearing looseness

Looseness due to improper fit between component parts will cause many harmonics due to non-linear response of loose parts to dynamic forces from the rotor [11]. This type of looseness is often caused by a loose bearing (Figure 2-12) liner in its cap, excessive clearance in either a sleeve or rolling element bearing or a loose impeller on a shaft. This type is often unstable and may vary widely from one measurement to the next, particularly if the rotor shifts position on the shaft from one startup to the next. For detecting bearing looseness, axial phase data can be taken in directions all the way around on the bearing housing (Figure 2-13). If axial phase readings are noticeably different at the four measurement points, this indicates that the bearing is "twisting" as illustrated.

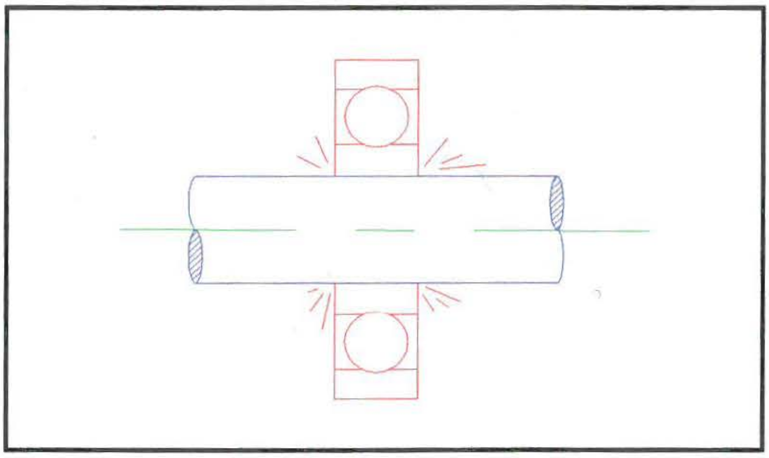


Figure 2-12: Loose bearing on shaft

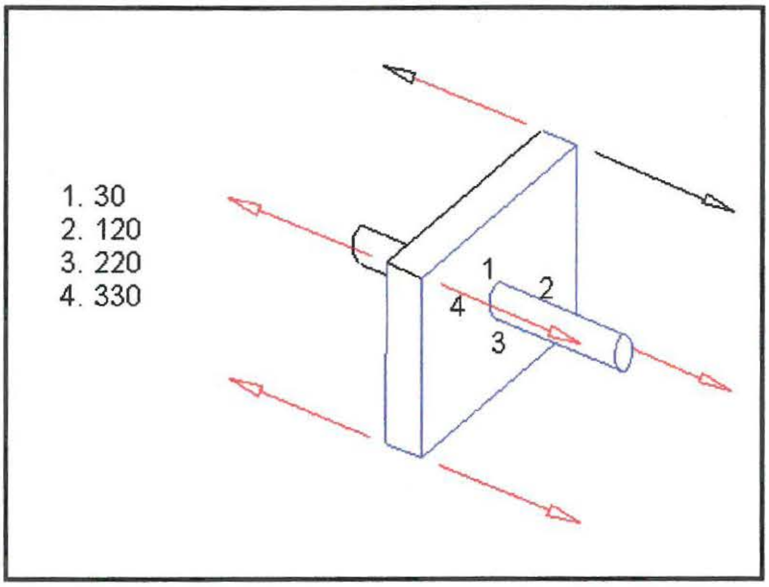


Figure 2-13: Axial phase readings for four positions on a bearing housing can indicate a loose bearing

2.2.5.4 Soft foot

Soft feet can cause increased vibration amplitude or vibration magnification. There are many combinations of conditions that produce a soft foot, most often the result of all the shimmed feet not being on the same plane. Figure 2-14 views some of the most common conditions for soft foot [14][16].

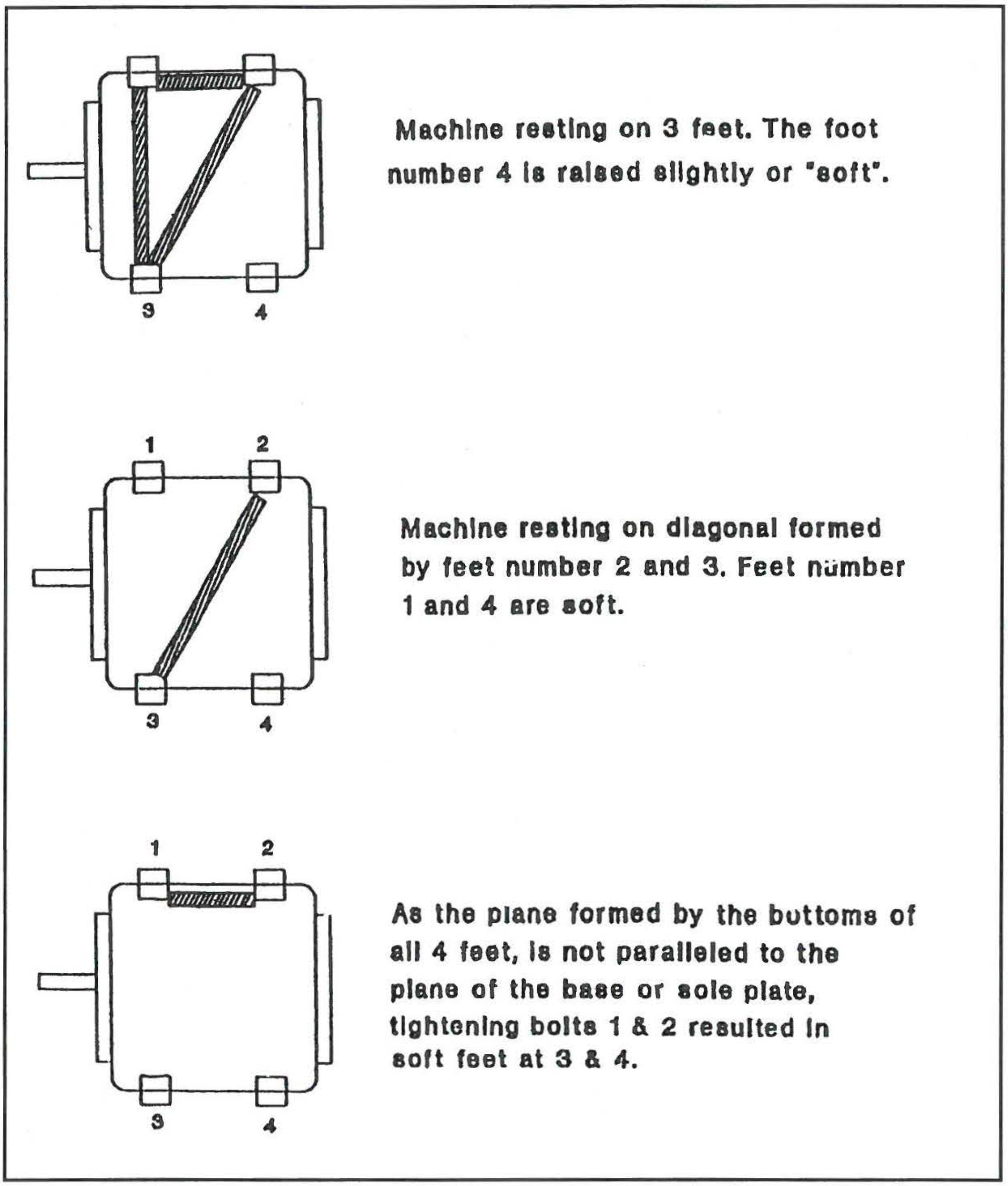


Figure 2-14: Combinations of soft feet

2.2.6 Sleeve Bearings

Sleeve, or fluid film, bearings can be divided into several subclasses: plain, grooved, partial arc and tilting-pad [19][20]. With the exception of the tilting-pad bearing, they do not generate a unique rotational frequency that would identify normal operation. The tilting-pad bearing has moving parts (pads); it will generate low-level vibration components at a pad-passing frequency that is equal to the number of pads multiplied by the shaft running speed.

The tilting-pad bearing forms a uniform thin film of lubricant between the bearing's babbit surface and the rotating shaft. If the shaft breaks through the lubricating film, a mechanical rub becomes evident in the vibration signature.

2.2.6.1 Wear/Clearance Problems

Later stages of sleeve bearing wear are normally evidenced by the presence of a whole series of running speed harmonics (Figure 2-15)[14]. Wiped sleeve bearings will often allow high vertical amplitudes compared to horizontal. Sleeve bearings with excessive clearance may allow a minor unbalance and/or misalignment to cause high vibration which would be much lower if bearing clearances were according to specifications[21][22].

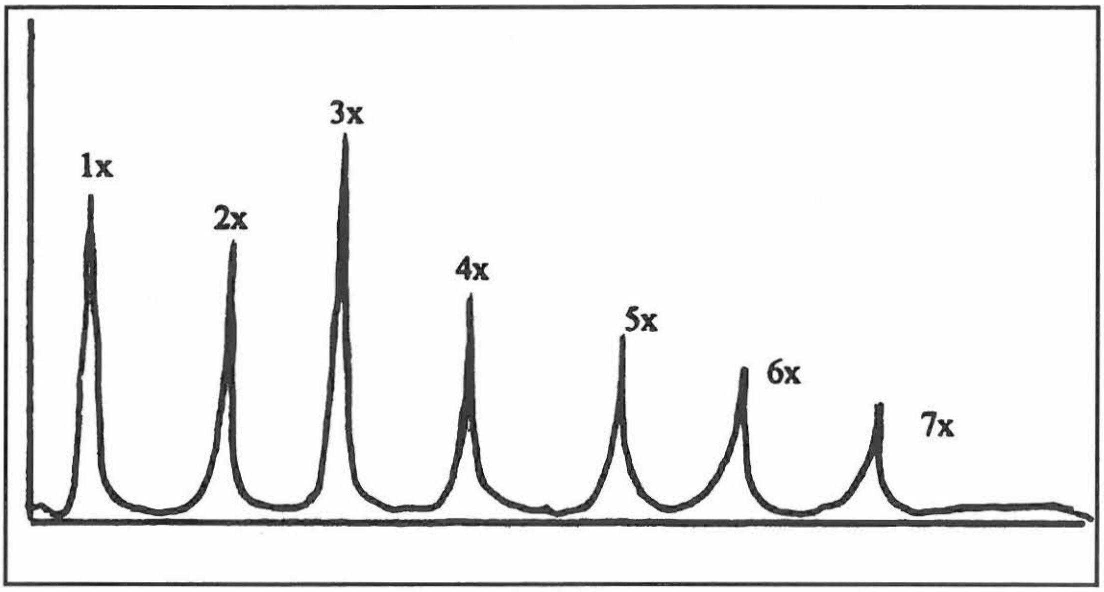


Figure 2-15: Sleeve bearing looseness

2.2.7 Defective Rolling Element Bearings

2.2.7.1 General

Defective rolling element bearings generate different types of spectra once they begin to develop defects. These include: high or ultrasonic frequencies, natural frequencies of bearing components and rotational defect frequencies [23].

2.2.7.2 High Frequency/Ultrasonic measurements

Vibration velocity readings help to identify bearing defects in the latter stages of bearing life. The earliest warning of impending trouble are the ultrasonic vibration levels in the 5 kHz - 40 kHz range[10][16][24]. This high frequency vibration can be measured by various techniques, such as spike energy, shock pulse, bearing defect energy or high-frequency detection (methods are discussed later). The terminology for the particular unit varies, but they all measure the same thing.

These high frequency measurements sense the low-energy, repetitive, metal-to-metal impacts that occur in the earliest stages of an incipient bearing failure [10]. The rate, amplitude and frequency are all combined to give a single numerical output.

The problem with these methods is that it is almost impossible to determine the bearing condition based on a single measurement. Measurements are very sensitive to external influences such as differences in installation, load, lubrication, pickup mounting, location and the number of interfaces. Other problems, such as cavitation, also have significant effects on the units.

a) Shock Pulse Monitoring

This is based on monitoring the mechanical impacts created in a damaged bearing. When initial bearing faults occur in the form of small pits in one of the races, high frequency shock waves are set up each time an element hits a pit [9]. The shock pulse meter is tuned to the resonant frequency of the accelerometer, so that when this resonant frequency is excited by the high frequency shock waves, the meter records the maximum value of the shock experienced. The shock pulse value is a function of the meter reading, the speed and geometry of the bearing and is related directly to the condition of the bearing.

b) ESP (Enveloping Signal Processing)

The contact stresses at the interface between the rolling elements and the races are very high and the abrupt changes in these stresses, when a rolling element passes over a defect, generates an impulsive force [25]. The impulsive force may excite resonances in the bearing races and the machine structure, causing them to ring at their natural frequencies in the kilohertz range up to several megahertz. The ESP (Enveloping Signal Processing) was developed to monitor the impulses generated by bearing defects. These impulses repeat periodically at a rate determined by the type of defect, the bearing geometry, and the running speed of the shaft. These repetition rates are called Bearing Defect Frequencies as discussed in section 2.2.7.2. The assumed method of signal generation is that a carrier frequency containing all the resonant frequencies of the machine is modulated by a signal containing the defect frequency and its harmonics. The resulting signal measured by the accelerometer contains side frequencies of the defect frequency and its harmonics as well as the running speed and its harmonics, of each resonant frequency.

The bearing defect frequencies tend to be "buried" in all these other resonances; it can be difficult to extract bearing information from a standard vibration velocity or acceleration spectra.

Enveloping Signal Processing is essentially a two-stage demodulation technique which extracts the high frequency range and obtains a signal containing only the impulses [25].

The first stage of the process is to pass the signal through a band-pass filter to reject the low frequency machine vibration and the very high frequency random noise, and leave the bursts of narrow band bearing signal. The band of the filter should be centred on the resonant or carrier frequency. After filtering, the signal consists of the chosen resonant frequency with side frequencies corresponding to the defect frequency and its harmonics, and the running speed and its harmonics.

The second stage of the process is the demodulation or Enveloping Detection process. The filtered signal is rectified which produces a signal consisting of the defect frequency and its harmonics, the running speed and its harmonics, and side frequencies corresponding to the resonant frequency. The signal is then passed through a low-pass filter which smoothes the signal and removes the last traces of the resonant frequency. Finally, this enveloped signal is frequency analyzed to give the enveloped spectrum, which contains less frequencies than the original signal. This spectrum shows peaks at the defect impact rate, and can pinpoint the fault to a specific part of the bearing.

As a consequence of the signal processing it is not possible to confidently define precise acceptance criteria in terms of ESP units [8]. ESP is normally interpreted in terms of decibels (dB), as the use of dB units ensures a very wide amplitude range and can enhance the significance of spectral peaks. ESP data is analyzed with reference to three major parameters:

i) The level of the highest defect frequency peak in ESP units.
ii) The dB carpet level.
iii) The ratio of defect frequency level to carpet or peak/carpet differential.

2.2.7.3 Bearing defect frequencies

Bearing defect frequencies differ from other vibration sources as they are defect frequencies and they are non-integer multiples of operating speed [26]. They should not be present at all in the normal vibration spectrum. When they are present, they signal at least an incipient problem. Other common frequencies such as 1 x RPM are always present, whether or not there is satisfactory balance or alignment and pumps will always show vibration at some amplitude for vane pass frequency.

Various formulas have been developed to detect specific defects in rolling element bearings, such as faults on the inner race, outer race, cage or rolling elements themselves [27][28]. The formulas depend on the bearing geometry, the number of rolling elements and the bearing rotational speed.

Formulas for bearing defect frequencies:

$$BPIR = Nb/2\ (1 + Bd/Pd\ \mathrm{Cos}\ \varnothing)$$
$$BPOR = Nb/2\ (1 - Bd/Pd\ \mathrm{Cos}\ \varnothing)$$
$$BSF = Pd/2Bd\ [1 - (Bd/Pd)^2\ (\mathrm{Cos}\ \varnothing)^2]$$
$$FTF = 1/2(1 - Bd/Pd\ \mathrm{Cos}\ \varnothing)$$

where:

BPIR = Ball Pass Inner Race
BPOR = Ball Pass Outer Race
BSF = Ball Spin Frequency
FTF = Fundamental Train Frequency (Cage)

Nb = Number of Balls or Rollers
Bd = Ball or Roller Diameter (mm)
Pd = Bearing Pitch Diameter (mm)
$\varnothing$ = Contact Angle (degrees)

2.2.7.4 Use of velocity spectra to determine bearing condition

Figure 2-16 presents the typical velocity spectra of four primary stages through which most rolling element bearings pass [14][23]. These spectra follow the bearing from the very onset of bearing problems in stage 1 through imminent failure of the bearing in stage 4.

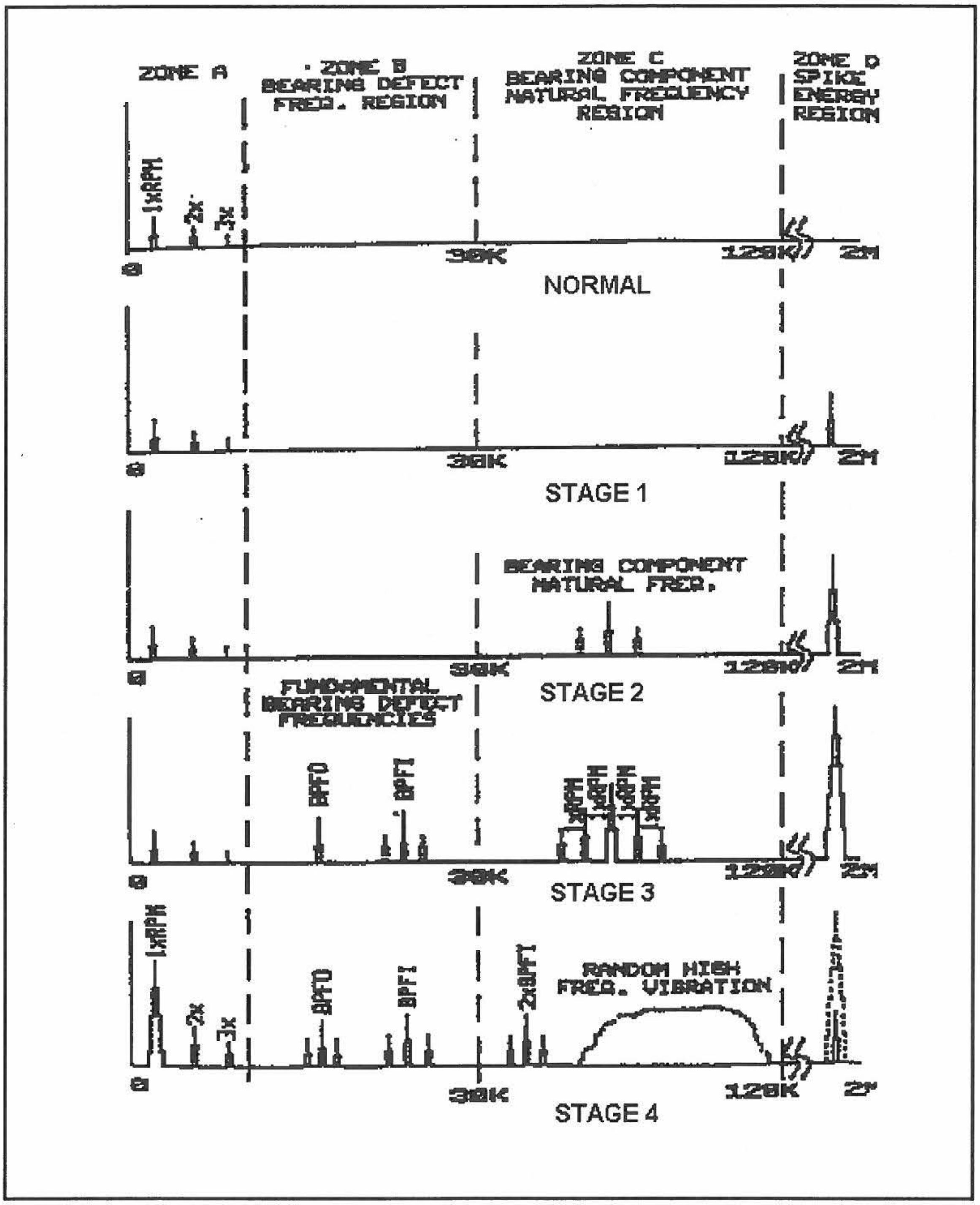


Figure 2-16: Primary failure stages through which most rolling element bearings pass

a) Stage 1

Stage 1 represents the velocity spectrum for the first stage of a bearing defect on it velocity spectra. There is no change in the vibration velocity spectrum and it only has the normal first 3 running speed harmonics.

An increase in spike energy (or Shock Pulse) units has occurred. During this stage, no sound will be detected by the human ear, indicating bearing damage, and no discernible change in temperature would be anticipated.

b) Stage 2

Stage 2 represents the velocity spectrum for the second stage of failure. Slight bearing defects begin to excite natural frequencies of the bearing component-natural frequencies which predominantly occur in the 30 000 - 120 000 CPM range. These frequencies are independent of the operating speed and are non-synchronous. Towards the end of stage 2, these frequencies will not only grow, but also become modulated with the running speed (1 x RPM side bands will later appear above and below these natural frequencies) as wear progresses. Modulation of these bearing component's natural frequencies most often occur at 1 x RPM, and side bands can also be spaced at bearing defect frequencies (BPFO or BPIR) of the bearing natural frequencies. The appearance of these side bands increases until a haystack has developed.

Spike energy units continue to increase. The defects themselves still may not yet be readily visible to the naked eye. Slight increases (roughly normal) in bearing noise and temperature will be observed.

c) Stage 3

Stage 3 represents the velocity spectrum for the third stage. For the first time, bearing defect frequencies appear in the velocity spectrum and can be associated with faults. Bearing defect frequencies and harmonics appear. As wear progresses, more defect frequency harmonics appear and the number of side bands grow, both around these and around bearing natural frequencies. The more harmonics of a bearing defect frequency appear, the greater the deterioration will be. At the end of stage 3, not only will 1 x RPM side bands appear around bearing defect frequencies, but more side band families will appear around the bearing component's natural frequencies.

Spike energy has reached a maximum value. Wear is usually visible and may extend throughout the periphery of the bearing, particularly when well-formed side bands accompany bearing defect frequency harmonics. Bearings should be replaced at this stage.

A word of caution is offered at this point [4]."When the bearings approach the conclusion of stage 3, the rate of wear becomes highly unpredictable. How much longer the bearing will last, will largely depend on its lubrication, temperature, cleanliness and dynamic loads being imposed upon it by vibration forces from unbalance, misalignment, etc."

d) Stage 4

Stage 4 represents the velocity spectrum for the fourth stage of failure. Towards the end, the amplitude of 1 x RPM normally begins to grow for the first time. It grows and normally causes growth of many running speed harmonics. Many 1 x RPM side bands will appear around bearing defect frequencies (indicating pronounced wear throughout the periphery of the bearing). Discrete bearing defects and component natural frequencies actually begin to "disappear" and are replaced by a random, broad band high frequency "noise floor". Due to the spread over a very large frequency range, and much broad band noise, this stage is often confused with symptoms of cavitation (see section of pumps).

Spike energy units have dropped. A significant increase in bearing housing temperature is noticeable.

2.2.7.5 Guidelines for analyzing bearings

a) Bearing components normally fail in the following order: race defects, ball or roller defects, cage defects (unless the bearing was defective when installed).

b) Inner race defects and failures occur at much lower amplitudes than outer race defects.

c) BSF is usually generated when a ball or roller is defective. When multiple balls are defective, multiples of the frequency appear. If four balls have defects, a peak will be seen at 4 x BSF.

d) The appearance of a ball spin frequency does not always necessarily mean that there is a defect in the rolling elements. It still means there is a problem present. In this case, it can indicate that a cage has broken at a rivet and that the balls are thrusting hard against the cage.

e) Internal looseness in a bearing has a component at 1 x RPM and several multiples of shaft RPM. A bearing turning on the shaft or in the housing shows 3 x RPM and possibly higher multiples of RPM.

f) Bearing misalignment can result in vibration at the number of balls or rollers times the shaft RPM.

g) Vibration amplitude increases as the bearing degrades, but may disappear prior to failure.

h) Baseline energy may increase across the entire spectrum and the band width broadens as the bearing degrades.

i) Allowable vibration at bearing defect frequency in previous research work has proven that up to now no absolute values/amplitudes could be given to allow vibration amplitudes at bearing defect frequencies. The author stated [3]: "The most important thing to look for indicating significant bearing wear is the presence of a number of bearing defect frequency harmonics, particularly if they are surrounded by side bands spaced at either 1 x RPM or side bands spaced at other defect frequencies of the bearing - independent of amplitude. If these are present in a spectrum, replace the bearing as soon as possible".

2.2.8 Electrical Fault Diagnosis

2.2.8.1 General

An important concept in understanding the induction motor vibration is the rotor magnetic frequency [17][23]. In a 2-pole motor, a spot on the rotor will line up with a stator pole twice during each revolution of slip. A rotor bar will be subjected to a maximum current flow twice per revolution of slip. The significance of this is the effect which a lack of rotor or stator symmetry will have on vibration. A magnetic problem in the rotor will manifest itself in vibration at twice rotor slip frequency.

In Southern Africa electricity is generated at 50 Hz. This creates a magnetic field that alternates (+ and -) at 2 x 50 Hz [16]. If some magnetic part is slightly loose (such as iron encapsulated by wires, or slightly loose armature laminations), the loose part will be pushed in one direction at the instant of the positive portion of the electrical cycle and then pushed in the negative direction at the instant of the negative portion of the cycle. The impacts at each end of the cycle cause a hum in the frequency of 2 x 50 Hz = 2 x 3000 cpm = 6000 cpm (also known as twice line frequency).

The most common electrical faults of induction motors causing problems will be discussed with reference to the following formulas [23]:

$2F_L = 2 \times 3000 = 6000$ CPM
$F_s = N_s - N_r$
$F_p = F_s \times P$
RBPF = Number of Rotor Bars x RPM

where:

F_L = Electrical Line Frequency
F_s = Slip Frequency
N_s = Synchronous Speed
N_r = Rotor Speed
F_p = Pole Pass Frequency
P = number of Poles
RBPF = Rotor Bar Pass Frequency

2.2.8.2 Uneven air gap (static eccentricity)

In a motor with the rotor perfectly centred in the stator, the magnetic forces on the rotor are equal, opposite and balanced [29]. Both magnetic poles are trying to pull the rotor into the stator. The only force that remains on the rotor is torque [14]. In Figure 2-17, a round rotor is off-set in the air gap [17][23]. In Figure 2-17a, the high flux zones of the rotating magnetic field of the stator are positioned as shown. In this position, the air gaps are even and radial forces on the rotor are balanced.

When the magnetic field of the stator has rotated 90 degrees (Figure 2-17b), the magnetic forces are unbalanced. The narrow air gap on the right hand side of the rotor tends to pull the rotor to the right.

When the magnetic field of the stator has rotated 90 degrees further (Figure 2-17c), the air gaps in the areas of high flux density are equal again. The magnetic forces are again balanced.

When the magnetic field of the stator has rotated 90 degrees further (Figure 2-17d), the magnetic forces are again unbalanced. The narrow air gap on the right side of the rotor tends to pull the rotor to the right.

The unbalance tug on the rotor occurs twice in one revolution of the rotating magnetic field. The frequency is at twice line frequency (100 Hz) for all motors, regardless of the number of poles [30]. When the power is cut, the vibration at twice line frequency disappears immediately.

It should be noted that since the stator is attracted to the stationary motor frame as well as to the rotor, twice line frequency vibration may also indicate a loose stator. The motor could also vibrate at 1x RPM and smooth out the instant the power is cut. As can be seen in Figure 2-18, this occurs twice for one revolution of slip. The vibration at 1x RPM is thus caused to modulate (rise and fall) at a rate equal to 2 times slip frequency.

For example, in a motor running at 2988 RPM, the modulation will occur 24 times per minute ($2Fs = 2(Ns - Nr) = 2(3000-2988) = 24$). Thus, an eccentric rotor will cause vibration to modulate at a rate equal to the number of poles times slip.

Eccentric rotors produce a rotating air gap between rotor and stator with induced pulsating vibration (normally between 2x Line frequency (F_L) and closest running speed harmonic) and at pole pass frequency side bands (F_p), as well as F_p side bands around running speed. F_p appears itself at low frequencies (Pole Pass Frequency = Slip Frequency x number of poles).

After an eccentric rotor has become suspected, soft foot checks must be made, because a motor with a soft foot may also exhibit the above symptoms [14]. This occurs when a motor is improperly shimmed and the tightening of the holding-down bolts distorts the stator, so that the air gap is no longer symmetric. This may be checked by loosening the motor hold-down bolts and observing the vibration level.

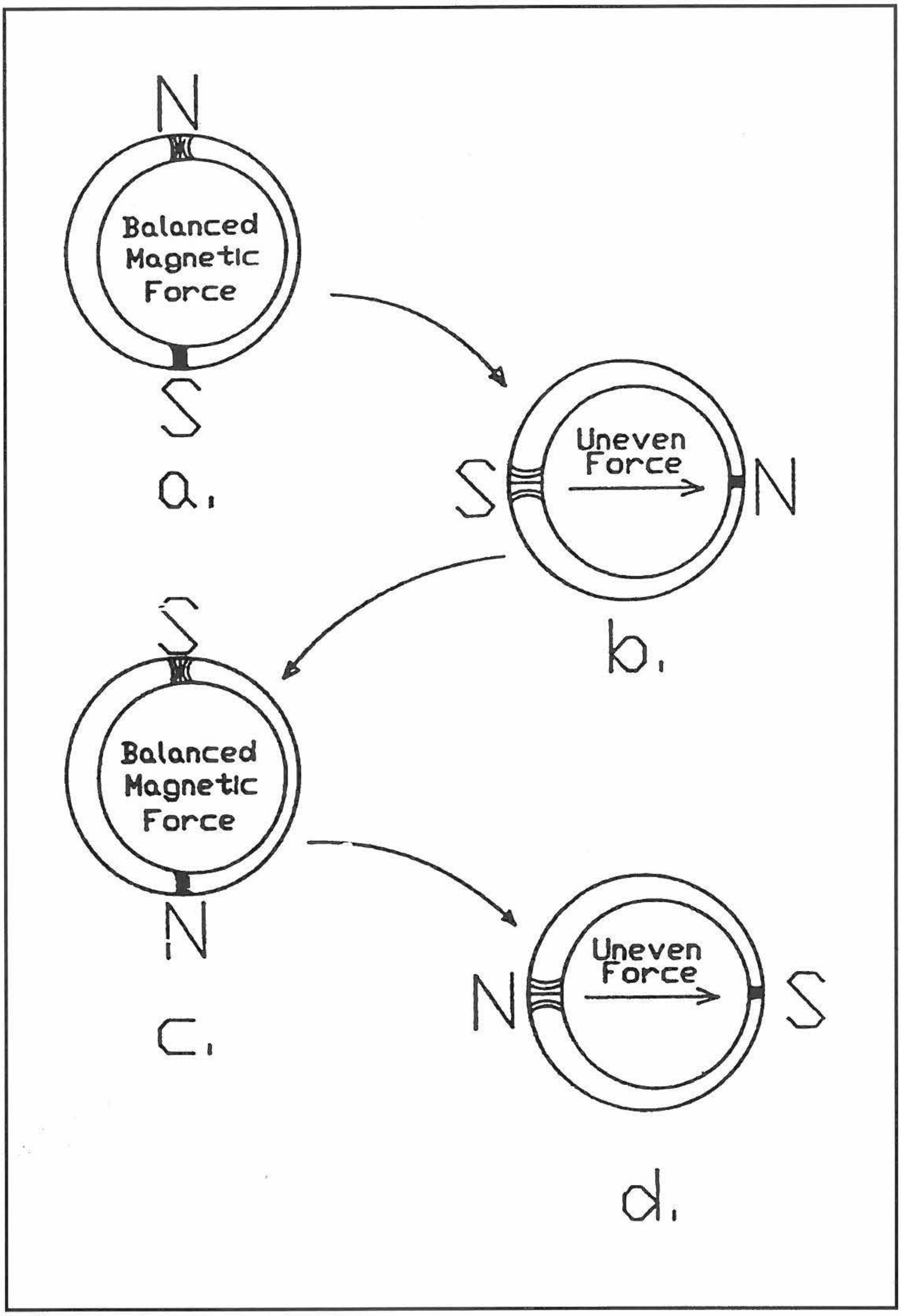


Figure 2-17: Uneven air gap rotor off-set in stator

2.2.8.3 Broken rotor bars

A motor with a broken rotor bar has symptoms similar to a motor with an eccentric rotor [29][30]. The vibration under no load will modulate at a rate equal to the number of poles times slip frequency. With a broken rotor bar, the amplitude of the vibration increases with load. When the motor is operated uncoupled, the broken rotor bar has virtually no effect and the motor will run smoothly.

The difference in comparison with an eccentric rotor is that a broken rotor bar cannot carry current. When it is in an area of high flux, the magnetic forces on the rotor are unbalanced. Since current flow through the rotor bar is proportional to slip, and at no load when the rotor current is low, the bar has virtually no effect. A cracked rotor bar may also cause localized heating of the rotor which causes uneven expansion and rotor bowing. This will result in unbalance and a strong 1x running speed vibration. The side bands which are related to slip frequency, will also be visible. A high resistance joint between a rotor bar and an end ring may exhibit the same symptoms.

Loose rotor bars may have similar symptoms and will also show vibration at rotor bar passing frequencies [31]. Broken or cracked rotor bars or shorting rings, or shorted rotor laminations will produce high 1x running speed vibration with pole pass frequency side bands (F_p). See vibration spectrum in Figure 2-18 for an illustration of broken rotor bars. In addition, cracked rotor bars will often generate F_p side bands around the third, fourth and fifth running speed harmonics. Loose rotor bars are indicated by 2x line frequency ($2F_L$) side bands surrounding the Rotor Bar Pass Frequency (RBPF).

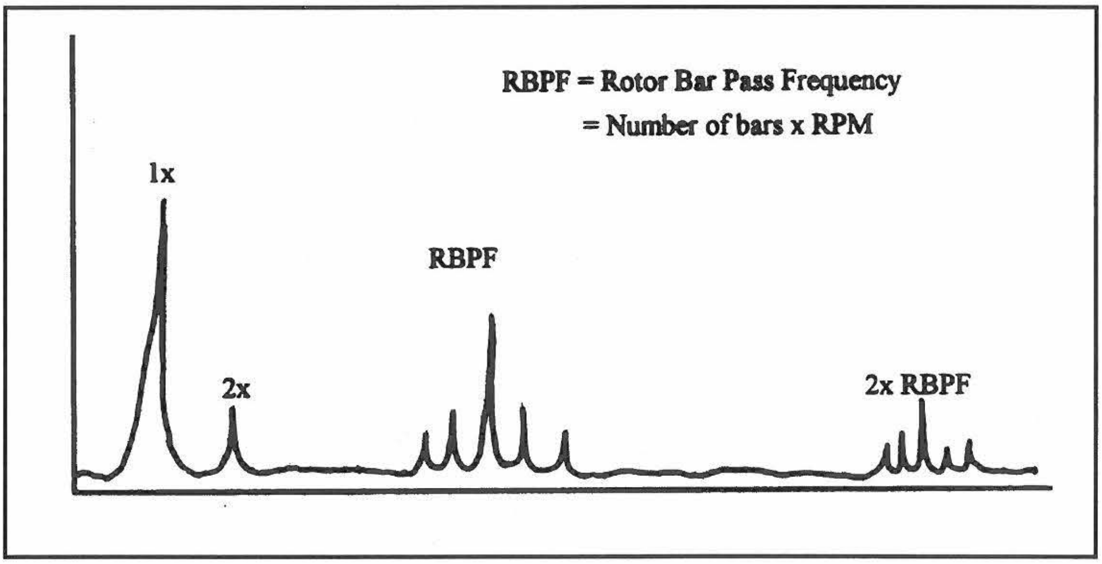


Figure 2-18: Vibration spectrum indicating broken rotor bars

2.2.8.4 Phasing Problems (Loose Connector)

Phasing problems due to loose or broken connectors can cause excessive vibration at 2x Line Frequency ($2F_L$) which will have side bands around it at one-third Line Frequency (1/3 F_L). Levels at $2F_L$ can exceed 25 mm/sec if left uncorrected. To identify phasing problems becomes a problem if the defective connector is only sporadically making contact [23].

2.2.8.5 Motor Current Analysis

A very useful diagnostic tool in the prediction of electrical faults is to measure the phase-current of an electric motor [30][32].

The basis of phase-current analysis is based on the theory that a rotor winding fault such as a broken rotor bar or high resistance joint produces harmonic fluxes in the motor's air gap which will cause a variation in current flow. A signal is obtained by connecting a current transformer to one of the motor phases [33]. A hand-held current transformer is clamped onto either the main lead to the motor, or onto a secondary (metering or protection) circuit. The signal from this temporary test circuit is connected to the spectrum analyzer.

Spectrum analysis of the line current has been found to be a useful indicator of motor condition, because an induction motor is essentially a large, sensitive transducer connected to the process it is driving. Every periodic or transient variation in torque, causes a back electromagnetic force (emf) in the stator winding that is easily measured. It monitors the motor's phase current to produce a pattern unique to motor-winding faults. This pattern allows an estimate of the severity of any fault. The frequency of the disturbance due to a rotor fault is equal to twice slip frequency. Slip frequency is the difference between the rotor speed and the speed at which the electric field rotates around the stator. For a 2-pole motor, typical slip speed is the synchronous speed minus running speed [33]. This is because the current through the region of the defect peaks when the break passes a pole in the stator field. The two times slip speed disturbance modulates line frequency, creating distinctive side bands.

The severity of rotor damage is determined by comparing the amplitudes of the side band components to the line frequency current amplitude [31]. As damage increases, the side band amplitudes become larger. Machine load and process variations may raise or lower the side band amplitudes. The amplitude of these side bands can also predict the number of broken rotor bars.

2.2.9 Pump and Hydraulic Vibration Problems

2.2.9.1 General

In every pump, dynamic forces of mechanical and hydraulic origin are present and a certain amount of vibration is therefore inevitable. To ensure the safety of a pump, the vibration must be kept within the acceptable limits [34]. If there is a high level of vibration, or if it increases markedly during the course of time, problems of mechanical or hydraulic nature are indicated [35]. There are many potential causes of vibration. Some of the causes were discussed in the previous sections. The pump design and construction characteristics must be evaluated and the history of the pump's operational reliability considered before positive conclusions can be drawn as to the cause of the vibration and safe operational levels of vibration. Detailed measurements and analysis of the vibration characteristics, the pump, and the pumping system are often needed for a reliable diagnosis. An important step in an analysis programme is to draw a distinction between vibrations that are system-related and vibrations from a mechanical or hydraulic problem in the pump [36].

2.2.9.2 System-related problems

Some typical system-related problems include the following [36]:

a) Unfavourable dynamic behaviour of the foundation, supporting structures or piping.

b) Excitations from the coupling area, especially due to misalignment of the driver or eccentrically bored coupling hubs.

c) Excitations from the vibrations of the driver (electrical motor).

d) Unfavourable incoming flow conditions such as cavitation, intake vortex, or suction recirculation.

e) Flow disturbances in the suction piping due to poor design and layout, especially in valves.

f) High pressure pulsations due to hydraulic instability of the entire pumping system.

g) Control system/pump interaction during startups or other periods of low flow.

2.2.9.3 Mechanical or hydraulic-related problems within the pump itself

Typical vibration problems of the pump include unfavourable dynamic behaviour of the rotor due to an excessive wearring, bushing, or seal clearances, poor support of the rotor because of loose fits on the shaft or housing in the case of ball bearings [37]. Another vibration problem is mechanical unbalance of the rotating parts which is due to poor balancing, careless assembly, or operational influences. Some increase in vibration is normal when departing from the best efficiency flow rate. However, increased radial forces occur when the pump is operated outside of the design flow range. The most common causes of vibration are [16][17]:

a) Vane Pass Frequencies

Every time a rotating blade passes near a discontinuity such as the tongue of a volute, one expects an impact [10]. If one has a six-bladed impeller, a blade frequency equal to six times the shaft speed of the impeller will be observed (Figure 2-20). Multiples or harmonics of the blade frequency could also exist.

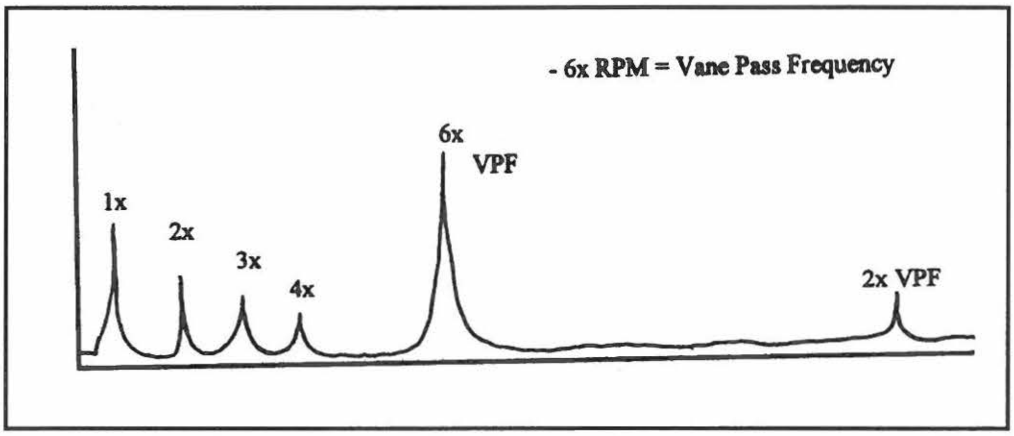


Figure 2-20 Vane pass frequency with harmonics

Large amplitude VPF and harmonics can be generated in a pump if the gap between rotating vanes and stationary diffusers is not kept equal all around or if the pump is stalled. VPF (or harmonics) sometimes can coincide with a system's natural frequency, causing high vibration. High VPF can be generated if an impeller wear ring ceases on the shaft or if weld fastening diffusers fail. High VPF can also be caused by abrupt bends in the pipe.

b) Cavitation

On a centrifugal pump, cavitation is caused mainly by a lack of Net Positive Suction Head (NPSH). It normally generates random, higher frequency broad-band energy which is sometimes superimposed with blade pass frequency harmonics. It normally indicates insufficient suction pressure (starvation).

2.2.10 Further Research

2.2.10.1 Turbomachinery

Most common problems detected on turbomachinery, including centrifugal compressors, high-speed centrifugal pumps, turbogenerators, steam and gas turbines are: unbalance, misalignment, oil whirl, rubbing, looseness, aerodynamic/hydraulic forces, internal friction, aerodynamic cross-coupling, surging and choking (stone-walling) [17].

2.2.10.2 Gears and Gearboxes

Gears and gearboxes have unique vibration signatures that identify both normal and abnormal operation. Gear mesh, gear excitation and backlash are dynamic forces generated in gearboxes that can be identified by vibration analysis [15].

2.2.10.3 Reciprocating machines

Routine vibration monitoring applied to reciprocating pumps, compressors, gasoline and diesel engines are generally quite effective for diagnosing mechanical problems such as rotating unbalance, misalignment and looseness. Reciprocating machines have an inherent vibration which is the result of inertia of the reciprocating components plus varying pressures on the pistons which cause torque variations. Excessive wear of rod and main bearings, piston slap, valve clash, compression leaks, faulty ignition, leaking valves, worn cam bearings, worn timing gears and chains cause high frequencies of vibration and can be detected [17].

CHAPTER 3

MEASUREMENT AND DATA PROCESSING

3.1 General

In chapter 2 the theory of predictive maintenance, and especially of vibration analysis, was outlined. This section provides a brief description of the principle of operation of instrumentation used in the measurement of parameters. It deals with parameters which characterize the total (broad band) signal and signal analysis.

3.2 Transducers

3.2.1 Transducer Types

Process transducers produce either a voltage or current signal corresponding to an engineering quantity such as temperature, pressure and speed. Dynamic vibration transducers include displacement probes, velocity pickups and accelerometers [10].

3.2.1.1 Displacement Transducer

Most common displacement transducers are non-contact proximity probes or eddy current devices and are usually used to monitor the motion of the journal relative to its bearing shell (sleeve bearing only) or to monitor thrust positions [17]. For sleeve bearings the transducers are usually mounted in the vertical and horizontal planes or in planes 90° apart, but off-set from the vertical and horizontal axes. The probe set-up in Figure 3-1 measures the position of the shaft relative to the bearing housing.

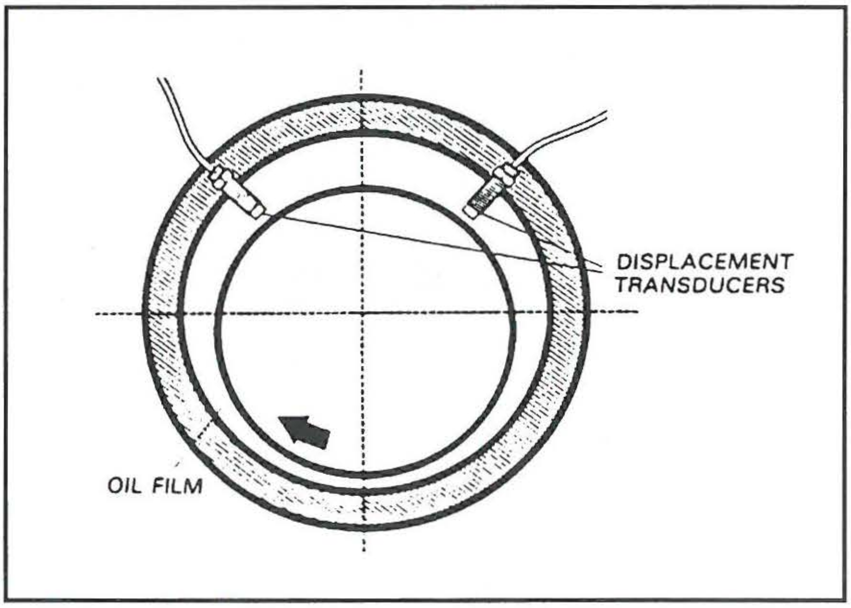


Figure 3-1: Proximite probe set-up

Displacement probes have excellent response at frequencies down to DC (zero Hertz) which makes them useful for measuring static position (both radial and axial), as well as vibration motion. The advantages of displacement probes are that they are highly sensitive at low frequencies and can provide true DC or static position values and that they are non-contacting.

The disadvantages of these probes are that the usable frequency range rarely exceeds 1000 Hz, making it impractical to measure high frequency vibration. The probes are also difficult to install.

3.2.1.2 Velocity Transducer

Velocity transducers were the first vibration transducers used in vibration analysis due to their ease of operation and the fact that the transducer signal is self-generating. They measure the absolute velocity of a measurement surface. They provide a voltage proportional to velocity without requiring additional signal conditioning [14]. There are three basic velocity probes that are presently used in industry, the inductive non-contacting pickup, the internal integrating accelerometer and the moving coil pickup. The most common type is the moving coil pickup.

In the moving coil pickup, an armature is suspended by springs inside a magnet as shown in Figure 3-2. Axial movement of the armature changes the flux of the magnetic field and generates a voltage proportional to velocity.

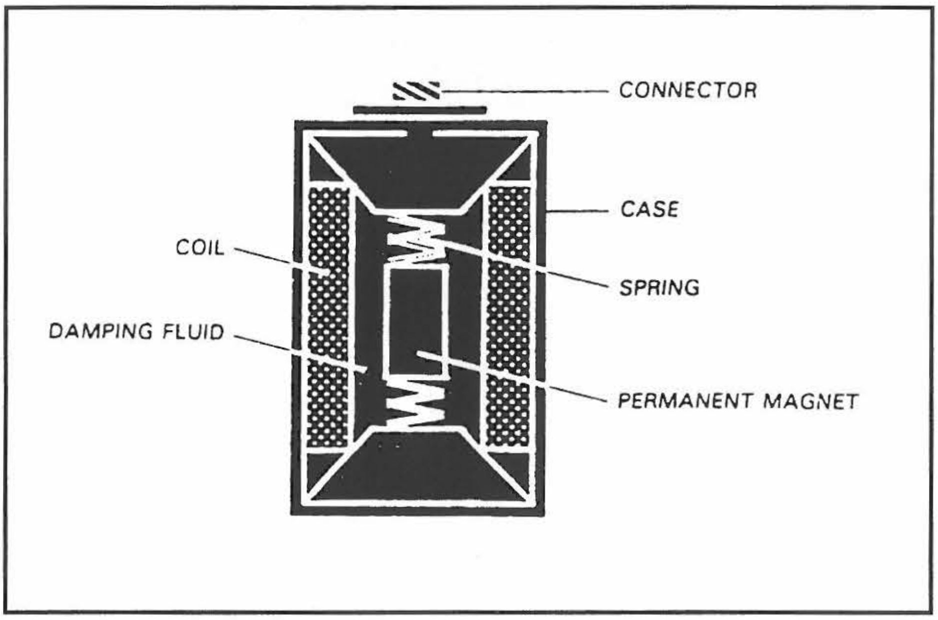


Figure 3-2: Velocity transducer

Velocity probes can be mounted directly onto machines with magnetic bases or stud-mounted to machined surfaces. The transducers can be "hand-held" but it may be difficult to obtain repeatable readings. The disadvantages of the moving coil pickup are size and frequency range limitations.

3.2.1.3 Accelerometers

Accelerometers offer the broadest range of types, sizes and frequency ranges and are the most widely used transducers in the field of vibration analysis [7][10]. They have frequency ranges higher than 20 kHz and as low as zero Hz.

The most common type is the piezo-electric accelerometer. As shown in Figure 3-3, a mass is placed on top of a crystal. When the base of the crystal is moved in the sensitive axis of the accelerometer, the crystal either expands or contracts which creates a charge. This charge is amplified into a signal voltage, either in the signal conditioning amplifier or with a small preamplifier located within the accelerometer housing. These accelerometers are known as charge and ICP (Integrated Circuit Preamplifiers) models respectively.

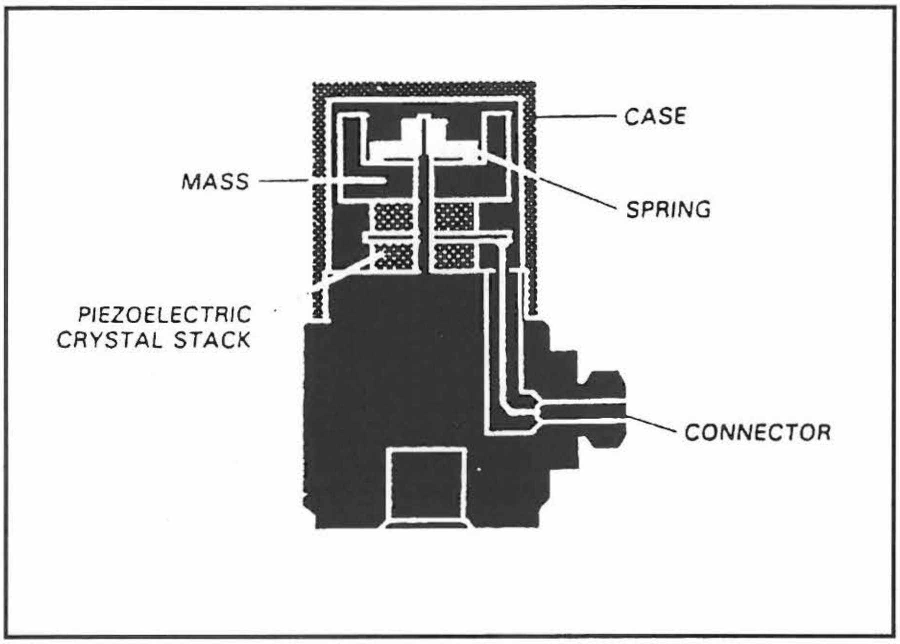


Figure 3-3: Piezoelectric Accelerometer

Accelerometers have many advantages, including size, cost, wide frequency range and temperature resistance. The disadvantages are that because of the wide frequency response, transducer mounting (Table 3-1) becomes critical and their excellent frequency response generates much data. For example, low frequency unbalance signal can be overwhelmed by high frequency gear mesh and bearing data. Accelerometer resonance can cause false high frequency readings unless properly filtered.

The mounting of these transducers have a significant effect on the frequency range. Table 3-1 employs the frequent response plots of the same accelerometer with different mounting configurations [7]. Trying to take repeatable high frequency readings with improper mounting techniques will lead to inconsistencies in the data.

Table 3-1: Mounting affects the usable frequency range of an accelerometer

Type of Mounting	Rating	Frequency Range	Notes
stud mount	best	120 - 600 000 cpm	Upper frequency limit depends on resonant frequency of accelerometer.
quick-lock	good	120 - 240 000 cpm	Attach transducer to permanent mounting blocks, using quick-lock connector.
magnetic	good	120 - 300 000 cpm	Maximum frequency depends on the physical characteristics of the magnet, transducer and mounting surface.
hand-held	fair	600 - 30 000 cpm	Transducer must be held with constant pressure.

3.2.2 Tachometer and Phase Reference Transducers

Tachometers provide rotational speed data and signals for analysis of events within a rotation, and data for systems that change speeds throughout a cycle [10][38].

Light tachometers use fiber optic technology to provide a voltage train related to shaft speed. A reflective surface is set on the shaft and a light source pointed at the shaft. As the shaft rotates, the reflective surface will cause a pulse shift. The biggest advantage of these systems is the ease of set-up. The machine will only be down for a short period of time to set up the reflective surface (white paint or reflective tape).

Strobe lights are also used to determine speed and in certain systems, phase can also be measured. The strobe is pointed at the shaft and the flash rate is adjusted to "freeze" the shaft. To measure phase, the flash rate is provided by the analyzer. Phase markings placed on the shaft are documented versus a defined reference.

3.2.3 Transducer Mounting, Location and Position

Considerations in a transducer mounting include accessibility, surface condition, environment, desired frequency range and speed of acquisition [10].

Bad mountings result in bad data or damage to the transducer. Although permanent and stud-mounted transducers give the best results, magnetic mounting should be considered, because it increases the speed of collecting data. The use of magnetic base accelerometers does not lower the frequency range appreciably, and makes repetitive measurements possible. Making consistent measurements within a short period of time is the most important factor for trending data.

The transducer must be held at the exact same location for each measurement. All the measurement points should therefore be permanently marked on the machines.

Measurements must be taken radially (horizontally and vertically) and axially on each bearing cap. The transducer must be mounted on a solid surface such as the bearing housing, as closely as possible to the bearing itself. A weak surface such as the flange or fin will lead to degradation in performance such as amplitude errors and inconsistent readings.

3.3 Signal Conditioning

All transducers require some level of amplification, attenuation or conditioning to generate a calibrated signal. In modern data collectors, most transducer interfaces are internal [7].

3.3.1 Filters

Filters are used to eliminate unwanted data from a signal. The four most common types of filters are [38]:

3.3.1.1 Low Pass Filters

A low pass filter passes all frequency information below the cut-off frequency of the filter. If an accelerometer has a resonance at 10 kHz, a low pass filter would be set to a cut-off frequency of 5 kHz to pass information only within the linear range of the transducer. Another common application is the prevention of alaising which will be discussed in more detail later in the section.

3.3.1.2 High Pass Filter

High pass filters pass frequencies above the cut-off frequency. In vibration analysis, high pass filters are often used to remove dominant lower frequency vibration from the measurement. When the vibration amplitude due to unbalance is 10 times greater than the bearing frequencies, the cut-off frequency can be set above running speed which will filter the unbalance data and allow a more precise measurement of the higher frequency components.

3.3.1.3 Band Pass Filters

Low and high pass filters are set to frequencies below and above a frequency range, or band. This filter is useful to identify the contribution of specific frequency bands.

3.3.1.4 Band Reject

The opposite of band pass, low and high pass filters are set above and below the cut-off frequencies respectively. A band reject filter could be set-up, for example to reject 50 Hz electrical noise from a vibration signal.

3.4 Digital Signal Processing

3.4.1 Analog to Digital Conversion

Digital signal processing has made modern vibration analysis possible. It deals with the process of converting an analog signal from a transducer to a digital signal to form a frequency spectrum [7].

Before a computer can operate on numbers from vibration data, it must be converted into a format that the computer can understand. Time history data from a vibration transducer is analog data. This is continuous in time and at any point in time the data can be looked at to determine at which amplitude and time it occurred. Digital data is not continuous. Snapshots of the analog data are taken in fixed intervals over a finite period of time and are stored in memory bins. From a continuous analog signal, a series of numbers are generated, which is exactly what the computer needs to operate on.

The process of digitizing the analog data is called Analog to Digital Conversion (ADC) which is shown in Figure 3-4. The spectrum analyzer is an example of an ADC device. Two of the most important errors introduced by the ADC are aliasing and leakage (discussed later in the section).

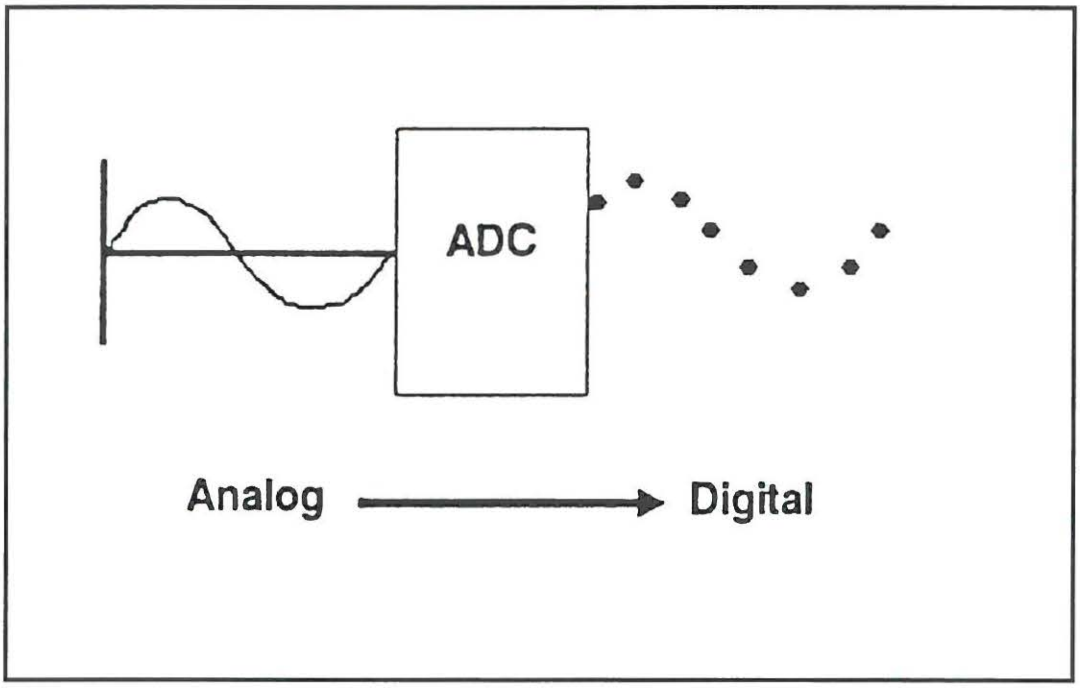


Figure 3-4: Analog to Digital Conversion

3.4.2 Digital Sampling

Sampling refers to the digital time increments needed for the ADC. Analog-to-digital converters sample at constant increments. By using a constant time increment, the information can be stored in half the space. The spacing between increments is Δt and refers to the resolution. Figure 3-5 illustrates digital sampling.

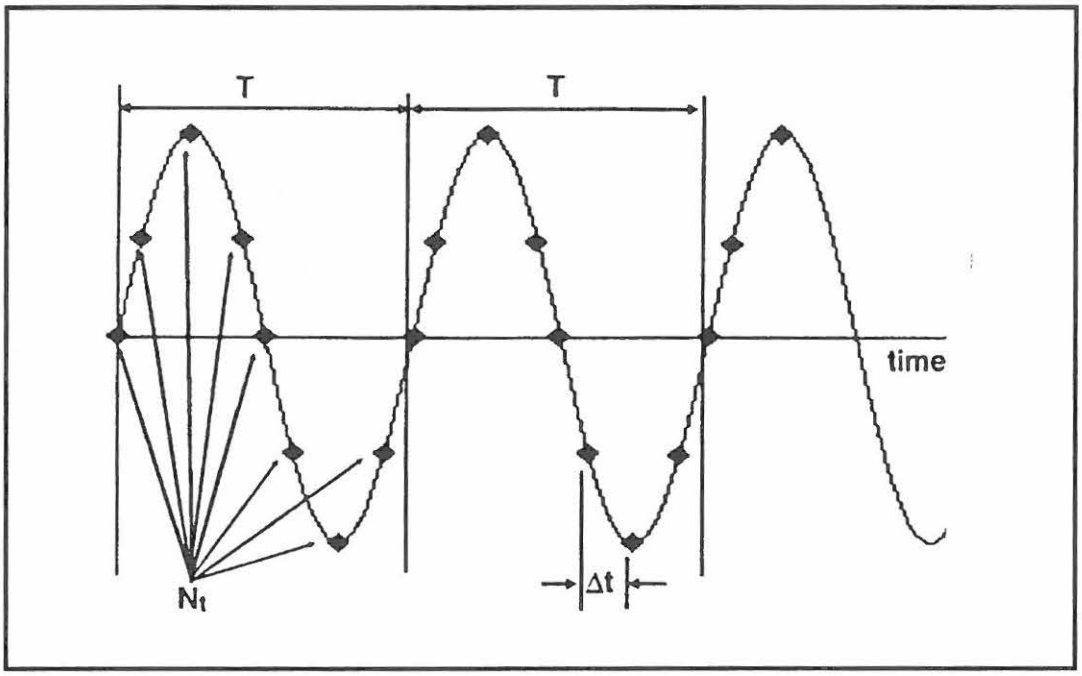


Figure 3-5: Digital Sampling

3.4.3 Fast Fourier Transforms (FFT)

The Fast Fourier Transforms are used for transforming digital time data to the frequency domain, resulting in a digital spectrum [38]. The spectrum is stored as a series of amplitude values as the incremental frequency spacing between them. This spacing is constant and is called Δf; it refers to the "resolution" of the measurement.

3.4.3.1 Fast Fourier Transform Errors

Fast Fourier Transforms have two conditions that must be met if it is to provide accurate results [7]. The first is that the signal must be periodic within the time window and the second is that Shannon's Sampling Theorem cannot be violated. The two errors that result are leakage and aliasing.

3.4.3.2 Leakage Errors

Leakage is more easily pictured in the frequency domain. At each spectrum line of resolution, a sine wave at that frequency will have an integer number of samples in the time sample. A signal that does not match one of the frequency bins is non-periodic and the energy will be distributed (leaked) into adjacent frequencies as shown in Figure 3-6.

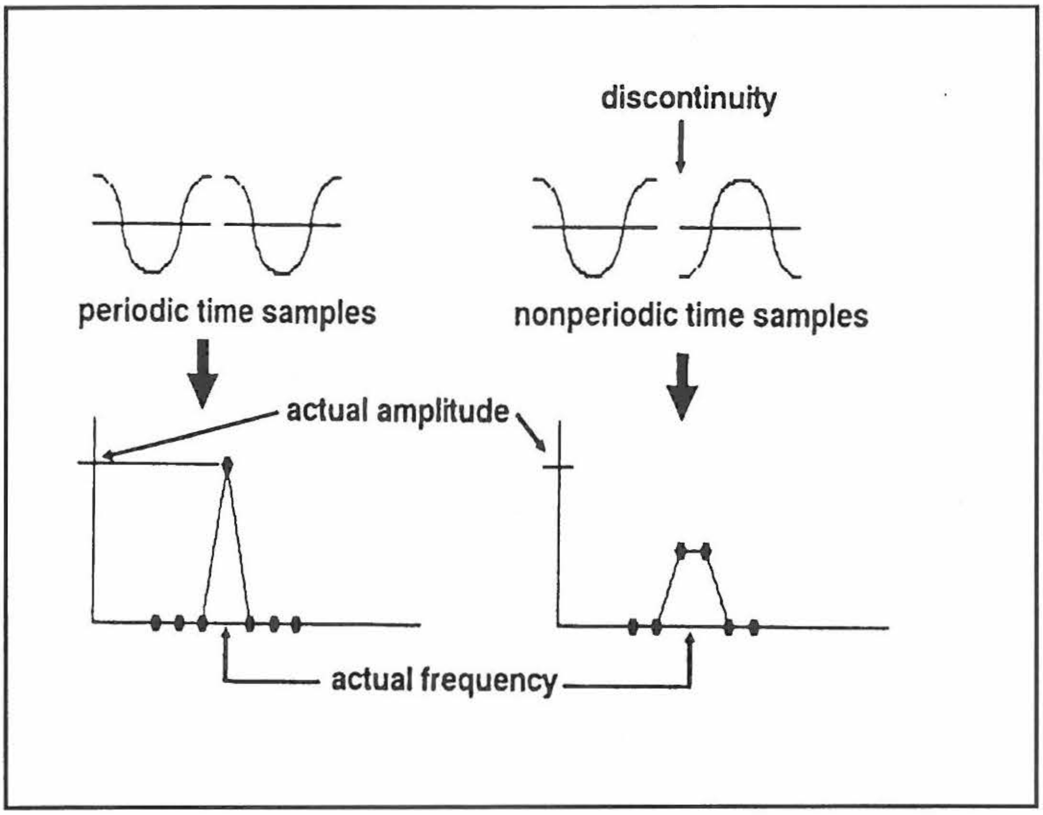


Figure 3-6: Effects of Leakage

3.4.3.3 Aliasing

Spectrum analyzers rarely display the total number of spectrum lines from the Fast Fourier Transform because of a phenomen called aliasing. Aliasing errors occur when Shannon's Sampling Theorem is violated [7]. Shannon's Sampling Theorem states that the sampling frequency of any signal must be greater than twice the maximum frequency which is to be uniquely resolved in the spectrum of the data.

3.4.4 Averaging

Most machinery contains significant levels of background noise that are not associated with the machine. In successive spectrum samples, all of the spectrum amplitudes would vary from sample to sample. Therefore, a single spectrum measurement is not statistically accurate, because of the random nature of the signal. The more averages collected, the lower the random errors will be, but the measurement time increases.

There are two methods available to average the spectrum amplitudes and reduce any background noise.

3.4.4.1 Sum or RMS average

Successive spectrum measurements are summed and then either divided through the number of samples taken, or the RMS (root mean square) of the summation is computed.

3.4.4.2 Time averaging

Time Averaging is a technique that is used to reduce noise not associated with machine rotation. This aids in the identification of low level signals that may have been hidden in the noise.

3.5 Data Analysis

3.5.1 Frequency Range

The frequency range selected for a measurement requires careful planning as it has a direct impact on any spectrum data that is acquired, and may directly affect overall vibration readings. Machines produce a wide spectrum of frequencies, many of which are important indicators of the health of the machine. Due to the limitations of digital signal processing, compromises are required to ensure that the important frequencies are measured to the highest accuracy available in the instrument.

A common mistake in predictive maintenance data collection is to select the highest frequency range available under the assumption that more is better. The first consideration for the selection of frequency range is the operating condition of the machine to be analyzed. The operating speed and typical fault frequencies of the machine components are very important. Secondly, the frequency limitations of the acquisition hardware (transducer, signal conditioning and analyzer) must be accounted for.

3.5.2 ADC Input Range

Amplitude accuracy in a measurement is a function of the dynamic range of the analog to digital converter and the input voltage range. If the correct input range is not selected, or unwanted data is not filtered, important vibration data may be lost in the noise floor of the spectrum. The dynamic range of a spectrum is fixed and cannot be improved without replacing the analyzer. Therefore, it is best to set the ADC input range as low as possible to maximize the resolution that is available without overloading the input. Figure 3-7 illustrates the effect the input range setting has on a simple 1 volt sine wave.

For AC measurements the process of selecting the input range is relatively straightforward. Using the spectrum analyzer is the best method, because all internal filters and settings are activated. For spectrum measurements, select the desired frequency range first and then adjust the input range to a point just above the overload level. Any high frequency energy contained in the signal is removed by the anti-aliasing filters above the frequency range of interest. Many analyzers offer an autorange feature that automatically selects the optimum ADC input setting. When the analyzer is in the autoranging mode it is important that the transducer is not disturbed as the input range could be set, based on the movement of the transducer rather than the actual vibration.

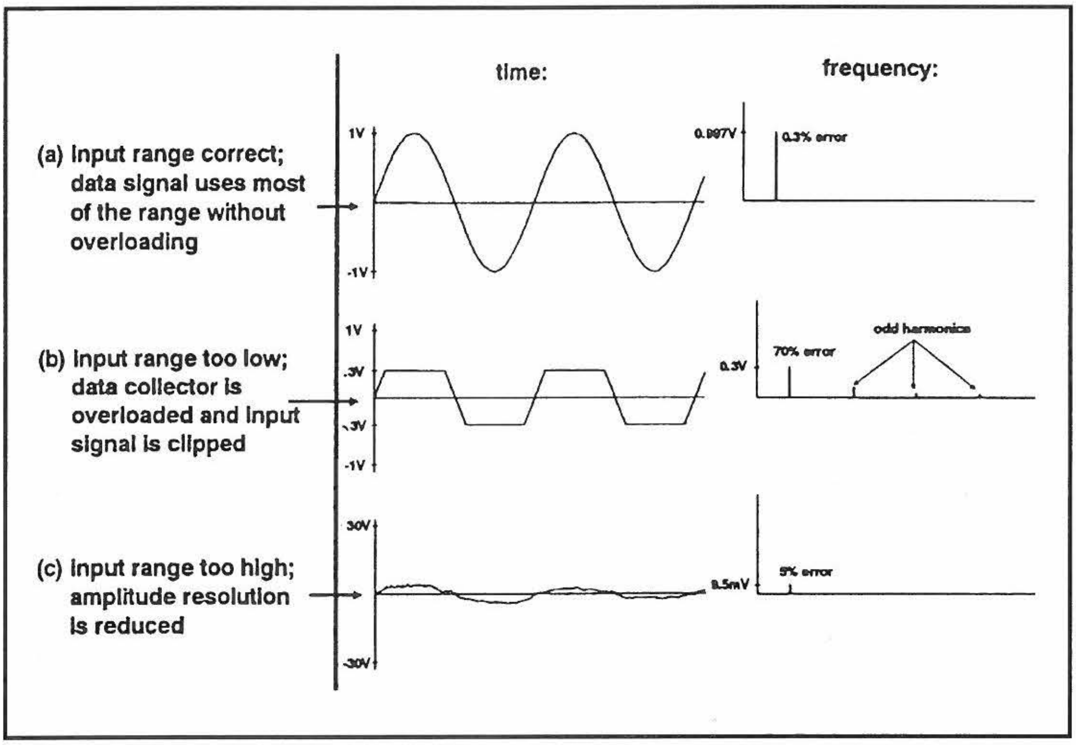


Figure 3-7: Selecting Correct ADC Range

3.5.3 Overall Value Measurement Methods

The object of this technique is to express the vibrations measured at any selected point on the machine in terms of a single vibration-level value. The vibration analyzer used two methods for measuring overall values:

3.5.3.1 Hardware Filtered Overall Measurement

The analyzer has a built-in frequency range for the overall measurement. This may be a filtered or unfiltered measurement. For filtered measurements, the analyzer usually has a large frequency range, for example, 0 - 20 kHz. For unfiltered measurements, the overall value is limited by the frequency responses of the transducer and analyzer.

3.5.3.2 Overall Calculations from Frequency Spectra

The analyzer measures a spectrum and computes the overall value from the spectrum's frequency range. Depending on the frequency content of your data signal, varying overall values can be obtained by changing the spectrum frequency range. This method is also called "band limited overall" or "power".

3.5.3.3 Overall Values Versus Spectrum Amplitudes

When viewing spectrum amplitudes, it will be noticed that the spectrum amplitudes are usually less than the overall value. Only signals with no leakage errors will have equal spectrum amplitudes and overall values.

The relationship between the spectrum amplitudes and overall value is best seen by considering a sine wave at 501 Hz. The amplitude of the sine wave is 2 V_{rms} as measured with a voltmeter. A 400 line spectrum is measured over a frequency range of 0-1000 Hz. The frequency resolution (Δf) is 1000/400 = 2.5 Hz. A Hanning window (line shape is 1.5) is used when measuring the spectrum. Figure 3-8 contains an expanded view (490-510 Hz) of the spectrum.

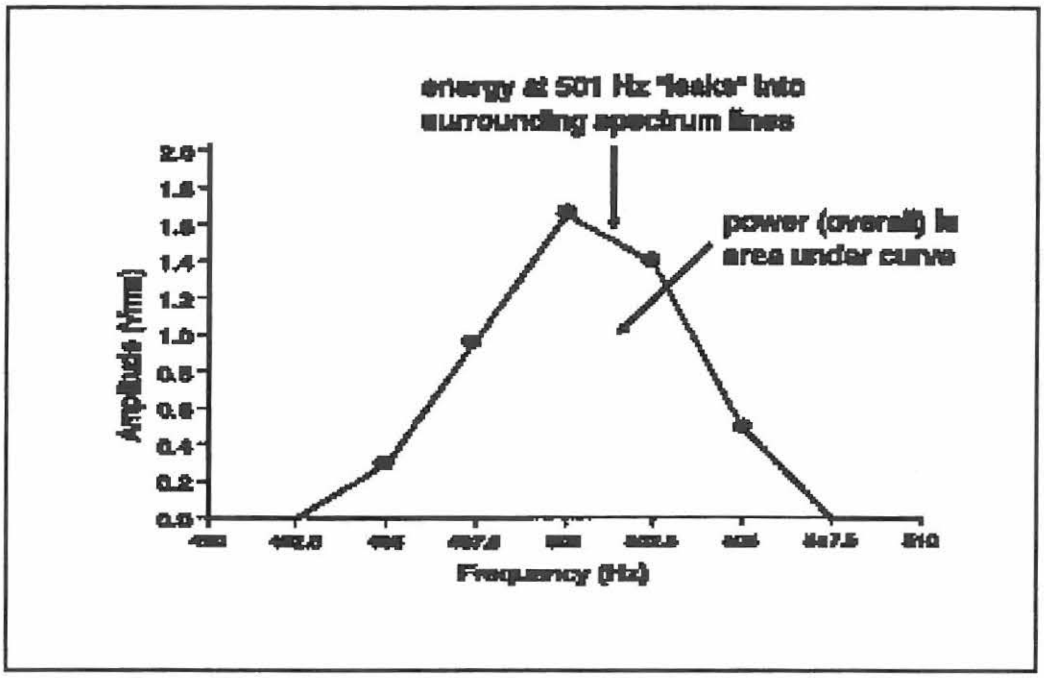


Figure 3-8: Relationship between the spectrum amplitude and overall amplitude

The maximum spectrum amplitude is 1.66, not the 2V_{rms} as measured with the voltmeter. This is due to leakage errors. The energy at 501 Hz has "leaked" into the surrounding spectrum lines (500 and 502.5 Hz). If the sine wave was at 500 or 502.5 Hz (falling directly on a line of resolution), then the spectrum line amplitude would be 2 V_{rms}.

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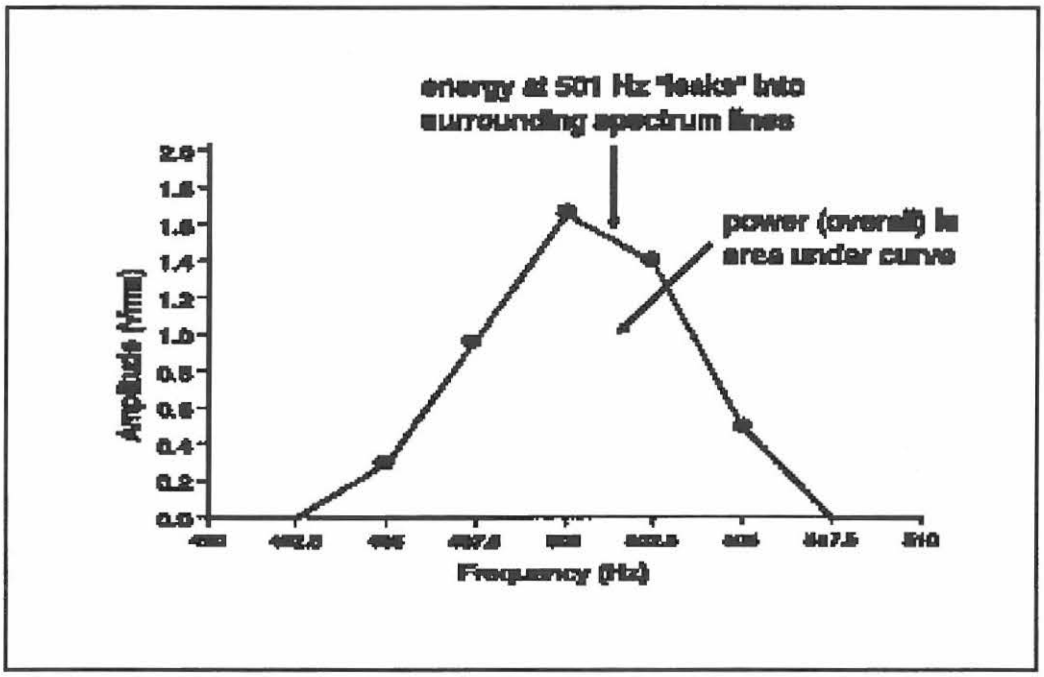

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3.5.4 Integration and Differentiation

Integration and Differentiation techniques are used to convert between acceleration (A), velocity (V) and displacement (D). There are two methods for integrating or differentiating the data:

a) Time Domain Conversion - It uses the analog signal from the transducer. This is done in the analyzer during measurements.
b) Frequency Domain Conversion - This is done by differentiation of a spectrum that takes place in the analyzer or computer after making the measurement.

3.5.5 Linear and Logarithmic Amplitude Scales

When monitoring equipment, gradual vibration changes must be detected before equipment failure. Overall vibration levels may be easy to measure, but they often only identify general vibration problems. Vibration data can be displayed on a linear or logarithmic scale. Logarithmic (log) amplitude scales are very useful for viewing small vibration signals in the presence of larger signals. Viewing the logarithmic scale, the vibration of a bearing can be identified. Figure 3-9 contains an example of linear and logarithmic scaling on the same data (the units are arbitrary for this figure).

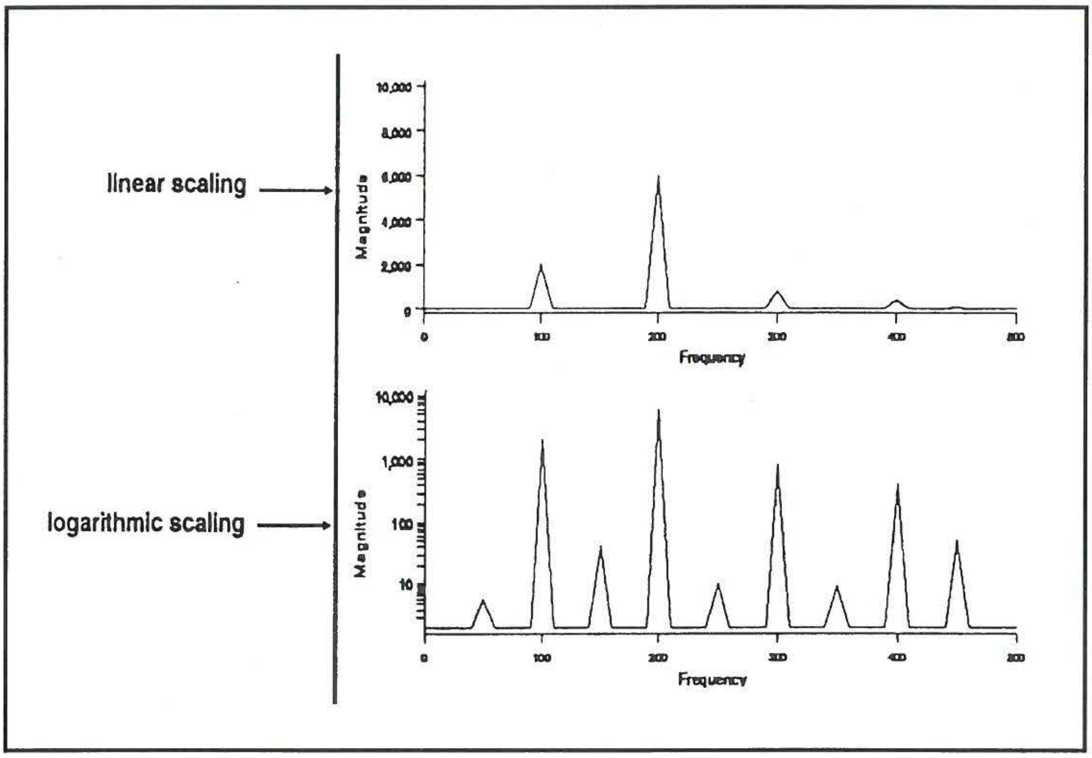

Figure 3-9: Linear and Logarithmic Scaling

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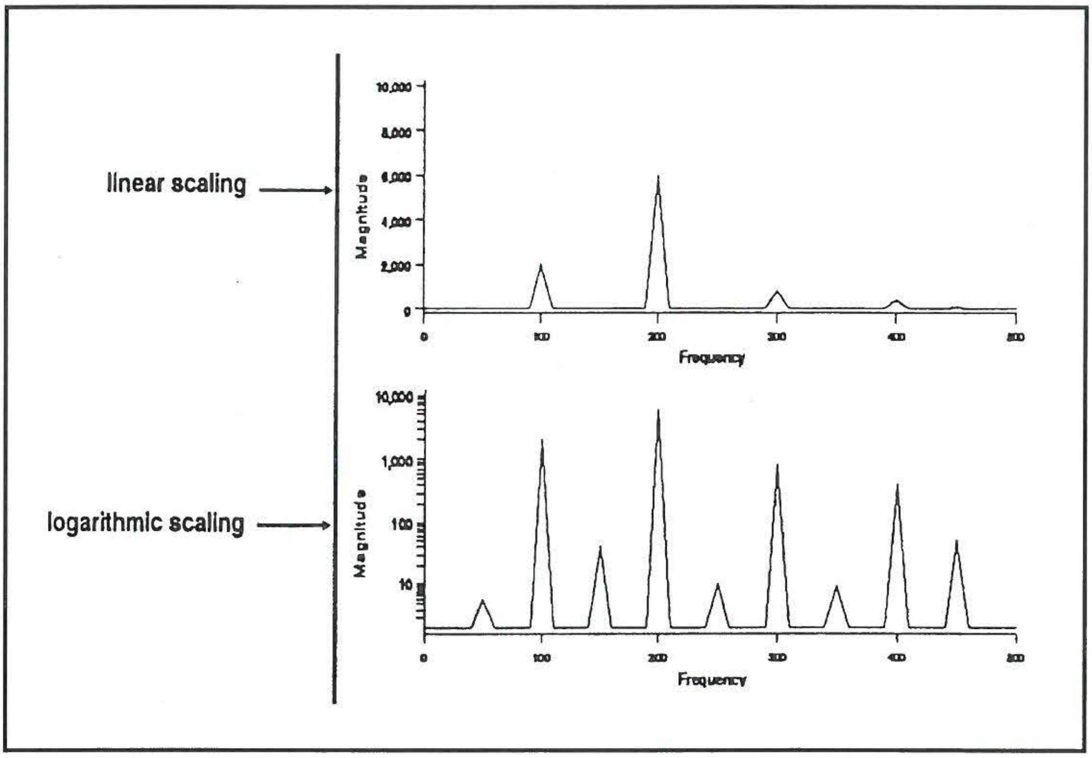

Figure 3-9: Linear and Logarithmic Scaling

CHAPTER 4

DATA ACQUISITION

4.1 General

This chapter provides the test data used to set up the predictive maintenance system, how problems were diagnosed and how to solve vibration problems. The vibration generated by various electrical motors and centrifugal pumps were measured. In all cases the pumps were driven directly via a coupling by electric motors.

4.2 Machinery Chosen For Experiment/Project

For practical reasons it was necessary to reduce the number of types of machinery being studied. Therefore a selected number of machinery based on various criteria (80 pump sets, ranging from 45 to 2400 kW) were selected for the project. The total number of vibration measurement points were approximately 800. Appendix B lists the general description of the electric motors and pumps studied with their mechanical descriptions. These descriptions are necessary to find the fault frequencies and process efficiencies. Appendix C lists the bearing frequencies.

4.3 Initial Condition-Monitoring Activities

4.3.1 A Viable Database

The main objectives for the establishment of a database for the project were:

a) To establish and build up a data base and systematically analyze measured condition data for selected machinery. This includes:-

i) selection of machinery and survey of defects,
ii) measured vibration response due to different occurrences and types of defects,
iii) recording of process parameters and operating conditions,
iv) a summary of machine condition and maintenance history describing events and causes.

b) To improve and develop the experience and expertise necessary to obtain a reliable and rapid diagnosis of deviation from normal conditions or unsatisfactory machine operation. This includes the investigation of methods for:

i) fault detection, location and identification,
ii) concise evaluation of events causing deviation from normal conditions,
iii) to evaluate the effectiveness of alarms to give advance warning of such conditions at an early stage, so that damage may be limited or faults can be corrected at the earliest opportunity, or other precautions be taken.

c) To promote the effective utilization of condition-monitoring data for machinery operation and maintenance planning.

4.3.2 Description of Vibration Equipment

When condition monitoring was started in 1992, monitoring was carried out by using the Shock Pulse Technique and an overall vibration meter giving only an RMS (Root Mean Square) measurement. It could have been sufficient to record overall vibration levels and upon analyzing detect a trend, but could not be used in an analysis to attempt to identify the cause of the vibrations. This information was not sufficient to trend the condition of a machine.

After a few months of feasibility studies into vibration analysis, a spectrum analyzer (IRD Model 350 Vibration Analyzer/Dynamic Balancer) was purchased. The instrument has a frequency range of 500 000 CPM (Cycles Per Minute) and uses integrated circuitry to provide a complete instrument for accurate vibration measurement, analysis and in-place balancing. An additional outstanding feature of the Model 350 is that it provides DC voltage outputs of amplitude and filter frequency to permit direct plotting of machinery vibration signatures and, when used with an optional XY Recorder (Model 1080), vibration signatures could be produced on-site for evaluation. The health of the machine was more noticeable with the help of the vibration spectrums and it allowed vibration associated with particular parts of the machine to be identified.

4.3.3 Collection of Vibration Data

The initial activities were concentrated on plotting data directly in the field, analyzing the data and initiating corrective actions. Back at the office the spectrum plots were filed. The database was established by manually reading the "overall" vibration measurements into database files in the computer. This whole process was very time-consuming. In this way a database was created and being developed, up to the implementation and upgrading of the system.

4.4 Implementation of an Automatic Acquisition/ Data Analysis System

4.4.1 The Complete System

After developing the requirements of the programme and the necessary experience was gained, the whole system/project was upgraded. A data acquisition-based system consisting of four main components: the analyzer (acquisition instrument), host computer, software programme and transducers were chosen. Figure 4-1 illustrates the automatic data acquisition/analysis scheme as implemented.

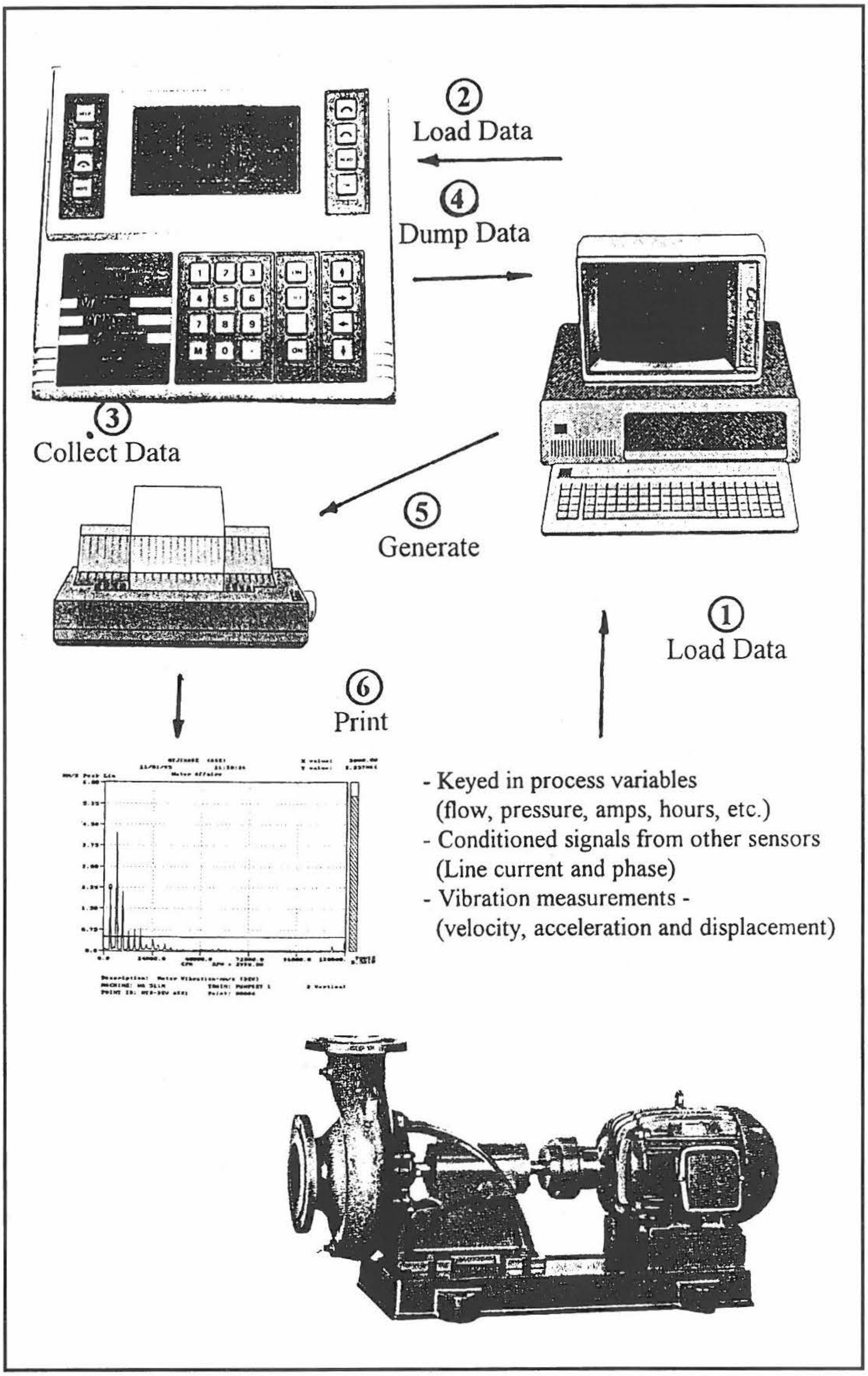

Figure 4-1: Data Acquisition/Analysis Scheme

The amount of data that was collected was massive and will continue to increase over the life of the programme. The system must be able to store, trend and recall data in multiple formats to monitor, trend and analyze the condition of equipment included in the project.

The system that was incorporated simplifies data acquisition and allows data to be transferred to the computer in a more rapid and less labour-intensive way. Vibration measurements were taken with a PL31 machine analyzer from Diagnostic Instruments. The acquisition-based data capture instrument can also compute the vibration spectrum and display it on an LCD (Liquid Crystal Display) on-site. The instrument is capable of storing in non-volatile memory around 300 spectra. Thus, on return to the office, this data can be transferred directly in a processed form to the computer for long-term storage and analysis. In association with this instrument a software package has been developed which automatically scans the data, identifying significant changes in the spectrum as they occur. In this way the researcher was relieved of the task of inspecting 300 or more plots at the end of the trip and just analizing the spectrums and overall readings which have been identified as significantly changed. When the analysis is completed, the data is stored on disk.

4.4.2 Measurement of Parameters

The mechanical vibration measurements were taken, using Piezoelectric type 327 ICP accelerometers, which have a resonant frequency greater than 12 kHz. Line current was measured with a clamp-on current transformer connected to the vibration instrument. Reliable data was clearly required since, without it, significant changes in the vibration characteristics can be lost in the uncertainty of the data. As discussed in section 3, attention was paid to the attachment of the vibration transducer to the machine. Deviation from the exact point or orientation will effect the accuracy of acquired data. It was also noted for accuracy of data, that a direct mechanical link to the machine's casing or bearing cap is absolutely necessary. Slight deviation in load will induce errors in the amplitude of vibration and may also create false frequency components that have nothing to do with the machine.

4.4.3 Data Acquisition

The data logger cannot be used for long-term storage. Therefore data was transferred into the host computer for further processing, presentation and storing. The actual time required to transfer the data into the host computer was the only non-productive time of the analyzer. Therefore, transfer time should be kept to a minimum. By using RS 232 communication protocol cable, data was transferred at a baud rate of up to 38 400 b/s. At this rate, long routes incorporating a large amount of equipment can be dumped in about 20 minutes. The data acquisition system was also implemented on the basis to support modem communications.

Measurements are carried out at regular intervals, averaging about once every six weeks. In order to gain as complete a picture as possible of a unit's performance it was comprehensively measured in the radial and axial directions on both driven and non- driven ends of each machine being monitored. Full vibration signatures and

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Measurements are carried out at regular intervals, averaging about once every six weeks. In order to gain as complete a picture as possible of a unit's performance it was comprehensively measured in the radial and axial directions on both driven and non- driven ends of each machine being monitored. Full vibration signatures and spectral components in the line current supplying the motor were acquired to verify the accuracy of the database set-up and to determine the initial operating condition of the machinery. As the project developed, trending and projected time-to-failure, as well as multiple readings, were required to provide sufficient data for the data base to develop trend statistics. Alarm generation for overall and spectrum bands were developed as discussed in the next section. If the rate of change of a specific machine indicated rapid degradation, the problem had to be identified, corrective actions carried out or at least initiated and the monitoring frequency was increased to prevent catastrophic failure. This increase of the monitoring frequency was one of the problems faced, because the equipment was located far from the condition-monitoring base station. It is not cost-effective to closely monitor the problem.

Using the information in Chapter 3, the boundaries and limits were set for each measurement point. The upper limit (Fmax) for each vibration spectrum was set high enough (120 000 CPM) to capture and display enough data to determine the machine condition. For most measurements, except where better resolution was required, the resolution was limited to 800 lines of resolution. The effect of this choice was that for a vibration signature with a maximum frequency of 120 000 CPM, taken at a resolution of 800 lines, the space between displayed lines would be equal to 150 RPM.

In the initial phases of implementation the maximum frequency (Fmax) was set too low which completely missed potentially serious developing bearing wear, particularly during the earlier stages. When this problem was detected, Fmax was again set too high and this again resulted in poor frequency resolution which caused misdiagnosis of data. The other problem experienced due to the high Fmax was that potentially valuable information on subsynchronous vibration was buried at the left-hand side of the spectrums.

4.4.4 Defining Alert and Alarm Limits

Years of experience and data accumulation has resulted in many vibration standards and severity charts like the ISO 2372 & 3945 (Appendix D) and the Rathbone chart, developed in the 1930s (Appendix E) [10][39]. These charts were derived from field measurements and they provide general guidelines to determine machine condition.

The problem when using industrial standards is that it applies to the average machine, and one is never sure whether a particular machine is average or not. It should already be suspicious when the modified Rathbone chart indicates the same levels for all machines, while other industrial standards divide machines into different classes, with a quite large spread.

When the initial alarm limits were set in the project, these published vibration limits were used because little information was available other than nameplate data of the machines.

4.4.5 Baseline Comparison

Baseline measurements are the set of vibration measurements of a machine against which future measurements can be compared to determine whether there has been any change in the machine's mechanical condition. It is a set of vibration measurements indicative of a machine in good mechanical condition, during normal operation and installed in its final configuration and measurements taken under actual operating conditions. Baseline comparisons are most effective when the machine operates in a consistent or constant environment. Changes in operating conditions can change the vibration levels of a machine.

A few sets of readings were taken to determine actual operating conditions of the equipment. Irrespective of the condition of the equipment, this was assumed as the baseline measurement for that particular machine until more up-to-date data was available.

Another method was used to get baseline data, namely to calculate the average overall vibration level for the same type of equipment. The level for each point on each similar machine was summed and divided by the total number of samples.

4.4.6 Spectral Alarms

Spectral alarms are a set of alarm bands incorporated on a vibration spectrum to help visualize potential fault frequencies. The main purpose for setting spectral alarms was to predict and detect possible specific problems in the machines as follow:

a) Specific mechanical faults such as unbalance, misalignment, etc. generate mechanical vibration in a well-defined frequency band or pattern. Problem detection was performed by:

   i) Breaking each measurement point into frequency bands of interest.
   ii) Applying multiple evaluation criteria to warn of the severity and nature of problem.

b) Specific mechanical faults such as unbalance or bearing damage generate different energies or amplitudes of vibration which may be considered severe.

The most possible effective guidelines for setting-up initial spectral alarms were found in Appendix F [40]. This allowed the spectrums to be broken up into 6 individual bands which the present software could also accommodate. Each of these bands can be set at any span of frequencies, and at any alarm level for each individual band.

Looking at item (i) in Appendix F, "General Rolling Element-Bearing Machine Without Rotating Vanes", it specified a maximum frequency range of 80x RPM (120 000 CPM) for a nominal speed of 1500 RPM. It was found, using a maximum of 120 000 CPM, that the spectrum of the machine would detect a rolling element bearing in all four failure stages through which it would normally pass. As discussed in section 2, the natural frequencies of bearing components will be excited in the second stage with a range from 30 000 to 120 000 CPM for most bearings. It is important to keep the Fmax sufficiently high to detect these when excited.

The specific purpose of item (iii) in Appendix F on each motor is to detect the presence of 1X and 2X rotor bar pass frequency (6000 CPM) side bands, and even slip frequency side bands. The rotor bar pass frequency (RBPF) is equal to the number of rotor bars times motor RPM. High amplitudes at rotor bar pass frequencies suggest rotor bar looseness and rotor eccentricity, particularly when these frequencies are accompanied by the 2X line and 2X slip frequency sidebands.

The same applies to item (iv) in Appendix F where mechanical and electrical vibration frequencies, particular in the area of 1X RPM, line frequency (3000 CPM or 50 Hz), and 2X line frequency (6000 CPM or 100 Hz). The spectrum will show high vibration at a frequency of 6000 CPM, which might imply electrical problems.

The primary purpose for specifying bands for machines in item (v) Appendix F specifically for centrifugal pumps from Types 1 through 4 was to separate the blade pass frequency band from the bearing defect frequency band. The problem here is that amplitudes which would be acceptable to blade pass frequency would normally be excessive for a bearing-defect frequency.

It was found that fluid-film bearings have relatively high damping coefficients when compared to anti-friction bearings. The higher the damping, the lower the vibration in the bearing cap due to the decrease in transmitted force from the rotor to the bearing cap. Bearing cap vibration measurements were indicative enough of machinery condition only when rotor forces were sufficiently transmitted through the bearing to induce bearing cap movement.

Each set of spectral alarm bands for each machine category was re-evaluated at six monthly intervals. These spectral alarms are very important to detect machine deterioration as it occurs, especially early in the failure process, so that corrective measurements can be planned orderly and that the required replacement parts can be acquired.

Figure 4-2 shows an example for the set-up of spectral bands for a 1500 rpm electric motor with rolling element bearings.

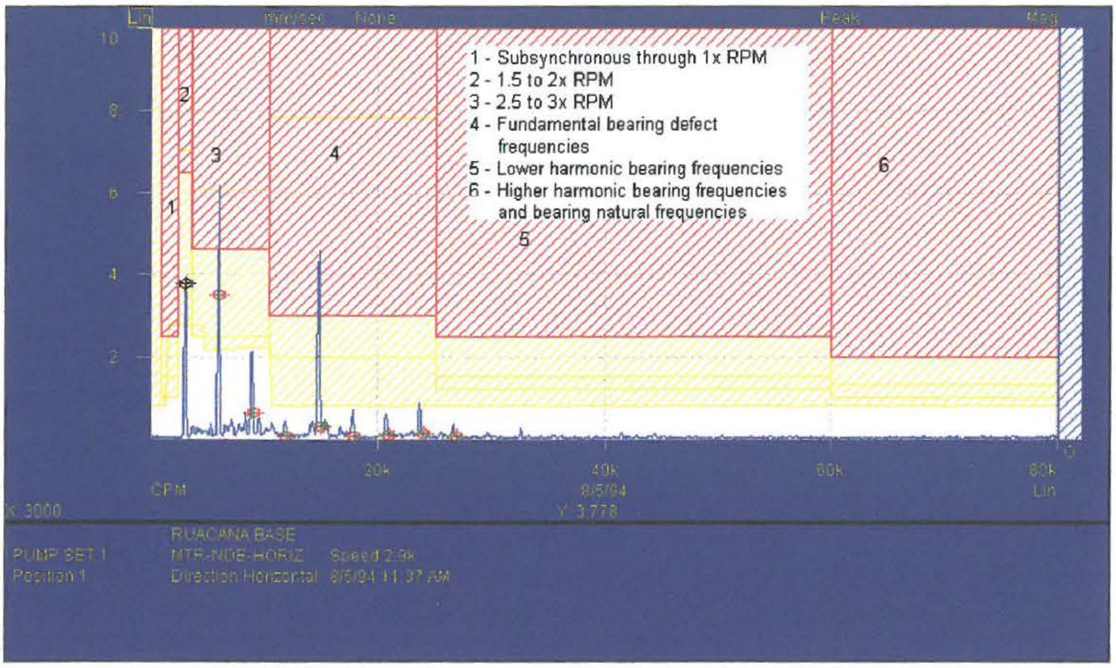

Figure 4-2: Examples of Spectral Alarms

4.5 Data Input Module

Spectrum plots are plots of amplitude versus frequency. They are vibration signatures because they are specific to a particular piece of equipment and are used to identify the vibration of components in the machine. The X (horizontal) axis units are frequency in Hertz, CPM or Orders (multiples of the operating speed). The Y (vertical) axis amplitude units are typically acceleration, velocity, displacement or voltage.

The input module of the software was used for initialization and management of the system. All input data was entered, namely: probe and machine identifications, signal calibration constants and alarm levels. This information was required for detailed analysis of data and listing or graphical output of results in report format.

The measured data is stored in the computer as a data file in the form of a time history, frequency spectrum or trend data. A data file head associated with each file contains all the information required to describe the measured signals and also indicates where the relevant administration data is stored in the system management.

Vibration points were created and set up for equipment in database files. Figure 4-3 shows the frame of the database hierarchy. The database was set-up as follows:

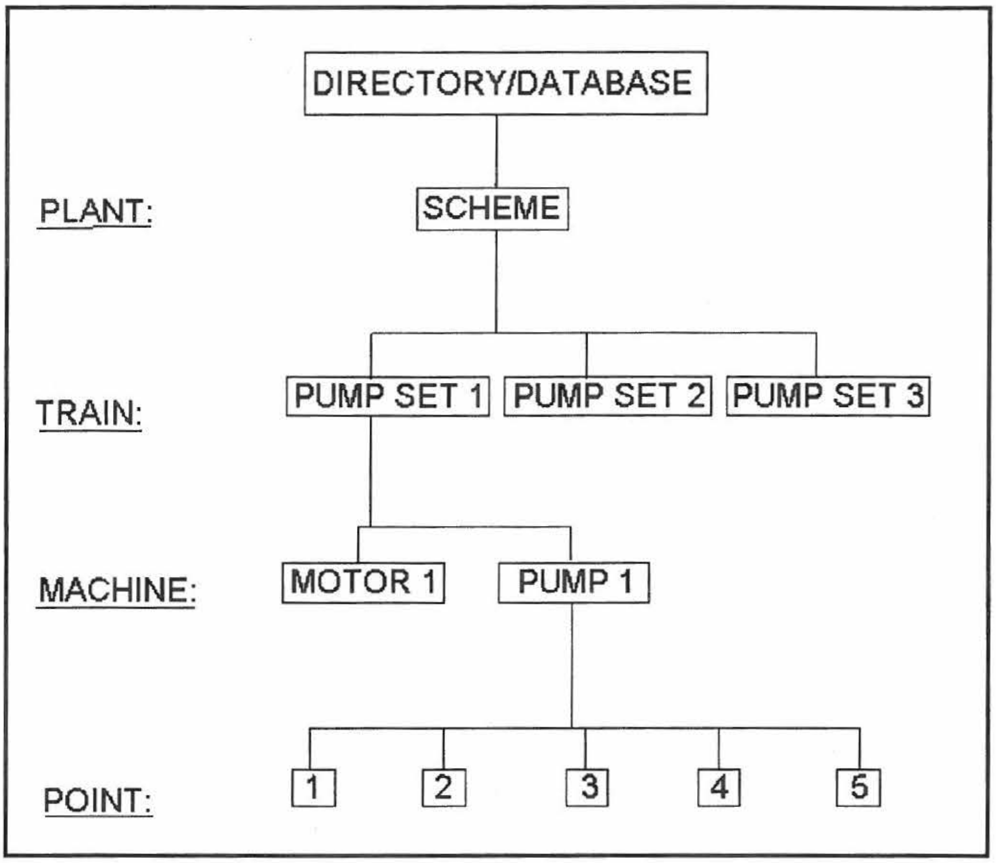

Figure 4-3: Database File Hierarchy

a) The Plant is the first level in the database hierarchy. It is a group of measurement points located in one geographical area of a Water Supply Scheme (Figure 4-3).

b) The Train database file is the second level in the database hierarchy. It consists of production line numbers (pump sets) in the plant.

c) Machine database files are the third level in the database hierarchy. It consists of the electrical motors and pumps to be studied.

d) The Point is the lowest level in the database file hierarchy. Points are used for data collection and plotting.

e) The Frequency Bands database file contains information for plotting frequency trend plots.

4.6 Data Output Module

4.6.1 General

Measured vibration data is stored on the computer until required. Plots of frequency spectra, time or process parameters may be obtained when required.

4.6.2 Trend Plots

Trends were used for monitoring the change in vibration level with time. The results were plotted graphically for the comparison of total vibration levels at different measurement points on a particular machine. A check module scans through all results and gives a condition summary report, indicating if alarm levels are exceeded. This module was used to list vibration levels measured on each machine or to list vibration levels measured that exceeded the alarm level as previously set.

Trend plots were printed to keep track of any changes in the machine's operating parameters. These plots are two-dimensional displays of amplitude versus time. The X (horizontal) axis units are in terms of date/time and the Y (vertical) axis units are amplitudes of overall vibration, temperature, voltage, pressure, flow and operating hours. Alarms were plotted and labelled on the trend plots as specified. Overall trend, average trend and single frequency trend plots including curve fitting were generated from the most recent data to predict the alarm date.

4.6.3.1 Overall Trend Plot

The overall trend plot in Figure 4-4 was used to display the overall values collected for single measurement points over a specified period of time. These plots are useful for keeping track of a machine's overall vibration level and for predicting when those levels will exceed the safe operating range. The curve fit predicts when an alarm will be exceeded if the current trend is followed.

4.6.3.2 Frequency Trend Plot

The Frequency trend in Figure 4-5 displays up to six different single frequency trend plots at one time. Each of the bands can be labelled with a band name or band number. These plots are useful for keeping track of the vibration of individual machine components and predicting when those levels will exceed the safe operating range.

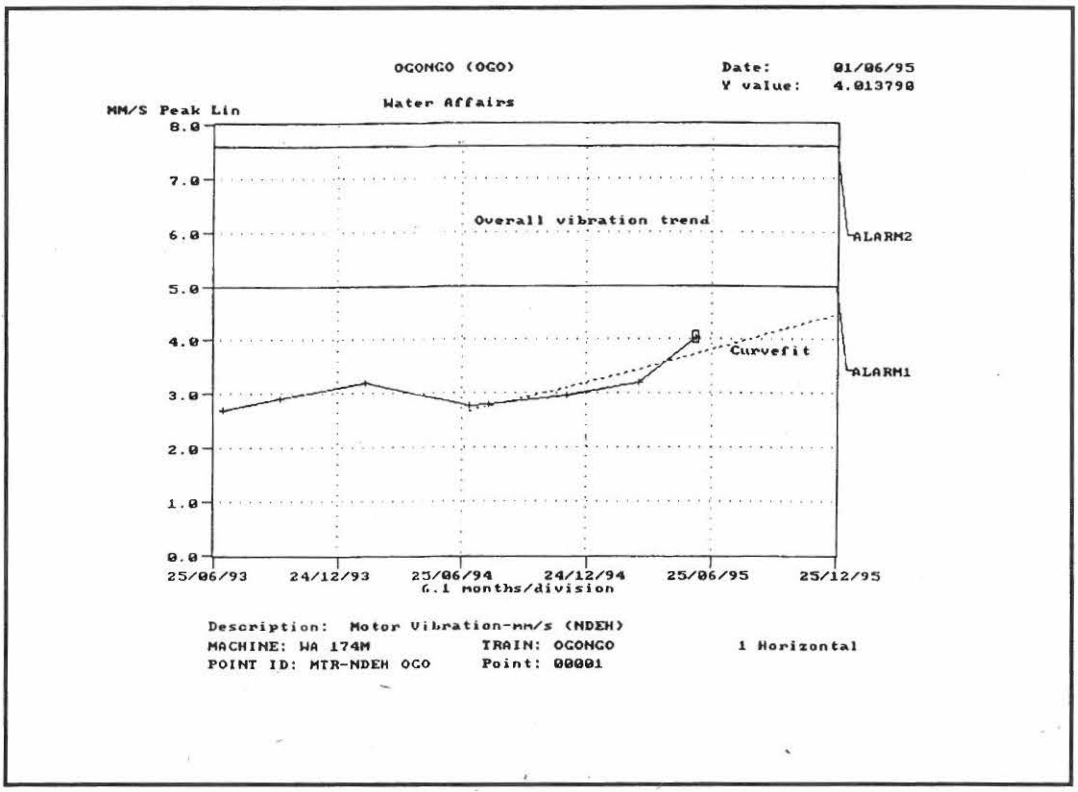

Figure 4-4: Overall Trend Plot

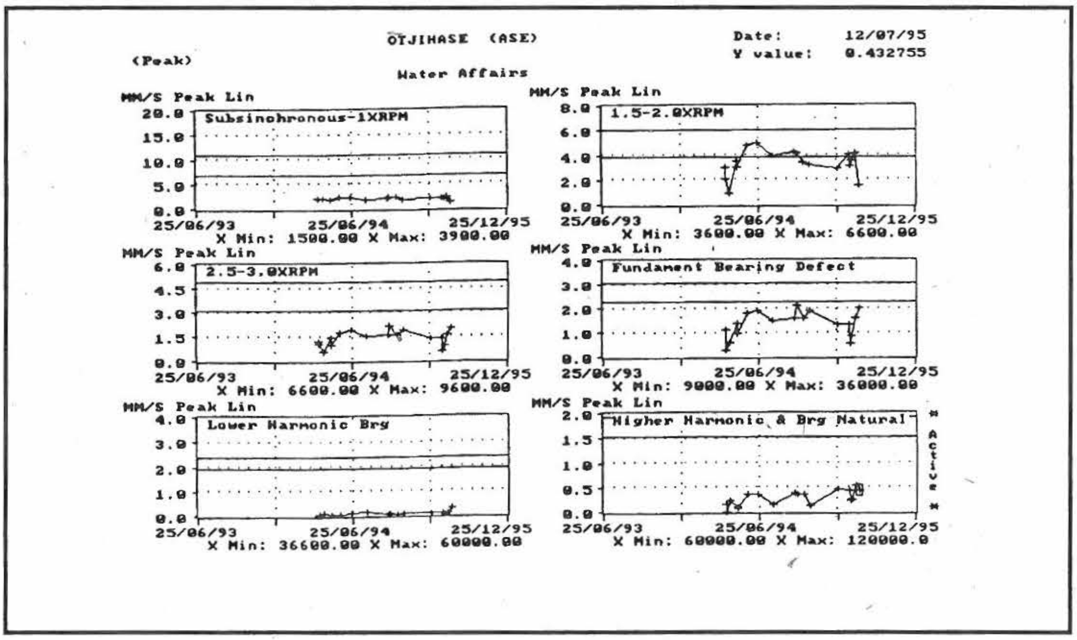

Figure 4-5: Frequency trend plot

4.6.3.3 Single Spectrum Plot

The single spectrum plot in Figure 4-6 displays a spectrum for a single measurement point. These plots identify important frequency bands and peaks exceeding alarm levels.

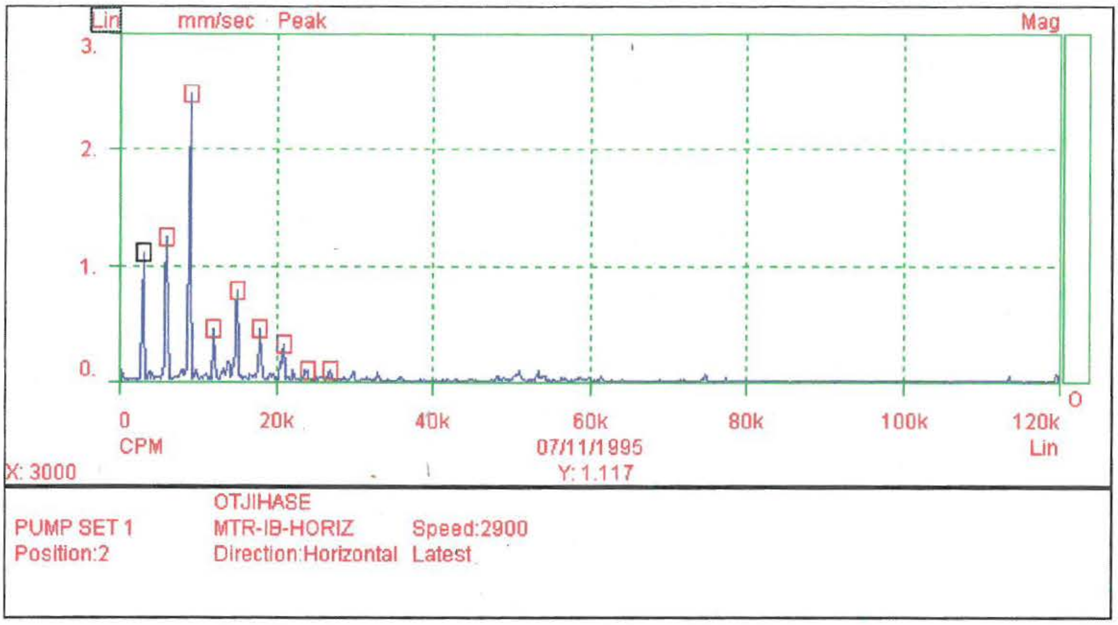

Figure 4-6: Single Spectrum Plot

4.6.3.4 Spectrum Map Plot

The spectrum map plot is used to collect single spectra arranged in a three-dimensional format. Plot 4-7 is an example of a spectrum map plot of all spectra taken at a specific measurement point.

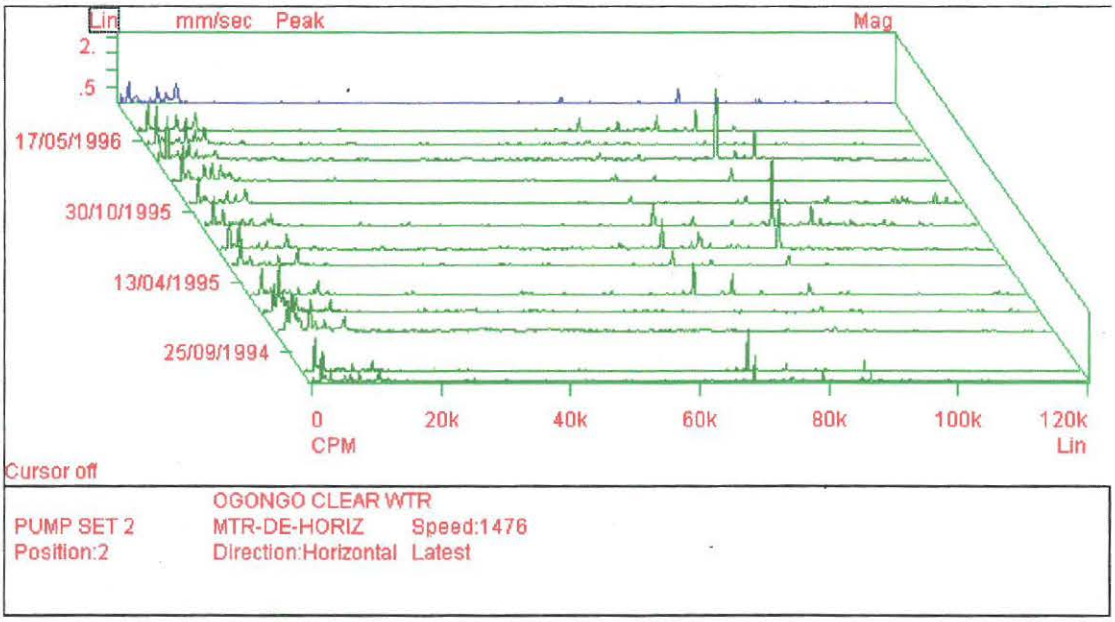

Figure 4-7: Spectrum Map Plot

4.6.3 Report Generation

Reports were generated and printed after each set of data was processed. Standard report formats used were:

4.6.3.1 Overall exception

Report of points in the plot list whose most recent overall trend value was an exception (Table 4-1). The following was reported:

a) Plant, Train and Machine.
b) Point ID, point number and measurement units.
c) Current and last measured overall trend values.
d) Alarm type, alarm 1 and alarm 2 values.
e) Percentage change between the two most recent overall trend values.
f) Ratio (percentage) of the most recent overall trend value to alarm 1.
g) Ratio (percentage) of the most recent overall trend value to alarm 2.
h) Date on which the last overall trend value was collected.
i) Exception status for the most recent overall trend value collected.

4.6.3.2 Last Measurement

Report of all points in the plot list and their last measured value (Table 4-2) as follows:

a) Plant, Train and Machine.
b) Point ID, point number and measurement units.
c) Current and last measured overall trend values.
d) Lower overall threshold alarm value.
e) Percentage change between the two most recent overall trend values.
f) Date on which the last overall trend was collected.

4.6.3.3 Peak Spectrum Value

Report of the peak spectrum value in the spectrum (frequency and amplitude) for the most recent spectrum collected for each point on the point list (Table 4-3) as follows:

a) Plant, Train and Machine.
b) Spectrum measurement units.
c) Amplitude and frequency of highest peak in spectrum.
d) Date on which spectrum was collected.

Table 4-1: Overall Exception Report

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OVERALL EXCEPTION REPORT

POINT ID	POINT	UNITS	LAST VALUE	CURRENT VALUE	ALARM 1	ALARM 2	ALARM TYPE	%CHANGE	%A1	%A2	DATE	OVER
PMP-DEV RCO/PU	00007	MM/S	6.537440	6.497000	5.000000	7.600000	Above Alarm	-1	130	85	30/05/95	YES
PLANT	: RUACANA CLEARWATER (RCO)											
TRAIN	: RCO/PURIFICATION											
MACHINE	: WA 625M											
ELECTRICAL DEH	00047	MM/S	0.000000	8.157970	5.000000	7.600000	Above Alarm	***	163	107	17/01/95	YES
MTR-DEV RCO/PU	00014	MM/S	9.647660	7.677310	5.000000	7.600000	Above Alarm	-20	154	101	17/01/95	YES
MTR-DEH RCO/PU	00013	MM/S	7.018110	8.019500	5.000000	7.600000	Above Alarm	14	160	106	17/01/95	YES
MTR-NDEH RCO/P	00011	MM/S	8.548610	6.464200	5.000000	7.600000	Above Alarm	-24	129	85	17/01/95	YES
ROTOR BAR NDEH	00048	MM/S	0.000000	6.407360	5.000000	7.600000	Above Alarm	***	128	84	17/01/95	YES
MTR-DEA RCO/PU	00015	MM/S	19.922400	12.277500	5.000000	7.600000	Above Alarm	-38	246	162	17/01/95	YES
PLANT	: RUACANA CLEARWATER (RCO)											
TRAIN	: RCO/BOOSTER/CLEA											
MACHINE	: WA 326C											
ESP-NDEV RCO/B	00079	ESP	0.000000	2.981350	1.300000	2.500000	Above Alarm	***	229	119	10/03/95	YES
PMP-NDEV RCO/B	00078	MM/S	3.883480	3.199560	2.500000	4.000000	Above Alarm	-18	128	80	10/03/95	YES
PMP-NDEH RCO/B	00076	MM/S	7.172610	5.772190	3.000000	5.500000	Above Alarm	-20	192	105	10/03/95	YES
ESP-DEV RCO/BO	00075	ESP	0.000000	11.346100	1.300000	2.500000	Above Alarm	***	873	454	10/03/95	YES
PMP-NDEA RCO/B	00080	MM/S	4.551220	4.591810	2.500000	4.000000	Above Alarm	1	184	115	10/03/95	YES
ESP-DEH RCO/BO	00073	ESP	0.000000	12.576000	1.300000	2.500000	Above Alarm	***	967	503	10/03/95	YES
PMP-DEH RCO/BO	00072	MM/S	7.294680	11.276200	5.000000	7.600000	Above Alarm	55	226	148	10/03/95	YES
PMP-DEV RCO/BO	00074	MM/S	4.608140	9.297840	5.000000	7.600000	Above Alarm	102	186	122	10/03/95	YES
PLANT	: RUACANA CLEARWATER (RCO)											
TRAIN	: RCO/BOOSTER/CLEA											
MACHINE	: WA 683C											
PMP-NDEH RCO/B	00028	MM/S	6.236450	19.853000	5.000000	7.600000	Above Alarm	218	397	261	05/08/94	YES
PMP-NDEA RCO/B	00030	MM/S	113.628000	4.355750	3.000000	5.500000	Above Alarm	-96	145	79	06/07/94	YES
PMP-DEV RCO/BO	00027	MM/S	10.877100	12.000600	5.000000	7.600000	Above Alarm	10	240	158	05/08/94	YES
ESP-DEV RCO/BO	00042	ESP	0.000000	3.439580	2.500000	4.000000	Above Alarm	***	138	86	05/08/94	YES
ESP-NDEV RCO/B	00043	ESP	0.000000	19.138100	2.500000	4.000000	Above Alarm	***	766	478	05/08/94	YES
ESP-NDEH RCO/B	00044	ESP	0.000000	23.352700	2.500000	4.000000	Above Alarm	***	934	584	05/08/94	YES
PMP-NDEV RCO/B	00029	MM/S	10.022200	13.889700	5.000000	7.600000	Above Alarm	39	278	183	05/08/94	YES

Table 4-2: Last Measurement Report

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17/07/95

L A S T   M E A S U R E M E N T   R E P O R T

POINT ID	POINT	UNITS	LAST VALUE	CURRENT	ALARM NO1		%CHANGE	DATE
PLANT	: RUACANA CLEARWATER (RCO)							
TRAIN	: RCO/BOOSTER/CLEA							
MACHINE	: WA 282M							
ESP-NDEH RCO/B	00057	ESP	1.312410	1.25803	2.500000		-4	10/03/95
MTR-DEA RCO/PU	00070	MM/S	4.469740	3.28104	5.000000		-27	10/03/95
MTR-DEV RCO/PU	00069	MM/S	3.199560	2.84142	5.000000		-11	10/03/95
ROTOR BAR NDEH	00056	MM/S	0.000000	2.97158	5.000000	***		12/10/94
MTR-DEH RCO/PU	00068	MM/S	3.712570	3.17514	5.000000		-14	10/03/95
ESP-NDEV RCO/B	00058	ESP	0.575273	1.04440	2.500000		82	10/03/95
ELECTRICAL DEH	00055	MM/S	0.000000	3.75331	5.000000	***		12/10/94
ESP-DEH RCO/BO	00060	ESP	2.806930	3.74095	2.500000		33	10/03/95
MTR WHITE PHAS	00071	Vac	0.000000	0.00000	6.000000	***		/ /
ESP-DEV RCO/BO	00059	ESP	2.598040	2.90887	2.500000		12	10/03/95
MTR-NDEV RCO/P	00067	MM/S	3.289130	3.00424	5.000000		-9	10/03/95
MTR-NDEH RCO/P	00066	MM/S	2.987910	3.17514	5.000000		6	10/03/95
PLANT	: RUACANA CLEARWATER (RCO)							
TRAIN	: RCO/BOOSTER/CLEA							
MACHINE	: WA 326C							
PMP-NDEV RCO/B	00078	MM/S	3.883480	3.19956	2.500000		-18	10/03/95
PMP-DEH RCO/BO	00072	MM/S	7.294680	11.2762	5.000000		55	10/03/95
PMP-DEV RCO/BO	00074	MM/S	4.608140	9.29784	5.000000		102	10/03/95
ESP-DEH RCO/BO	00073	ESP	0.000000	12.5760	1.300000	***		10/03/95
PMP-NDEH RCO/B	00076	MM/S	7.172610	5.77219	3.000000		-20	10/03/95
ESP-DEV RCO/BO	00075	ESP	0.000000	11.3461	1.300000	***		10/03/95
ESP-NDEH RCO/B	00077	ESP	0.000000	1.10483	1.300000	***		10/03/95
ESP-NDEV RCO/B	00079	ESP	0.000000	2.98135	1.300000	***		10/03/95
PMP-NDEA RCO/B	00080	MM/S	4.551220	4.59181	2.500000		1	10/03/95
PLANT	: RUACANA CLEARWATER (RCO)							
TRAIN	: RCO/BOOSTER/CLEA							
MACHINE	: WA 683C							
PMP-NDEH RCO/B	00028	MM/S	6.236450	19.8530	5.000000		218	05/08/94
PMP-NDEV RCO/B	00029	MM/S	10.022200	13.8897	5.000000		39	05/08/94
ESP-DEV RCO/BO	00042	ESP	0.000000	3.43958	2.500000	***		05/08/94
ESP-DEH RCO/BO	00041	ESP	0.000000	0.00000	2.500000	***		/ /
ESP-NDEH RCO/B	00044	ESP	0.000000	23.3527	2.500000	***		05/08/94
PMP-DEH RCO/BO	00026	MM/S	6.586650	3.19147	5.000000		-52	05/08/94
PMP-DEV RCO/BO	00027	MM/S	10.877100	12.0006	5.000000		10	05/08/94
ESP-NDEV RCO/B	00043	ESP	0.000000	19.1381	2.500000	***		05/08/94
PMP-NDEA RCO/B	00030	MM/S	113.628000	4.35575	3.000000		-96	06/07/94
PLANT	: RUACANA CLEARWATER (RCO)							
TRAIN	: RCO/BOOSTER/CLEA							
MACHINE	: WA 97M							
MTR-DEA RCO/BO	00025	MM/S	7.669300	7.17261	5.000000		-6	05/08/94
MTR WHITE PHAS	00031	Vac	0.000000	0.23037	6.000000	***		06/07/94
MTR-DEH RCO/BO	00023	MM/S	3.761400	4.61622	5.000000		23	05/08/94
MTR-DEV RCO/BO	00024	MM/S	4.099240	3.83465	5.000000		-6	05/08/94
MTR-NDEH RCO/B	00021	MM/S	2.157500	3.10357	5.000000		44	05/08/94
MTR-NDEV RCO/B	00022	MM/S	0.000000	0.00000	5.000000	***		/ /
ESP-DEH RCO/BO	00040	ESP	2.302920	3.57341	2.500000		55	05/08/94
ESP-NDEH RCO/B	00037	ESP	1.235140	1.53038	2.500000		24	05/08/94
ESP-NDEV RCO/B	00038	ESP	0.000000	0.00000	2.500000	***		/ /

Table 4-3: Peak Spectrum Value Report

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17/07/95

M A X I M U M   A M P L I T U D E   I N   S P E C T R U M

POINT	POINT ID	UNITS	PEAK	FREQ CPM	DATE	MACHINE
PLANT	: RUACANA CLEARWATER (RCO)					
00001	MTR-NDEH RCO/PU1	MM/S	1.32564	6000.00	10/03/95	WA 248M
00002	MTR-NDEV RCO/PU1	MM/S	1.13513	6000.00	10/03/95	WA 248M
00003	MTR-DEH RCO/PU1	MM/S	1.75984	6000.00	10/03/95	WA 248M
00004	MTR-DEV RCO/PU1	MM/S	1.16992	6000.00	10/03/95	WA 248M
00005	MTR-DEA RCO/PU1	MM/S	1.00024	6000.00	10/03/95	WA 248M
00006	PMP-DEH RCO/PU1	MM/S	6.66104	3000.00	30/05/95	WA 551C
00007	PMP-DEV RCO/PU1	MM/S	5.70848	2962.50	30/05/95	WA 551C
00008	PMP-NDEH RCO/PU1	MM/S	2.56904	5925.00	30/05/95	WA 551C
00009	PMP-NDEV RCO/PU1	MM/S	2.27500	3000.00	30/05/95	WA 551C
00010	PMP-NDEA RCO/PU1	MM/S	1.40537	2962.50	30/05/95	WA 551C
00011	MTR-NDEH RCO/PU2	MM/S	3.51039	3000.00	17/01/95	WA 625M
00012	MTR-NDEV RCO/PU2	MM/S	3.13081	3000.00	17/01/95	WA 625M
00013	MTR-DEH RCO/PU2	MM/S	5.06874	3000.00	17/01/95	WA 625M
00014	MTR-DEV RCO/PU2	MM/S	6.11018	3000.00	17/01/95	WA 625M
00015	MTR-DEA RCO/PU2	MM/S	11.4299	3000.00	17/01/95	WA 625M
00016	PMP-DEH RCO/PU2	MM/S	2.31803	5925.00	10/03/95	WA 272C
00017	PMP-DEV RCO/PU2	MM/S	1.25156	5925.00	10/03/95	WA 272C
00018	PMP-NDEH RCO/PU2	MM/S	1.47938	8925.00	10/03/95	WA 272C
00019	PMP-NDEV RCO/PU2	MM/S	0.30000	6000.00	10/03/95	WA 272C
00020	PMP-NDEA RCO/PU2	MM/S	0.57222	5925.00	10/03/95	WA 272C
00021	MTR-NDEH RCO/BO	MM/S	2.83272	3000.00	05/08/94	WA 97M
00022	MTR-NDEV RCO/BO	No Spect	No Spec	No Spect	No Spect	WA 97M
00023	MTR-DEH RCO/BO	MM/S	4.34476	3000.00	05/08/94	WA 97M
00024	MTR-DEV RCO/BO	MM/S	3.18811	3000.00	05/08/94	WA 97M
00025	MTR-DEA RCO/BO	MM/S	6.80066	3000.00	05/08/94	WA 97M
00026	PMP-DEH RCO/BO	MM/S	1.90038	6000.00	05/08/94	WA 683C
00027	PMP-DEV RCO/BO	MM/S	10.8344	3000.00	05/08/94	WA 683C
00028	PMP-NDEH RCO/BO	MM/S	15.0120	450.00	05/08/94	WA 683C
00029	PMP-NDEV RCO/BO	MM/S	8.96176	3000.00	05/08/94	WA 683C
00030	PMP-NDEA RCO/BO	MM/S	47.4868	112.50	02/02/94	WA 683C
00031	MTR WHITE PHASE	Vac	0.22983	3000.00	06/07/94	WA 97M
00032	MTR WHITE PHASE	Vac	0.25280	3003.75	11/10/94	WA 248M
00033	ESP-NDEH RCO/PU1	ESP	0.46236	6000.00	10/03/95	WA 248M
00034	ESP-NDEV RCO/PU1	ESP	0.13224	12000.00	10/03/95	WA 248M
00035	ESP-DEV RCO/PU1	ESP	1.00741	6000.00	10/03/95	WA 248M
00036	ESP-DEH RCO/PU1	ESP	0.67248	6000.00	10/03/95	WA 248M
00037	ESP-NDEH RCO/BO	DEG-F	0.25617	35625.00	05/08/94	WA 97M
00038	ESP-NDEV RCO/BO	No Spect	No Spec	No Spect	No Spect	WA 97M
00039	ESP-DEV RCO/BO	DEG-F	0.51424	375.00	05/08/94	WA 97M
00040	ESP-DEH RCO/BO	DEG-F	0.68606	6000.00	05/08/94	WA 97M
00041	ESP-DEH RCO/BO	No Spect	No Spec	No Spect	No Spect	WA 683C
00042	ESP-DEV RCO/BO	DEG-F	0.52293	5925.00	05/08/94	WA 683C
00043	ESP-NDEV RCO/BO	DEG-F	2.42622	2962.50	05/08/94	WA 683C
00044	ESP-NDEH RCO/BO	DEG-F	4.04065	12375.00	05/08/94	WA 683C
00045	ELECTRICAL DEH	MM/S	2.43156	2977.50	06/07/94	WA 97M
00046	ROTOR BAR NDEH	MM/S	1.68080	3000.00	06/07/94	WA 97M
00047	ELECTRICAL DEH	MM/S	4.95467	2962.50	17/01/95	WA 625M
00048	ROTOR BAR NDEH	MM/S	3.64696	3000.00	17/01/95	WA 625M
00049	ELECTRICAL DEH	MM/S	1.55148	6000.00	10/03/95	WA 248M
00050	ROTOR BAR NDEH	MM/S	1.27330	6000.00	10/03/95	WA 248M
00051	ESP-NDEH RCO/PU1	DEG-F	0.20154	28312.50	17/01/95	WA 625M
00052	ESP-NDEV RCO/PU1	DEG-F	0.16764	2962.50	17/01/95	WA 625M
00053	ESP-DEV RCO/PU1	DEG-F	0.13596	20475.00	17/01/95	WA 625M
00054	ESP-DEH RCO/PU1	DEG-F	0.66872	6000.00	17/01/95	WA 625M
00055	ELECTRICAL DEH	MM/S	1.38019	6000.00	10/03/95	WA 282M
00056	ROTOR BAR NDEH	MM/S	3.04528	6000.00	10/03/95	WA 282M

4.6.4.4 Band Exception, Peak

Report of all points in the plot list with at least one frequency band whose peak value exceeds Alarm 1 (Table 4-4). The following is reported:

a) Plant, Train and Machine.
b) Point ID and point number.
c) Spectrum measurement units.
d) Frequency bands alarm file name.
e) Exception status for each frequency band peak value exceeding the first Alarm.

Table 4-4: Band Exception, Peak

```
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17/07/95
           B A N D   E X C E P T I O N   R E P O R T   (P E A K)

POINT ID           POINT UNITS    BAND NAME                        1 2 3 4 5 6

PLANT           : RUACANA CLEARWATER  (RCO)
TRAIN           : RCO/BOOSTER/CLEA
MACHINE         : WA 326C
PMP-DEV RCO/BO     00074 MM/S     Pump 3000RPM ROLL                - - Y - - -
PMP-DEH RCO/BO     00072 MM/S     Pump 3000RPM ROLL                - Y - - - -

PLANT           : RUACANA CLEARWATER  (RCO)
TRAIN           : RCO/BOOSTER/CLEA
MACHINE         : WA 683C
PMP-NDEV RCO/BO    00029 MM/S     Pump 3000RPM ROLL                Y - - - - Y
PMP-DEV RCO/BO     00027 MM/S     Pump 3000RPM ROLL                Y - - - - -
PMP-NDEH RCO/BO    00028 MM/S     Pump 3000RPM ROLL                - - - - - Y
PMP-NDEA RCO/BO    00030 MM/S     Pump 3000RPM ROLL                Y Y Y - - -

PLANT           : RUACANA CLEARWATER  (RCO)
TRAIN           : RCO/PURIFICATION
MACHINE         : WA 1059M
MTR-DEH RCO/PU     00087 MM/S     Motor 3000RPM ROLL               - Y - - - -
MTR-NDEH RCO/PU1   00082 MM/S     Motor 3000RPM ROLL               - Y - - - -
ELECTRICAL DEH     00089 MM/S     Motor ELECTRICAL 3000 RPM        - - Y - - -
MTR-DEA RCO/PU     00092 MM/S     Motor 3000RPM ROLL               - Y - - - -

PLANT           : RUACANA CLEARWATER  (RCO)
TRAIN           : RCO/PURIFICATION
MACHINE         : WA 271C
PMP-DEH RCO/PU1    00061 MM/S     Pump 3000RPM SLEEVE (7)          - - Y - - -
PMP-DEV RCO/PU1    00062 MM/S     Pump 3000RPM SLEEVE (7)          - - Y Y - -

PLANT           : RUACANA CLEARWATER  (RCO)
TRAIN           : RCO/PURIFICATION
MACHINE         : WA 551C
PMP-DEH RCO/PU1    00006 MM/S     Pump 3000RPM SLEEVE (7)          - - Y - - -

PLANT           : RUACANA CLEARWATER  (RCO)
TRAIN           : RCO/PURIFICATION
MACHINE         : WA 625M
MTR-DEA RCO/PU2    00015 MM/S     Motor 3000RPM ROLL               Y - - - - -
MTR-DEH RCO/PU2    00013 MM/S     Motor 3000RPM ROLL               - Y - - - -
ELECTRICAL DEH     00047 MM/S     Motor ELECTRICAL 3000 RPM        - - Y - - -
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CHAPTER 5

EXPERIMENTS ON ELECTRIC MOTORS AND PUMPS

5.1 General

In this chapter the vibration measured on equipment with the vibration characteristics associated with the different types of problems detected/analysed are discussed. Vibration amplitude versus frequency signatures and other methods of data presentation are included.

After the vibration data has been obtained, the characteristics of vibration typical of various types of faults are compared. Since the problems on the machines have similar characteristics and several problems may be present in a machine simultaneously, it was often necessary to choose between several likely possibilities.

5.2 Overall Vibration Measurements

Tables 5-1 & 5-2 list the average overall vibration of the motors and pumps with the quantity in each kilowatt range that was measured over a period of 3 years. Each range was given a letter for reference purposes. As discussed in Section 4, this group of machines was chosen due to their importance in water supply. The reason for the classification is to see if there is a correlation between the size and speed of the machine.

The histograms in figures 5-1 & 5-2 display the overall vibration measured on the different sizes of electrical motors and pumps. It shows the low and high average of vibration on each size. The high average vibration on each histogram indicates mostly where vibration was found before corrective actions were taken or where deterioration had already taken place.

The overall vibration levels measured were compared with existing machine vibration standards in order to check the acceptability of these vibrations and hence the machine condition. The modified Rathbone Chart and ISO standards were used (Appendix D & E). The purpose of these standards is to evaluate the vibration of machines with respect to reliability, safety and human perception and to provide an initial basis for condition assessment. This base must be modified as soon as experience is gained. The standards are not intended to evaluate the vibration of machines with respect to smooth running. According to ISO standards the vibration data in Table 5-1 & 5-2 will be listed as Class II. These standards indicate that vibration between 1.5 - 2.5 mm/s (when converted from RMS to Peak) is satisfactory. Vibration between 4 and 6.3 mm/s is no longer satisfactory. An overall vibration of 7.5 mm/s is considered as the dangerous operating level.

The end result was that when the overall vibration showed an increase, some problem was developing and therefore fault diagnosis was necessary to pinpoint the problem. Specific fault diagnosis case histories are discussed now.

Table 5-1: Average Vibration on Electrical Motors

Group	kW	Speed (rpm)	Qty	*H/V	Average
A1	15	970	5	H	1.34
A2	15	2980	3	H	3.6
B	45	1470	5	H	1.6
C1	55	980	1	H	1.83
C2	55	1480	5	H	1.4
D	75	2990	2	H	4.7
E	90	1476	2	H	2.2
F	100	2990	2	H	7.8
G	110	1479	3	H	2.55
H	160	1485	6	H	2.43
I	200	1483	6	H	2.4
J	250	1483	3	H	1.45
K	275	2990	3	H	4.4
L	315	1490	13	H	2.1
M	375	1488	3	H	1.76
N	560	1480	4	V	2.3
O	650	484	2	V	3.5
P	1900	987	6	V	1.6
Q	2400	985	4	H	1.34

*H/V = Horizontal / V = Vertical

Table 5-2: Average Vibration on Centrifugal Pumps

Group	kW	Speed (rpm)	Qty	*H/V	Average
Single Stage					
A	15	970	5	H	2.4
B	45	1480	3	H	2.05
C	55	980	5	H	0.85
D	55	1485	2	H	2.3
E	90	1476	3	H	2.05
F	110	1479	3	H	3.4
G	250	1483	9	H	3.15
Multi-Stage					
H	45	1470	2	H	2.0
I	100	2980	3	H	4.4
J	160	1485	6	H	2.5
K	200	1450	6	H	2.6
L	270	2990	3	H	3.2
M	375	1488	16	H	2.4
N	2400	1485	4	H	1.8

*H/V = Horizontal /Vertical

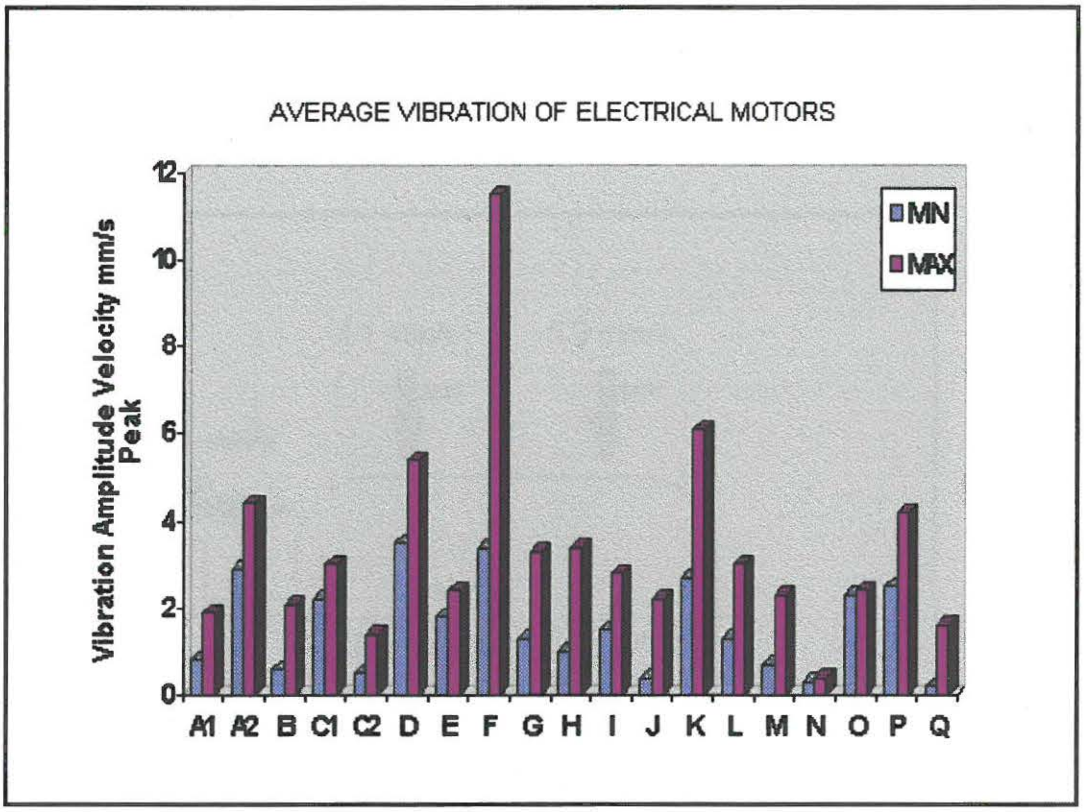


Figure 5-1: Histogram of average overall vibration on electrical motors

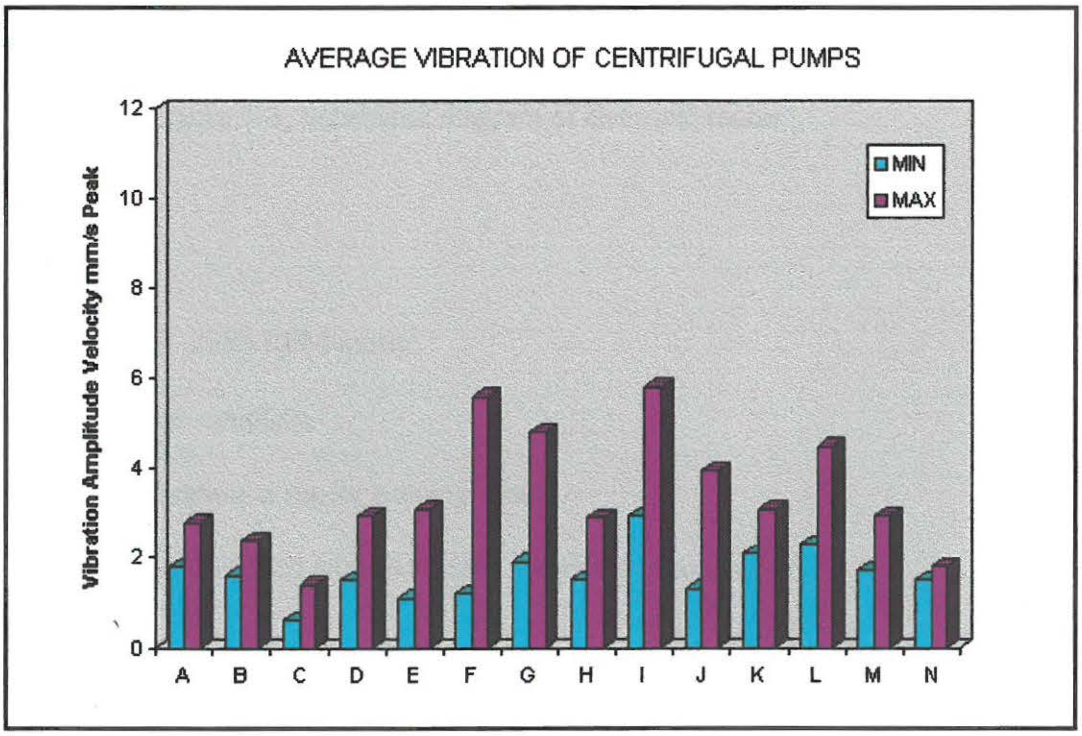


Figure 5-2: Histogram of average overall vibration on centrifugal pumps

5.3 Unbalance

5.3.1 Case 1

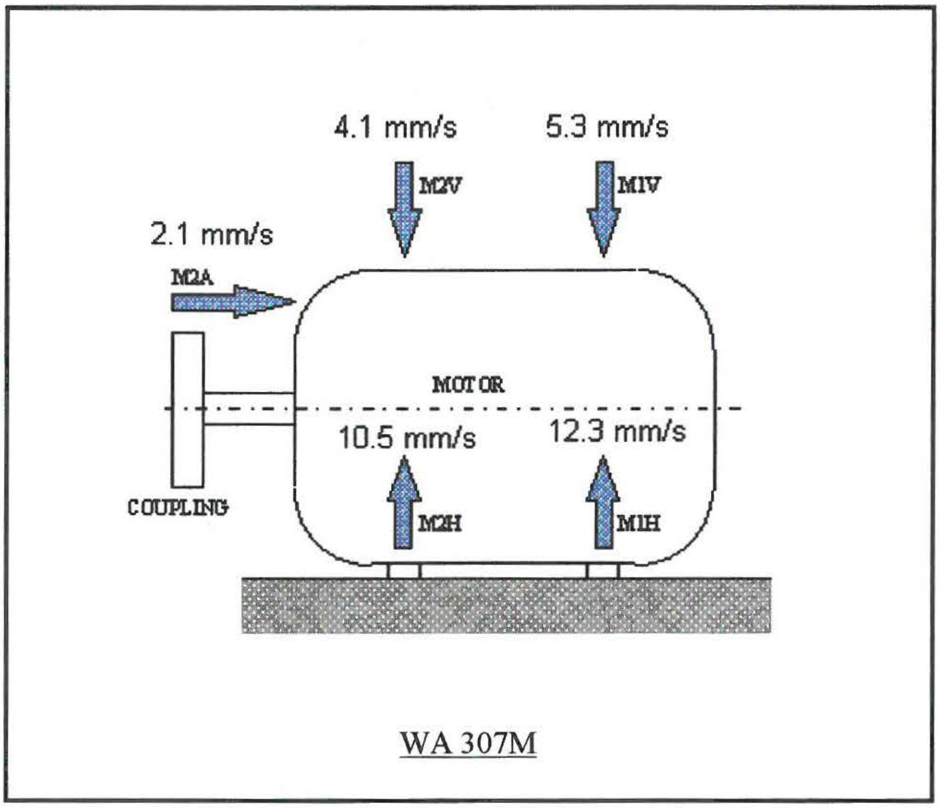


Figure 5-3: Schematic diagram of electrical motor

1. 275 kW, 2985 RPM motor.
2. Gear-type coupling.
3. High vibration at motor running speed.
4. Data was collected with the motor coupled and uncoupled.

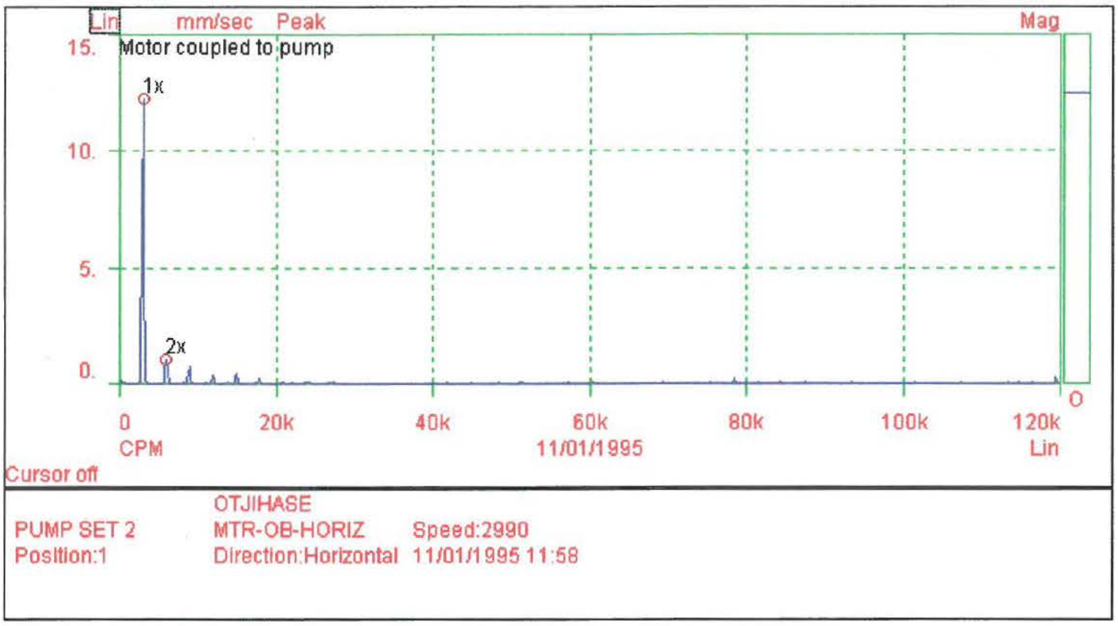


Figure 5-4: Spectrum at point M1H indicating unbalance

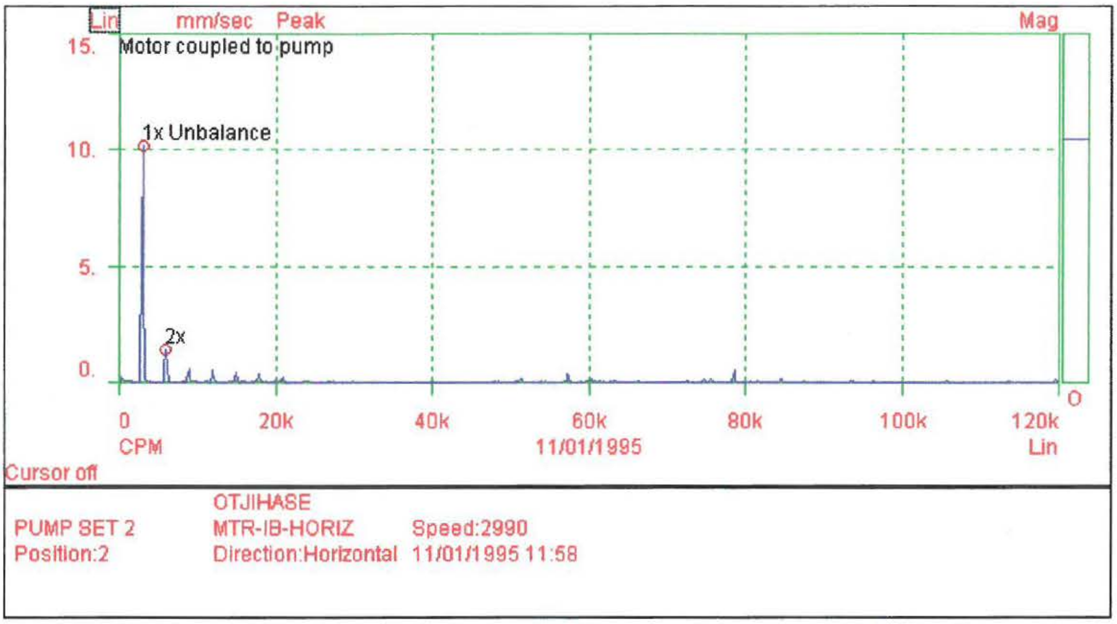


Figure 5-5: Spectrum at point M2H indicating unbalance

The spectrum for the motor outboard horizontal (M1H) and motor inboard horizontal (M2H) is shown above. The running speed is marked and shows 12.2 mm/s. It appears that the highest amplitudes are in the horizontal directions. Small amplitudes of vibration are present at 2, 3 and 4x RPM.

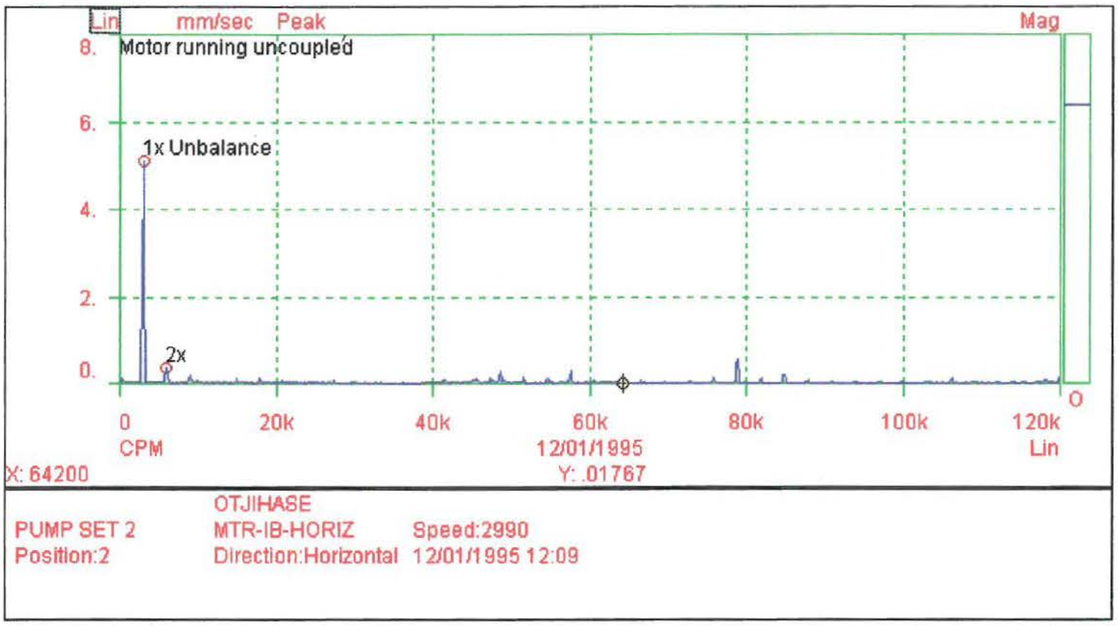


Figure 5-6: Spectrum at point M2H indicating unbalance (uncoupled)

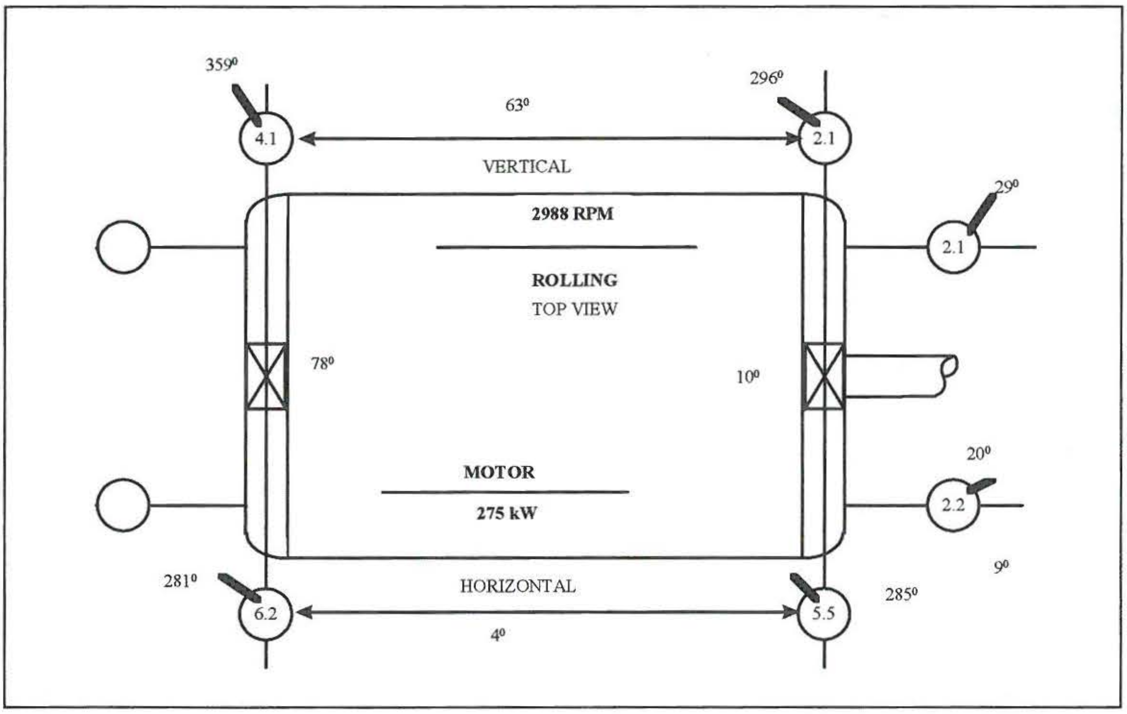


Figure 5-7: Running speed amplitude and phase data

The spectrum in figure 5-6 was taken after the motor was disconnected from the pump. The vibration decreased, but was still not acceptable for an electric motor running on no load. Phase readings in figure 5-7 revealed a shaking mode of 63° vertically and 4° horizontally. It is not a perfect unbalance situation, but it is suspected that some other source like mechanical looseness caused a shift in phase. If the phase is compared at the same bearing, 78° (left) and 10° (right), an unbalance is suspected more on the outboard bearing (M1H).

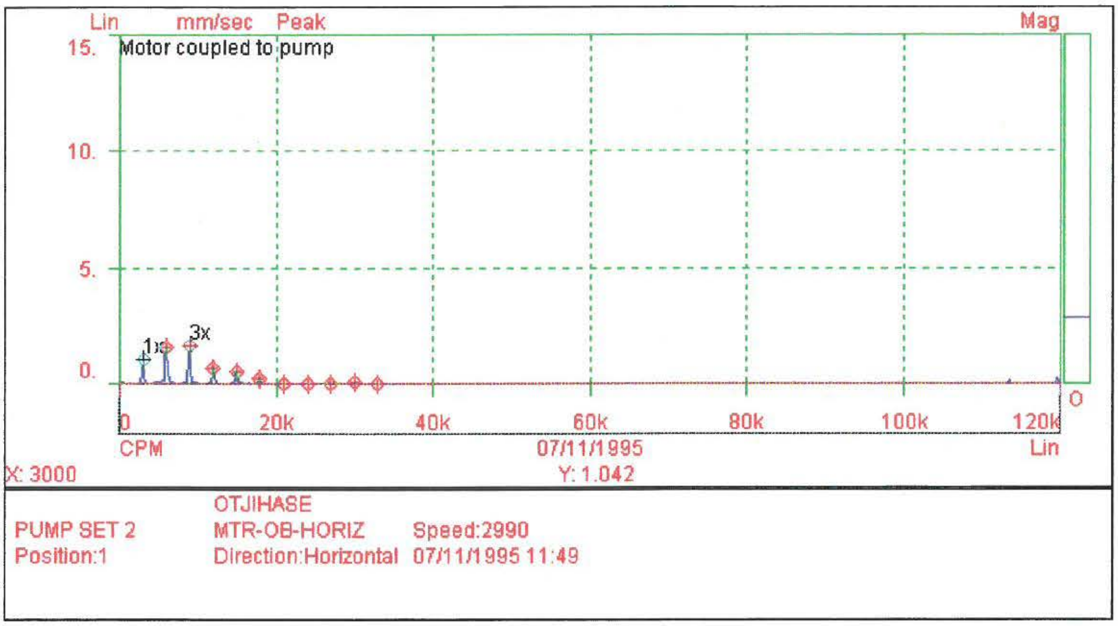


Figure 5-8: Spectrum at point M1H after rotor was balanced

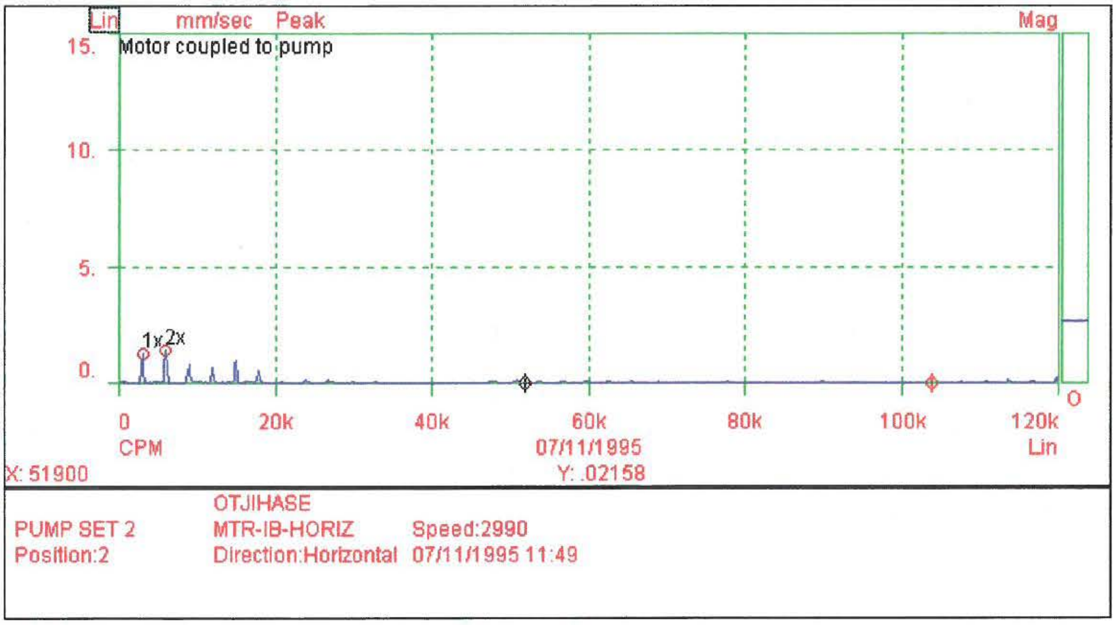
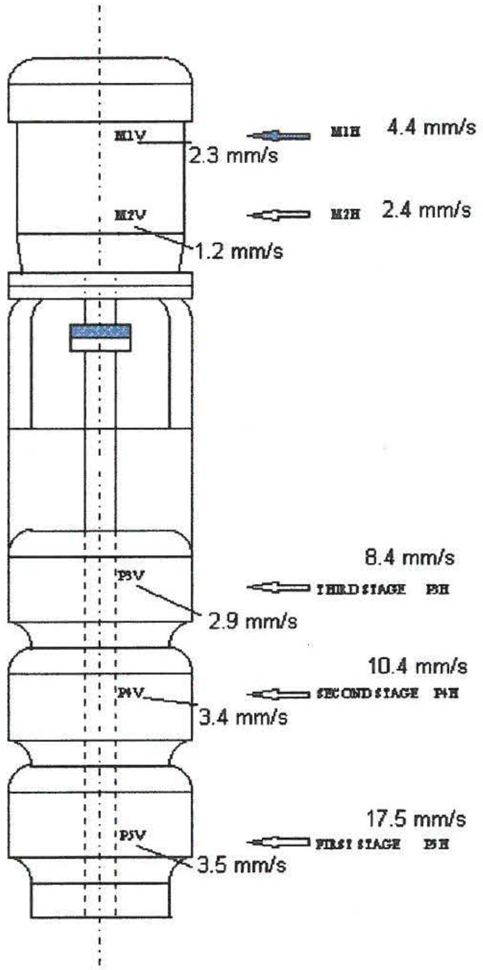


Figure 5-9: Spectrum at point M2H after rotor was balanced

The above spectrums are for the motor outboard horizontal (M1H) and motor inboard horizontal (M2H) after the rotor was balanced. The running speed amplitude decreased from 12.2 to 1.04 mm/s.

5.3.2 Case 2



WA 68C

Figure 5-10: Schematic diagram of vertically mounted pump

1. 1900 kW, 980 RPM vertically mounted pump.
2. Gear-type coupling.
3. "Cutlass" type bearings.
4. High vibration at running speed.
5. Small amplitudes at 2x rpm and higher harmonics.
6. Highest vibration measured on first stage of pump.

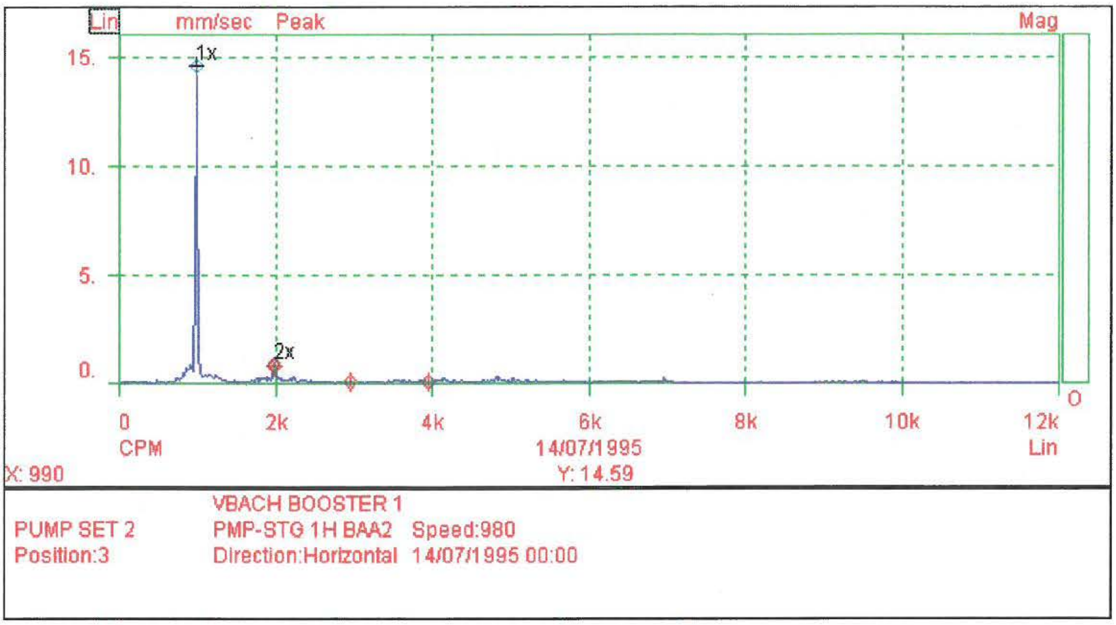


Figure 5-11: Spectrum at point P5H indicating unbalance

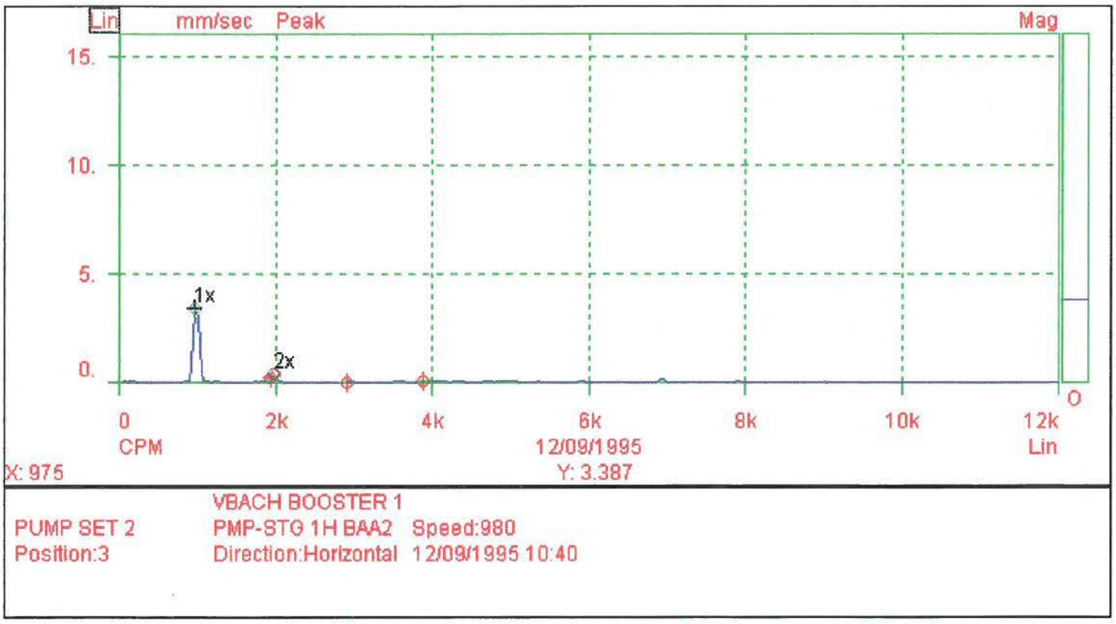


Figure 5-12: Spectrum at point P5H after impellers was balanced

The spectrum for the pump first stage (P5H) is shown in figure 5-11. The high 1x rpm of 14.5 mm/s indicates a possible unbalance problem. The spectrum in figure 5-12 was taken after the impellers were balanced. Vibration decreased from 14.5 to 3.3 mm/s.

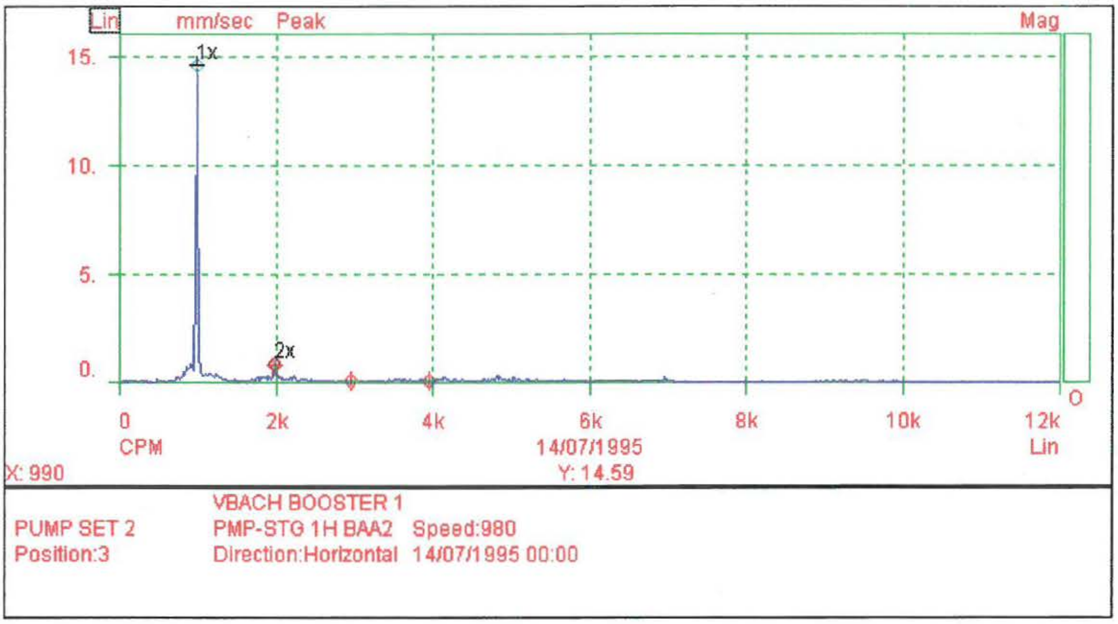


Figure 5-11: Spectrum at point P5H indicating unbalance

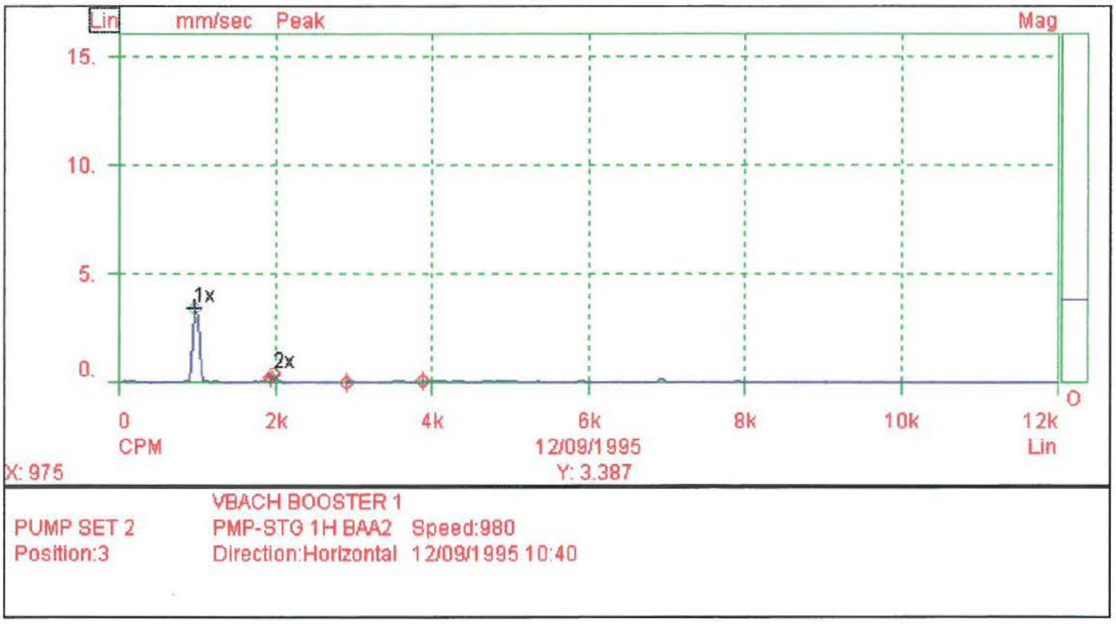


Figure 5-12: Spectrum at point P5H after impellers was balanced

The spectrum for the pump first stage (P5H) is shown in figure 5-11. The high 1x rpm of 14.5 mm/s indicates a possible unbalance problem. The spectrum in figure 5-12 was taken after the impellers were balanced. Vibration decreased from 14.5 to 3.3 mm/s.

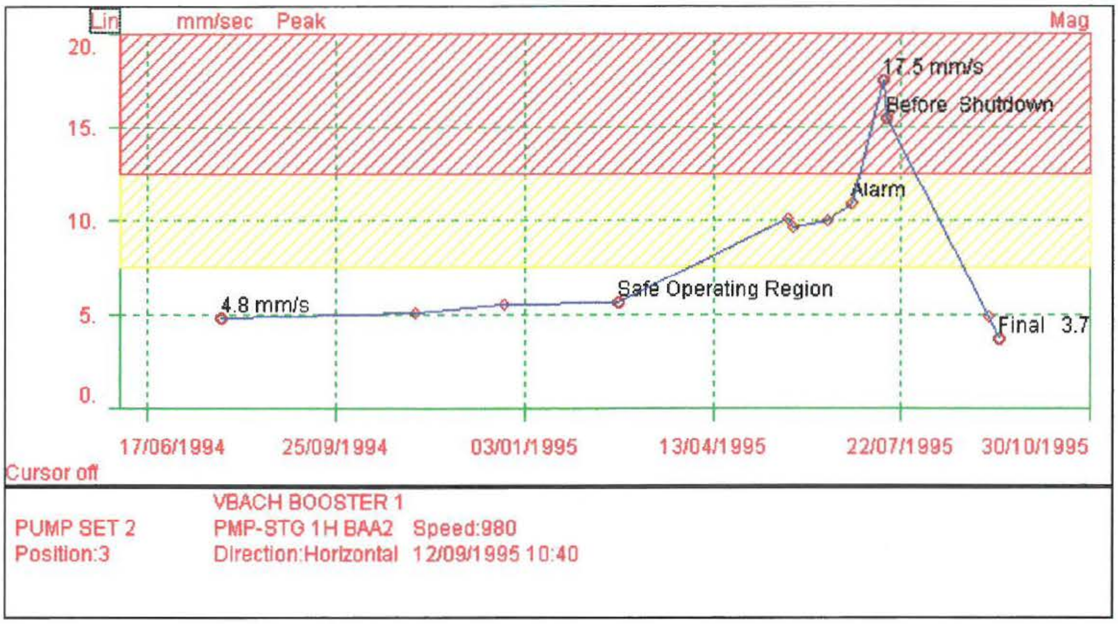


Figure 5-13: Overall vibration plot at point P5H after rotor was balanced

This overall vibration plot shows measurements taken on the pump (P5H) over a period of 14 months. The overall vibration increased from 4.8 to 17.5 mm/s over a period of 12 months. After corrective action was taken, the vibration decreased to 3.7 mm/s.

5.3.3 Discussion

When unbalance was the primary problem, a 1x rpm had appeared in the spectral data. The harmonics of 1x rpm was normally low in amplitude with low axial vibration levels. Case 1 was such an example except for the higher than normal axial vibration of 2 mm/s. This was caused (after opening) by a misaligned bearing. Phase data had also revealed an unbalance. After the rotor was balanced, the vibration at 1x rpm decreased from 10.2 to 1.3 mm/s. Case 2 is another example where unbalance was detected on a vertical pump. Vibration at 1x rpm decreased from 14.5 to 3.3 mm/s after the impellers were balanced.

5.4 Misalignment

5.4.1 Case 3

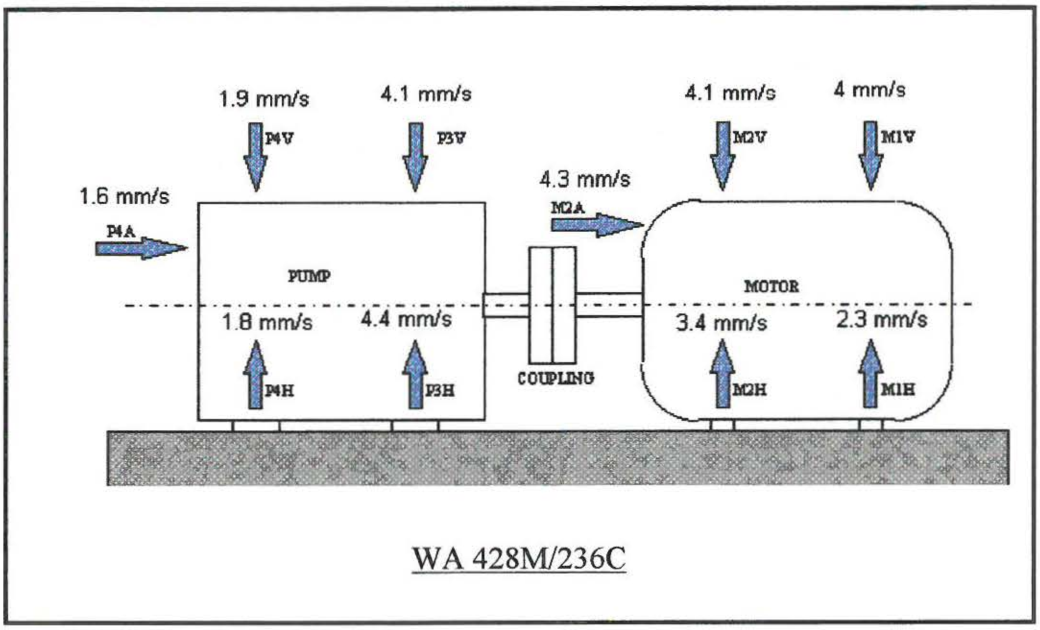


Figure 5-14: Schematic diagram of pump set

1. 200 kW, 1485 RPM
2. Gear-type coupling.
3. Newly installed pump set.
4. Vibration at 1x and 2x running speed.
5. High axial vibration.
5. This vibration was detected on both inboard bearings of motor and pump.
6. Loose baseframe bolt on foundation.
7. Excessive bearing temperatures.

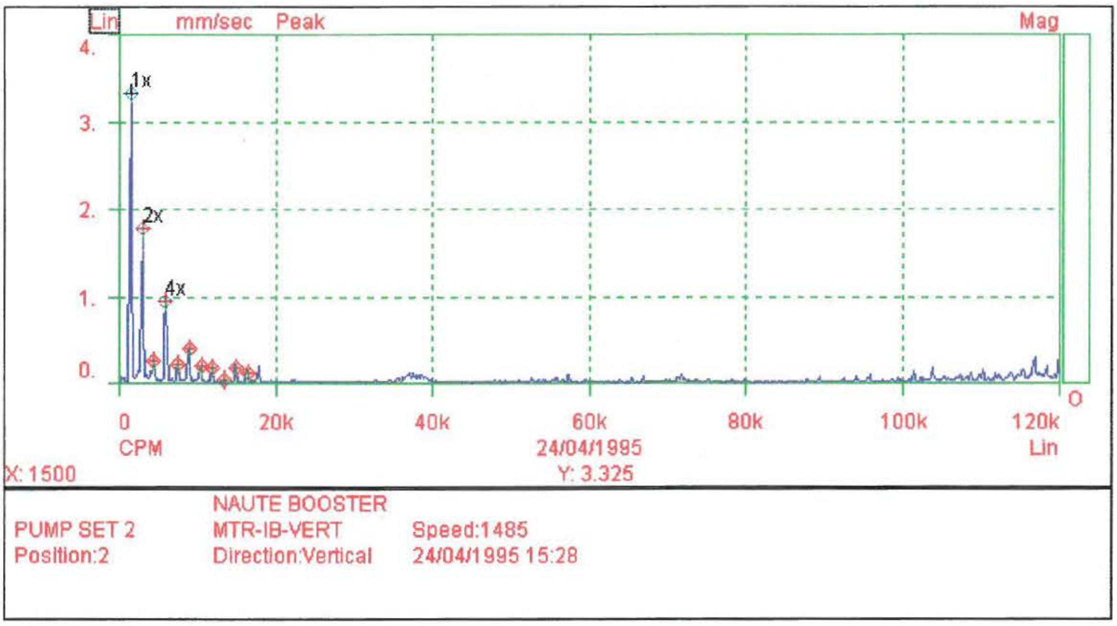


Figure 5-15: Spectrum at point M2V indicating misalignment

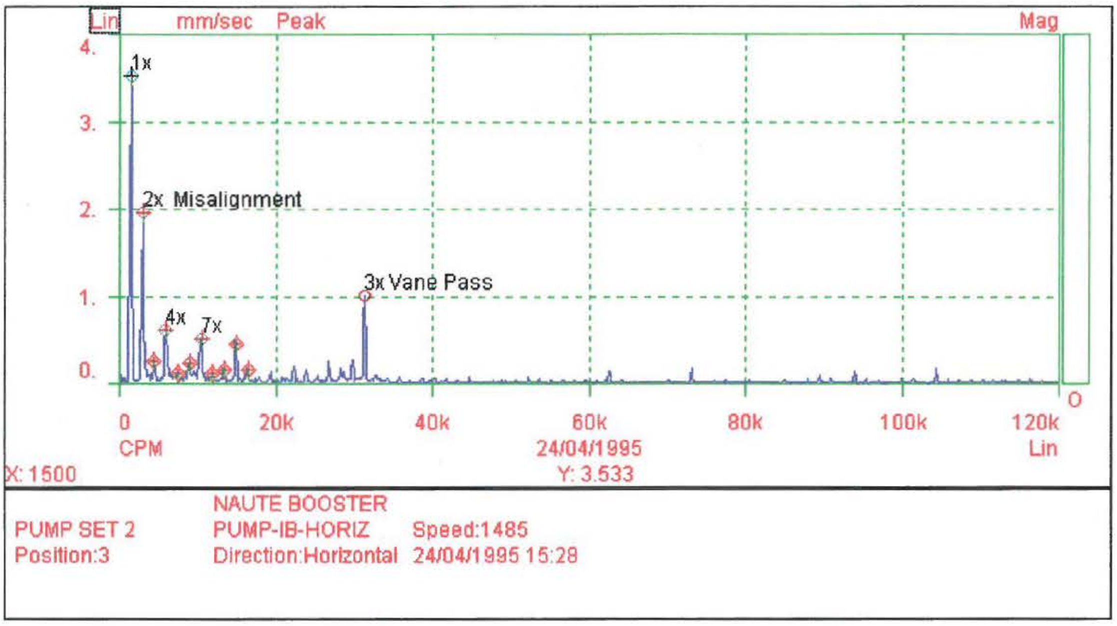


Figure 5-16: Spectrum at point P3H indicating misalignment

After the baseframe bolts on the foundation were tightened, the vibration decreased from 8 to 4 mm/s which was still rough. Figure 5-15 shows the spectrum for the motor inboard vertical (M2V) and figure 5-16 the pump inboard horizontal (P3H). Notice the higher than usual 2x rpm frequency indicating misalignment.

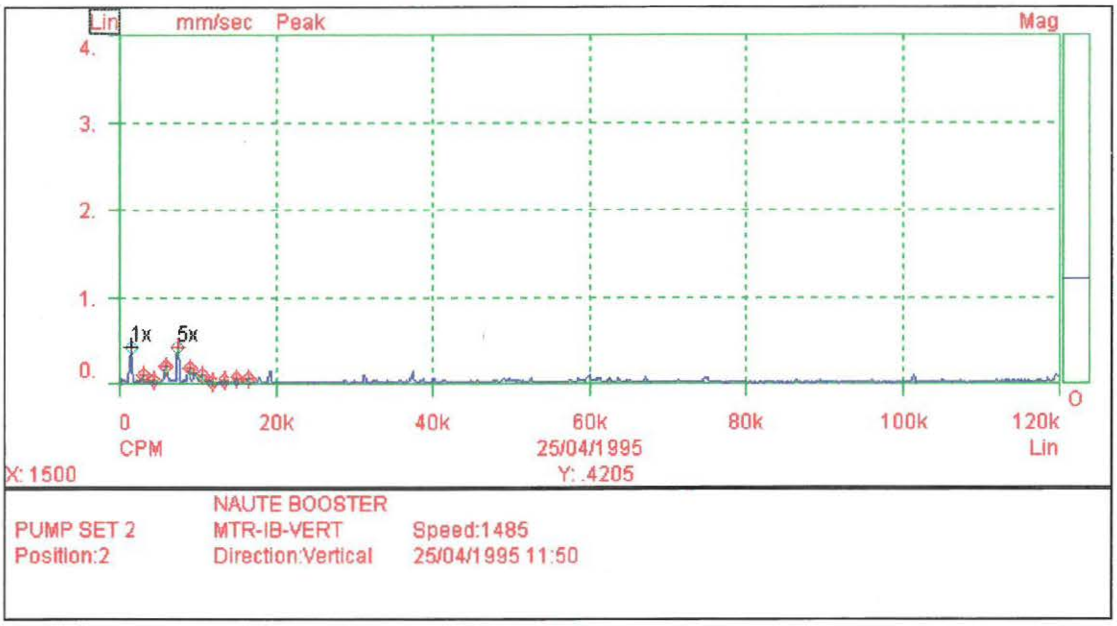


Figure 5-17: Spectrum at point M2V after re-alignment

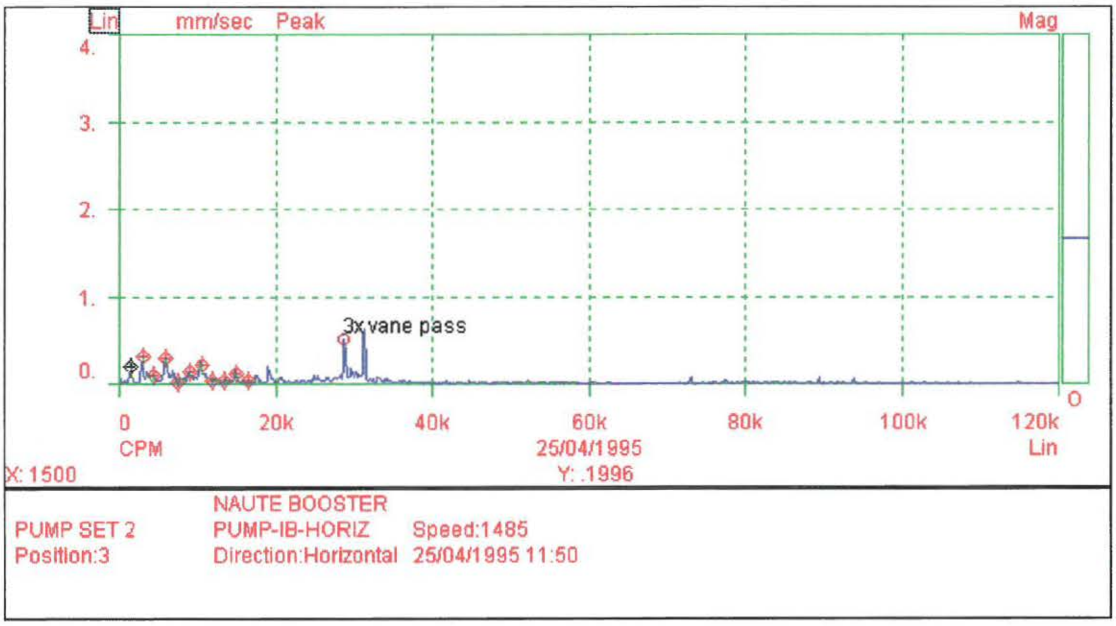


Figure 5-18: Spectrum at point P3H after re-alignment

Unfortunately, phase readings were not obtained, but misalignment was suspected. After the pump set was re-aligned, the vibration decreased to 0.4 mm/s (M2V) and 0.2 mm/s (P3H) as seen above. The root cause for the rise in bearing temperature and deterioration was caused by severe misalignment.

5.4.2 Discussion

Misalignment was detected when a predominant vibration occurred at a frequency of 1x RPM with a very substantial vibration at 2x RPM. This was normally accompanied by high axial readings. The primary cause of vibration in Case 3 was misalignment. The radial reading was 3.3 mm/s and the axial 4.3 mm/s. After re-alignment of the pump set, the vibration decreased to 0.2 and 1.3 mm/s.

5.5 Combination of Unbalance and Misalignment

5.5.1 Case 4

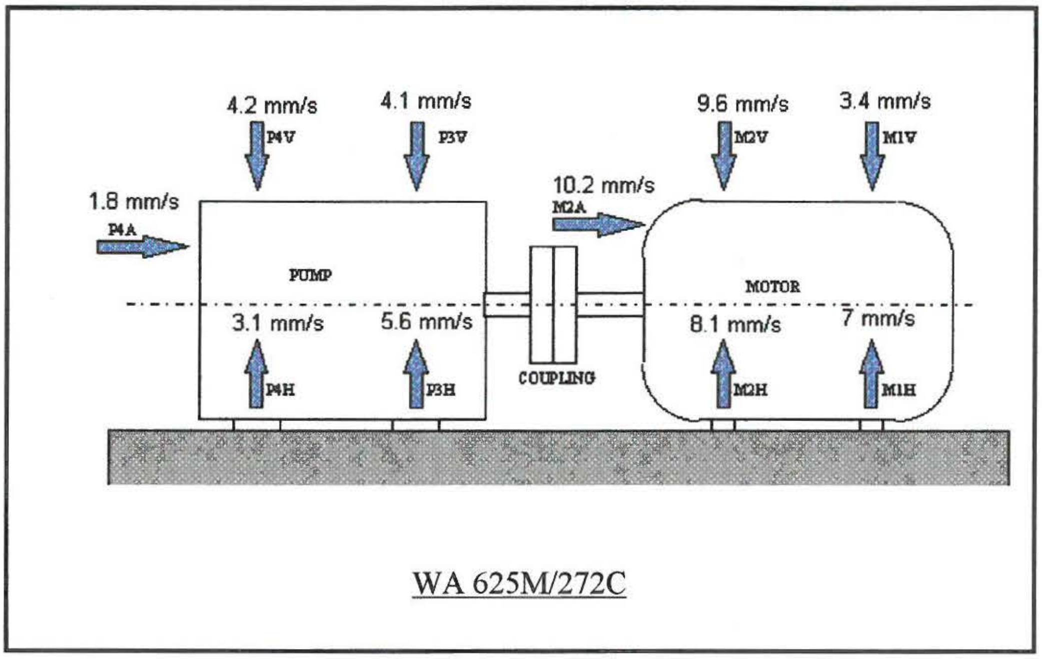


Figure 5-19: Schematic diagram of pump set

1. This is a 100 kW, two-pole motor.
2. Flexible-type coupling.
3. High vibration at motor running speed present on all measurement points.
4. Substantial vibration at 2x rpm with strong harmonics up to 6x rpm.
5. This vibration was both in the horizontal and vertical directions.
6. High axial vibration.

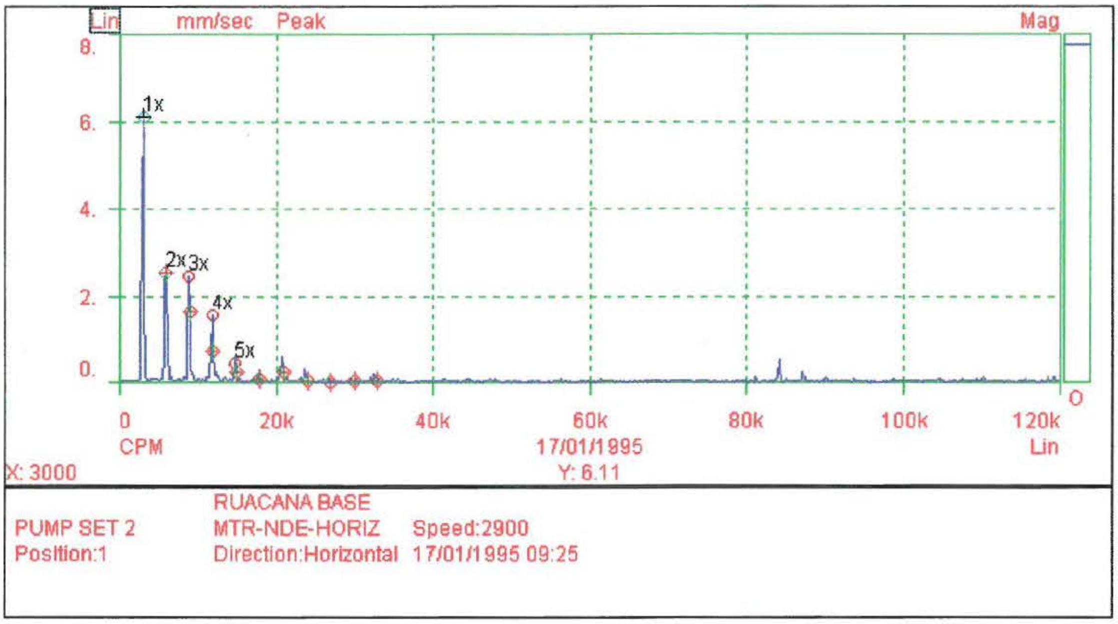


Figure 5-20: Spectrum at point M1H indicating unbalance and misalignment

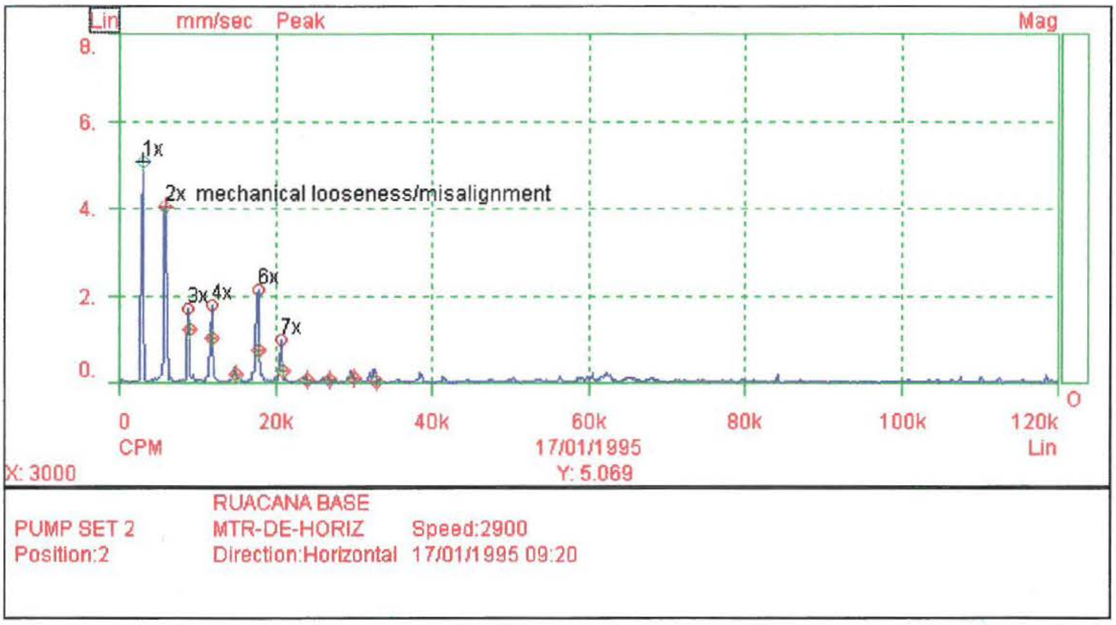


Figure 5-21: Spectrum at point M2H indicating unbalance and misalignment

The spectrum for the motor outboard horizontal (M1H) and motor inboard vertical (M2H) are shown above. Note the 1x rpm amplitude between the horizontal and vertical reading. The stronger than usual 2x rpm with harmonics indicates possible misalignment or looseness.

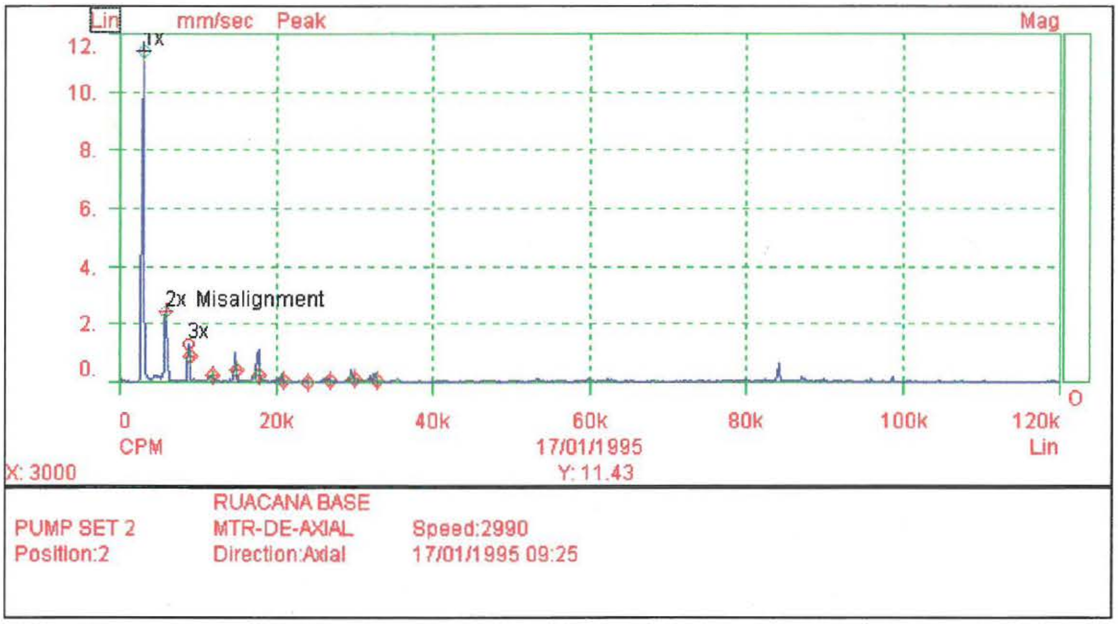


Figure 5-22: Spectrum at point M2A indicating misalignment

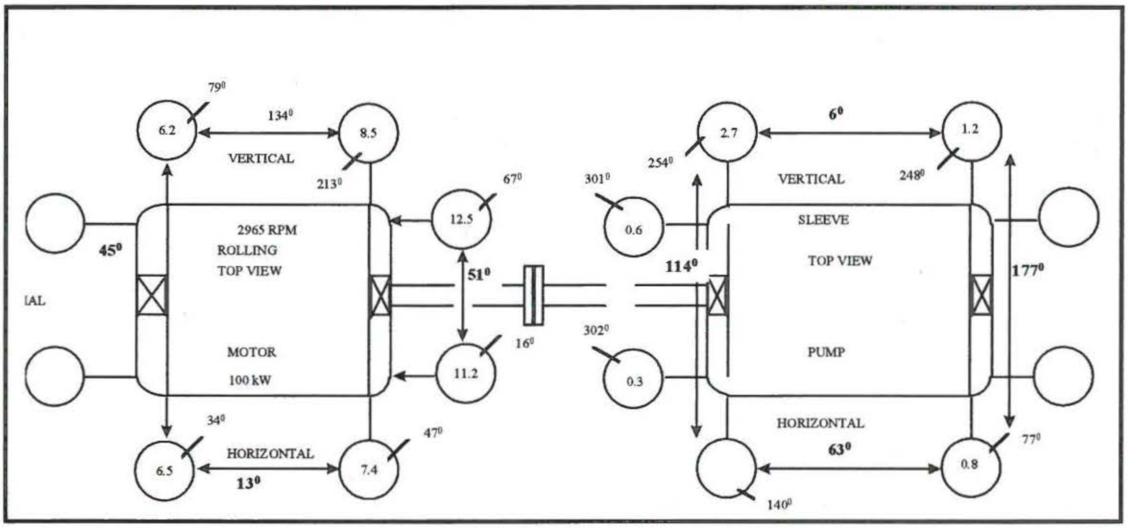


Figure 5-23: Running speed amplitude and phase data

The spectrum in figure 5-22 display the axial reading (M2A). This amplitude is more than half the amplitude of the radial readings. This is a strong symptom of misalignment. Phase data shows 134 degrees out-of-phase vertically and 13 degrees horizontally. When the horizontal and vertical phases at the same bearing are evaluated they show 45° (left) and 162° (right). This indicates stronger symptoms of misalignment than unbalance. In addition to the phase readings, amplitude readings must be compared. If the source is unbalanced, the horizontal amplitude at a bearing would be similar to the vertical amplitude. The outboard bearing of the motor in the vertical direction (M1V) shows 6.2 and the inboard vertical (M2V) is 8.5. The same applies to the amplitudes at the same bearing, namely 6.2 and 6.5 which look the same.

5.5.2 Case 5

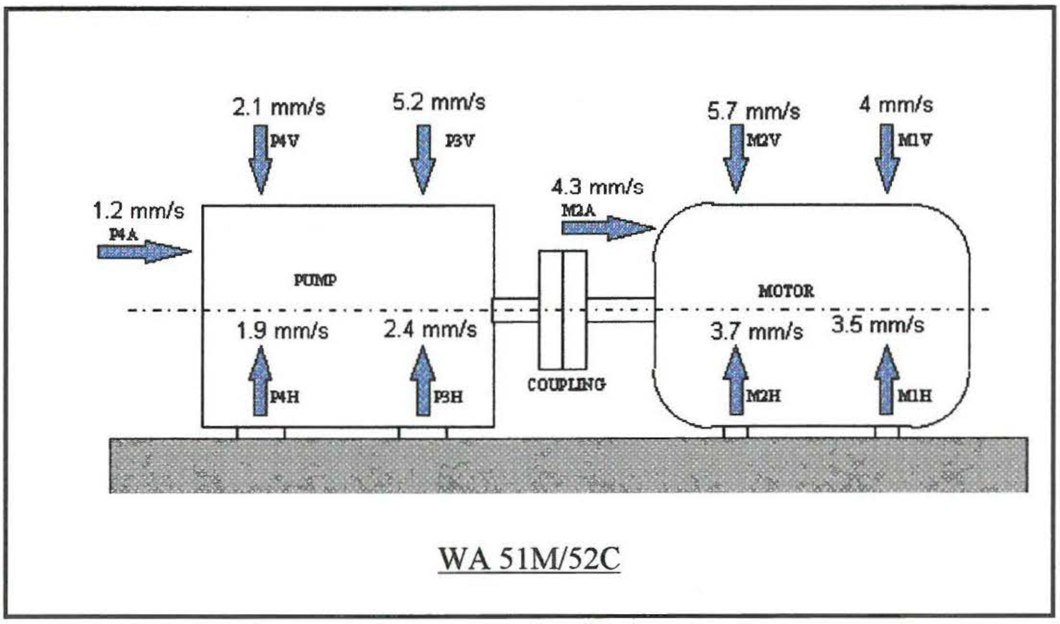


Figure 5-24: Schematic diagram of pump set

1. 275 kW, 2980 RPM motor.
2. The unit has a gear coupling.
3. High vibration amplitudes at 2x and 3x rpm.
4. High axial readings.

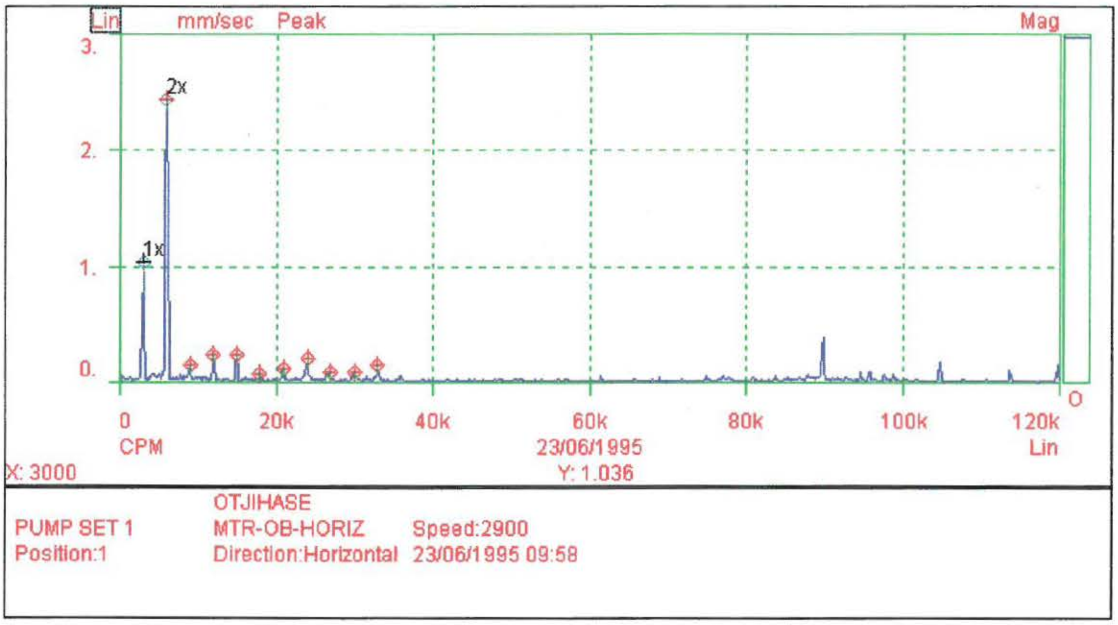


Figure 5-25: Spectrum at point M1H indicating misalignment

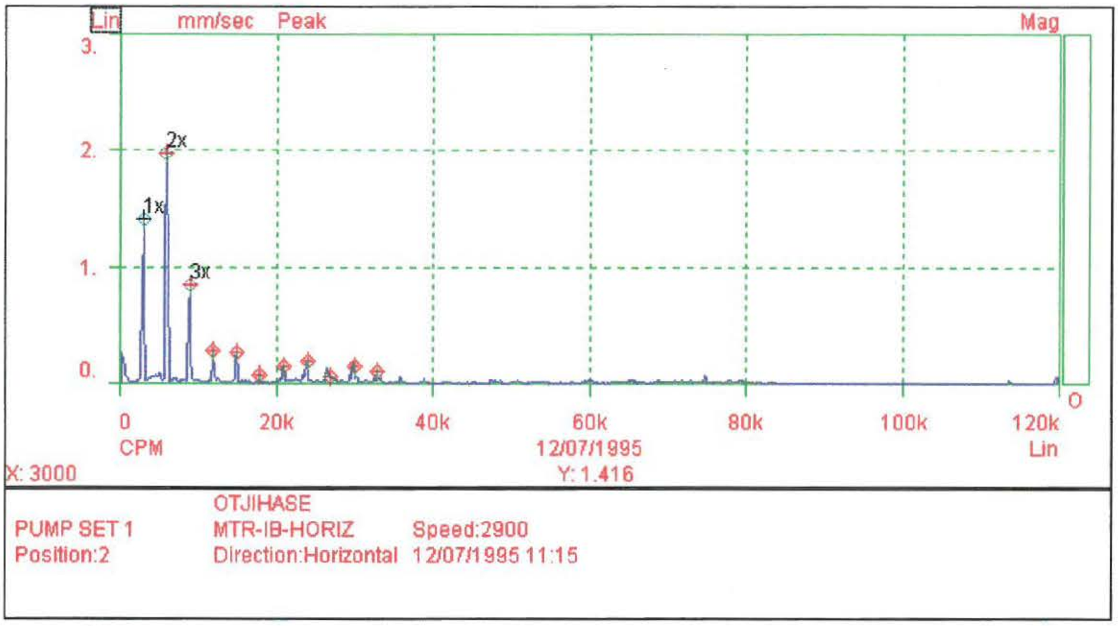


Figure 5-26: Spectrum at point M2H indicating misalignment

The spectrum for the motor outboard horizontal (M1H) and motor inboard horizontal (M2H) are shown above. The 2x and 3x rpm components exceed the running speed frequency at 1x rpm. These frequencies are an indication of misalignment or looseness.

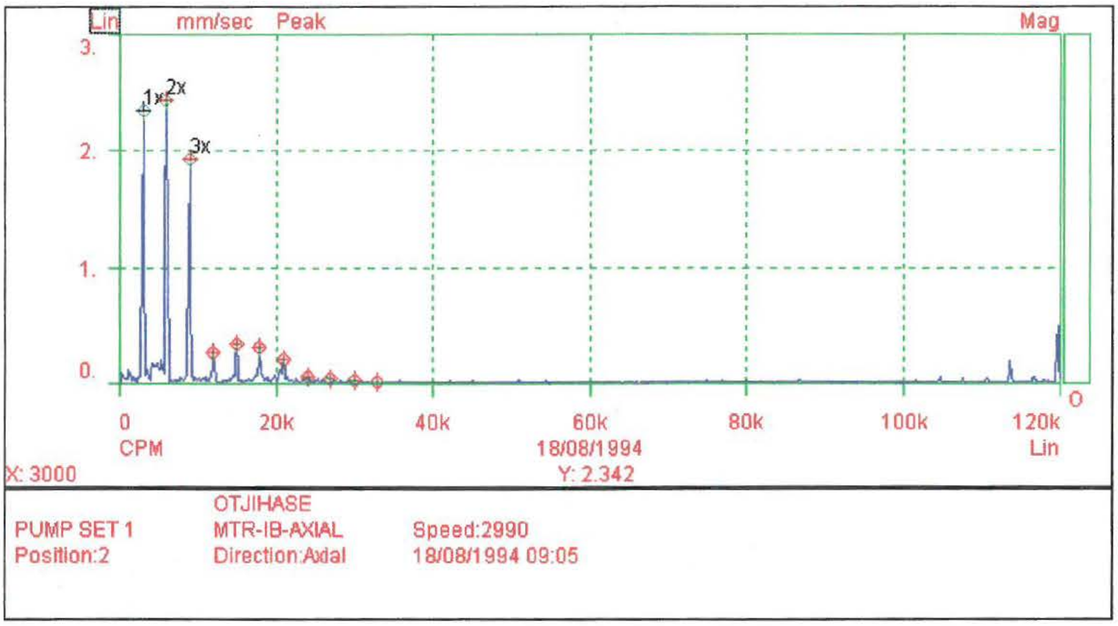


Figure 5-27: Spectrum at point M2A indicating misalignment

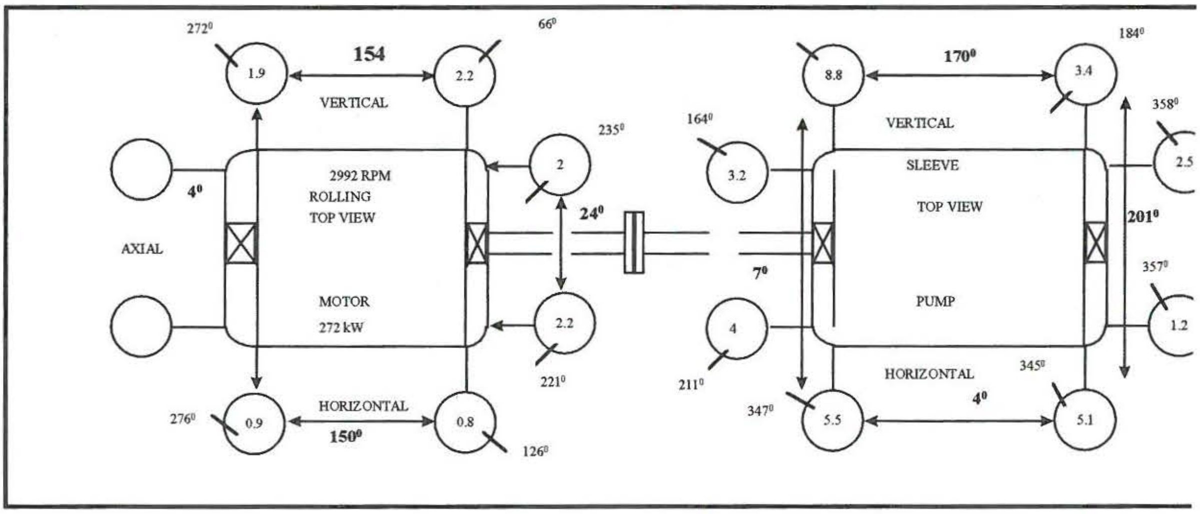


Figure 5-28: Running speed amplitude and phase data

The axial reading also shows the misalignment defect (more than half the amplitude). If phase data is reviewed on the pump it shows an out-of-phase shaking (170° vertically and 4° horizontally) mode diagnosed as misalignment, whereas the motor shows an in-phase shaking (154° and 150°) mode. When the horizontal and vertical phases at the same bearing on the pump are compared, it shows 7° (left) and 201° (right) which also makes it a strong symptom of misalignment.

5.5.3 Combination of Unbalance and Misalignment

Cases 4 & 5 were examples of where a combination of unbalance and misalignment was detected. Both spectrum and phase data indicated a combination of misalignment and unbalance.

For misalignment there is another symptom based on phase and shaking mode that can also be used for evaluation. This is to compare the shaking of one half of the coupling relative to the other half of the coupling. In Case 2 the horizontal phase readings measured at the machine's left inboard bearing is 126°, while on the other side of the same coupling, the horizontal phase is at 347°. This shows that when the motor side of the coupling moves one way in the horizontal direction, the other side of the same coupling is moving in the opposite direction. This is an indication of misalignment. For misalignment the modes must show either 0° or 180° out-of-phase at the same bearing. For unbalance at the same bearing it must show approximately 90° out-of-phase.

5.6 Unbalance, Misalignment and Looseness

5.6.1 Case 6

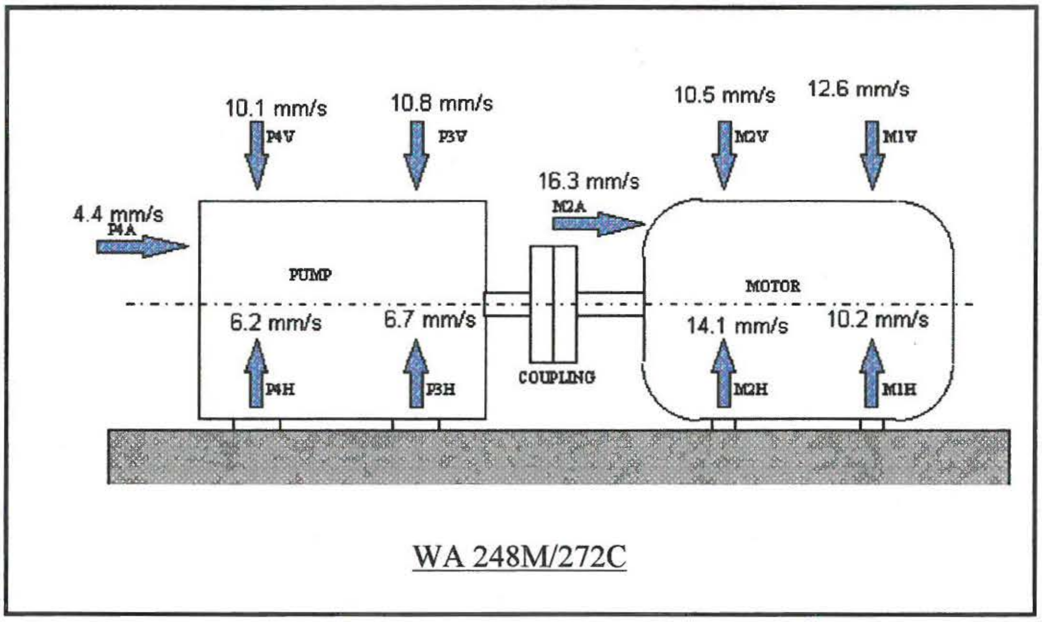


Figure 5-29: Schematic diagram of pump set

1. 100 kW, 2985 RPM motor.
2. Flexible-type coupling.
3. High vibration at 1x rpm with strong harmonics on motor.
4. Higher than usual amplitude at 5x rpm on motor.
5. High vibration at 2x and 3x rpm with harmonics on pump.
6. High axial vibration.

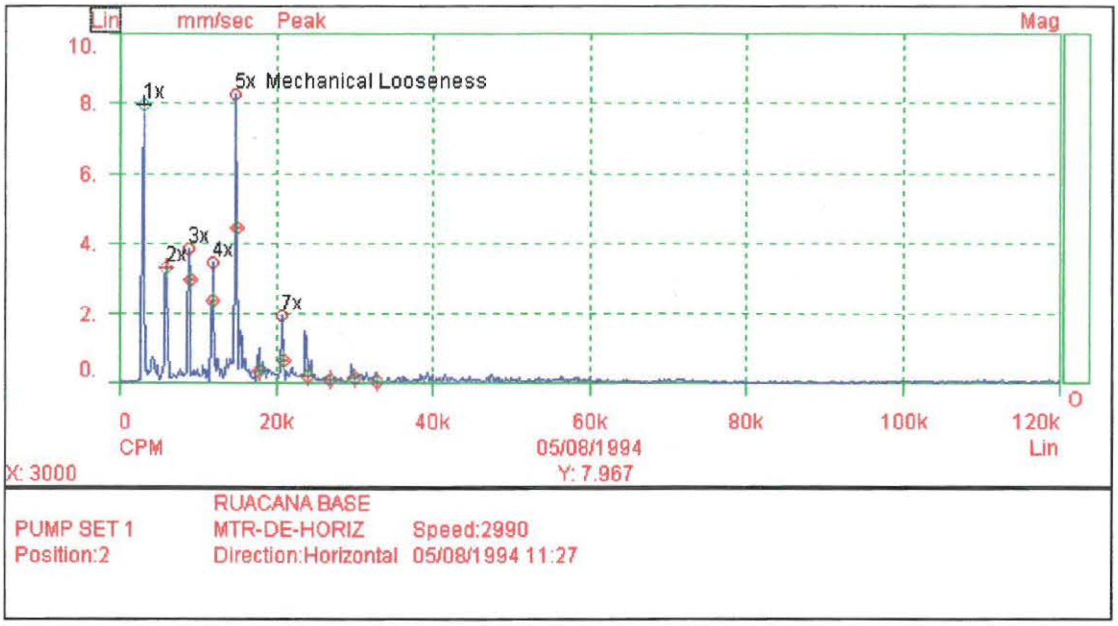


Figure 30: Spectrum at point M2H indicating unbalance, misalignment & looseness

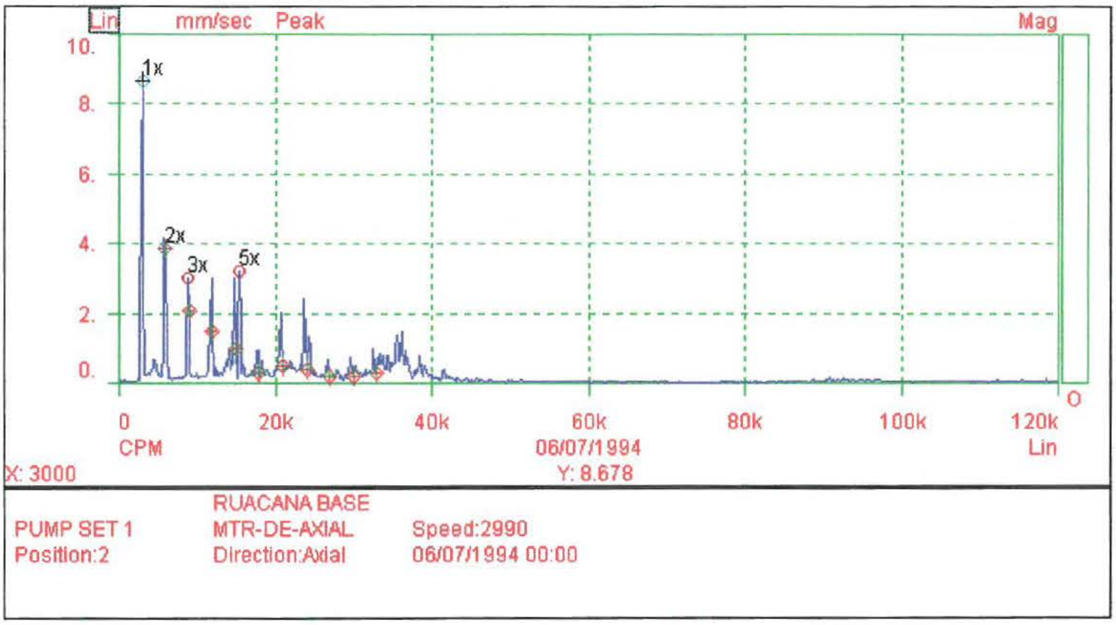


Figure 31: Spectrum at point M2A indicating misalignment & looseness

The spectrum for the motor inboard horizontal (M2H) and motor inboard axial (M2A) are shown above. High amplitudes can be noticed at 1x and 5x RPM. Based on the presence of a series of motor speed harmonics (multiples of rpm), definite mechanical looseness is indicated in all the spectrums taken. The 2x rpm component exceeds the running speed frequency on the pump indicating possible misalignment. The spectrum in figure 5-31 shows a high frequency peak at running speed with harmonics. This is also a strong symptom of misalignment and looseness.

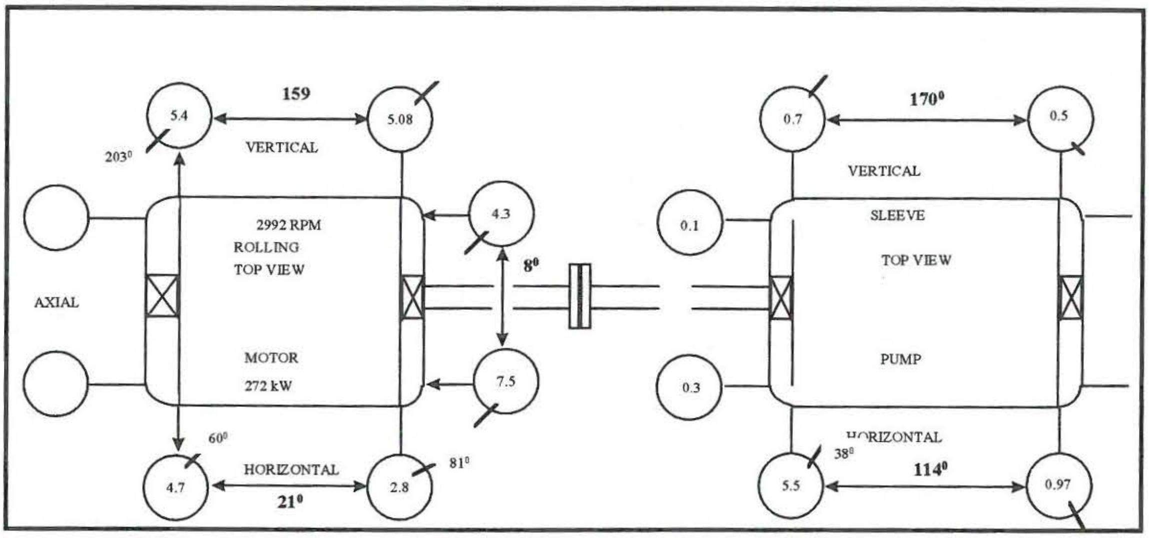


Figure 5-32: Running speed amplitude and phase data

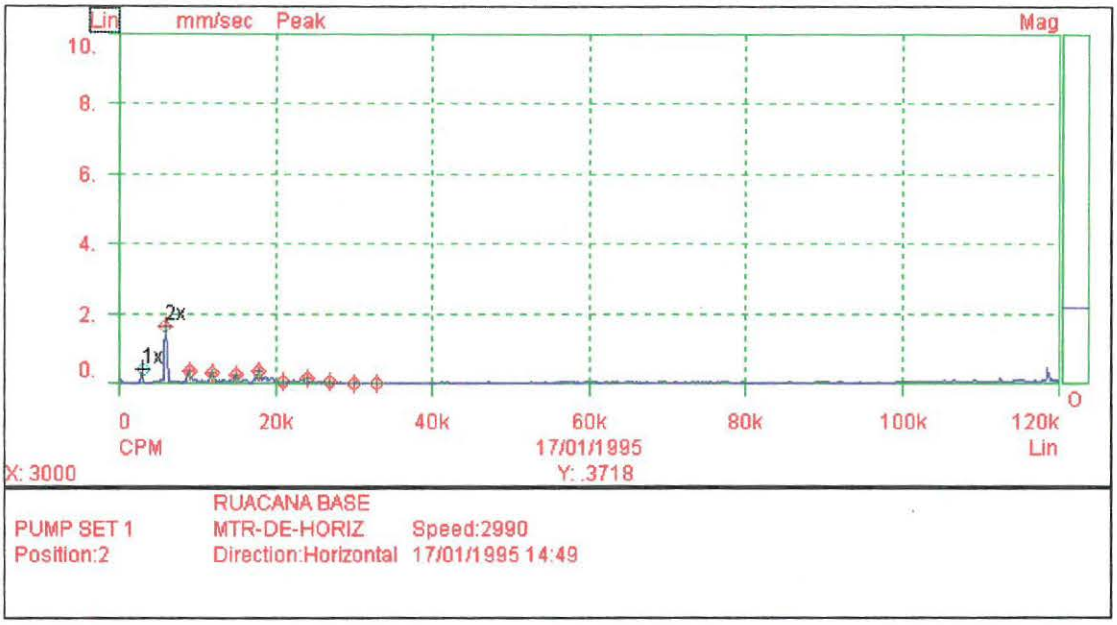


Figure 5-33: Spectrum at point M2H after motor was repaired

If phase data is reviewed in figure 5-32, it shows an out-of-phase shaking (159° vertically and 21° horizontally) mode diagnosed as misalignment, whereas the pump shows an in-phase shaking (104° and 114°) mode. When the horizontal and vertical phases at the same bearing on the motor are compared, it shows 143° (left) and 37° (right) which make it also a strong symptom of misalignment. When the motor bearing covers were opened for inspection, it was found that the source of looseness was from a loose bearing, turning in the housing of the outboard bearing housing. Unbalance and misalignment were also found. The rotor of the motor was balanced and the pump set re-aligned. See figure 5-33 for improvement in vibration for the motor inboard horizontal (M2H).

5.6.2 Discussion

Mechanical looseness results in a vibration at a frequency of twice the rotating speed (2x RPM), but may also result in higher harmonic frequencies (3, 4, 5 & 6x RPM). Mechanical looseness will not occur unless there is another force such as unbalance or misalignment to excite it. For example, in Case 8 a loose bearing turning in the housing was detected. The most dominant vibration was pinpointed and corrected. It shows how a 2 x RPM vibration can be generated by a combination of misalignment and mechanical looseness. After corrective action was taken the vibration decreased from 17.5 to 2.2 mm/s.

5.7 Mechanical Looseness

5.7.1 Case 7

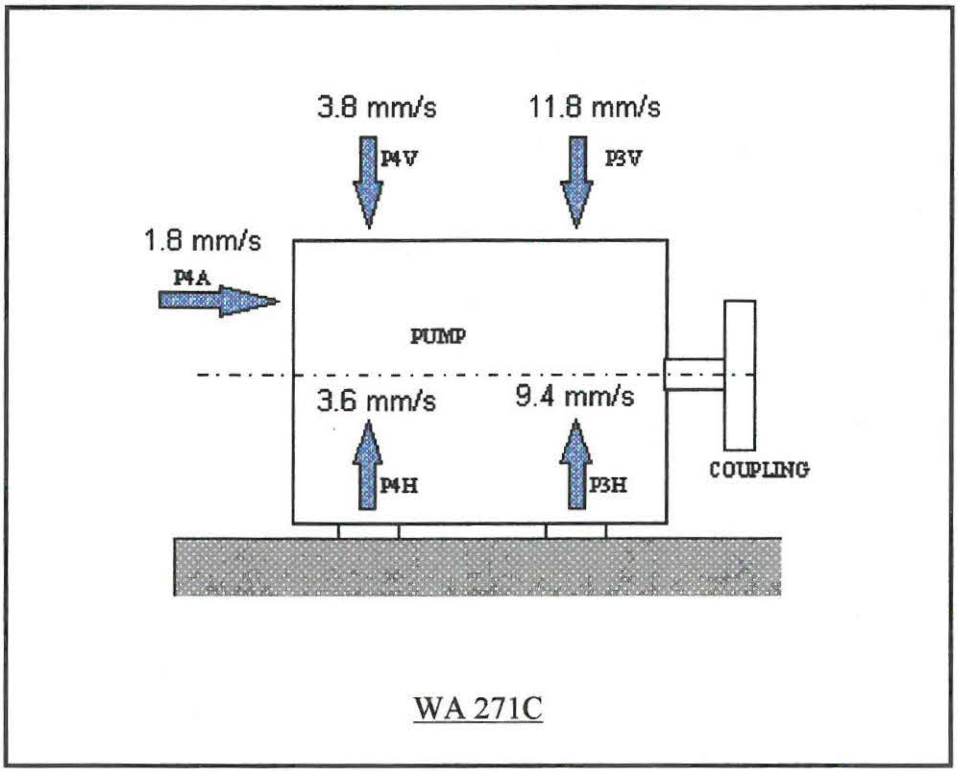


Figure 5-34: Schematic diagram of pump

1. 110 kW, 2985 RPM multi-stage pump.
2. Sleeve type bearings.
3. High vibration at 1x & 2x rpm with many harmonics.

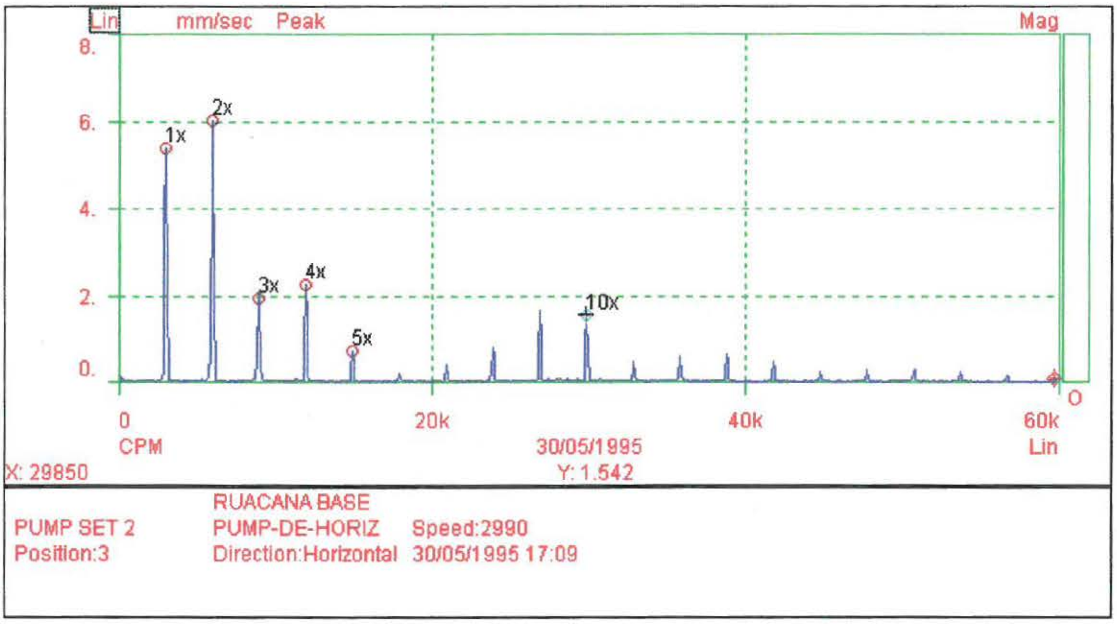


Figure 5-35: Spectrum at point P3H indicating mechanical looseness

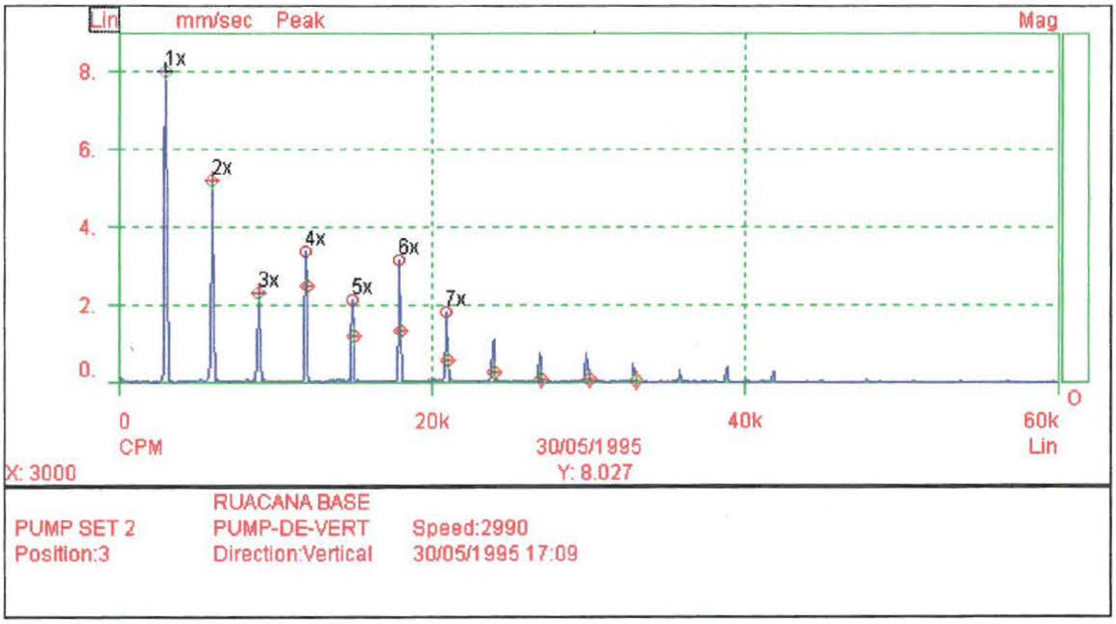


Figure 5-36: Spectrum at point P3V indicating mechanical looseness

The spectrum for the pump inboard horizontal (P3H) and pump inboard vertical (P3V) are shown above. Notice the running speed frequency with the harmonics up to 9x rpm. This is a clear indication of looseness on the pump bearings (incorrect bearing clearances).

5.7.2 Case 8

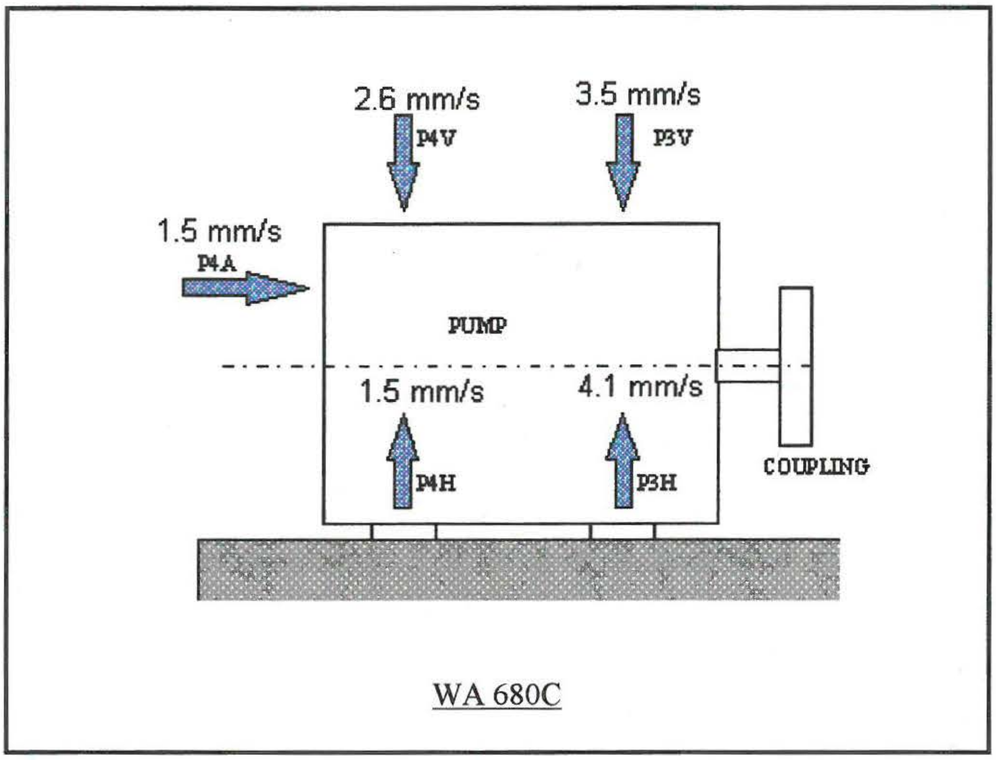


Figure 5-37: Schematic diagram of pump

1. 160 kW, 1485 RPM multi-stage pump.
2. Sleeve-type bearings.
3. High vibration between 12 and 30 KCPM.

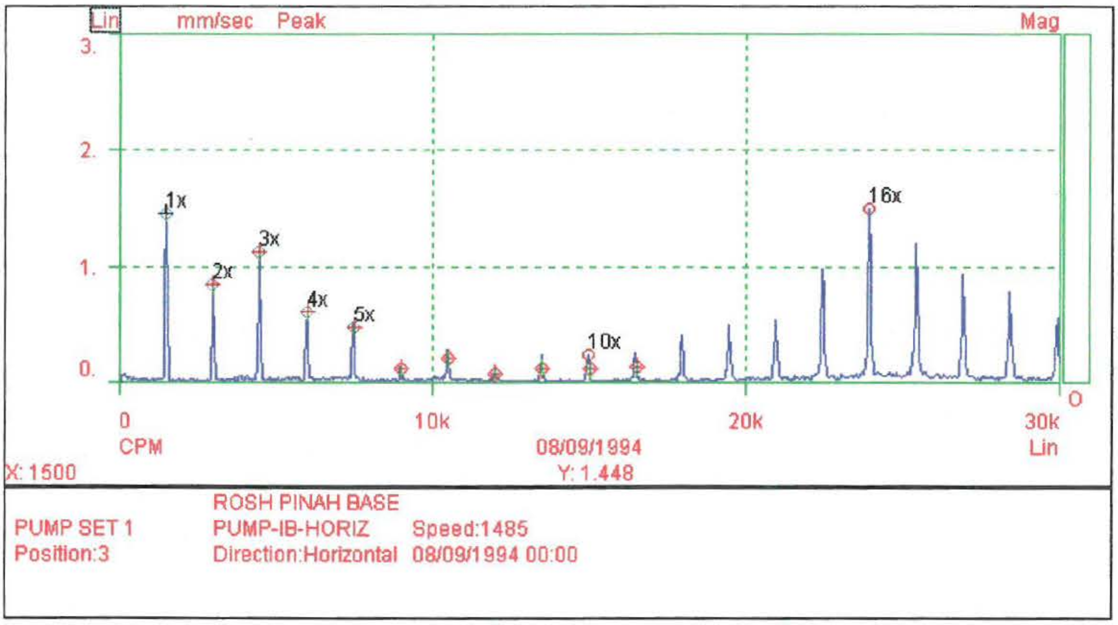


Figure 5-38: Spectrum at point P3H indicating mechanical looseness (rubbing)

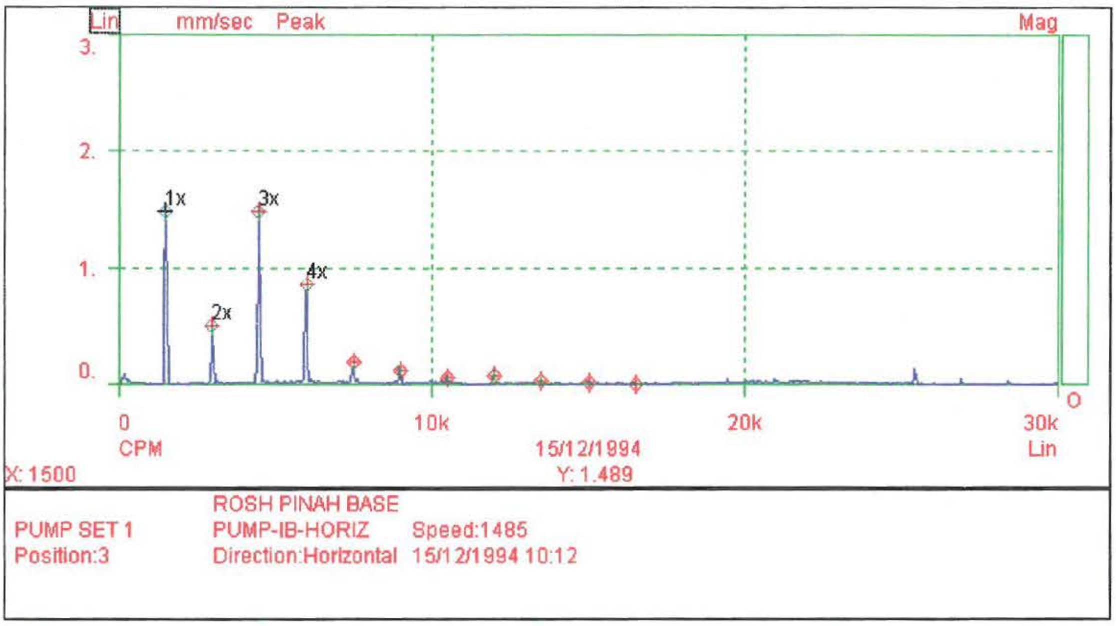


Figure 5-39: Spectrum at point P3H after replacement of sleeve bearing

Figure 5-38 displays the spectrum for the pump inboard horizontal (P3H) bearing. The running speed frequency with the lower harmonics and the appearances of frequencies between 12 000 to 30 000 CPM is a clear indication of looseness on the pump bearings (incorrect bearing clearances). This type of spectrum is an indication of a stationary part rubbing against a rotating component. After opening it, it was found that the shaft was rubbing against the bearing. The spectrum in figure 5-39 was taken after the bearing was replaced.

5.7.3 Discussion

As discussed in section 5.6.2, mechanical looseness results in a vibration at a frequency twice rotating speed (2x RPM), but may also result in higher harmonic frequencies (3, 4, 5 & 6x RPM). Vibration analysis data in Case 7 shows the vibration resulting from excessive clearance in a sleeve-type bearing. It can be seen that significant amplitudes of vibration occurred at frequencies of both 1x and 2x rotor rpm with harmonics. Case 8 experienced the same type of problem on a sleeve-type bearing. For comparison purposes the bearing was replaced with one having proper radial clearances. The exciting force was not removed (misalignment). It can be seen that by eliminating excessive looseness, the vibration decreased.

5.8 Rolling Element Bearings

5.8.1 Case 8

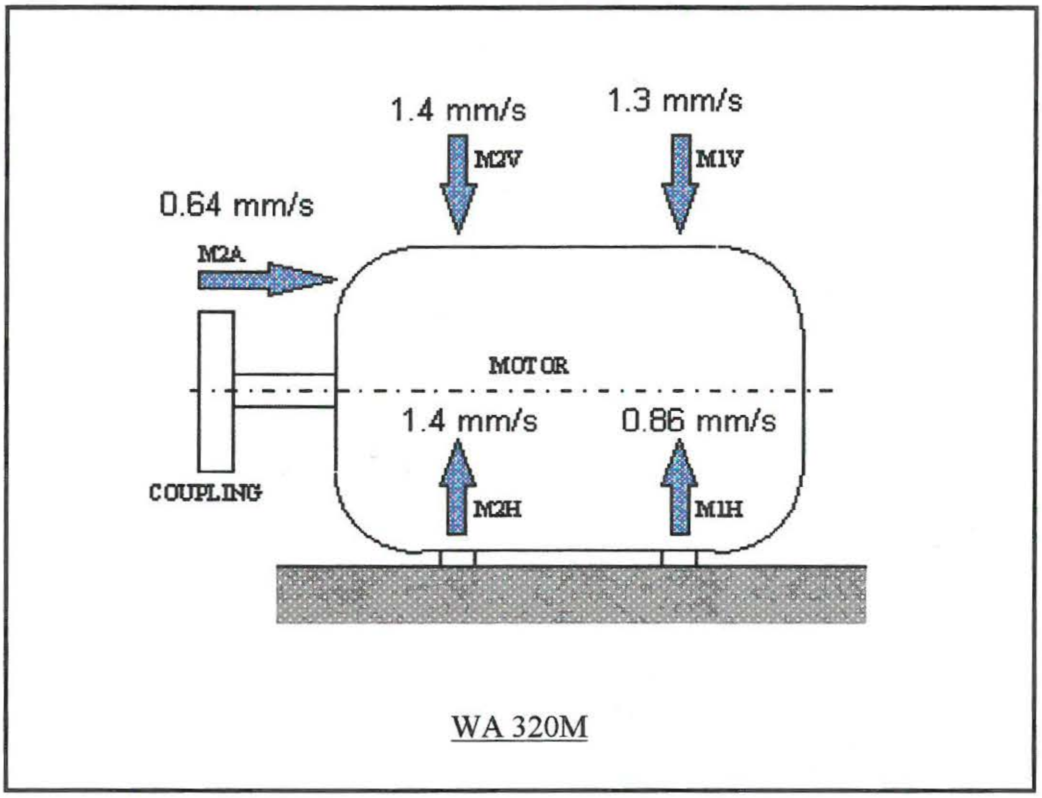


Figure 5-40: Schematic diagram of electrical motor

1. 45 kW, 1470 RPM motor.
2. Frequencies present between 60 000 and 120 000 CPM.

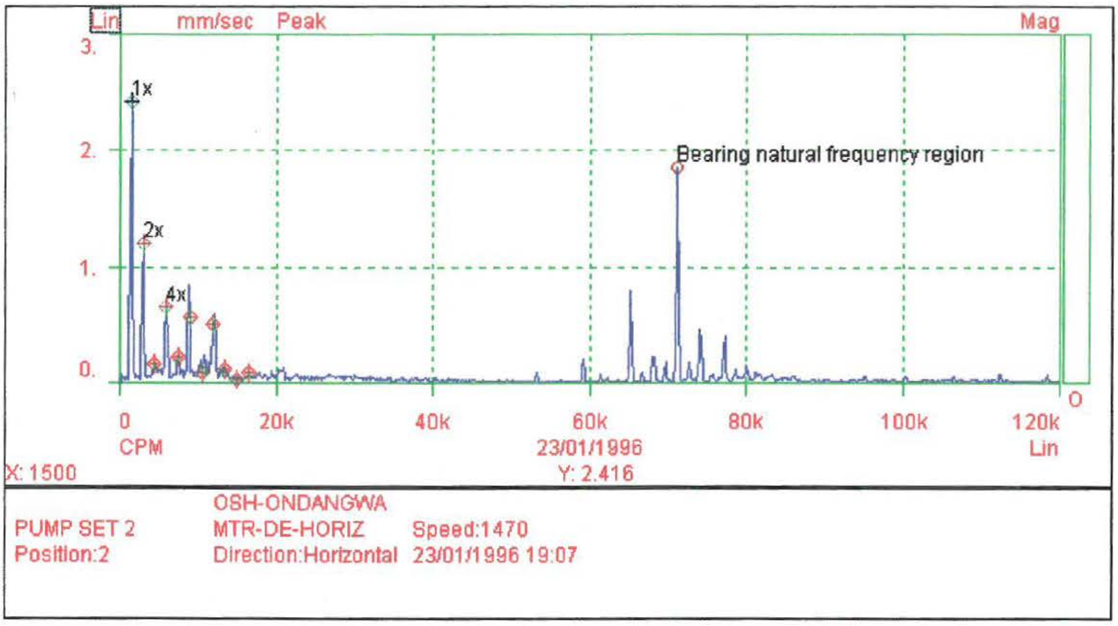


Figure 5-41: Spectrum at point M2H shows bearing natural frequencies

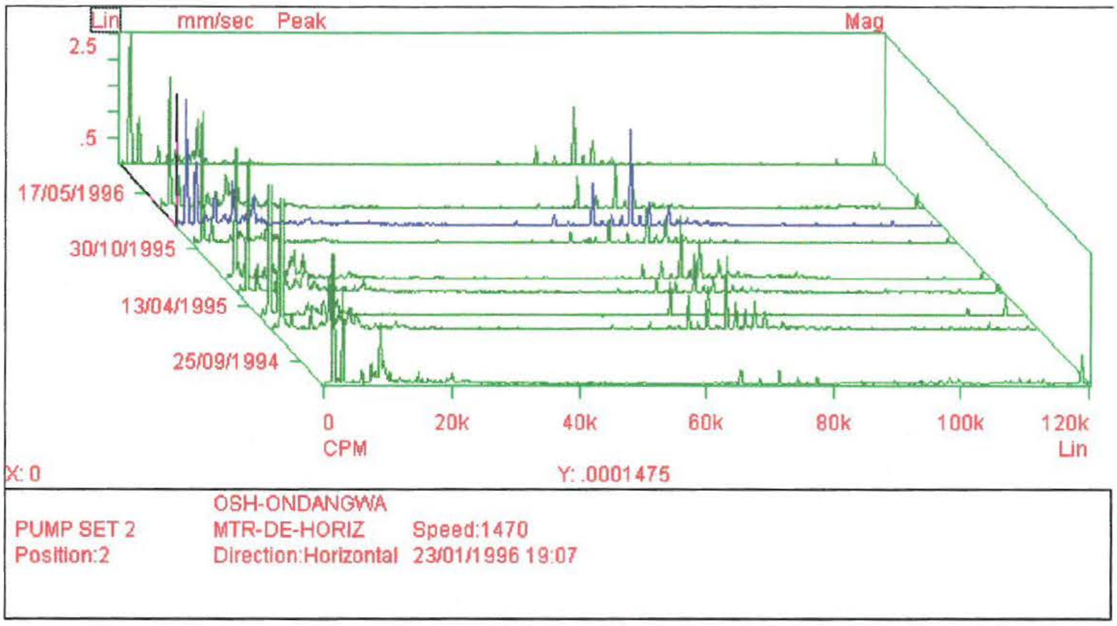


Figure 5-42: Multiple spectrum plot at point M2H

The spectrum in figure 5-41 shows the motor inboard vertical (M2H) bearing. Notice the bearing activities in the natural frequency range. The next plot in figure 5-42 is a multiple spectrum plot showing the motor inboard vertical (M2V) bearing over a period of 16 months. The purpose of these plots is to show the indication of a bearing problem on a velocity spectrum as indicated by the bearing components natural frequency. These frequencies are independent of operating speed and non-synchronous.

5.8.2 Case 9

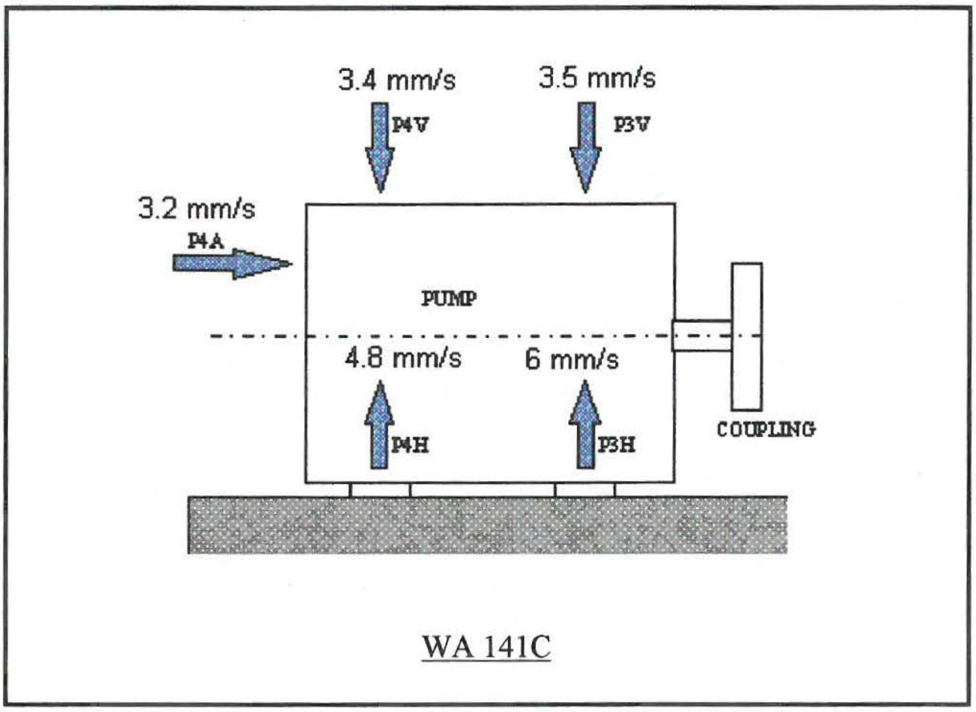


Figure 5-43: Schematic diagram of pump

1. 250 kW, 1483 RPM horizontal split-casing pump.
2. High vibration on the inboard bearing.
3. Broad band frequencies present

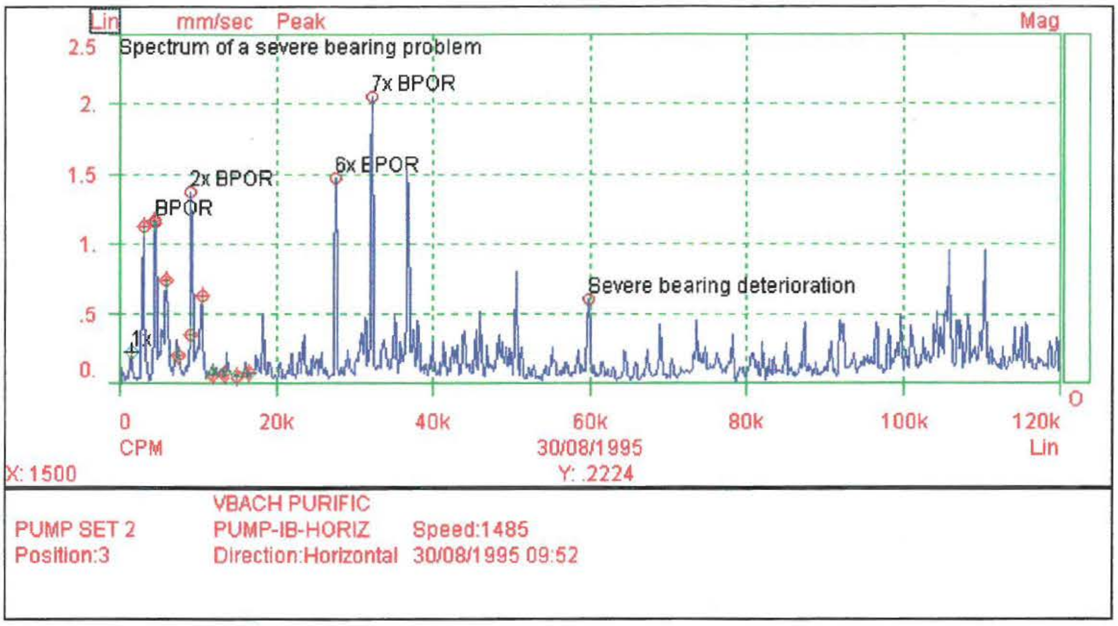


Figure 5-44: Spectrum at point P3H shows a severely damaged bearing

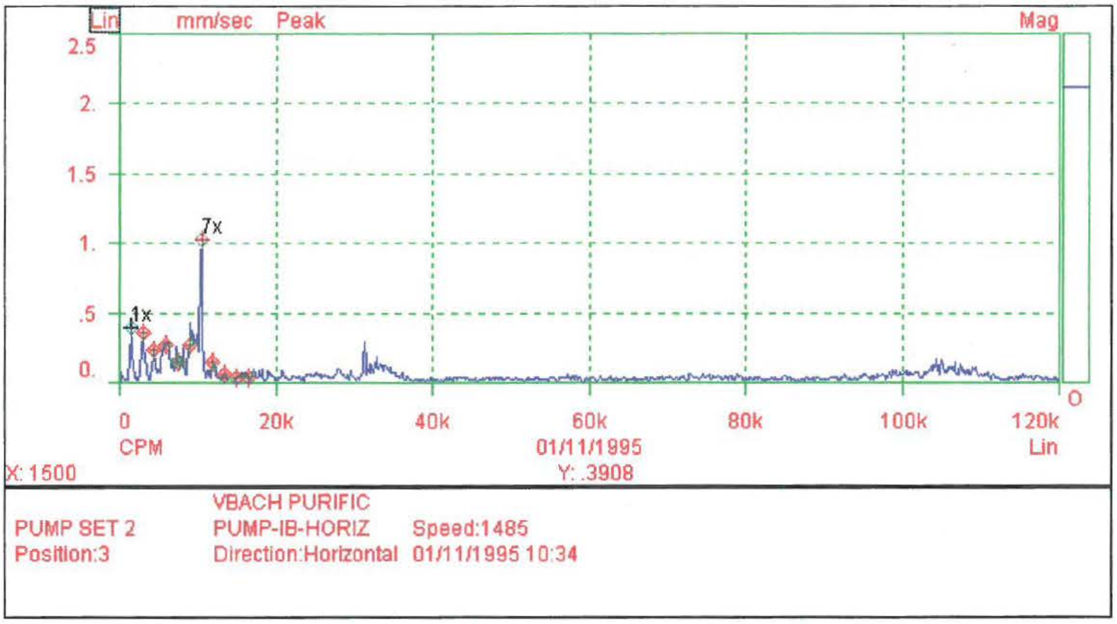


Figure 5-45: Spectrum at point P3H after the bearing was replaced

The spectrum in figure 5-44 shows the pump inboard horizontal (P3H) bearing. The fault frequencies for a SKF 6314 were calculated (Table 3, Appendix I). Note the primary BPOR (Outer ring defect frequency) which is situated at 121.7 Hz (4562 CPM) is present with its harmonics. This type of spectrum indicates a bearing which is close to failure. Note the many discrete peaks and side bands combined to form the characteristically raised noise floor over the whole range of the spectrum. The next spectrum in figure 5-45 was taken after the bearing was replaced.

![Central University of Technology, Free State logo]

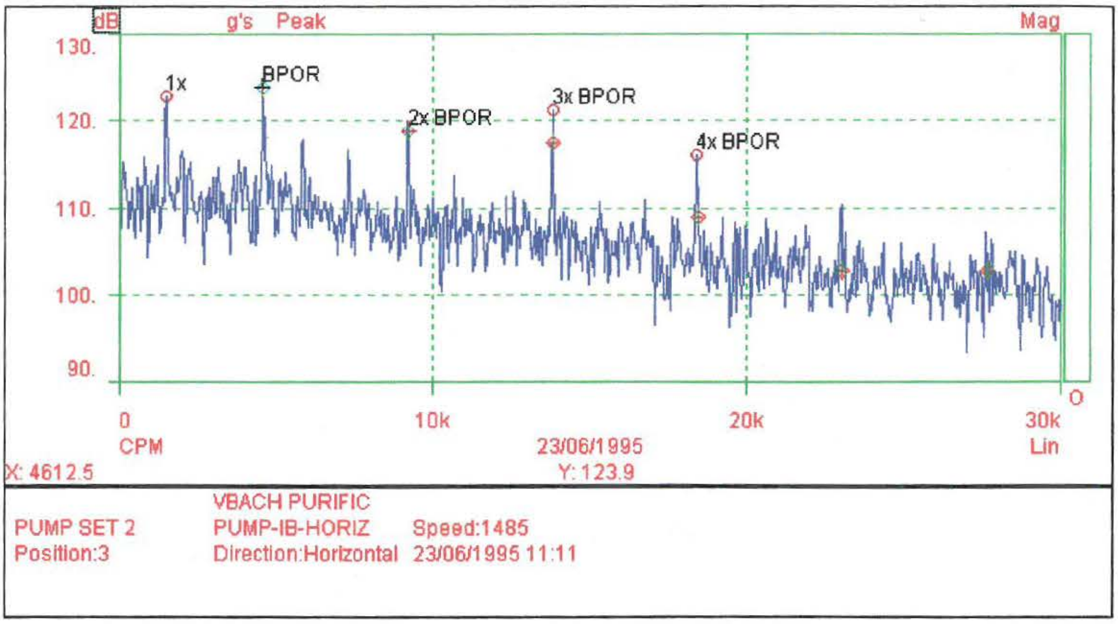


Figure 5-46: Enveloped acceleration spectrum at point P3H before bearing replacement

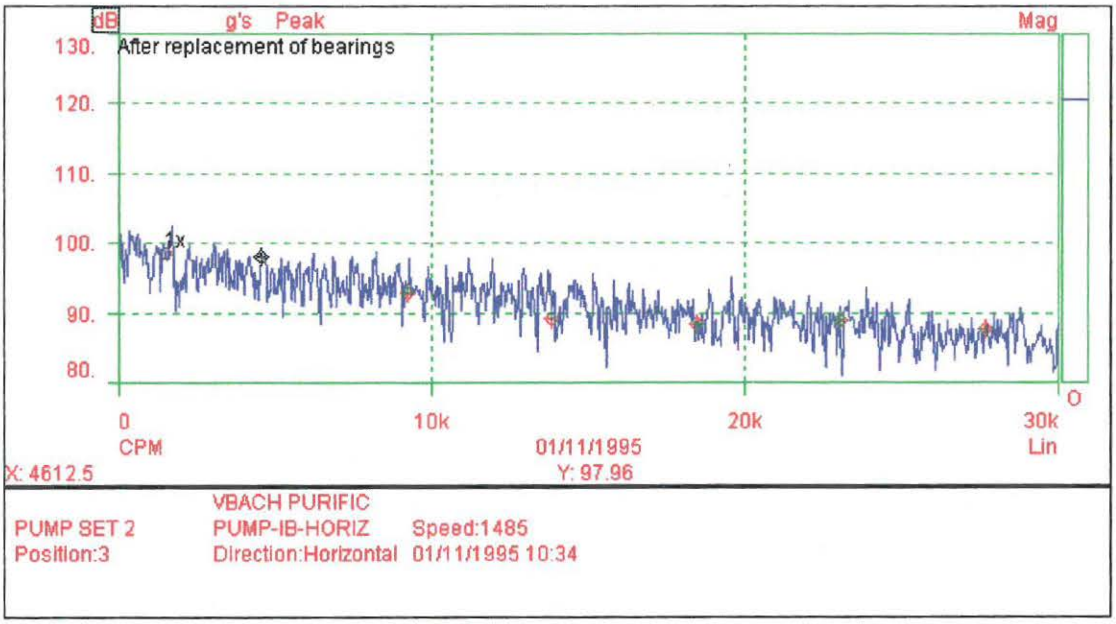


Figure 5-47: Enveloped acceleration spectrum at point P3H after bearing replacement

The plot in figure 5-46 shows the enveloped acceleration spectrum taken on the same point (P3H) before the bearing was replaced. To view the frequencies better the spectrum was displayed logarithmically in Decibels (dB). Note the prominent BPOR with its harmonics on the spectrum. The next enveloped acceleration spectrum in figure 5-47 shows the improvement after the bearing was replaced. Note the decrease in carpet level.

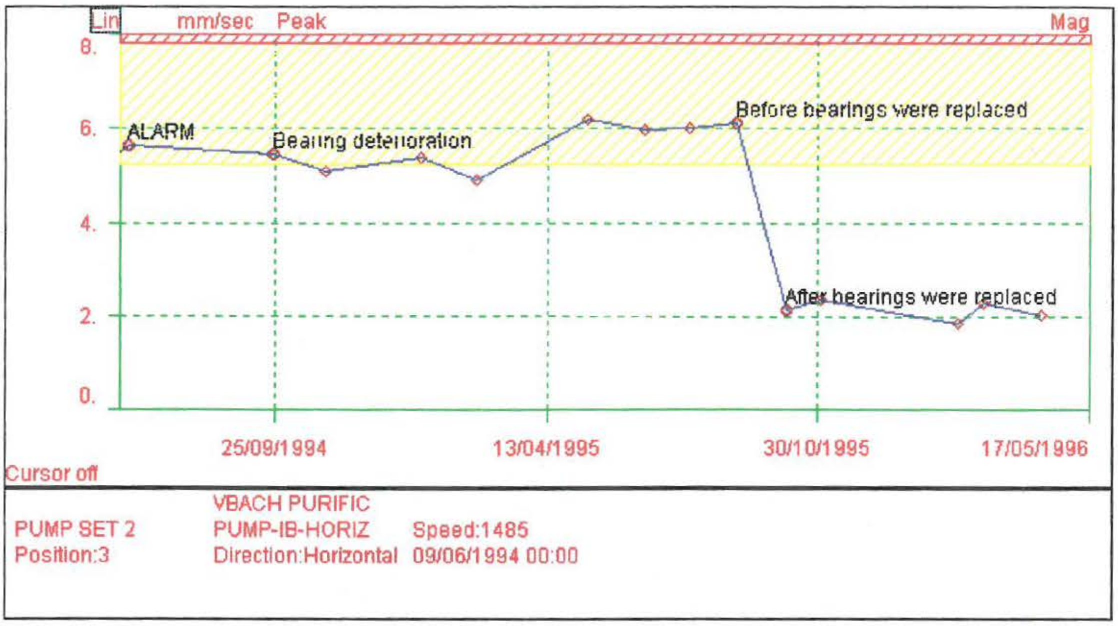


Figure 5-48: Overall vibration trend

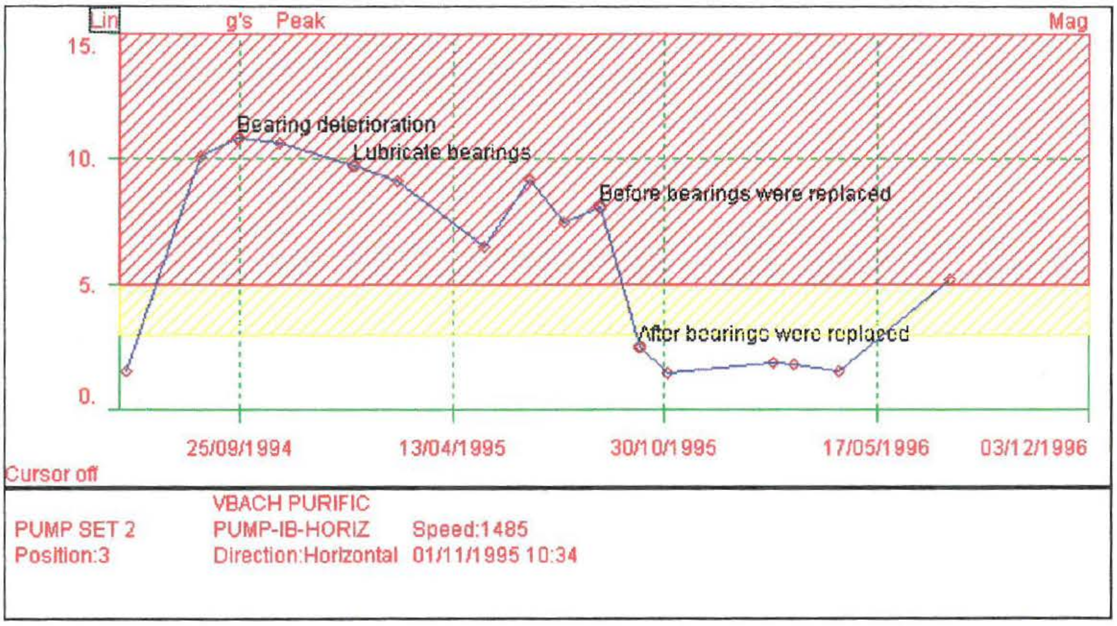


Figure 5-49: Overall vibration trend

Figure 5-48 shows the overall vibration trend (mm/s) and Figure 5-49 the high energy values (measured in g's) before and after replacement of the bearings. Note the improvement in both the overall vibration and the high energy value.

5.8.3 Case 10

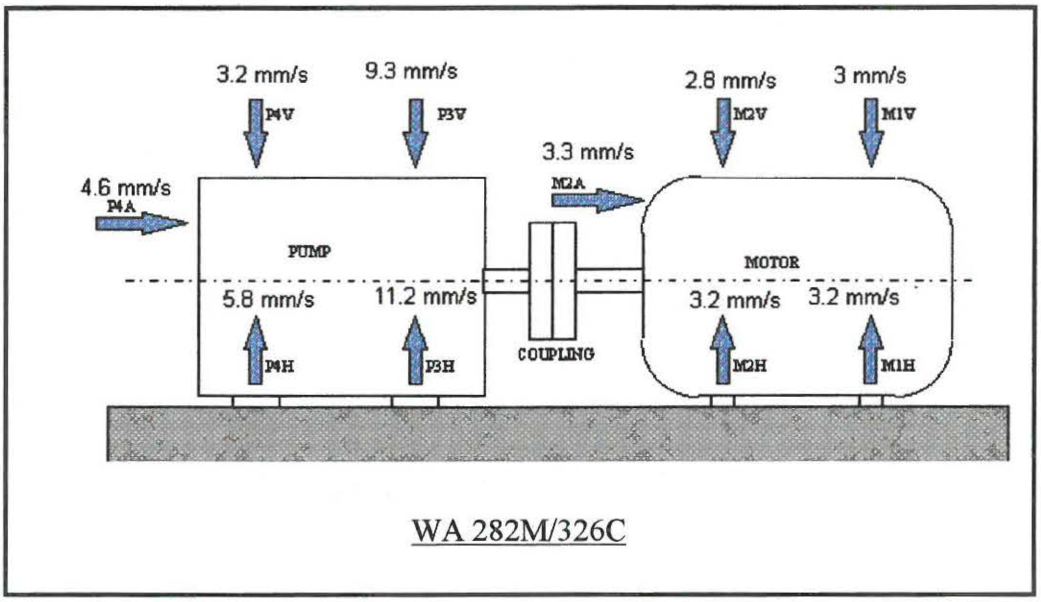


Figure 5-50: Schematic diagram of pump set

1. 90 kW, 2985 RPM multi-stage pump.
2. High overall vibration of 11.3 mm/s.
3. Low harmonic frequencies with a higher than usual 3x rpm.
4. Broad band frequencies present.
5. Huge amount of energy from 36 000 CPM.

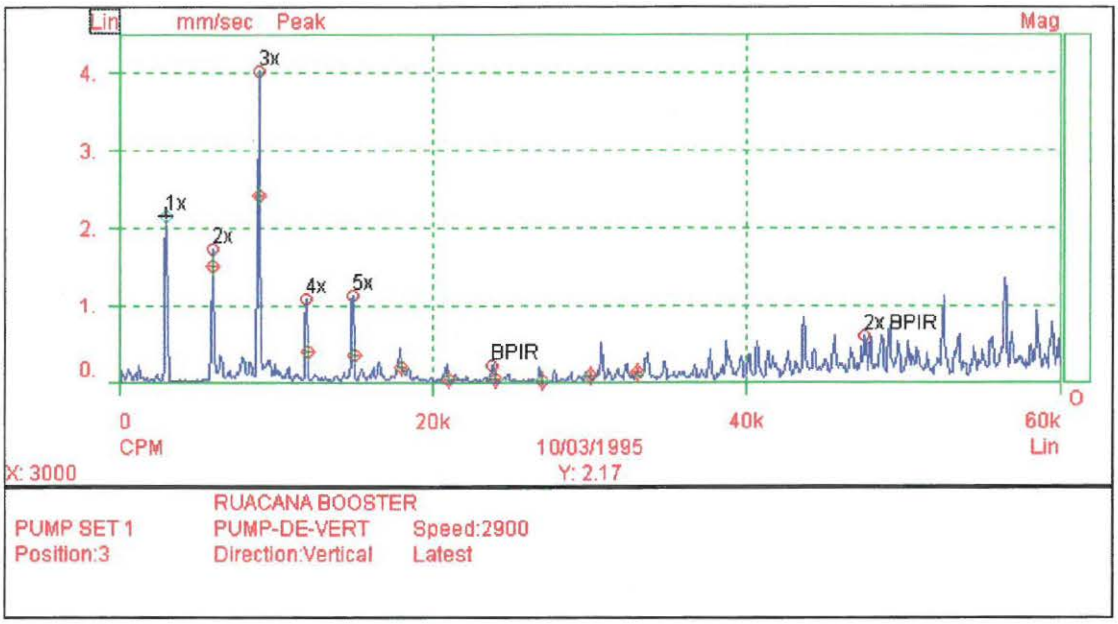


Figure 5-51: Spectrum at point P3V shows a damaged bearing with excessive clearances

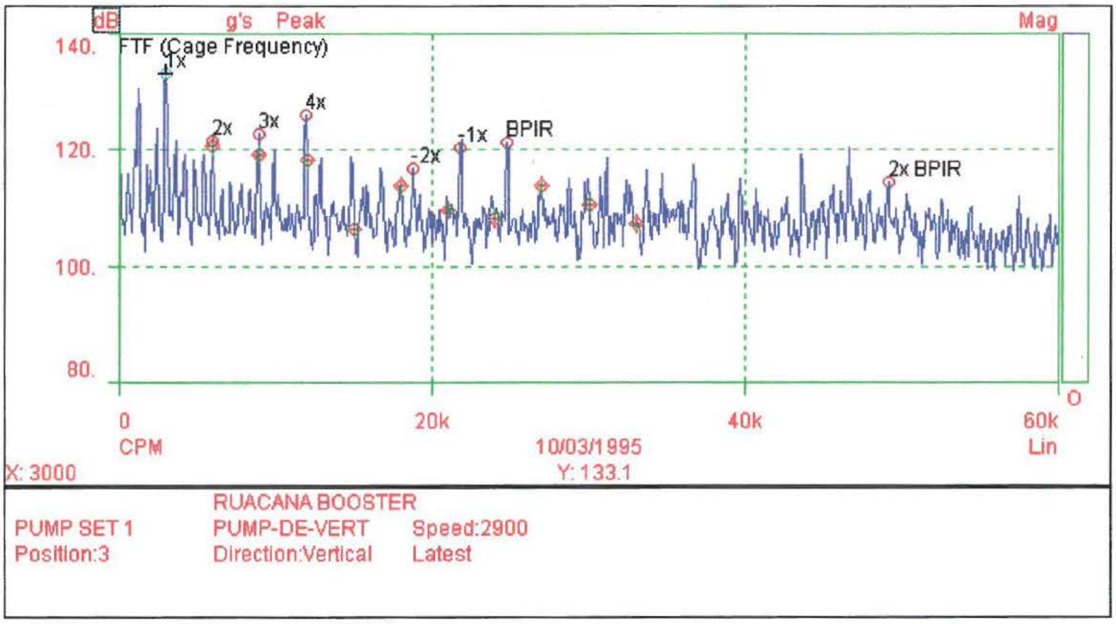


Figure 5-52: Enveloped acceleration spectrum at point P3V shows the damaged bearing with side bands

The velocity spectrum in figure 5-51 shows the pump inboard vertical (M3V) bearing. Note the small peak of the primary BPIR (rotating ring defect frequency) which is situated at 24 735 CPM (SKF NU 208E bearing). This bearing has excessive clearances between the rollers and a huge amount of energy from 36 000 CPM can be seen. The next plot in figure 5-52 shows the enveloped acceleration spectrum taken at the same point (M3V). Note the fundamental BPIR with side bands marked with -1x & -2x on the spectrum.

5.8.4 Case 11

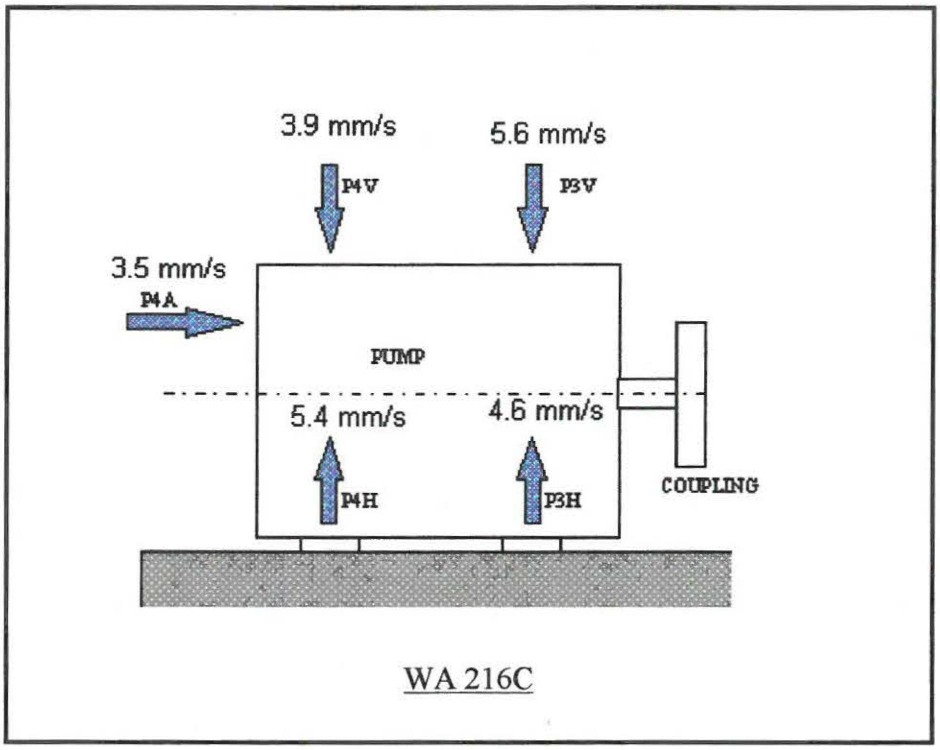


Figure 5-53: Schematic diagram of pump

1. 55 kW, 970 RPM low-pressure single-stage pump.
2. Broad band frequencies present.
3. Bearing noise was noticeable.

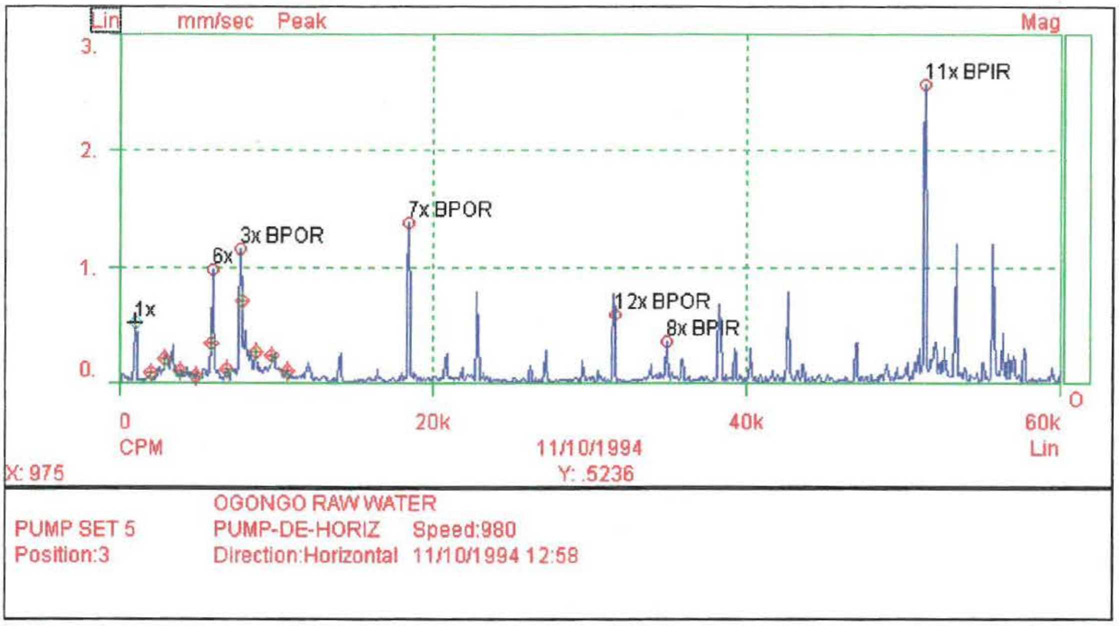


Figure 5-54: Spectrum at point P3H shows a damaged bearing

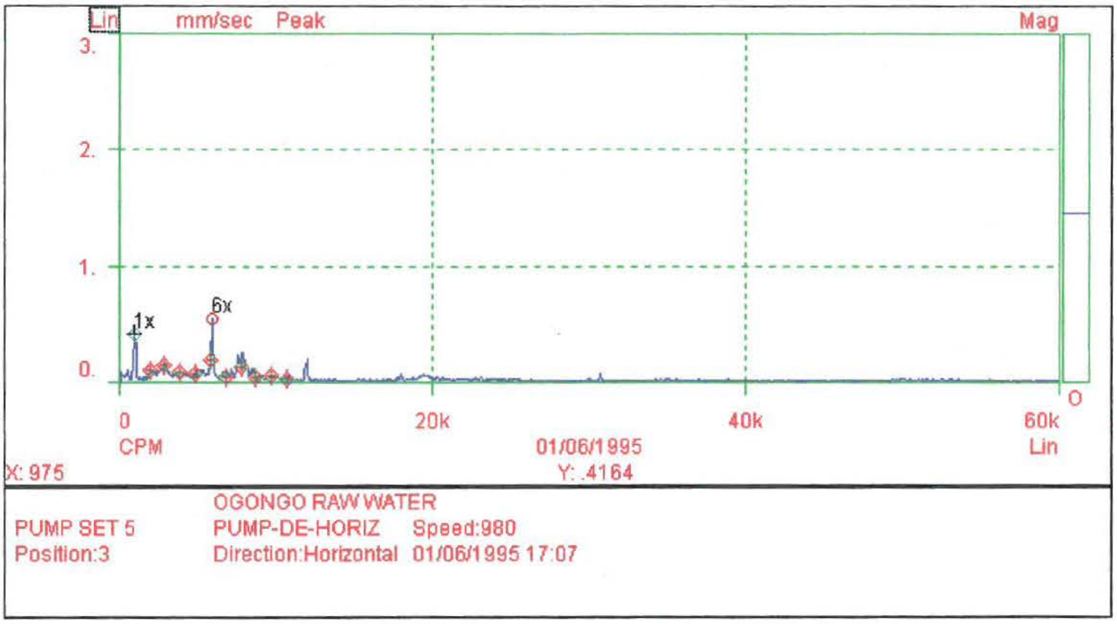


Figure 5-55: Spectrum at point P3H after the bearing was replaced

The spectrum in figure 5-54 shows the pump inboard horizontal (M3H) bearing. Note the harmonics of BPIR and BPOR frequencies for a SKF 6411 ball bearing. This type of spectrum has severe bearing deterioration. Many 1x rpm side bands appear around bearing defect frequencies indicating pronounced wear throughout the periphery of the bearing. Note the random high frequency noise floor which extends far down into the spectrum. The next spectrum in figure 5-55 was taken at the same point after the bearing was replaced.

5.8.5 Discussion

It was found that bearing defect frequencies differ from other frequencies as they are non-integer multiples of operating speed. They are defect frequencies which should not be present at all.

It was experienced that the increases due to bearing defects when compared to amplitude increases due to other vibration sources, such as unbalance, misalignment, etc. were of an entirely different nature. For example, a bearing is considered bad at 1 mm/s, but if this amplitude originates from unbalance, it is usually considered as acceptable. It was stated that each component of an anti-friction bearing develops a unique characteristic vibration signature. The four components of bearing defect frequencies were calculated in Table 2, Appendix I, to identify which bearing component was defective. Other data contained in the vibration signature that was investigated included information on internal clearances, looseness and adequacy of lubrication.

The shock pulse monitoring technique was the first technique that was used in the project. Not much experimental work was done to recognize a certain pattern of data (trending) over a period of time. It was experienced that this technique was very sensitive to transducer location and mounting. Small changes in location produced changes in the readings. This technique can be useful as a detecting device, but cannot be used for pinpointing the problem.

In Case 8 bearing defects began to excite the natural frequencies of the bearing which occur from 30 to 120 000 CPM. These frequencies have already started to modulate with the running speed (1 x rpm side bands appear above and below these natural frequencies) as wear progresses. There are several high frequencies generated by the faulty bearing that are random and unsteady, and, as wear progresses, a haystack will develop.

Cases 9, 10 & 11 show vibration spectrums of bearings with progressive failure modes. This is the appearance of the bearing defect frequencies themselves. As a bearing defect progresses, several different failure paths may be followed. The defect frequency may drop, but increased side band and multiples will give the spectrum a very characteristic shape often referred to as a raise in baseline or noise floor.

For Case 10 it was interesting to note the appearance of the cage frequency (FTF) with harmonics marked with the cursor on the spectrum. The cage frequency will appear if the inner race rotates with the shaft. The bearing defects were so advanced that the bearing defect frequencies started to develop in harmonics and modulated with the running speed (1 x rpm side bands appear above and below these natural frequencies) as wear progresses. The bearing failed one week after data collection. After opening the pump, it was found that the inner race of inboard bearing rotated with the shaft and it showed severe wear, indicating cracking and flaking of the race.

5.8.5 Discussion

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5.9 Resonance

5.9.1 Case 12

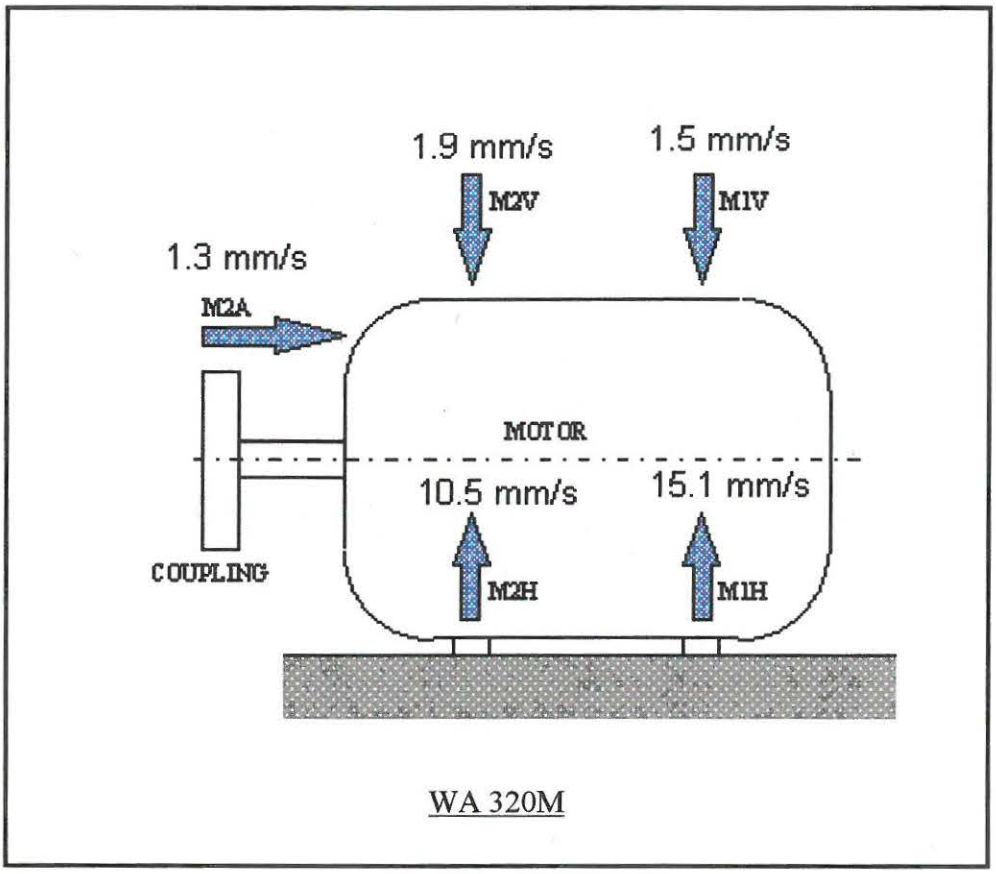


Figure 5-56: Schematic diagram of motor

1. 45 kW, 1470 RPM motor.

2. High vibration at 1x rpm (running speed) in the horizontal direction.

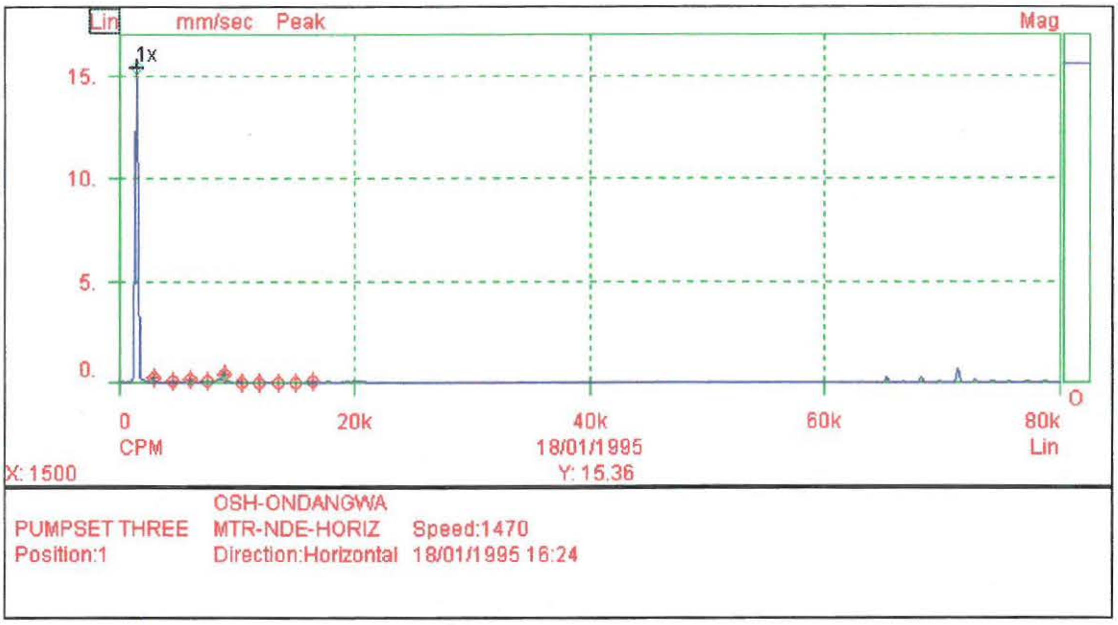


Figure 5-57: Spectrum at point M1H shows a high running speed frequency

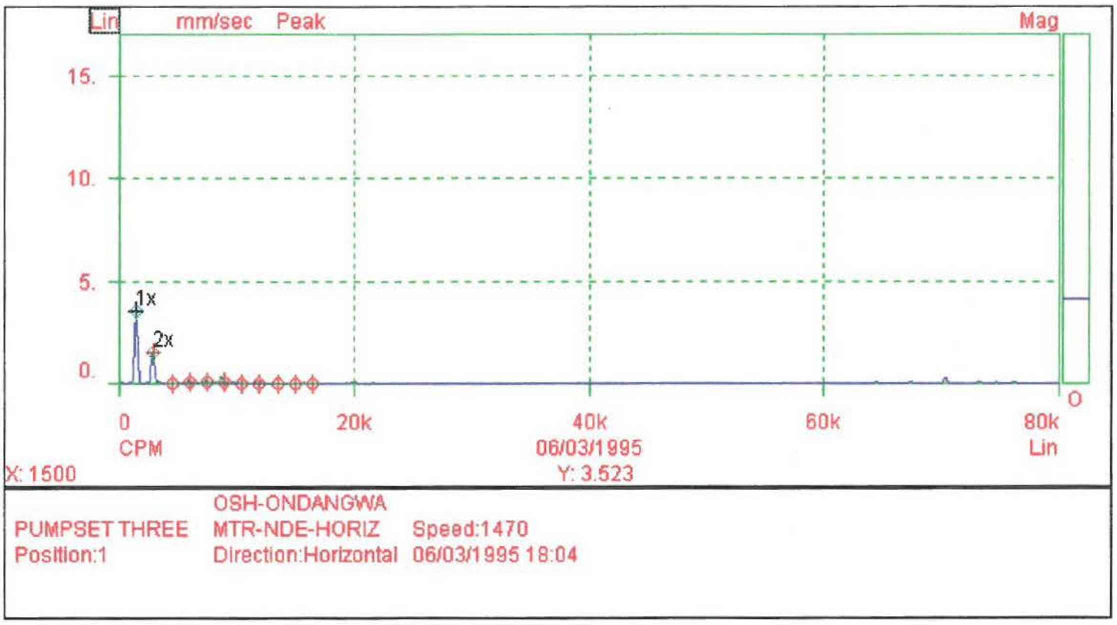


Figure 5-58: Spectrum at point M1H shows a decrease in the running speed frequency

The spectrum for the motor outboard horizontal (M1H) is shown in figure 5-57. Note the high running speed frequency at 1x RPM. When one of the baseframe bolts was loosened, the vibration decreased from 15.3 to 3 mm/s (Figure 5-58).

5.9.2 Case 13

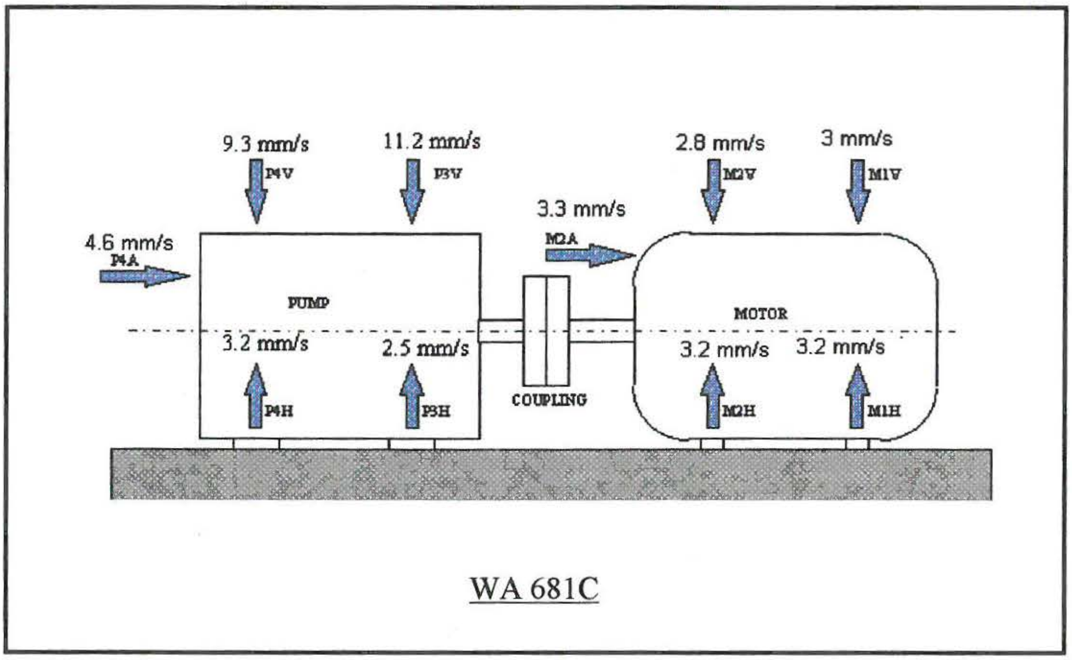


Figure 5-59: Schematic diagram of pump set

1. 160 kW, 1480 RPM multi-stage pump.
2. Sleeve-type bearings.
3. High vibration at 1x rpm in the vertical direction.
4. Overall vibration of 37 mm/s is measured in the discharge pipe.

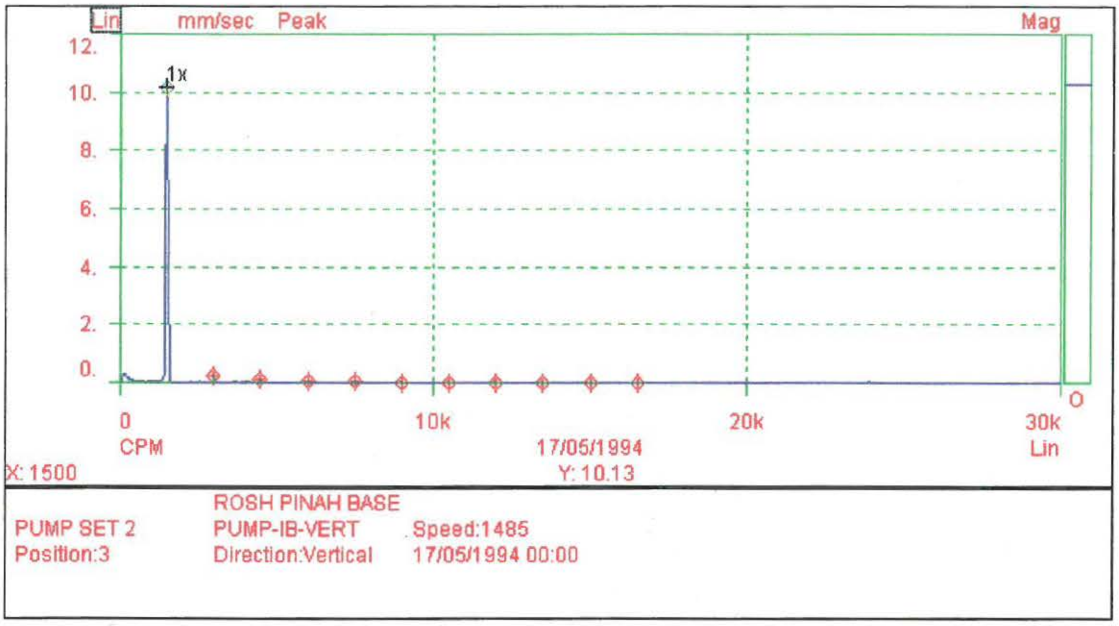


Figure 5-60: Spectrum at point P3V shows a high running speed frequency

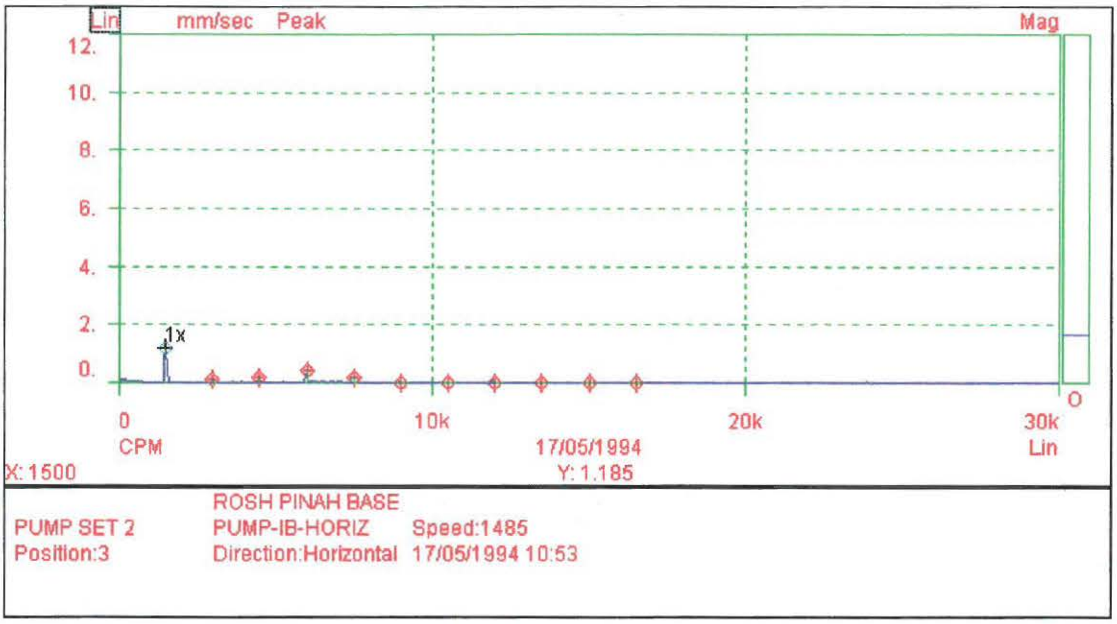


Figure 5-61: Spectrum at point P3H shows a small running speed frequency

The above spectrums shows the pump inboard vertical (P3V) and horizontal (P3H) bearing. It seems to be unbalanced due to the high 1x RPM component, but when the vertical spectrum (high vibration of 10 mm/s at 1x RPM) was compared to the horizontal (1.1 mm/s at 1x rpm) reading at the same bearing, resonance was suspected.

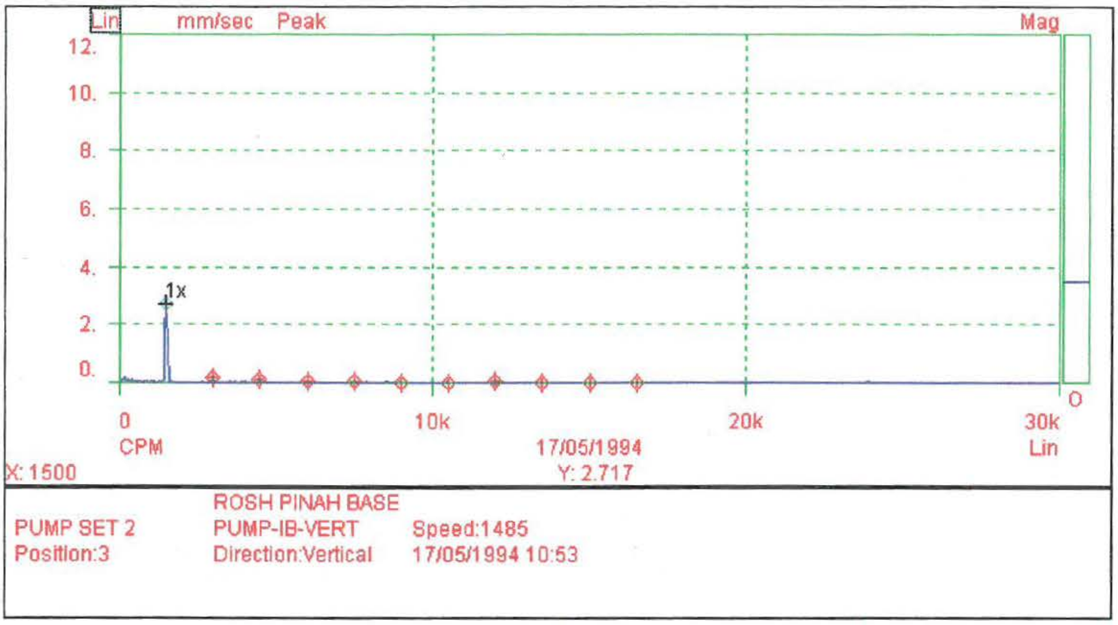


Figure 5-62: Spectrum at point P3V shows a decrease in the running speed frequency

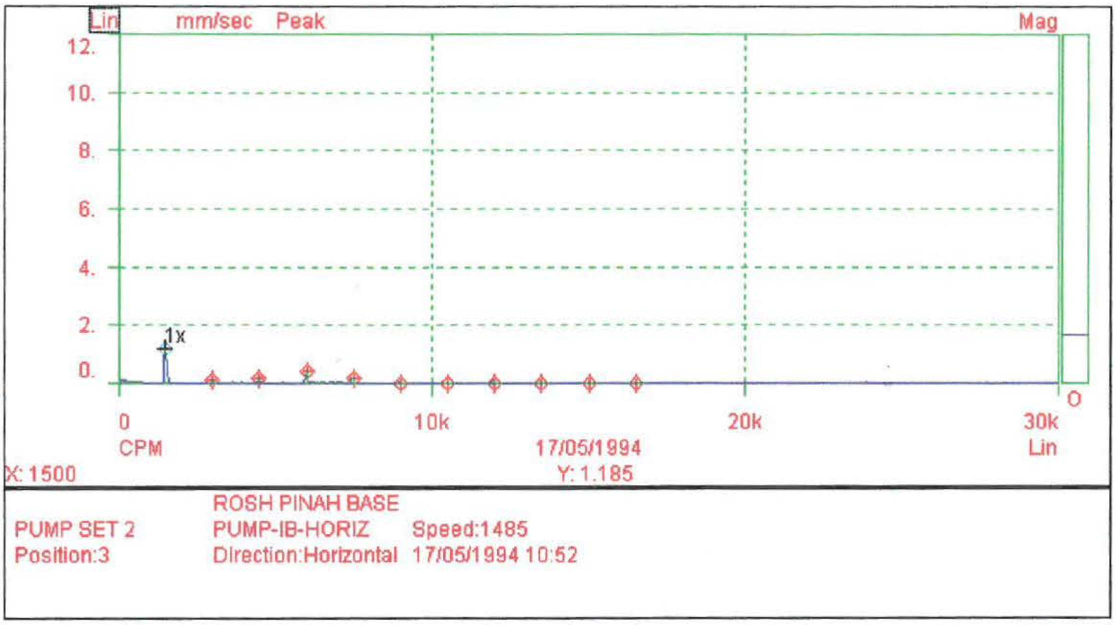


Figure 5-63: Spectrum at point P3H shows no increase in the running speed frequency

The above spectrums show the pump inboard vertical (P3V) and horizontal (P3H) bearing after stiffness on the pipe was altered. The 1x rpm vibration at point P3V decreased from 10 to 2.7 mm/s, and point P3H from 1.1 to 0.6 mm/s. The overall vibration on the pipe decreased from 37 to 5 mm/s.

5.9.3 Case 14

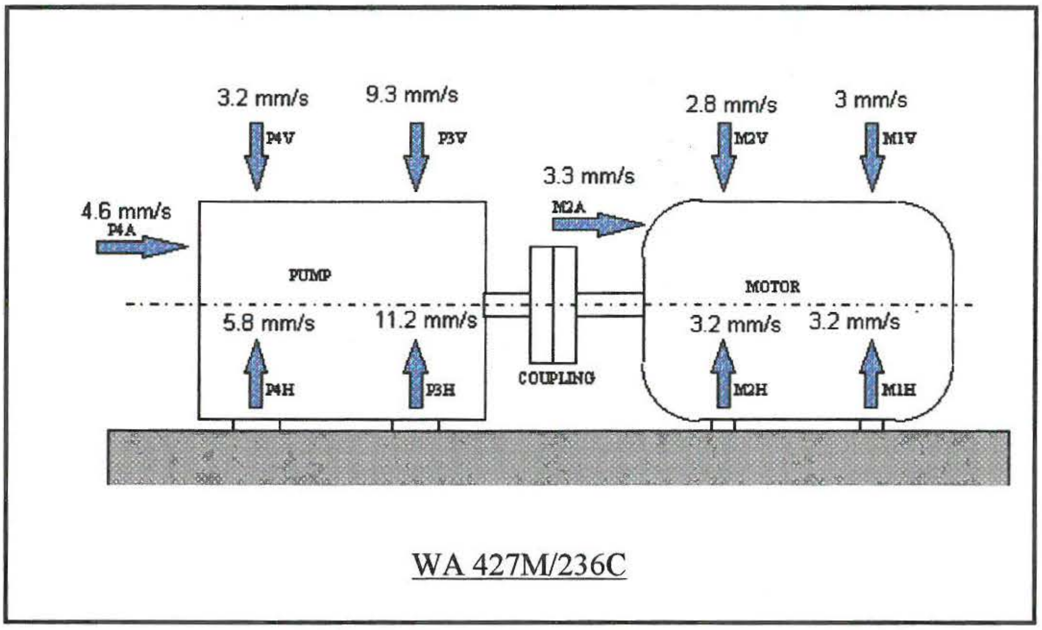


Figure 5-64: Schematic diagram of pump set

1. 200 kW, 1480 RPM multi-stage pump.
2. Vibration of 8 mm/s at a frequency of 31200 CPM.
3. This frequency appears in the vertical direction.

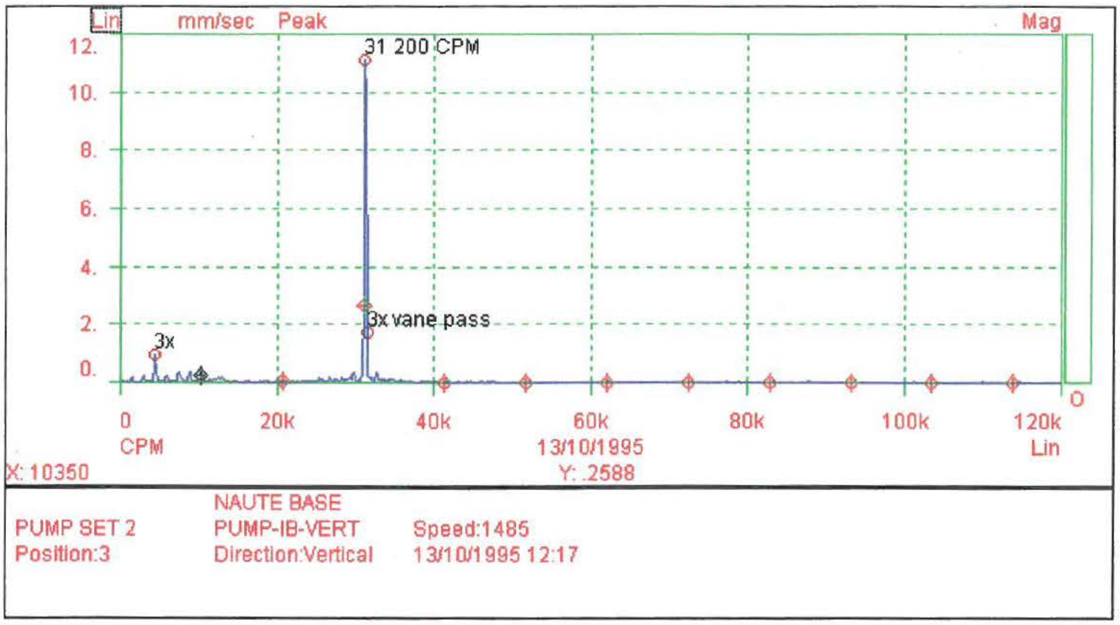


Figure 5-65: Spectrum at point P3V shows a high amplitude at 32 200 CPM

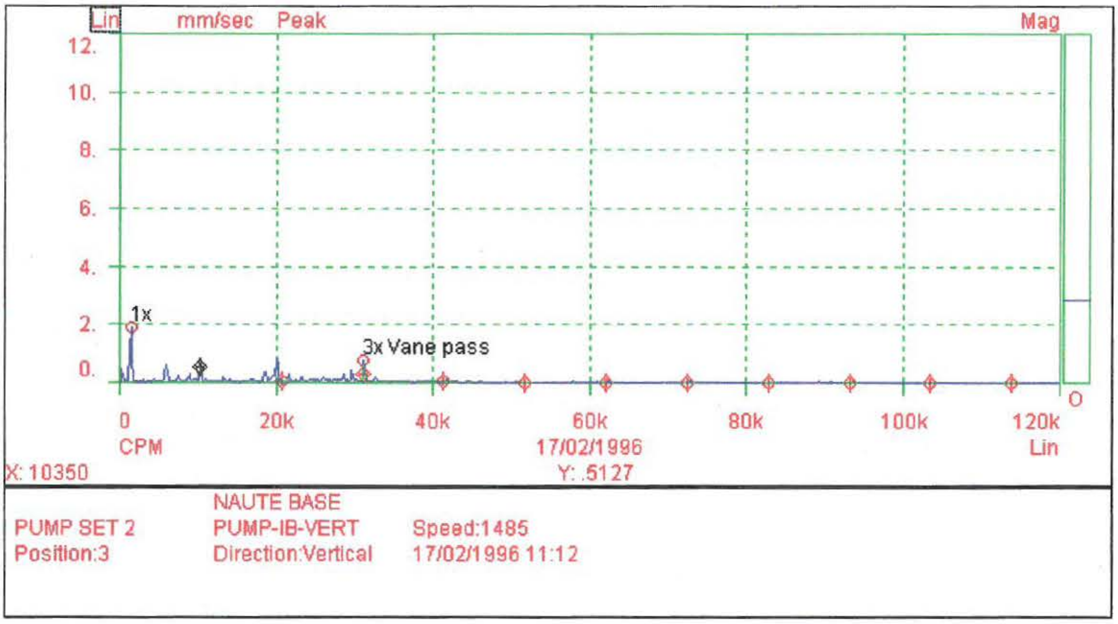


Figure 5-66: Spectrum at point P3V shows the amplitude at the same frequency after the discharge valve was throttled

The spectrum for the pump inboard vertical (P3V) is shown in figure 5-65. Notice the high amplitude of vibration at a frequency of 32 200 CPM. This peak is not a harmonic of the running speed. The next spectrum in figure 5-66 was taken after corrective action was taken.

5.9.4 Discussion

The phenomenon of resonance has been covered in Section 2 with ways to confirm whether or not a part is vibrating at resonance. It was stated that to design a machine installation with no natural frequencies, coincident with any significant exciting force generated by the machine, is extremely difficult. As a result, resonance will appear at some stage in equipment. This is true, because resonance was detected in various installations. One of the symptoms of resonance was that it produced directional vibration.

Such a resonance problem was experienced in Case 13. A high vibration of 10 mm/s was detected at the running speed (1x rpm) in the vertical directions of the pump. A vibration reading of 37 mm/s was measured on the pipe. When a part is in resonance, it can be determined by slightly changing the part's natural frequency. For non-rotating parts such as an attached pipe, the suspected part can have its natural frequency changed by applying a load with a hydraulic jack or a support.

When more strain was added on the resonant point, the vibration increased both on the pipe (resonant part) and bearing housing. When strain was removed from the resonant point, the vibration decreased on the pipe from 37 to 5 mm/s. The vibration on the bearing housings decreased from 10 to 2.7 mm/s.

When the resonant part's natural frequency is altered, for example by making it more rigid, the resonant part can or will no longer resonate. Therefore, its amplitude decreases and its resonance symptoms disappear. The vibration problem was solved by working on the resonant part and not on the source. Further unnecessary deterioration of the pump bearings was prevented, thereby reducing the chances of possible premature pump failure.

Case 14 displays another example where resonance was involved. High vibration of 8 mm/s was detected on the driven end of the pump. The vibration is caused by a frequency at 31 200 CPM. The third vane pass frequency was magnified by the natural frequency of the impeller. After corrective action was taken, the vibration decreased from 8 to 2.7 mm/s.

In Case 12 high vibration of 15.3 mm/s was detected at one of the measurement points. Viewing the vibration spectrum, it was found that the vibration was generated by the running speed (1x RPM). In this case the 1x RPM was magnified by the holding-down bolts. A possible explanation for the change in vibration was due to a change in resonance of the baseframe. It is stated in theory that soft feet is experienced when a machine frame or feet deflect slightly when the hold-down bolts are tightened or loosened. When the vibration increases, it is usually due to misalignment or a change in gap concentricity. There were several resonances on the total structure. These resonant frequencies were changed appreciably when the frame was bolted down as compared to when it was totally free. It was experienced that when one of the foot bolts was loosened, the vibration decreased from 15.3 to 3 mm/s.

5.10 Electrical

5.10.1 Case 15

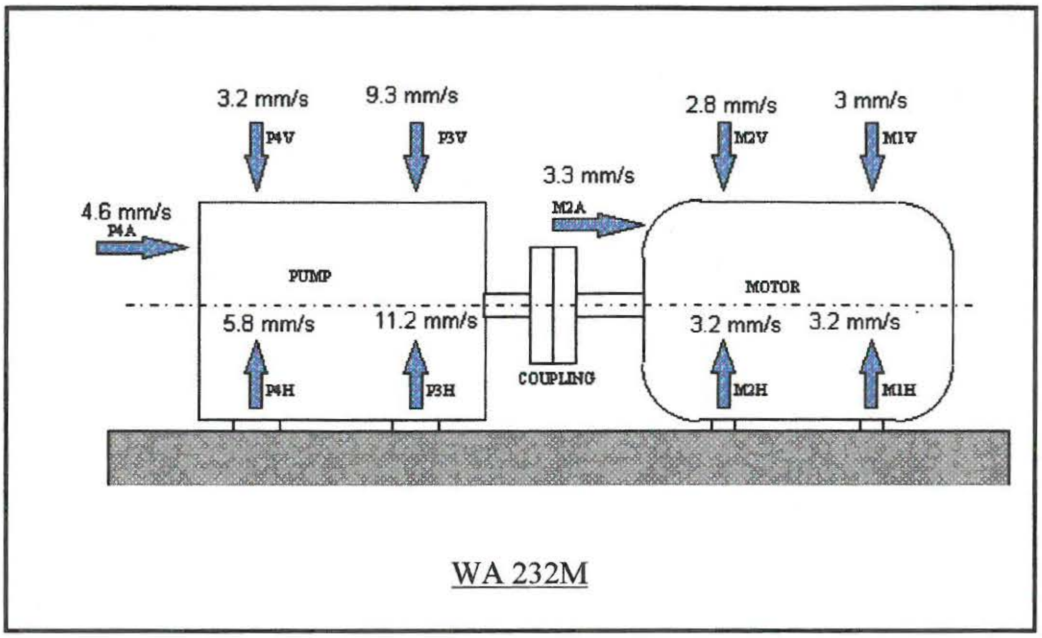


Figure 5-67: Schematic diagram of electrical motor

1. 200 kW, 1480 RPM 4 pole motor.

2. High vibration at 4x rpm..

5.10 Electrical

5.10.1 Case 15

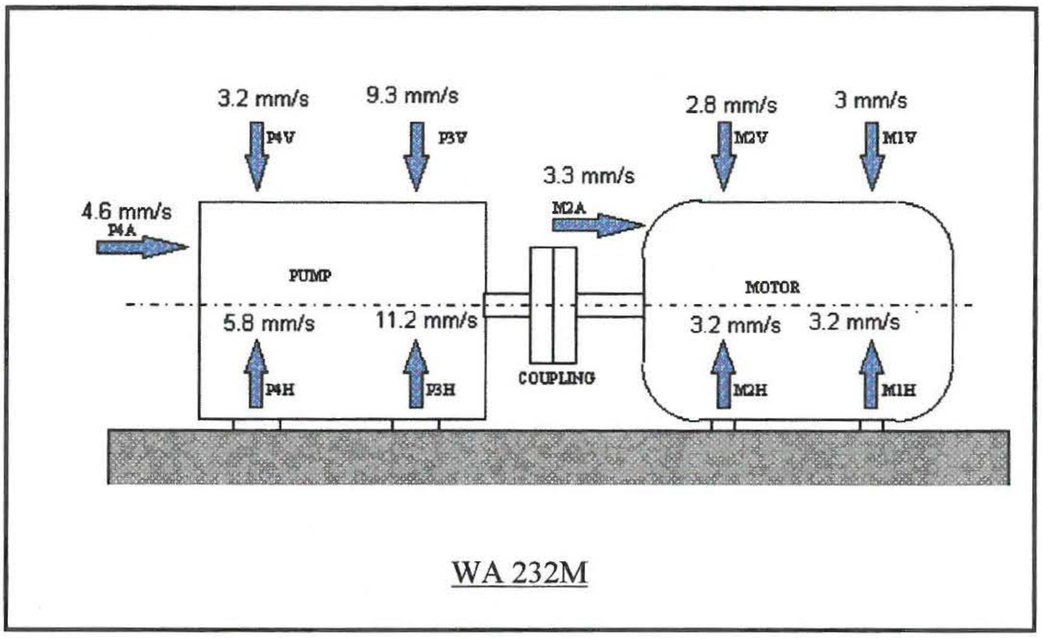


Figure 5-67: Schematic diagram of electrical motor

1. 200 kW, 1480 RPM 4 pole motor.
2. High vibration at 4x rpm..

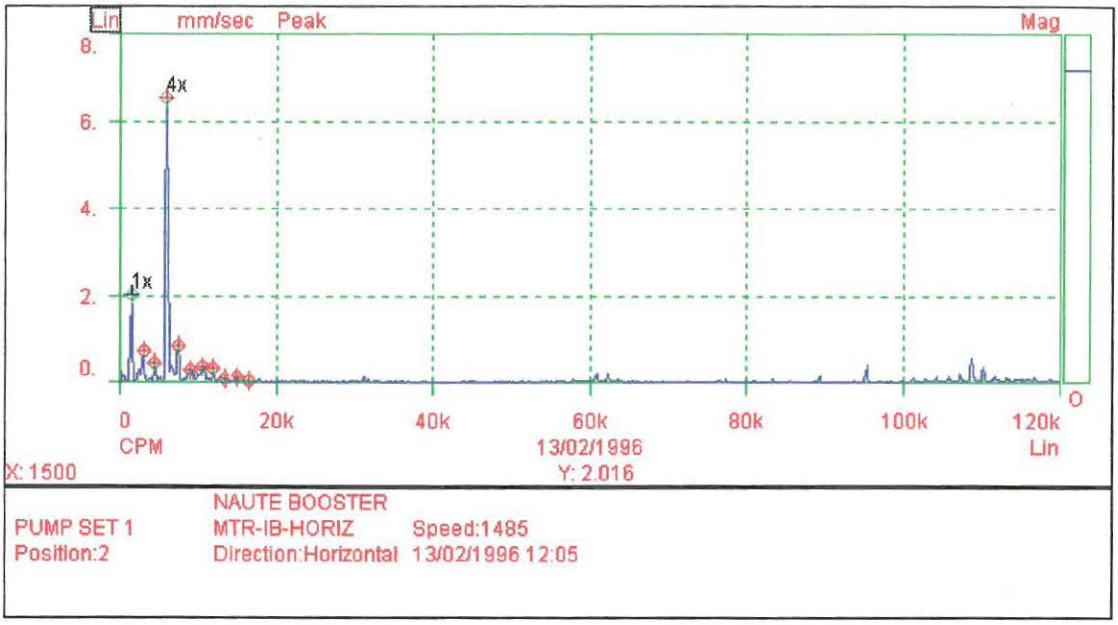


Figure 5-68: Spectrum at point M2H shows a high 4x RPM frequency

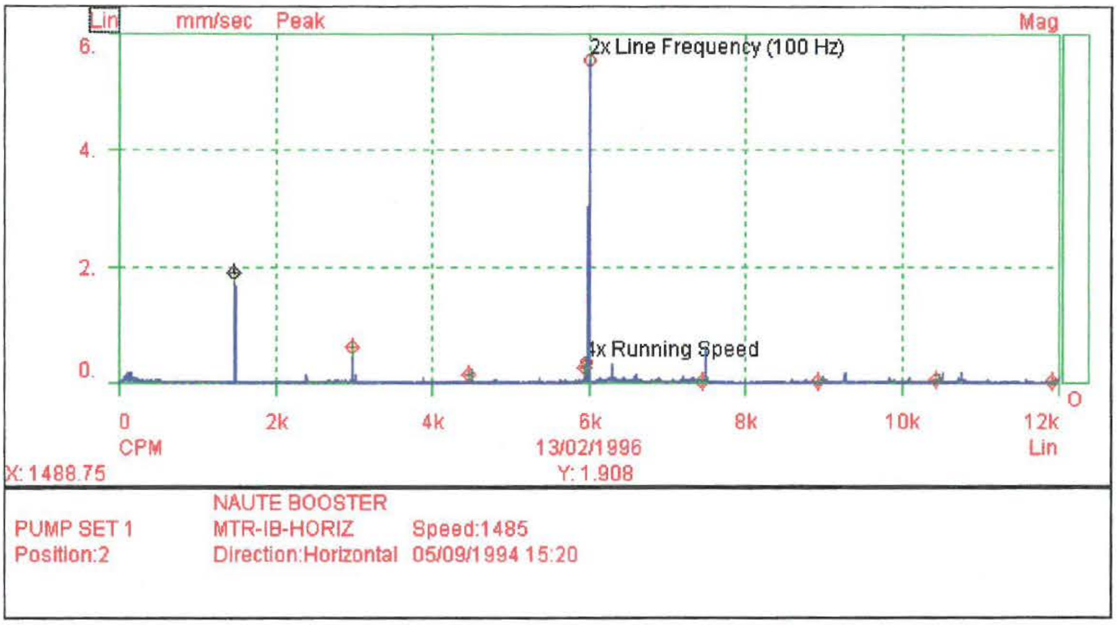


Figure 5-69: Spectrum at point M2H shows a "zoom" spectrum around four times running speed

The spectrum in figure 5-68 shows the motor inboard horizontal (M2H) bearing with a high frequency at 4x running speed. By "zooming" into this frequency (figure 5-69), it is possible to separate the 4x running speed with the 2x line frequency. This indicates a possible electrical problem.

5.10.2 Case 16

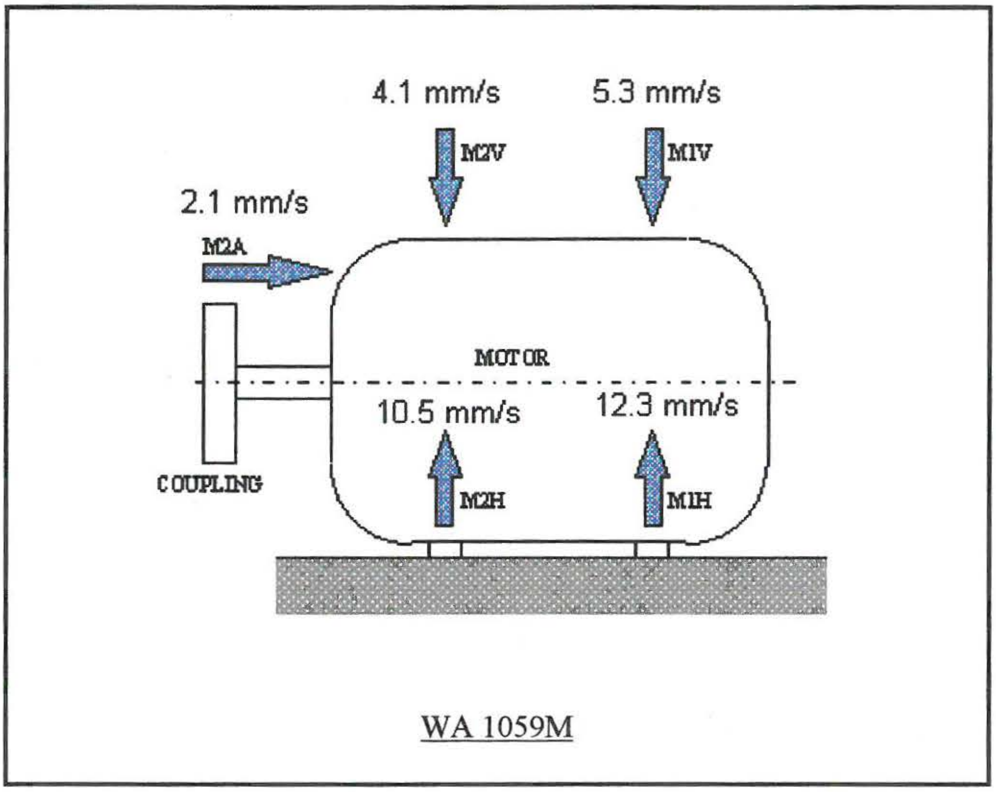


Figure 5-70: Schematic diagram of electrical motor

1. 110 kW WEG, 2950 RPM 2 pole motor.
2. Newly installed motor.
3. From installation the motor experienced vibration
4. A high 2x rpm frequency appearing on all spectrums.

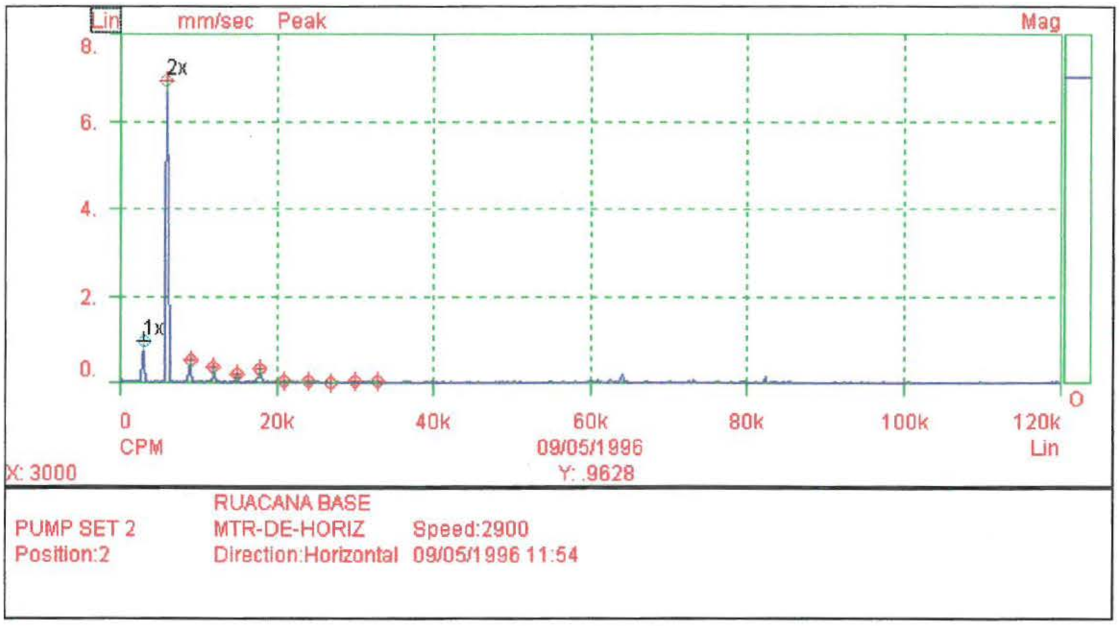


Figure 5-71: Spectrum at point M2H shows a high 2x running speed frequency

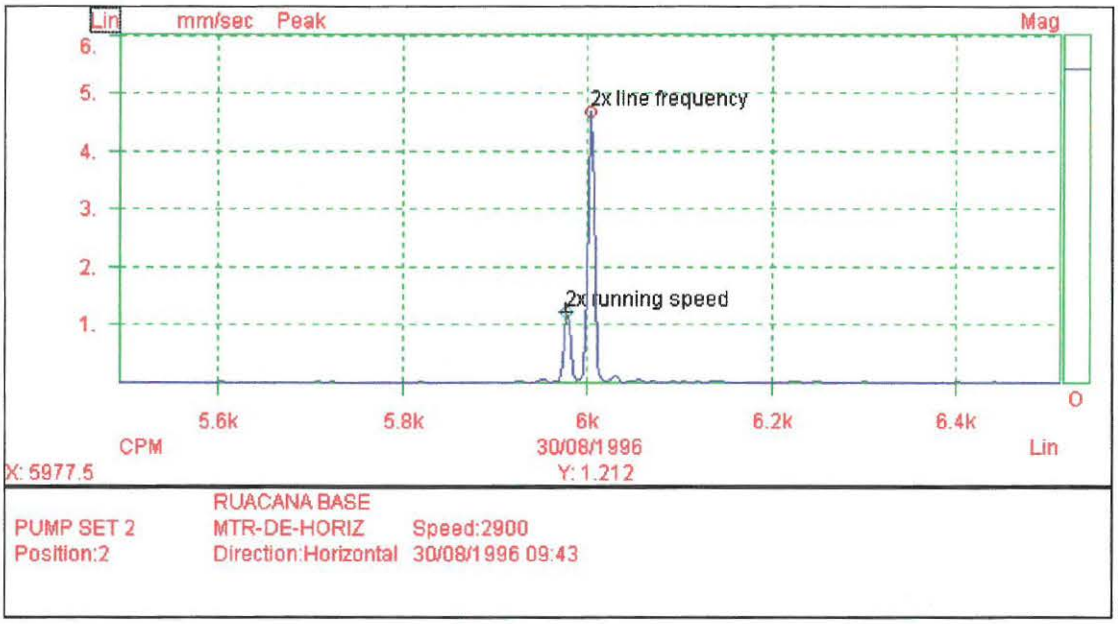


Figure 5-72: Spectrum at point M2H shows a "zoom" spectrum around the two times running frequency

Figure 5-71 shows the spectrum of the motor inboard (M2H) bearing. Notice the high 2x running speed frequency. The spectrum in figure 5-72 shows a zoom spectrum around 2x rpm. It can clearly be seen that it is caused by a 2x line (electrical) frequency component and not the 2x running speed frequency.

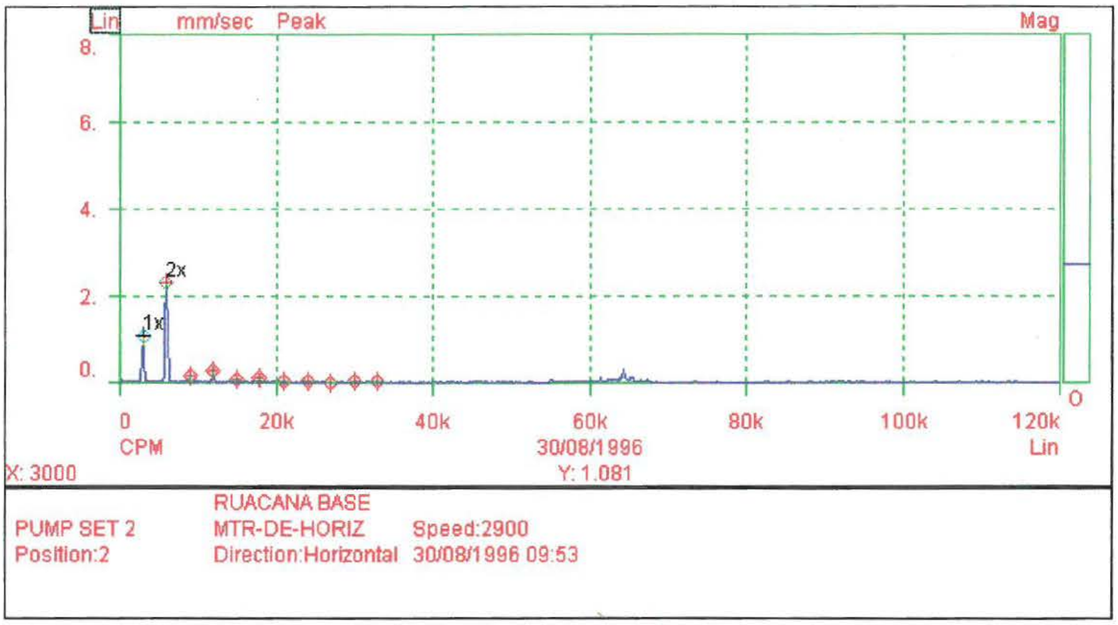


Figure 5-73: Spectrum at point M2H shows the 2x running speed frequency after corrective action was taken

Soft foot checks were performed and it was found that when one of the rear bolts was loosened, the vibration decreased substantially. The spectrums in figure 5-73 show the improvement in vibration. Notice the decrease in the 2x line frequency.

5.10.3 Case 17

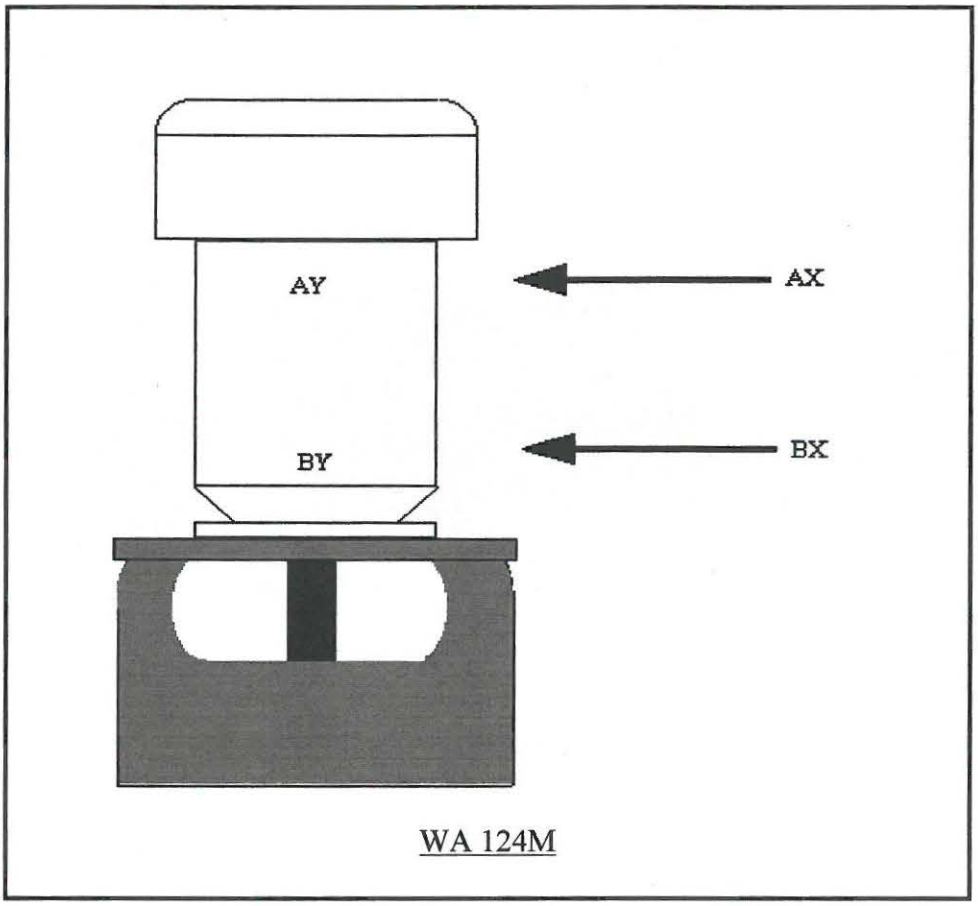


Figure 5-74: Schematic diagram of vertical motor

1. 650 kW, 1485 RPM 4 pole vertical mounted motor.
2. High vibration at 1x rpm.
3. Subsynchronous vibration at approximately 0.6x rpm.
4. Vibration peaks at approximately 120 000 CPM.

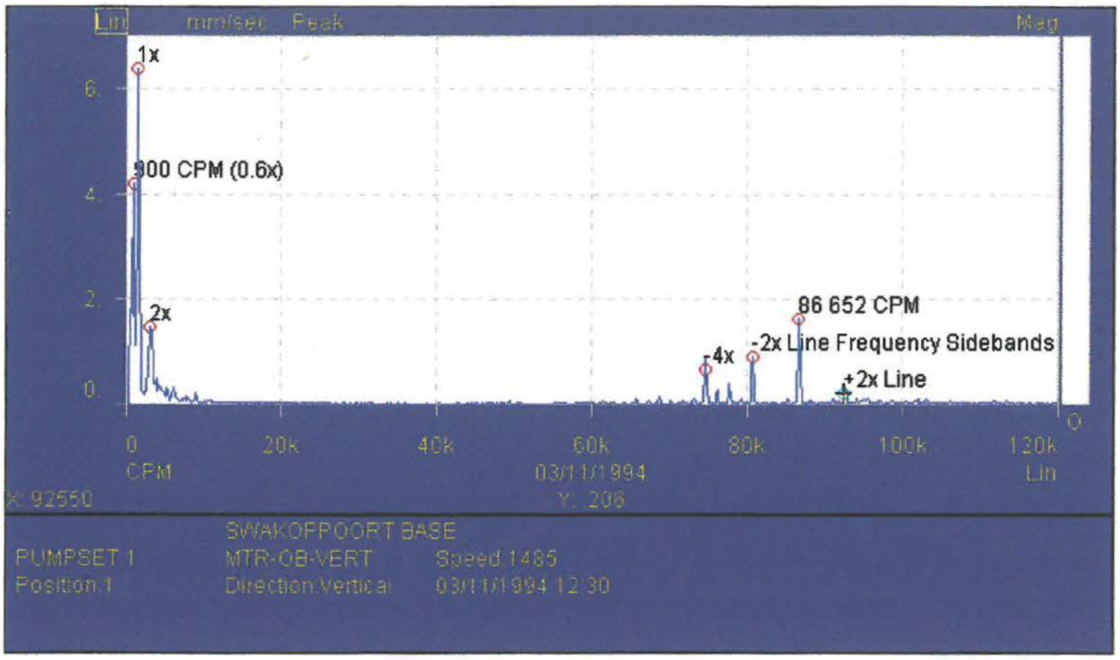


Figure 5-75: Spectrum at point AX shows the rotor bar frequency with 2x line frequency side bands

The above spectrums show the motor outboard vertical (AX) bearing. Notice the rotor bar frequency at 86 652 CPM (85 x 1494) in figure 5-75. Side bands equal to 2x line frequency are also evident. The second and third harmonic of rotor bar frequency was also evident.

5.10.4 Discussion

Although not many electrical problems were experienced during the study, electrical faults can be separated from mechanical problems. Vibration spectrums were examined in great detail and frequencies that were very close together, were differentiated. Areas of interest were centred around multiples of running speed, twice line frequency and rotor bar passing frequency.

Most of the electrical problems detected caused a periodic swing in vibration. The swing was normally caused by an interaction of two frequencies very close together (e.g. 100 Hz and 2x rpm for the 2-pole motors), or by a modulation of the amplitude of vibration at 1x rpm (for eccentric air gap or rotor).

Cases 15 & 16 are examples where the 2x line frequency was detected. The cause of Case 16 was probably a distorted frame caused by a screw base frame, causing an uneven air gap inside the motor which produced the 2x line frequency. The motor in Case 17 was opened and the rotor bar problem was confirmed. The only explanation for the frequency at 0.6x RPM was found to be a possible rub, exciting a subharmonic resonance of a shaft or rotor.

5.11 Flow-Related Problems

5.11.1 Case 18

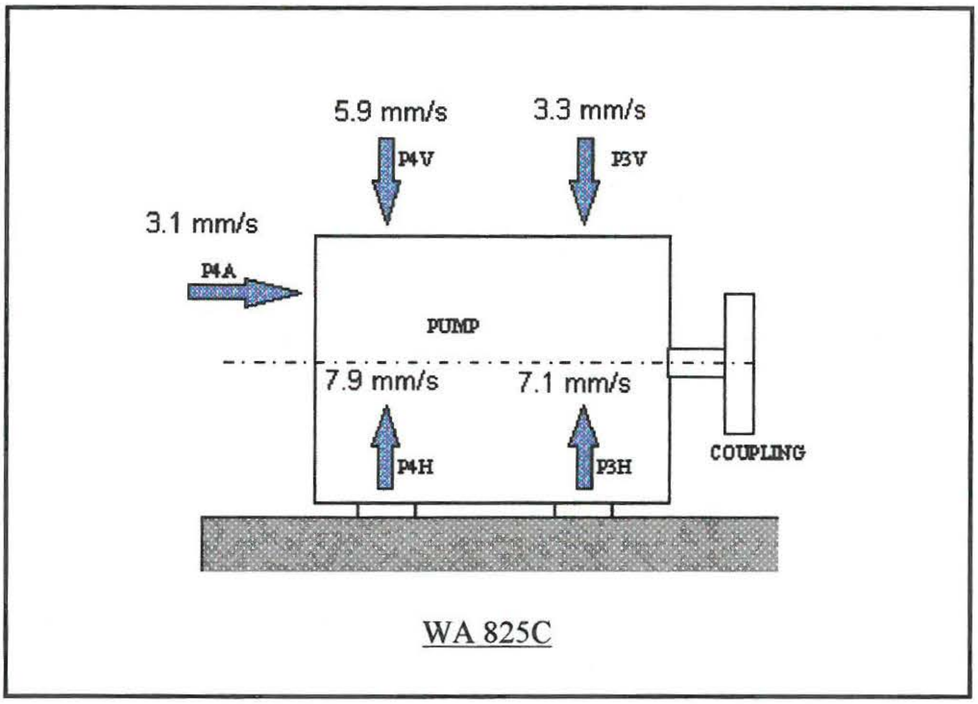


Figure 5-76: Schematic diagram of pump

1. 100 kW, 1485 RPM horizontal split casing pump.
2. High vibration at 5x rpm.

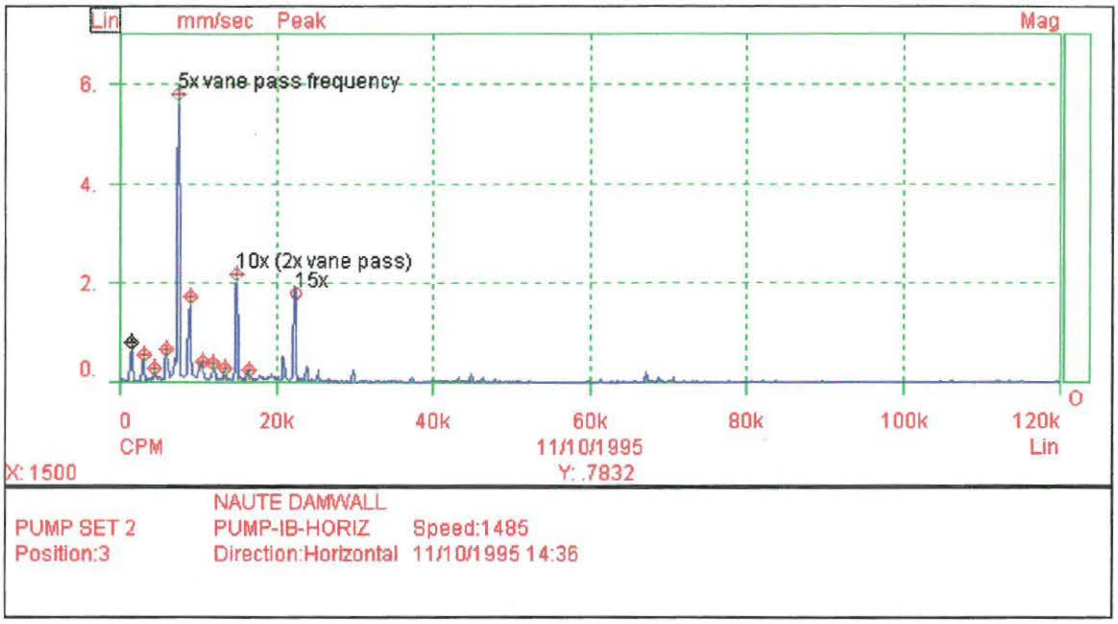


Figure 5-77: Spectrum at point P3H shows the vane pass frequency with its second and third harmonic

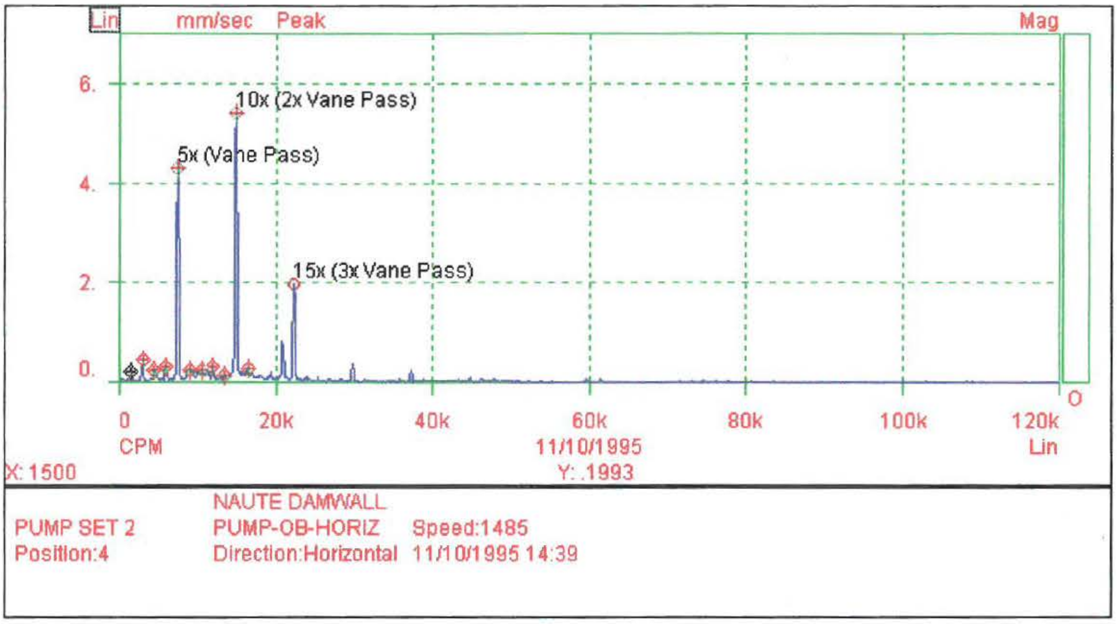


Figure 5-78: Spectrum at point P4H shows the vane pass frequency with its second and third harmonic

The above spectrums show the pump inboard horizontal (M3H) and outboard horizontal (M4H) bearing. Excessive vibration at the number of vanes times rpm (5 x 1485) is visible with its second and third harmonic. This is an indication of a defect on the impeller of the pump.

5.11.2 Case 19

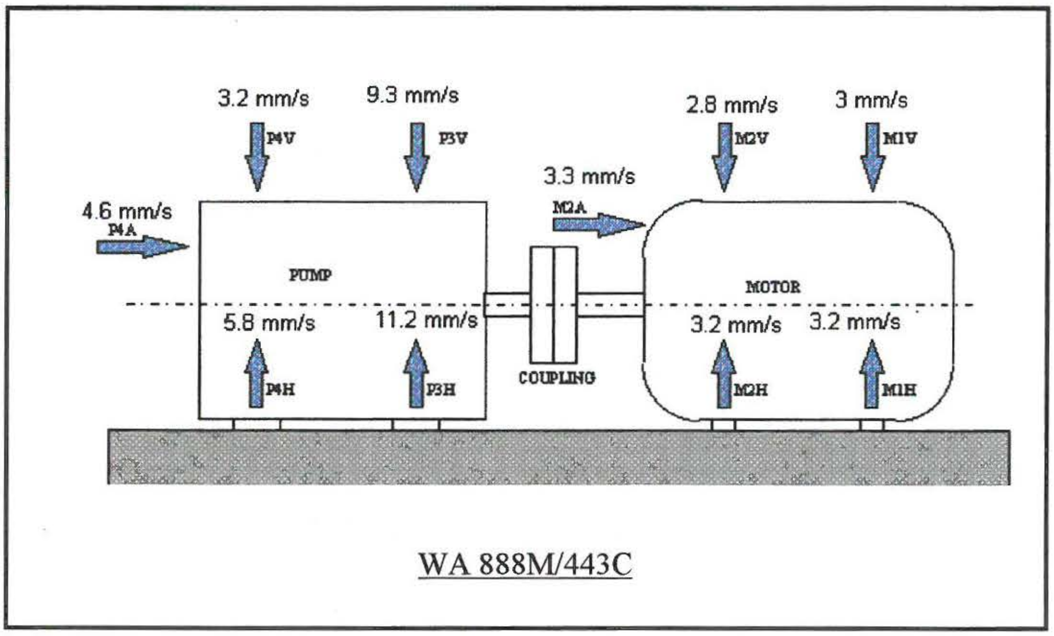


Figure 5-79: Schematic diagram pump set

1. 375 kW, 1485 RPM multi-stage pump.
2. Vibration analysis carried out using IRD 350 Vibration Analyzer with HP plotter.
3. Vibration at 1x rpm with random activity at 9x rpm.

5.11.2 Case 19

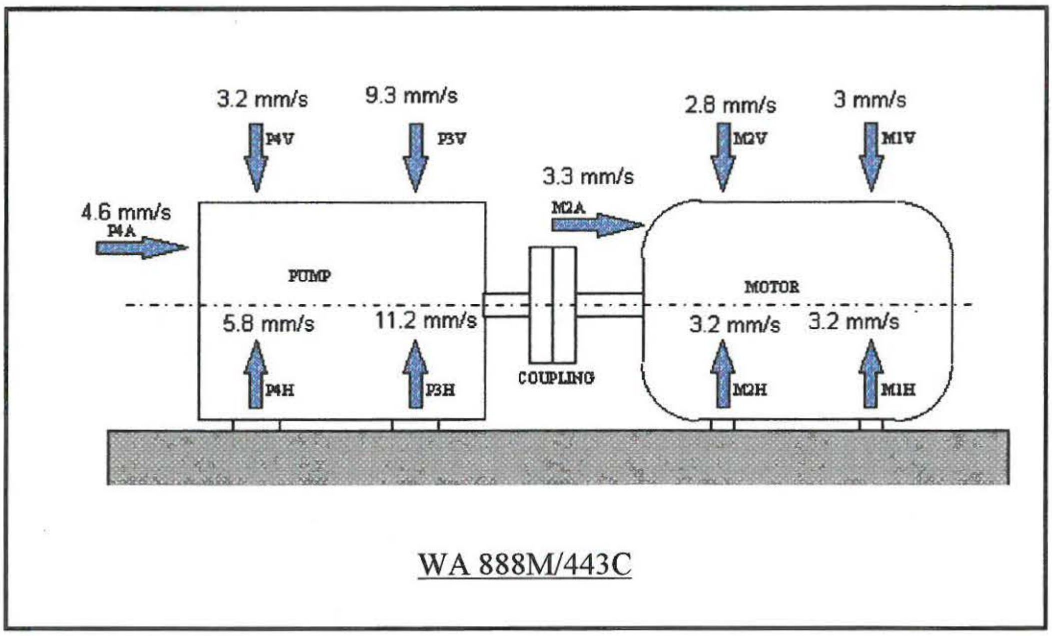


Figure 5-79: Schematic diagram pump set

1. 375 kW, 1485 RPM multi-stage pump.

2. Vibration analysis carried out using IRD 350 Vibration Analyzer with HP plotter.

3. Vibration at 1x rpm with random activity at 9x rpm.

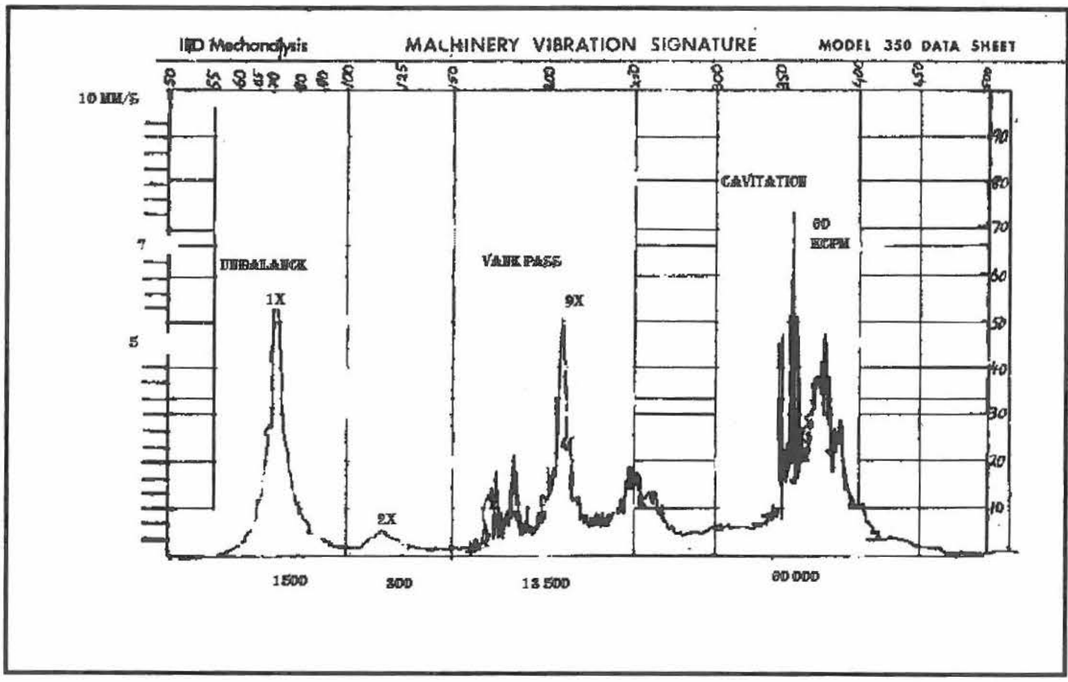


Figure 5-80: Spectrum at point P3V shows the vane pass frequency

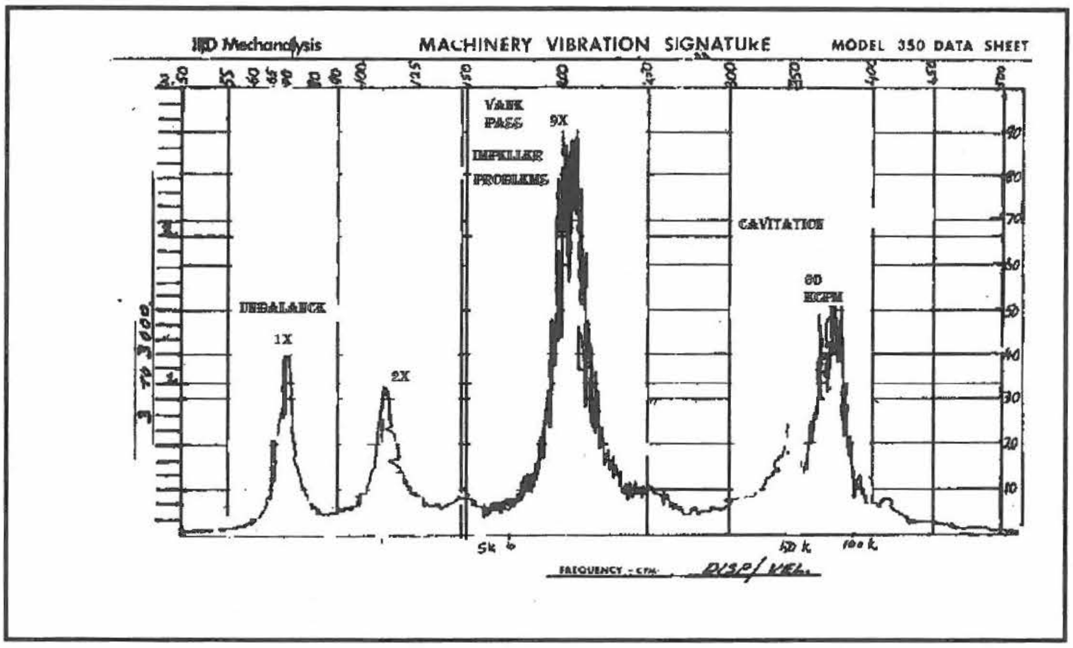


Figure 5-81: Spectrum at point P4V shows the vane pass frequency

The above spectrums show the pump inboard vertical (M3V) and outboard vertical (M4H) bearings. Excessive vibration at the number of vanes times rpm (9 x 1485) is visible. Note the presence of the 1x rpm frequency. This is an indication of a possible defect and unbalance on one of the impellers.

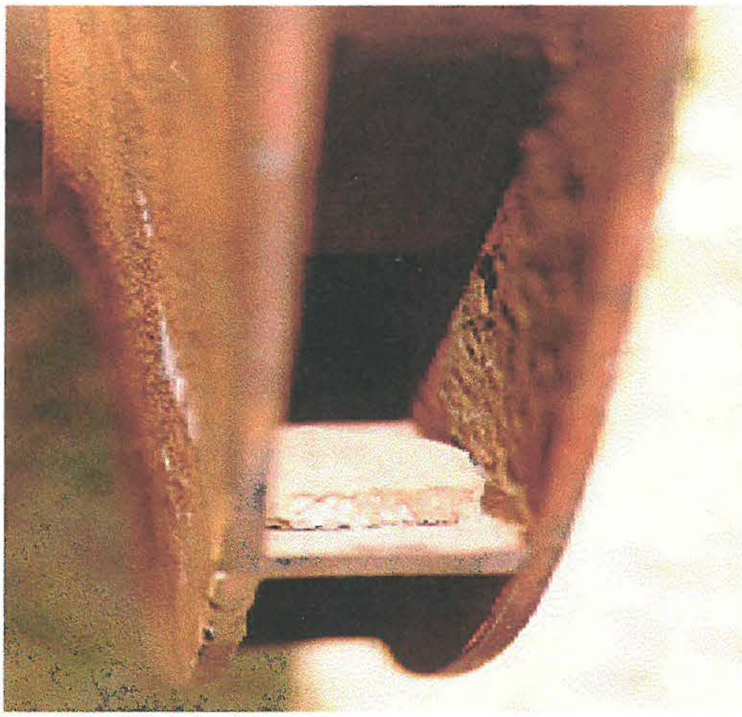


Figure 5-82: Picture shows a small piece of bronze found in the exit of the impeller

Figure 5-83: Picture shows cracks on the impeller shroud

The above pictures show the defects found after the pump was opened. A small piece of bronze (broken off from one of the diffuser vanes) was found in the exit of one of the impeller vanes (Figure 5-82) and it was responsible for the unbalance force generated at 1x rpm. Further investigations showed that the impeller had cracked on its shroud (Figure 5-83) and that some diffuser vanes were damaged.

5.11.3 Case 20

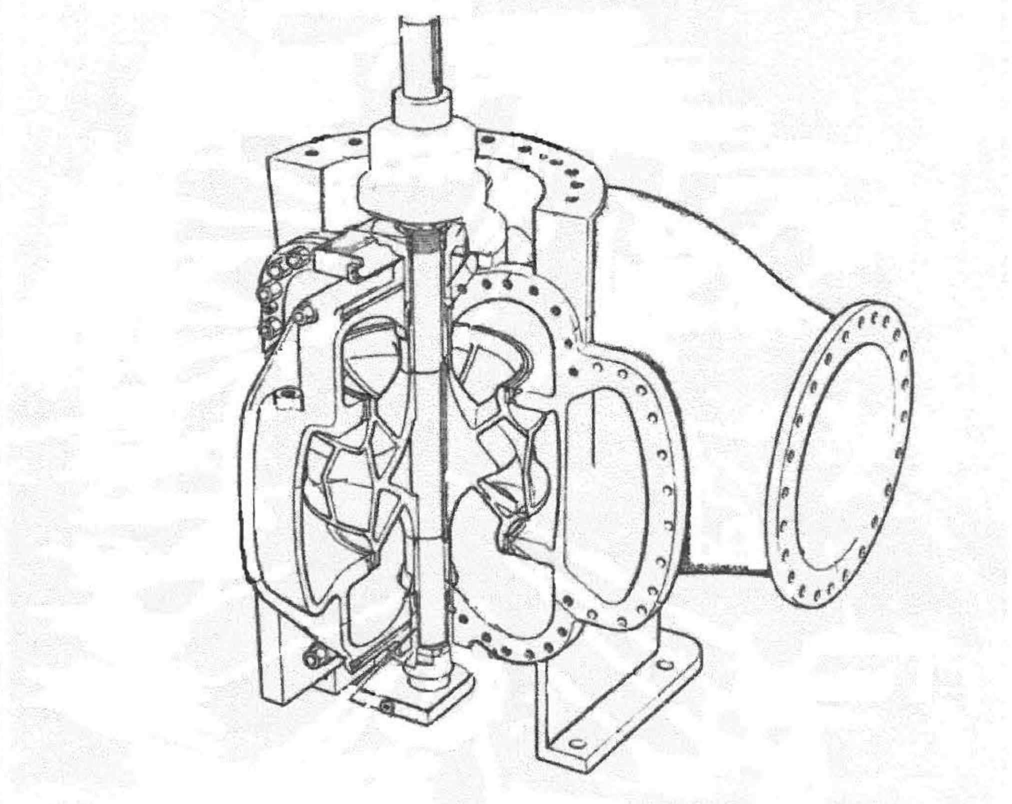


Figure 5-84: Schematic diagram of cut-away section of vertical split casing centrifugal pump

1. 650 kW, 495 RPM vertical low pressure single stage pump.

2. Random vibration.

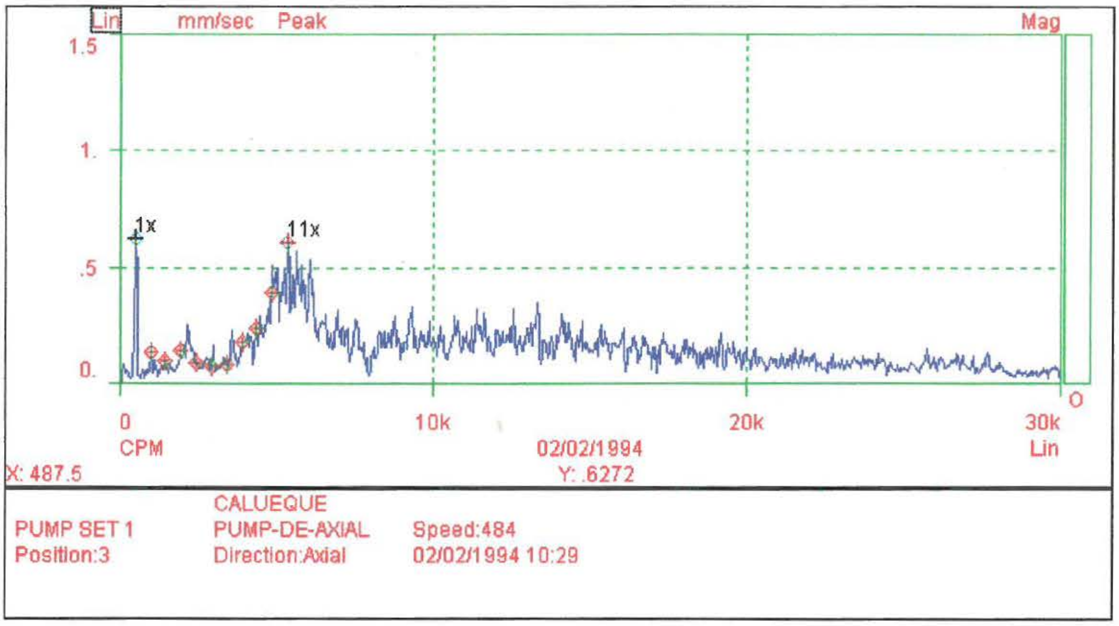


Figure 5-85: Spectrum at point P3A shows cavitation

The above spectrum shows the pump inboard axial (P3A) bearing. Notice the broadband vibration over the whole range of the spectrum. This is a typical example of cavitation (sounds as if sand was pumped). Almost all peaks are non-synchronous.

5.11.4 Case 21

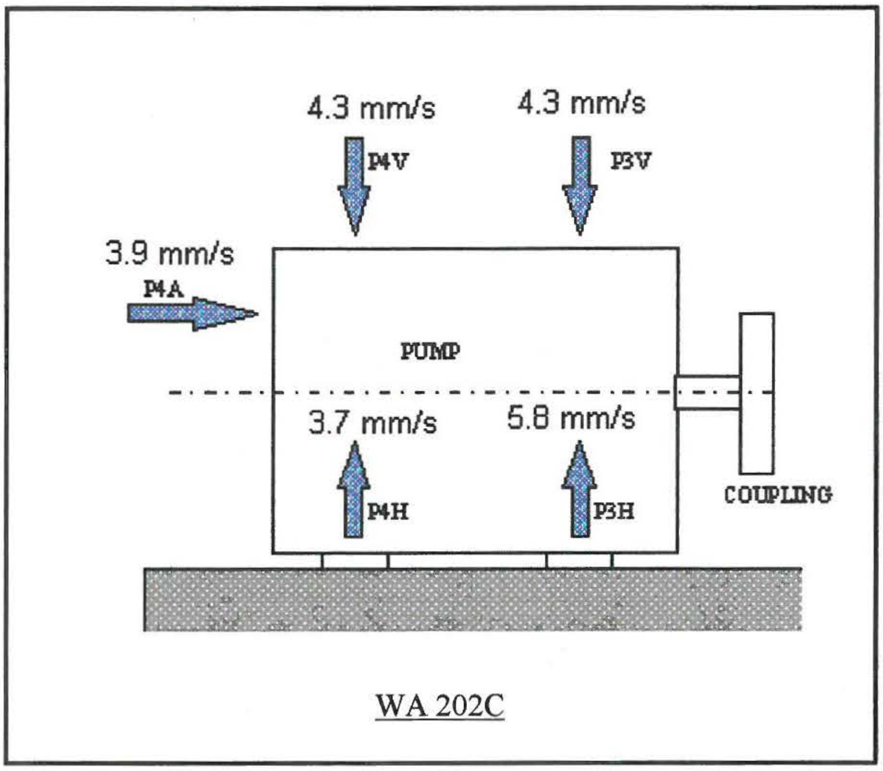


Figure 5-86: Schematic diagram of pump

1. 110 kW, 1485 RPM single stage-pump.
2. Random vibration.

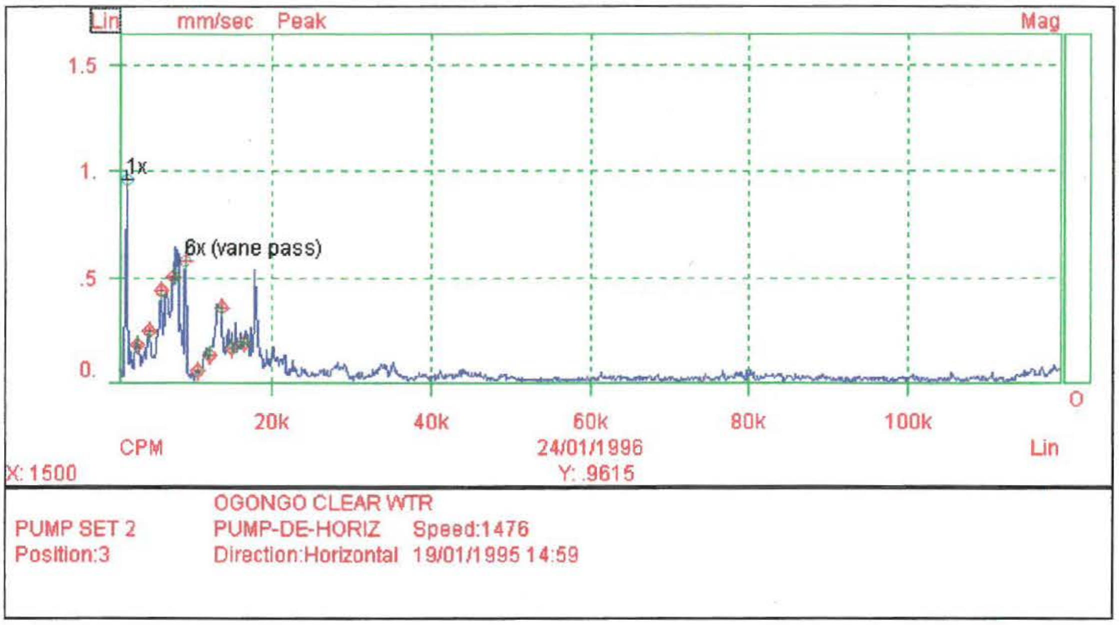


Figure 5-87: Spectrum at point P3H shows cavitation

The above spectrum shows the pump inboard horizontal (M3H) bearing. Notice the broad band vibration between 0 - 30 000 CPM on the spectrum. When the pump was dissembled for visual inspection, it was found that the pump impeller was severely damaged by cavitation.

5.11.5 Discussion

The most common cause of vibration of fluid (water) passing through centrifugal pumps was found to be inherent hydraulic forces caused by cavitation and recirculation. In most cases these vibrations occur at a frequency equal to the number of impeller vanes multiplied by shaft rpm. This vibration is simply the result of pressure pulsations within the pump, created whenever an impeller vane passes a stationary diffuser.

If the impeller is centrally located within the pump housing and is properly aligned with the pump diffusers, the hydraulic pulsations will be balanced and the amplitudes of vibration from inherent hydraulic forces will be minimal.

Vane pass frequencies (rpm x number of vanes) were detected on many of the pumps. The majority was of such low amplitude that it caused no trouble and therefore it was not noticed. However, when the output of the pump was restricted, the fluid flow was disturbed and the vane pass frequency was higher than normal. When it approached a resonant frequency, the vibration had risen well beyond an acceptable limit. Excessive vibration at rpm x vanes could also be caused by a flow rate that was too low. Excessive vane pass vibration amplitude often occurred when the flow rate was lower than approximately 40 percent of the best efficiency flow. An excessively high vane pass frequency could also have meant that the impeller had a defect or there could be a strong possibility that a resonant condition could magnify the vibration.

Case 19 was the first vibration problem detected with vibration analysis. The analysis was made because of a substantial increase in pump vibration and noise. It can be noted that the predominant vibration was found to be 13 365 CPM, which is 9x rpm. This corresponds directly with the number of vanes (9 vanes on impeller), indicating that hydraulic forces are responsible for this vibration. When the vibration suddenly increased, the pump was shut down and opened for visual inspection. This inspection revealed that a piece of one of the diffuser tips had broken off and had been thrown into one of the impeller vanes. It was also noted that the impeller had cracked on its shroud. The rotating element was therefore no longer concentric with the stationary diffuser causing vibration and noise.

An example of a very high vane pass frequency was found on a horizontal split casing pump in Case 18. This centrifugal pump was used for the pumping of raw water to a purification plant. A frequency of 6 mm/s was situated at 7 425 CPM with its second and third harmonic. Tests for resonance were carried out on the pipework, i.e. jacking, but no natural frequencies were found which corresponded to the objectionable 7425 CPM vibration frequency. Another possible cause for the high vane pass vibration generated can be due to too little clearance between the impeller face and pump casing in open face impellers. When the pump was dissembled for inspection, it was discovered that the clearance of the wear ring was so excessive that it resulted in the recirculation of water in the pump, causing this vibration at 5x RPM with harmonics.

Cavitation was detected in Cases 20 & 21 due to pumps operating below their designed capacity and inadequate suction pressure. It was found that these impacts caused by cavitation excited the natural frequencies of the pump housing, impeller and other parts. These symptoms were analysed and appeared as random noise and broad vibration areas of the spectrum. The amplitudes and frequencies were changing constantly and it was found that the amplitudes did not exceed alert levels. It was also noted that broad band vibration peaks at frequencies not related to running speed harmonics were excessively high as resonant frequencies of some parts appeared. The spectrums obtained from cavitation produced very erratic and varied frequencies and low amplitudes with no sharp peaks, but instead the peaks looked like haystacks.

CHAPTER 6

CONCLUSION

6.1 General

Vibration checks were done on 80 pump sets of vital importance for water supply. The pump sets comprised mainly of electric motor/pump assemblies up to 2 400 kilowatt. A 10% fault detection rate occurred over a period of 60 days, and about 8 or 10 items of running equipment were found to be in need of attention. This indicates that regular vibration monitoring leads to improved availability of running equipment. The detection rate will decrease over years if the condition of machines can be improved due to better workmanship and training of maintenance personnel.

6.2 Overall Vibration Measurements

It was found on some overall vibration plots that vibration showed a uniform degradation over time and the values could be used to predict the time of failure. For example, the trend in Figure 6-1 was taken on a pump over a period of two years. The pump was replaced when it reached 8.6 mm/s. The projected time was at 7.5 mm/s. The pump had severe wear and needed an overhaul. It can be seen that the lead-time-to-failure which can be determined using overall-level measurements, is dependent on the rate of deterioration of the machine and the level at which diagnostic or corrective action is considered. On some trending plots no uniform degradation was indicated, but instead it showed a variation in amplitude. This was possibly due to a variation in load, or the machine had not reached operating temperature yet.

When a measurement enters the alert level, no action is taken until the next set of readings are received. If the vibration stays a defect is developing. It was found that in most cases the trending plots were not accurate enough to determine maintenance schedules. However, it was found that when a measurement entered an alarm condition, some impending trouble was present.

An increase in overall vibration is a good indication that a machine is developing a problem, but it cannot be evaluated on its own. Overall vibration measurements can be used for global checks and trend monitoring. The overall vibration measurement provides an estimation of machine condition. It was found in cases of no significant mechanical or operational defect, that the overall vibration was low in amplitude. If the readings are high, or if the trend plot shows an increase, a complete vibration analysis must be carried out to identify the specific cause of vibration.

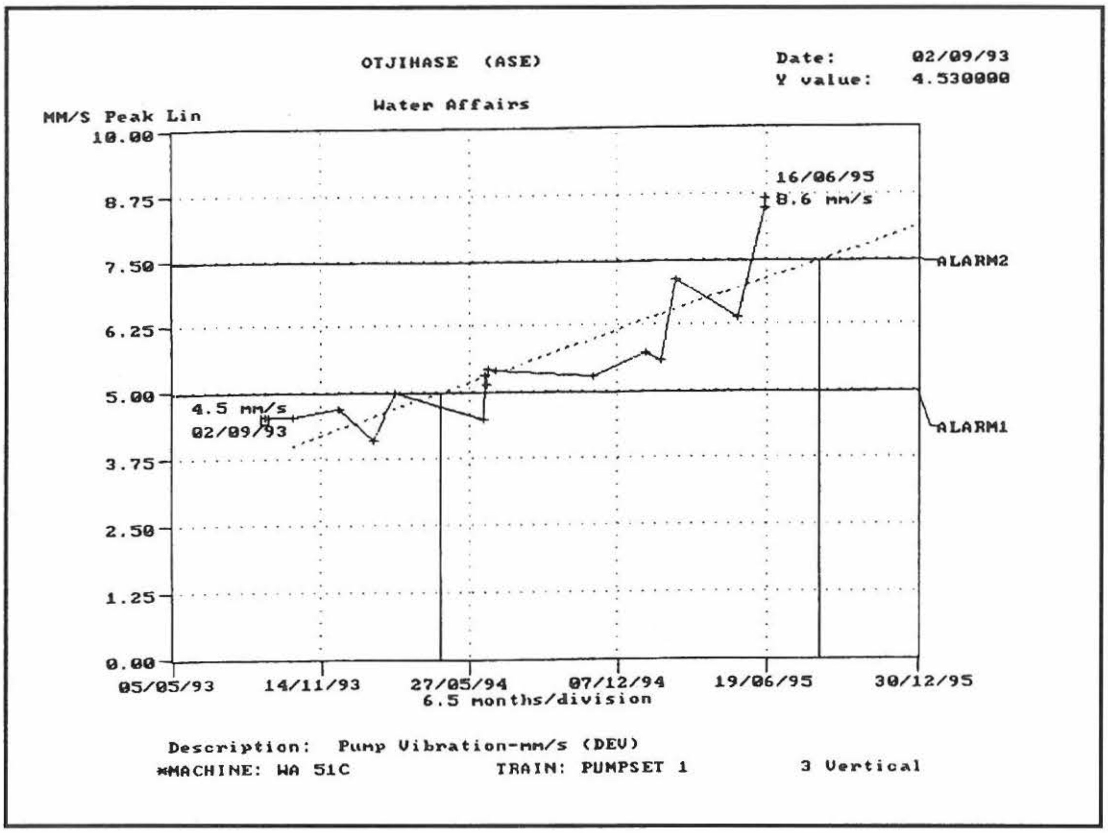


Figure 6-1: Overall trend plot

6.3 Equipment Vibration Review

6.3.1 General

All machines vibrate to some extent. It was noticed in this project that in most cases combinations of unbalance, misalignment and looseness were present. The dominant vibration could be identified and corrected. Time and experience is needed to identify vibration problems.

There are various causes and combinations of causes for vibration. In some cases good alignment was performed, but due to other factors like pipe stress it was not possible to eliminate misalignment. New installations must undergo a full vibration analysis before going into operation. Vibration analysis therefore, has started to be a powerful tool for quality control in the Department of Water Affairs.

The inboard bearing of a pump set is the most prone to deterioration, misalignment and unbalance being the main causes of bearing defects. The inboard bearing is particularly susceptible, presumably because of the contribution from coupling misalignment. Most of the failures are due to failure of the drive-end bearing.

6.3.2 Unbalance and Misalignment

Unbalance was normally observed as the fundamental (1x RPM) frequency component. It was found that other defects could also appear at 1x RPM. Misalignment was found at 2x, 3x and 4x RPM. When a predominant vibration occurred at a frequency of 1x rpm with a very substantial vibration at 2x RPM, misalignment could be suspected. High axial vibration was a good indicator of misalignment, but it could also originate from other sources or be magnified by resonance.

It was found that spectrum analysis must not be evaluated alone in pinpointing unbalance or misalignment but must be used in conjunction with phase analysis. Phase can be a valuable tool (reveals the shaking mode) to help distinguish between an unbalance or a misalignment problem. For phase analysis it was found that the best method was to analyze the machine with the highest vibrations first. Clear symptoms will not always show, because the 1x RPM vibration is not always the result of only the primarily source of the vibration, but is combined with small amounts from other sources. For example, if misalignment exists, a small vibration resulting from an unbalanced or bent shaft can also exist. These sources change the resulting phases slightly and result in shaking modes that are not perfectly in-phase or perfectly 180° out-of-phase. The effect of these other sources usually changes the phase with less than 30°. This would still allow a rotor that was for example, 170° out-of-phase, to be diagnosed as essentially 180° out-of-phase.

With misalignment there is no relationship between vertical vibration and horizontal vibration. For unbalance, the vertical phase difference between one end of the rotor and the other will be similar to the phase difference measured horizontally. In addition to the phase readings, amplitude readings must be compared. If the source is unbalance, then the horizontal amplitude at a bearing would be reasonably similar to the vertical amplitude.

6.3.3 Mechanical Looseness

Mechanical looseness occurs when machinery components do not fit properly. It is characterized by a large number of running speed (1x rpm) harmonics. The highest amplitude typically occurs radically in the vertical direction.

6.3.4 Bearings

The type of vibration spectra that was produced by a damaged bearing, how the vibration was measured and analyzed, and how the results of that analysis could be used to determine the condition of the bearing, were also examined. In applying spectral analysis for the detection of bearing faults, several frequencies can be predicted as being associated with particular defects. One technique which has been tested, is the enveloping acceleration spectrum. It was found to be a useful diagnostic tool in addition to the vibration velocity spectrum.

6.3.5 Resonance

Resonance problems were detected far more than suspected. It was found that resonance magnified the amplitude of vibration. In most cases when the resonance problem was solved, it resulted in a great reduction of vibration. Resonance was detected where vibration in one direction was higher compared to the other direction.

The vibration in Case 15 was magnified by resonance. Although the sound was similar as for cavitation, the vibration spectrums displayed no symptoms of cavitation. This is an example of the importance of vibration analysis to find individual faults.

6.3.6 Electrical Problems

A few cases of vibration which were caused by electrical problems, were detected. Normally this was the result of unequal magnetic forces acting on the rotor or stator. These unequal forces were due to unbalanced phases, unequal air gaps and broken rotor bars.

6.3.7 Flow-Related Problems

A centrifugal pump produces at least two types of flow-related vibrations. The first is found at impeller blade frequency with harmonics and is associated with the inevitable unsteady flow of the pumping fluid through the pump. The second is cavitation, producing broad band noise over the whole range of the vibration spectrum.

6.4 Most common reasons for the increase in vibration

6.4.1 General

Table 6 in Appendix A is an approximation of faults detected in equipment. It was also used, as discussed under the section on cost analysis, for cost-saving purposes. As noticed in the test data of Histogram 1 & 2 (p81), machines can vibrate to some extent. Therefore it is important to determine the baseline or level of vibration for the safe operation of each machine. If the vibration increases from this safe operating zone or baseline, some impending trouble is on its way. One of the problems faced with, was that the equipment monitored was already 5 years and longer in operation. Therefore it took longer than expected to determine baseline levels.

It was noticed during the analysis that in almost all cases combinations of unbalance, misalignment and looseness were present. As clear indications of the existence of specific problems are not always visible, it is difficult to analyze the problems. It was possible though to identify the dominant vibration and correct it. Sometimes, more time and experience was needed to solve a vibration problem.

6.4.2 Most common causes for vibration increase

a) Increase in the 1x rpm vibration

i) Unbalance change
ii) Hold-down bolts are loose, creating a resonant situation
iii) Possible electrical problem such as electrical short in motor armature or field

b) Increase in the 2x, 3x, 4x rpm or other harmonic vibration

i) Coupling alignment change or coupling itself may not be positioned well onto shaft.
ii) Loose bearing or other looseness

c) Higher frequency vibration increase

i) Defective or improperly lubricated rolling element bearings
ii) Erratic, random, very wide frequency range vibration increase and noise
iii) Cavitation developed with usual low amplitudes

6.5 Approaching a vibration problem

When a vibration or an increase in vibration was detected, a mental approach was followed to solve the problem. In most cases elementary corrections were performed, like re-alignment, soft foot checks, adding or removing stiffness on pipes to eliminate resonance. Balancing could not be done, because the Department did not have balancing facilities. Vibration problems, like high vane pass frequencies, require flow rate or resonance corrections to eliminate the problem.

It was mentioned in Case 14 that a sudden increase in vibration occurred in the pump. It was very important to determine if maintenance had been done recently on the equipment. Although maintenance was not performed on the pump itself, maintenance history indicated that one of the other two pumps connected on the same common line had been recently overhauled. The following step was to identify the cause of the problem by measuring vibration on attached pipework, basis and pedestals. Excessive vibration was measured on a portion of the discharge pipe approximately two metres from the pump. A resonant condition was immediately suspected. Concentrating on the high vibration amplitude, all the possible sources that could create a high 1x rpm was investigated. The machine's history was reviewed again, and those items that could have caused vibration at the trouble frequency was marked. For example, if the pump was rebuilt, balancing could be suspected or the re-alignment might have gone wrong.

From the analysis, the cause was determined. After the problem was solved it was found that when they re-installed the other pump and connected the discharge pipe to the common line, they put more strain on the discharge pipe. This caused a change in the stiffness of the common line and changed the stiffness (one of the parameters of resonance) of the pipe by magnifying a resonance about 20% around 1x rpm.

In many cases, recurring machine-train problems can be traced to a system- or process-related problem and not a mechanical problem in the pump set itself. Chronic mechanical problems do exist and can usually be traced to improper application, installation or poor maintenance. Chronic bearing failure on a particular machine or machines should not be ignored. If the bearings fail, other mechanical damages may also be present. Chronic bearing failure is a definite indicator that machine-train life is being adversely affected.

It was found in most instances, that these problems could be solved at a minimal expense, but they are usually ignored and machine failures continue to occur. The net result is that machine life is drastically reduced and maintenance costs are increased.

6.6 Cost Analysis

6.6.1 System and Operating Cost

The direct cost involved to implement vibration analysis for the predictive maintenance system is estimated at N$70 000. The real cost of implementing, maintaining and further upgrading was not simply the cost of the initial system. It included the annual labour (qualified skilled person) and overhead costs associated with acquiring, storing, trending and analysing the data required to determine operating conditions.

6.6.2 Financial Benefits

The success or failure of the programme can be measured by maintenance costs on the equipment. The researcher tried to evaluate the benefits obtained over a 36 month period from January 1992 to January 1995, concerning 81 documented cases (Appendix G). The following case will be used to discuss how these savings were calculated.

Case 6 provides an example of how worksheets (Appendix H) were developed to trend the cost savings via a Machine Improvement Report. The repair savings that were achieved as described in Case 6 by preventing premature motor and pump failure, was N$16 346.

Except for the corrective actions taken on equipment, the main aim of the project was to justify cost for vibration analysis and to achieve savings on the maintenance budget. This included a complete description of the possible extent of damage where the repair savings allowed for actual cost savings as compared to historical cost information.

Costs savings were calculated as achieved in the direct maintenance costs of repairs as a result of machine failure, and the costs which would have arisen if the fault been allowed to develop until a breakdown had occurred.

In some cases the actual cost of the repair from a previous repair (historic cost) was compared with the cost of the most recent (actual) one. The Machine Improvement Report (Appendix H) of Case 6 illustrates that the previous repair cost on the motor was N$7 846. Due to the early detection of a fault, it was repaired at a cost of N$2 500, resulting in a cost saving of N$5 346 on the motor. The same applied to the pump. The calculation of Repair Savings is:

Repair Savings = Historic Cost - Actual Cost

Another way to look at it, was to compare it with actual maintenance costs from workorders received. This was possible by asking the Maintenance Section to calculate the maintenance cost on jobcards completed from January 1992 up to January 1995. Figure 6-2 illustrates the actual maintenance expenses spent on equipment. (This is just the maintenance cost on the 80 pump sets included in the project). Following the curve, it can be seen that the curve shows a decrease in maintenance cost.

Most of the savings were achieved by preventing possible failures and preventing unscheduled overhauls. Minor corrective actions that were submitted to maintenance staff was not included for cost savings. For example, pump sets were re-aligned, thereby preventing further bearing wear and extending machine life.

It should be stressed that this is a minimum cost saving; the consequences of an electric motor failure, which can be very costly, have not been included. For example, a 275 kW motor showed a sudden and significant increase in vibration levels over the previously established baseline. Vibration analysis indicated bearing damage and immediate corrections were recommended. For operational reasons, the motor could not be spared from service and the bearings failed one week later with catastrophic consequences. The rewinding of the motor, which was unavoidable, cost about N$30 000. If corrective actions had been taken by replacing the bearing and eliminating the root cause, the failure could have been avoided. Corrective action would have cost approximately N$3 000 and a cost saving of about N$27 000 could have been achieved. The total cost savings achieved during this period was estimated at N$315 000 (Appendix G).

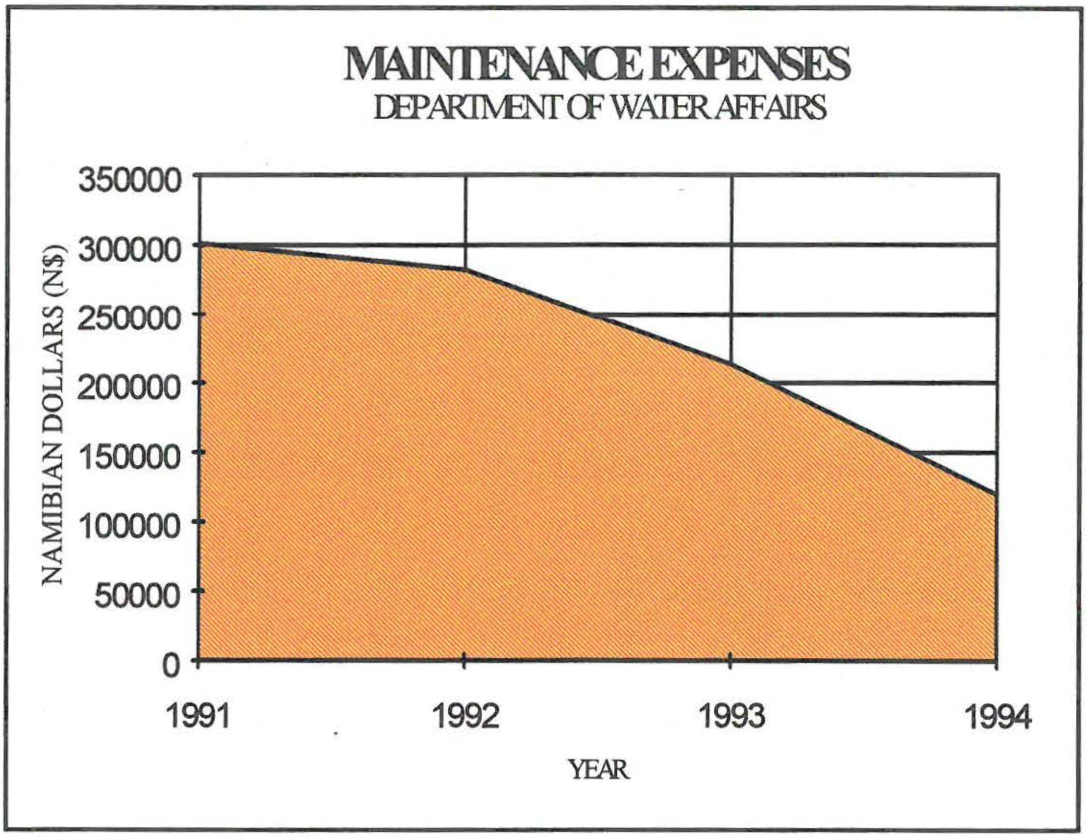


Figure 6-2: Actual cost from Maintenance Records

6.7 Reliability and Accuracy

The monitoring equipment has functioned satisfactorily for a period of three years, demonstrating the practicallity of long term monitoring. It was found that velocity (mm/s) as the parameter to measure vibration was acceptable in the case of the variety of equipment on the project. Decisions made on the condition of plant machinery are based on the data acquired and reported by means of the predictive maintenance system. Therefore, data collection is essential and must be accurate and continuous. During this period various problems were experienced with the analyzer and software. On one occasion the hard disk of the host computer failed and valuable time and data were lost.

To ensure continuous accurate measurements, data must be collected in the same place and direction every time. Data collection points should therefore be clearly marked on each machine. In many cases it was found that data was not collected under the same conditions. This was experienced in some of the data. For example, two pumps running at the same time will give different readings than when one pump is operating alone.

The selected hardware and software were proven in actual field conditions to ensure their reliability. It was important to evaluate the field history of equipment and systems before purchasing. This is a sure way to evaluate the strengths and weaknesses of a particular system before making a capital investment. The system functioned very well despite long distances that had to be travelled to monitor machinery.

6.8 The Predictive Maintenance System

The goals set by the researcher when implementing the monitoring system, such as reduced machine failures, better utilization of available assets and reducing maintenance costs, have been met. This dissertation has detailed a number of examples where the monitoring programme successfully identified and diagnosed machine faults. Not only have a number of possible catastrophic failures been prevented, but in many cases faults had been detected and monitored to ensure that significant damage was avoided without wasting money on premature repairs.

With the implementation of vibration monitoring, corrective maintenance was carried out on most of the machines included in the programme, based on the results of the periodic analysis of machine vibration. A vibration-monitoring system was installed and, if necessary, other available systems can now also be incorporated, such as oil analysis, thermography, etc. as a support to vibration analysis to develop a more complete condition monitoring system.

Due to the long distances to schemes from the maintenance centre, all maintenance, and especially unscheduled failures of equipment, were expensive. By reducing the regular maintenance cost by changing from preventive to predictive maintenance, and by minimizing the occurrence of unscheduled failure, the total expenditure on maintenance could be decreased considerably. This could also result in decreasing the capital expenditure of standby equipment.

Using vibration analysis, a machine's operating condition can be evaluated by using vibration spectrums and phase analysis. By measuring vibration spectra regularly and comparing them with a reference spectrum, faults can be detected in the early stages of failure. Comparison of vibration data shows that a deviation from a normal (baseline) spectrum is an indication of a machine developing trouble. It was noticed that when a machine exceeded its alarm settings, trouble was on its way and action had to be taken to prevent failure.

The vibration monitoring programme is recognized as a viable maintenance strategy in the Department. The project focused both on the technical aspects (tracking vibration levels), and cost savings were achieved. However, vibration levels have very little impact when reviewed by a managerial or financial team. Tracking the costs saved by reducing maintenance costs will be understood by almost anyone in the company. For a successful programme, the future programme must include financial and technical data to survive and produce the best long-term results. The predictive maintenance system as implemented should not only realize financial benefits, but also the importance of a constant water supply to communities. If maintenance can effectively be scheduled (preventing failures), water can be supplied without any interruption.

There are a variety of problems that surround the implementation of a successful condition-monitoring programme. Recent technological advances have served greatly in assisting the researcher with the successful implementation and analysis of machinery problems. What has been learnt from it is that in many cases technology is a double-edged sword, and if not applied with common sense, could lead to confusion and difficulty. In most cases the simple approach works the best, and just because there is a fancy feature does not mean that it has to be used.

It was found that root-cause failure analysis was the most accurate method of isolating the specific machine component that was degrading and also indicated the reason for the degradation. One of the more visible successes achieved in the programme was in identifying problems in time to prevent severe damage. However, to benefit more from the programme, it is not enough to predict failure in time to repair it without effective operation. Pump set failures do not occur without a reason. In order to solve the problem, the reason for premature failure must be found and corrected. If one or more machine-trains exhibit chronic problems, root-cause analysis is essential.

The analysis of vibration data and information was based on water supply equipment and the benefits in other types of plant equipment are expected to be similar with perhaps some change of emphasis. Predictive Maintenance using vibration analysis proved to be a viable tool for detecting faults on electrical motors and pumps.

APPENDIX A

CONDITION-MONITORING TECHNOLOGIES

Monitoring Technique	Equipment Monitored	Short-Term Benefits	Long-Term Benefits
Vibration Analysis	Rotating machinery: pumps, compressors, turbines, gearboxes and diesel/gasoline engines	Identifies deteriorating or defective bearings and gears, looseness, rubs, misalignment and imprecise balancing	Identifies improper alignment or balancing procedures, improper bearing or gear installation practices, excessive operating conditions
Shock Pulse Monitoring	Rotating machinery with anti-friction bearings, motors, large pumps, compressors and turbines	Identifies deteriorating or defective bearings, excessive wear, lack of lubricant	Identifies improper bearing installation practices, improper oil handling or dispensing practices , improper machine operation
Motor Current Signature Analysis	Motors, generators, gearboxes, pumps, air - conditioning equipment and blowers	Identifies worn or damaged bearings, seal/packing wear, misalignment, imprecise balancing	Identifies improper bearing installation practices, improper seal/packing replacement practices, poor alignment or balancing practices
Ultrasonic Analysis	Steam systems, plant air or pneumatic systems, hydraulic systems and vacuum systems	Identifies compressed air, vacuum or steam seal leaks: hydraulic leaks and bearing wear	Identifies improper seal/packing replacement parts, poor equipment operation or design
Thermographic Analysis	Roofs, boilers, furnaces, steam systems, electrical switching, motor control centres, high voltage lines	Identifies roof and insulation leaks, poor electrical contacts or connections, leaking steam traps, refractory deterioration	Identifies poor electrical, roofing, steam systems or insulation repair procedures, excessive operation, poor equipment design
Electrical Surge Comparison	Induction or synchronous motors, dc armatures, synchronous field poles	Identifies turn-to-turn and phase-to-phase shorting, short circuits, reversed or open coils	Identifies improper motor installation or repair, excessive operating condition, design differences between motor manufactures
Oil Analysis	Lubrication/ hydraulic systems, large pumps, compressors and diesel/gasoline engines	Identifies excessive wear of lubricated parts, deterioration of fluid or lubricant condition, the presence of contamination	Identifies improper fluid or lubricant handling or dispensing practices, excessive operating conditions
Wear Particle Analysis	Lubrication/ hydraulic systems, large pumps, compressors and diesel/gasoline engines	Identifies excessive wear of lubricated parts, deterioration of fluid or lubricant condition, the presence of contamination	Identifies improper bearing or gear installation practices, improper oil handling and dispensing practices, excessive operating conditions
Contamination Analysis	Lubrication/ hydraulic systems, large pumps, compressors and diesel/gasoline engines	Identifies excessive gas, liquid or particle contamination	Identifies improper fluid or lubricant or filtration practices, improper installation of seals/packing; poor filter design
Performance Trending	Heat exchangers, pumps, motors, compressors, refrigeration equipment, diesel/gasoline engines	Identifies losses in pumping efficiency, flow rates and heat transfer due to deteriorating components	Identifies improper maintenance or repair practices, poor maintenance design or installation, excessive equipment loading or operation
Visual Observations, Listening and touching	Virtually any machine shown above	Identifies loose or visibly worn parts, oil leaks, excessive noise/vibration and hot/couplings	Identifies improper maintenance or repair practices, poor quality parts and excessive operation

APPENDIX B

DESCRIPTION OF ELECTRIC MOTORS AND PUMPS

SCHEME: **OGONGO CLEAR WATER**

= Motor/Pump number
WA NO. = Water Affairs numbering
Cap = m^3/h, Press = m

B1

#	WA NO.	kW	A	V	RPM	SLOTS	COS θ	MODEL	MAKE	BEARINGS	
										DRIVE	NDE
2	WA 174M	90	172	380	1476	50			HAWKER-SIDLEY	NU 317	6314 / NU320
3	WA 175M	90	172	380	1476	50			HAWKER-SIDLEY	NU 317	6314
4	WA 176M	110	210	380	1479	50			HAWKER-SIDLEY	NU 317	6314
5	WA 177M	110	210	380	1479	50			HAWKER-SIDLEY	NU 317	6314
	WA NO.	**I/V**	**D/V**	**RPM**	**Q**	**PRESS**					
4	WA 202C	6	-----	1476				ETA 200-33	KSB	6411 C3	6411 C3
2	WA 82C	6	-----	1476				ETA 200-33D-	KSB	6411 C3	6411 C3
3	WA 83C	6	-----	1479				ETA 200-33D	KSB	6411 C3	6411 C3
5	WA 85C	6	-----	1479				ETA 200-33	KSB	6411 C3	6411 C3

*I/V = IMPELLER VANES *D/V = DIFFUSER VANES

SCHEME: **OGONGO RAW WATER**

B2

#	WA NO.	kW	A	V	RPM	SLOTS	COS θ	MODEL	MAKE	BEARINGS	
										DRIVE	N/DRIVE
1	WA 328M	15	31.5	380	970		0.82	1LA4 186	Siemens	6310 C3	6210 C3
2	WA 329M	15	31.5	380	970		0.82	1LA4 186	Siemens	6310 C3	6210 C3
3	WA 330M	15	31.5	380	970		0.82	1LA4 186	Siemens	6310 C3	6210 C3
4	WA 331M	30	59	380	980		0.84	1LA4 223	Siemens	6213 C3	6213 C3
5	WA 332M	30	59	380	980		0.84	1LA4 223	Siemens	6213 C3	6213 C3
6	WA 333M	30	59	380	980		0.84	1LA4 223	Siemens	6213 C3	6213 C3
	WA NO.	**I/V**	**D/V**	**RPM**	**CAP**	**PRESS**					
1	WA 210C	6	-----	970				ETA-B150-315	KSB	6409 C3	6409 C3
2	WA 211C	6	-----	970				ETA-B150-315	KSB	6409 C3	6409 C3
3	WA 212C	6	-----	970				ETA-B150-315	KSB	6409 C3	6409 C3
4	WA 213C	6	-----	980				ETA 250-33	KSB	6411 C3	6411 C3
6	WA 215C	6		980				ETA 250-33	KSB	6411 C3	6411 C3
5	WA 216C	6	-----	980				ETA 250-33	KSB	6411 C3	6411 C3

*I/V = IMPELLER VANES *D/V = DIFFUSER VANES

SCHEME: **RUACANA CLEAR WATER**

B3

#	WA NO.	kW	A	V	RPM	SLOTS	COS θ	MODEL	MAKE	BEARINGS	
										DRIVE	NDE
	WA 282M	75		380	2990	44		D 250M	GEC	NU 315 EC	6312 2RS1
	WA 97M	75		380	2990	44		D 250M	GEC	NU 315 EC	6312 2RS1
1	WA 248M	100	165	380	2990			D 280S	KAPAK		
	WA 1059M	110	200	380	2950		0.9	D 280S/M	WEG	6314 C3	6314 C3
2	WA 625M	100	165	380	2990			D 280S	KAPAK		
	WA NO.	**I/V**	**D/V**	**RPM**	**CAP**	**PRESS**					
	WA 326C	7	8	2990	30	60		WKL65-6 NA6	KSB	NU 208 C3	
	WA 683C	7	8	2990	30	60		WKL65-6 NA6	KSB	NU 208 C3	
	WA 271C	7	8	2990	70	380		HPH 18-8-30°	Sulzer	Sleeve	Sleeve
2	WA 272C	7	8	2990	70	380		HPH 18-8-30°	Sulzer	Sleeve	Sleeve
1	WA 551C	7	8	2990	70	380		HPH 18-8-30°	Sulzer	Sleeve	Sleeve

*I/V = IMPELLER VANES *D/V = DIFFUSER VANES

SCHEME: **OSHAKATI-ONDANGWA**

B4

#	WA NO.	kW	A	V	RPM	SLOTS	COS θ	MAKE	MODEL	BEARINGS	
										DRIVE	N/DRIVE
3	WA 319M	15	31.5	380	970		0.82	Siemens	1LA4 186	NU210	6210 2RS
3	WA 320M	45	85	380	1470		0.87	Siemens	1LA4 223	NU213	6213 2RS
4	WA 321M	15	31.5	380	970		0.82	Siemens	1LA4 186	NU210	6210 2RS
4	WA 322M	45	850	380	1470		0.87	Siemens	1LA4 223	NU213	6213 2RS
5	WA 323M	45	85	380	1470		0.87	Siemens	1LA4 223	NU213	6213 C3
6	WA 324M	55	108	380	980		0.85	Siemens	1LA4 281	NU217	6217
	WA NO.	**I/V**	**D/V**	**RPM**	**CAP**	**PRESS**					
3	WA 193C	6	-----	970/1485	32	32		KSB	ETA B150-315	6409 C3	6409 C3
4	WA 194C	6	-----	970/1485	32	32		KSB	ETA B150-315	6409 C3	6409 C3
5	WA 195C	6	-----	1485	32	32		KSB	ETA B150-315	6409 C3	6409 C3
6	WA 196C	6	-----	1485	32	32		KSB	ETA B150-315	6409 C3	6409 C3

* I/V = IMPELLER VANES * D/V = DIFFUSER VANES

SCHEME: **OSHAKATI**

B5

#	WA NO.	kW	A	V	RPM	SLOTS	COS θ	MAKE	MODEL	BEARINGS	
										DRIVE	NDE
1	WA 884M	55	31.5	380	1480		0.82	GEC	D 250S	NU 315	6315 2RS
2	WA 885M	55	31.5	380	1480		0.82	GEC	D 250S	NU 315	6315 2RS
3	WA 886M	55	31.5	380	1480		0.82	GEC	D 250S	NU 315	6315 2RS
	WA NO.	**I/V**	**D/V**	**RPM**	**CAP**	**PRESS.**					
3	WA 960C	6	-----	1480				KSB	ETA NEW 125-40C	6409 C3	6409 C3
1	WA 961C	6	-----	1480				KSB	ETA NEW 125-40C	6409 C3	6409 C3
2	WA 962C	6	-----	1480				KSB	ETA NEW 125-40C	6409 C3	6409 C3

* I/V = IMPELLER VANES * D/V = DIFFUSER VANES

SCHEME: **OTJIHASE**

B6

#	WA NO.	kW	A	V	RPM	SLOTS	COS θ	MAKE	MODEL	BEARINGS	
										DRIVE	N/DRIVE
1	WA 511M	270	485	380	2990	40		Siemens	1LA4 356	NU 215	NU215 / 6215 C4
2	WA 512M	270	485	380	2990	40		Siemens	1LA4 356	NU 215	NU215 / 6215 C4
3	WA 307M	270	485	380	2990	40		Siemens	1LA4 356	NU 215	NU215 / 6215 C4
	WA NO.	**I/V**	**D/V**	**CAP**	**RPM**	**PRESS**					
1	WA 51C	7	6	146	2990	380		Sulzer	HPH 24-12½-22½-7	SLEEVE	SLEEVE
2	WA 52C	7	6	146	2990	380		Sulzer	HPH 24-12½-22½-7	SLEEVE	SLEEVE
3	WA 57C	7	6	146	2990	380		Sulzer	HPH 24-12½-22½-7	SLEEVE	SLEEVE

* I/V = IMPELLER VANES * D/V = DIFFUSER VANES

SCHEME: **NAUTE**

B7

#	WA NO.	kW	A	V	RPM	SLOTS	COS θ	MAKE	MODEL	BEARINGS DRIVE	BEARINGS NDE
1	WA 224M	200	360	380	1485		0.88	Siemens	1LA2 246	NU 320	6318
2	WA 231M	200	360	380	1485		0.88	Siemens	1LA2 246	NU 320	6318
3	WA 427M	200	375	380	1483		0.85	Siemens	1LA0 313	NU 320	NU219 / 6219
2	WA 225M	200	360	380	1485		0.88	Siemens	1LA2 246	NU 320	6318
1	WA 232M	200	360	380	1485		0.88	Siemens	1LA2 246	NU 320	NU219 / 6219
3	WA 428M	200	375	380	1483		0.85	Siemens	1LA0 313	NU 320	NU219 / 6219
1	WA 163M	110	205	380	1485	40	0.87	Siemens	1LA4 310	NU 219	6219 C4
3	WA 165M	110	205	380	1485	40	0.87	Siemens	1LA4 310	NU 219	6219 C4
	WA NO.	**I/V**	**D/V**	**RPM**	**CAP**	**PRESS**					
3	WA 235C	7	12	1450	203	207		KSB	WL 150/6	NU 410	NU 410
1	WA 251C	7	12	1450	203	202		KSB	WL 150/6	NU 410	NU 410
2	WA 252C	7	12	1450	203	202		KSB	WL 150/6	NU 410	NU 410
3	WA 236C	7	12	1450	203	207		KSB	WL 150/6	NU 410	NU 410
2	WA 258C	7	12	1450	203	202		KSB	WL 150/6	NU 410	NU 410
1	WA 555C	7	12	1450	203	202		KSB	WL 150/6	NU 410	NU 410
3	WA 825C	5	-----	1470	155	27		KSB	RDL 150-400A	NU 308	6308
1	WA 827C	5	-----	1470	155	27		KSB	RDL 150-400A	NU 308	6308

* I/V = IMPELLER VANES * D/V = DIFFUSER VANES

SCHEME: **ROSH PINAH**

B8

#	WA NO.	kW	A	V	RPM	SLOTS	COS θ	MAKE	MODEL	BEARINGS DRIVE	BEARINGS NDE
3	WA 656M	160	305	380	1485		0.84	Siemens	1LA4 314	6314 C3	6214 C3
2	WA 660M	160	305	380	1485		0.84	Siemens	1LA4 314	6314 C3	6214 C3
1	WA 661M	160	305	380	1485		0.84	Siemens	1LA4 314	6314 C3	6214 C3
3	WA 657M	160	305	380	1485		0.84	Siemens	1LA4 314	NU 319	6219 C3
2	WA 658M	160	305	380	1485		0.84	Siemens	1LA4 314	NU 319	6219 C3
1	WA 659M	160	305	380	1485		0.84	Siemens	1LA4 314	NU 319	6219 C3
	WA NO.	**I/V**	**D/V**	**RPM**	**CAP**	**PRESS**					
3	WA 677C	6	5	1485	100	253		Harland Salweir	MRB 6	Sleeve	Sleeve
1	WA 680C	6	5	1485	100	253		Harland Salweir	MRB 6	Sleeve	Sleeve
2	WA 681C	6	5	1485	100	253		Harland Salweir	MRB 6	Sleeve	Sleeve
1	WA 676C	6	5	1485	100	253		Harland Salweir	MRB 6	Sleeve	Sleeve
3	WA 678C	6	5	1485	100	253		Harland Salweir	MRB 6	Sleeve	Sleeve
2	WA 679C	6	5	1485	100	253		Harland Salweir	MRB 6	Sleeve	Sleeve

* I/V = IMPELLER VANES * D/V = DIFFUSER VANES

SCHEME: **VON BACH PURIFICATION**

B9

#	WA NO.	kW	A	V	RPM	SLOTS	COS θ	MAKE	MODEL	BEARINGS DRIVE	BEARINGS NDE
1	WA 89M	250	445	380	1483		0.87	Siemens	1LA4 317	NU 320	NU220 / 6220
2	WA 157M	250	445	380	1483		0.87	Siemens	1LA0 317	NU 320	NU220 / 6220
3	WA 158M	250	445	380	1483		0.87	Siemens	1LA0 317	NU 320	NU220 / 6220
2	WA 514M	55	102	380	1485		0.88	Siemens	1LA4 252		
1	WA 515M	55	102	380	1485		0.88	Siemens	1LA4 252		
	WA NO.	**I/V**	**D/V**	**RPM**	**CAP**	**PRESS**					
2	WA 135C							Matter & Platt	8”-10”		
1	WA 136C							Matter & Platt	8”-10”		
3	WA 141C							Matter & Platt	8”-10		
2	WA 128C	5	-----	1485				KSB	ETA 250-29	6411	6411
1	WA 129C	5	-----	1485				KSB	ETA 250-29	6411	6411

* I/V = IMPELLER VANES * D/V = DIFFUSER VANES

SCHEME: **SWAKOPOORT**

B10

#	WA NO.	kW	A	V	RPM	SLOTS	COS θ	MAKE	MODEL	BEARINGS	
										DRIVE	NDE
1	WA 124M	560	165	3300	1480	58		Boyd Brown	DH 400LD	7324 B	6322
2	WA 561M	560	165	3300	1480	58		Boyd Brown	DH 400LD	7324 B	6322
	WA M	560	165	3300	1480	58		Boyd Brown	DH 400LD	7324 B	6322
	WA M	560	165	3300	1480	58		Boyd Brown	DH 400LD	7324 B	6322
	WA NO.	**I/V**	**D/V**	**RPM**	**CAP**	**PRESS**					
	WA 69C							APE	18 HC	Sleeve	Sleeve
2	WA 44C							APE	18 HC	Sleeve	Sleeve
1	WA 48C							APE	18 HC	Sleeve	Sleeve

* I/V = IMPELLER VANES * D/V = DIFFUSER VANES

SCHEME: **VON BACH-WINDHOEK**

B11

#	WA NO.	kW	A	V	RPM	SLOTS	COS θ	MAKE	MODEL	BEARINGS	
										DRIVE	NDE
1	WA 105M	1900	125	11000	987		0.9	Toshiba	TIKE		
2	WA 106M	1900	125	11000	987		0.9	Toshiba	TIKE		
1	WA 108M	1900	125	11000	987		0.9	Toshiba	TIKE		
2	WA 109M	1900	125	11000	987		0.9	Toshiba	TIKE		
1	WA 13M	1900	125	11000	987		0.9	Toshiba	TIKE		
2	WA 14M	1900	125	11000	987		0.9	Toshiba	TIKE		
	WA NO.	**I/V**	**D/V**	**RPM**	**CAP**	**PRESS**					
1	WA 33C	7	8	987		253		Sulzer	BPK 62-360	SLEEVE	SLEEVE
2	WA 35C	7	8	987		253		Sulzer	BPK 62-360	SLEEVE	SLEEVE
1	WA 159C	7	8	987		253		Sulzer	BPK 62-360	SLEEVE	SLEEVE
2	WA 160C	7	8	987		253		Sulzer	BPK 62-360	SLEEVE	SLEEVE
1	WA 66C	7	8	987		253		Sulzer	BPK 62-360	SLEEVE	SLEEVE
2	WA 68C	7	8	987		253		Sulzer	BPK 62-360	SLEEVE	SLEEVE

* I/V = IMPELLER VANES * D/V = DIFFUSER VANES

SCHEME: **OMATAKO**

B12

#	WA NO.	kW	A	V	RPM	SLOTS	COS θ	MAKE	MODEL	BEARINGS	
										DRIVE	N/DRIVE
2	WA 626M	2400		11000	985			Toshiba	TIKE	6314 C3	6214 C3
1	WA 627M	2400		11000	985			Toshiba	TIKE	6314 C3	6214 C3
2	WA 628M	2400		11000	985			Toshiba	TIKE	6314 C3	6214 C3
1	WA 629M	2400		11000	985			Toshiba	TIKE		
	WA NO.	**I/V**	**D/V**	**RPM**	**CAP**	**PRESS**					
2	WA 641C	7	15	985	3600	175		Sulzer	HPDH 450-690-3D	NU 334 M	2X 7334BGM
1	WA 642C	7	15	985	3600	175		Sulzer	HPDH 450-690-3D	NU 334 M	2X 7334BGM
2	WA 643C	7	15	985	3600	175		Sulzer	HPDH 450-690-3D	NU 334 M	2X 7334BGM
1	WA 644C	7	15	985	3600	175		Sulzer	HPDH 450-690-3D	NU 334 M	2X 7334BGM

* I/V = IMPELLER VANES * D/V = DIFFUSER VANES

SCHEME: **CALUEQUE**

B13

#	WA NO.	kW	A	V	RPM	SLOTS	COS θ	MAKE	MODEL	BEARINGS	
										DRIVE	NDE
1	WA 292M	650	146	11000	484		0.82	Brown Boveri	OWVG 5609 B12		
2	WA 293M	650	146	11000	484		0.82	Brown Boveri	OWVG 5609 B12		
	WA NO.	**I/V**	**D/V**	**RPM**	**CAP**	**PRESS**					
1	WA 244C			484	9000	27			**WKLN 65/6**	MIT-CHELL	NU322E TVPC3
2	WA 260C			484	9000	27			**WKLN 65/6**	MIT-CHELL	NU322E TVPC3

* I/V = IMPELLER VANES * D/V = DIFFUSER VANES

SCHEME: **SWAKOP-RöSSING BASE**

B14

#	WA NO.	kW	A	V	RPM	SLOTS	COS θ	MAKE	MODEL	BEARINGS	
										DRIVE	NDE
1	WA 444M	315	560	380	1490	56	0.89	Siemens	1LA4 404	NU 224	6220
2	WA 655M	375	690	380	1488	56	0.86	Siemens	1LA4 404	NU 324	NU320 / 6220
3	WA 887M	375	690	380	1488	56	0.86	Siemens	1LA4 404	NU 324	NU320 / 6220
4	WA 443M	315	560	380	1490	56	0.89	Siemens	1LA4 404	NU 224	6220
	WA NO.	I/V	D/V	RPM	CAP	PRESS					
1	WA 435C	9	8	1490	390	460		Sulzer	HPL 42-22.5 30°	Sleeve	Sleeve
2	WA 442C	9	8	1490	390	460		Sulzer	HPL 42-22.5 30°	Sleeve	Sleeve
3	WA 449C	9	8	1490	390	460		Sulzer	HPL 42-22.5 30°	Sleeve	Sleeve
4	WA 447C	9	8	1490	390	460		Sulzer	HPL 42-22.5 30°	Sleeve	Sleeve

* I/V = IMPELLER VANES   * D/V = DIFFUSER VANES

SCHEME: **SWAKOP-RöSSING BOOSTER1**

B15

#	WA NO.	kW	A	V	RPM	SLOTS	COS θ	MAKE	MODEL	BEARINGS	
										DRIVE	NDE
1	WA 434M	315	560	380	1490	56	0.89	Siemens	1LA4 404	NU 224	6220
2	WA 888M	375	690	380	1488		0.86	Siemens	1LA0 357	NU 324	NU320 /6220
3	WA 433M	315	560	380	1490	56	0.89	Siemens	1LA4 404	NU 224	6220
4	WA 446M	315	560	380	1490	56	0.89	Siemens	1LA4 404	NU 224	6220
	WA NO.	I/V	D/V	RPM	CAP	PRESS					
1	WA 446C	9	8	1490	390	460		Sulzer	HPL 42-22.5 30°	Sleeve	Sleeve
2	WA 443C	9	8	1490	390	460		Sulzer	HPL 42-22.5 30°	Sleeve	Sleeve
3	WA 438C	9	8	1490	390	460		Sulzer	HPL 42-22.5 30°	Sleeve	Sleeve
4	WA 450C	9	8	1490	390	460		Sulzer	HPL 42-22.5 30°	Sleeve	Sleeve

* I/V = IMPELLER VANES   * D/V = DIFFUSER VANES

SCHEME: **SWAKOP-RöSSING BOOSTER2**

B16

#	WA NO.	kW	A	V	RPM	SLOTS	COS θ	MAKE	MODEL	BEARINGS	
										DRIVE	NDE
1	WA 436M	315	560	380	1490	56	0.89	Siemens	1LA4 404	NU 224	6220
2	WA 447M	315	560	380	1490	56	0.89	Siemens	1LA4 404	NU 224	6220
3	WA 431M	315	560	380	1490	56	0.89	Siemens	1LA4 404	NU 224	6220
4	WA 437M	315	560	380	1490	56	0.89	Siemens	1LA4 404	NU 224	6220
	WA NO.	I/V	D/V	RPM	CAP	PRESS					
1	WA 439C	9	8	1490	390	460		Sulzer	HPL 42-22.5 30°	Sleeve	Sleeve
2	WA 436C	9	8	1490	390	460		Sulzer	HPL 42-22.5 30°	Sleeve	Sleeve
3	WA 433C	9	8	1490	390	460		Sulzer	HPL 42-22.5 30°	Sleeve	Sleeve
4	WA 445C	9	8	1490	390	460		Sulzer	HPL 42-22.5 30°	Sleeve	Sleeve

* I/V = IMPELLER VANES   * D/V = DIFFUSER VANES

SCHEME: **SWAKOP-RöSSING BOOSTER3**

B17

#	WA NO.	kW	A	V	RPM	SLOTS	COS θ	MAKE	MODEL	BEARINGS	
										DRIVE	N/DRIVE
1	WA 438M	315	560	380	1490	56	0.89	Siemens	1LA4 404	NU 224 C3	6220 C3
2	WA 445M	315	560	380	1490	56	0.89	Siemens	1LA4 404	NU 224 C3	6220 C3
3	WA 442M	315	560	380	1490	56	0.89	Siemens	1LA4 404	NU 224 C3	6220 C3
4	WA 439M	315	560	380	1490	56	0.89	Siemens	1LA4 404	NU 224 C3	6220 C3
	WA NO.	I/V	D/V	RPM	CAP	PRESS					
1	WA 448C	9	8	1490	390	460		Sulzer	HPL 42-22.5 30°	Sleeve	Sleeve
2	WA 441C	9	8	1490	390	460		Sulzer	HPL 42-22.5 30°	Sleeve	Sleeve
4	WA 440C	9	8	1490	390	460		Sulzer	HPL 42-22.5 30°	Sleeve	Sleeve
3	WA 434C	9	8	1490	390	460		Sulzer	HPL 42-22.5 30°	Sleeve	Sleeve

* I/V = IMPELLER VANES   * D/V = DIFFUSER VANES

APPENDIX C

BEARING FREQUENCIES

MAKE	Bearing number	Pitch dia. mm	Element dia. Mm	No. of elements	Contact angle	RPM	BPOR	BPIR	2 x BSF	FTF
SKF	NU 208 E	61	11	14	0	1 480	8 492	12 228	6 728	607
SKF	NU 210	71	11	16	0	1 480	10 006	13 674	8 073	625
SKF	NU 212	86	12	15	0	1 480	9 551	12 649	9 126	637
SKF	NU 213	94	15	16	0	1 480	9 951	13 729	7 795	623
SKF	NU 215 C3	103	14	16	0	1 480	10 231	13 449	9 408	639
SKF	NU 217	119	18	17	0	1 480	10 677	14 483	8 304	628
SKF	NU 219	133	19	16	0	1 480	10 149	13 531	8 880	634
SKF	NU 220	140	20	16	0	1 480	10 149	13 531	8 880	634
SKF	NU 224 C3	168	24	16	0	1 480	10 149	13 531	8 880	634
SKF	NU 308	66	14	12	0	1 480	6 996	10 764	5 497	583
SKF	NU 313	103	19	12	0	1 480	7 242	10 518	6 544	603
SKF	NU 314	110	20	12	0	1 480	7 265	10 495	6 660	605
SKF	NU 315	118	22	12	0	1 480	7 224	10 536	6 458	602
SKF	NU 317	132	24	12	0	1 480	7 265	10 495	6 660	605
SKF	NU 319	148	26	13	0	1 480	7 930	11 310	6 944	610
SKF	NU 320	158	28	13	0	1 480	7 915	11 325	6 872	609
SKF	NU 324	190	36	12	0	1 480	7 197	10 563	6 332	600
SKF	NU 334	265	45	15	0	1 480	9 215	12 985	7 236	614
SKF	NU 410 C3	71	20	10	0	1 480	5 315	9 485	3 774	532
SKF	3307	58	12.7	12	32	1 480	7 231	10 529	5 504	603
SKF	6210 C3/C4	70	12.7	10	0	1 480	6 057	8 743	6 678	606
SKF	6212 C3/C4	85	15.1	10	0	1 480	6 085	8 715	6 852	609
SKF	6213 C3	93	15.88	11	0	1 480	6 750	9 530	7 188	614
SKF	6214 C4	98	16.67	11	0	1 480	6 755	9 525	7 220	614
SKF	6215 C3	103	17.46	11	0	1 480	6 760	9 520	7 250	615
SKF	6217 C3	118	19.84	11	0	1 480	6 771	9 509	7 322	616
SKF	6219	133	23.81	10	0	1 480	6 075	8 725	6 788	608
SKF	6220 C3	140	25.4	10	0	1 480	6 057	8 743	6 678	606
SKF	6309 2RS C3	73	17.46	8	0	1 480	4 504	7 336	4 708	563
SKF	6310	80	19.05	8	0	1 480	4 510	7 329	4 735	564
SKF	6314	110	25.4	8	0	1 480	4 553	7 287	4 930	569
SKF	6318	140	31.75	8	0	1 480	4 577	7 263	5 046	572
SKF	6322	175	41.27	8	0	1 480	4 524	7 316	4 796	565
SKF	6409 C3	83	23.02	7	0	1 480	3 743	6 617	3 856	535
SKF	6411	98	26.99	7	0	1 480	3 753	6 607	3 894	536
SKF	6314	110	25.4	8	0	1483	4602	7302	6079	570
SKF	7213 B	93	16.67	16	40	1 480	10 214	13 466	7 124	638
SKF	7214 B	98	17.46	16	40	1 480	10 224	13 456	7 174	639
SKF	7215 B	103	17.46	17	40	1 480	10 946	14 214	7 598	644
SKF	7217 B	118	19.84	17	40	1 480	10 960	14 200	7 668	645
SKF	7219 B	133	23.81	16	40	1 480	10 216	13 464	7 134	639
SKF	7220 B	140	25.40	15	40	1 480	9 557	12 643	7 024	637
SKF	7324 B	190	41.27	13	40	1 480	8 019	11 221	5 680	617

BPOR = Ball Pass Outer Race
BPIR = Ball Pass Inner Race
BSF = Ball Spin Frequency
FTF = Fundamental Train Frequency

APPENDIX D

ISO STANDARDS 2372 & 3945

RANGES OF VIBRATION		EXAMPLES OF VIBRATION SEVERITY FOR SEPARATE CLASSES OF MACHINES			
V, RMS VELOCITY		CLASS I	CLASS II	CLASS III	CLASS IV
MM/S	INCHES/SEC				
0.28	0.01				
0.45	0.02	A			
0.71	0.03		A		
1.12	0.04	B		A	
1.80	0.07		B		A
2.80	0.11	C		B	
4.5	0.18		C		B
7.1	0.28	D		C	
11.2	0.44		D		C
18	0.71			D	
28	1.10				D
45	1.77				

MACHINERY CLASSES

CLASS I INDIVIDUAL COMPONENTS, INTEGRALLY CONNECTED WITH THE COMPLETE MACHINE IN ITS NORMAL OPERATING CONDITION

CLASS II MEDIUM-SIZED MACHINES

CLASS III LARGE PRIME MOVERS MOUNTED ON HEAVY, RIGID FOUNDATIONS

CLASS IV LARGE PRIME MOVERS MOUNTED ON RELATIVELY SOFT, LIGHTWEIGHT STRUCTURES

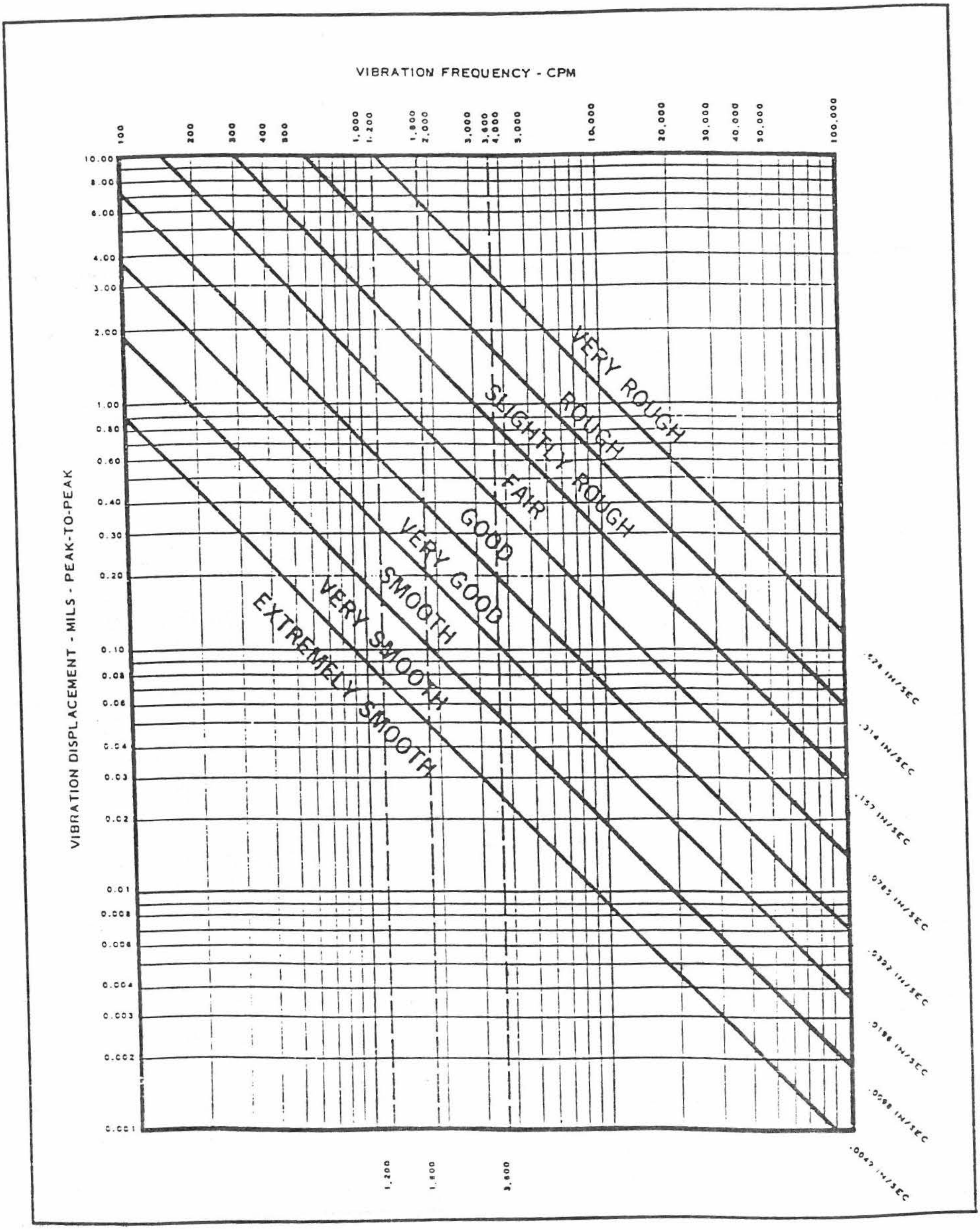
ACCEPTANCE CLASSES

A EXCELLENT

B SATISFACTORY

C UNSATISFACTORY

D UNACCEPTABLE


VIBRATION FREQUENCY - CPM
100
200
300
400
500
1,000
1,200
1,800
2,000
3,000
3,600
4,000
5,000
10,000
20,000
30,000
40,000
50,000
100,000
VIBRATION DISPLACEMENT - MILS - PEAK-TO-PEAK
10.00
8.00
6.00
4.00
3.00
2.00
1.00
0.80
0.60
0.40
0.30
0.20
0.10
0.08
0.06
0.04
0.03
0.02
0.01
0.008
0.006
0.004
0.003
0.002
0.001
VERY ROUGH
ROUGH
SLIGHTLY ROUGH
FAIR
GOOD
VERY GOOD
SMOOTH
VERY SMOOTH
EXTREMELY SMOOTH
.628 IN/SEC
.314 IN/SEC
.157 IN/SEC
.0785 IN/SEC
.0392 IN/SEC
.0196 IN/SEC
.0098 IN/SEC
.0049 IN/SEC
1,200
1,800
3,600

APPENDIX F

SPECTRAL ALARMS

(i) GENERAL ROLLING ELEMENT BEARING MACHINES WITHOUT ROTATING VANES

ITEM	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
BAND LOWER FREQ	1% Fmax	1.2 X RPM	2.2 X RPM	3.2 X RPM	12.2 X RPM	50% F max
BAND HIGHER FREQ	1.2 X RPM	2.2 X RPM	3.2 X RPM	12.2 X RPM	50% Fmax	100% Fmax
BAND ALARM	90% OA	50% OA	40% OA	30% OA	25% OA	20% OA
DESCRIPTION OF BAND COVERAGE	Sub-synchronous thru 1 X RPM	1.5 - 2.0 X RPM	2.5 - 3.0 X RPM	Funda. Bearing Defect Frequencies	Lower Harmonic Bearing Frequencies	Higher Harmonic Bearing Freqs & Brg. Natl. Freqs.

(ii) GENERAL SLEEVE-BEARING MACHINES WITHOUT VANES

ITEM	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
BAND LOWER FREQ	1% Fmax	0.8 X RPM	1.8 X RPM	2.8 X RPM	3.8 X RPM	10.2 X RPM
BAND HIGHER FREQ	0.8 X RPM	1.8 X RPM	2.8 X RPM	3.8 X RPM	10.2 X RPM	100% Fmax
BAND ALARM	30% OA	90% OA	50% OA	40% OA	35% OA	20% OA
DESCRIPTION OF BAND COVERAGE	Sub-synchronous Band	1X - 1.5 X RPM	2X - 2.5 X RPM	3X - 3.5 X RPM	4X - 10 X RPM	10.5 X RPM - Fmax

(iii) MOTOR ELECTRICAL ROTOR BAR PASS FREQUENCY POINT (MEASURE AT MOTOR OUTBOARD BEARING HORIZONTAL)

ITEM	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
BAND LOWER FREQ	30 000 CPM	75 000 CPM	120 KCPM	165 KCPM	210 000 CPM	255 000 CPM
BAND HIGHER FREQ	75 000 CPM	120 000 CPM	165 KCPM	210 KCPM	255 000 CPM	300 000 CPM
BAND ALARM	2.03 mm/s	1.52 mm/s	1.52 mm/s	1.52 mm/s	1.52 mm/s	1.52 mm/s
DESCRIPTION OF BAND COVERAGE	30K - 75K	75K - 120K	120K - 165K	165K - 210K	210K - 255K	255K - 300K

(iv) MOTOR ELECTRICAL 12 000 CPM MEASUREMENT POINT AT MOTOR INBOARD BEARING HORIZONTAL

ITEM	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
BAND LOWER FREQ	240 CPM	2 000 CPM	4 000 CPM	6 500 CPM	8 000CPM	10 000 CPM
BAND HIGHER FREQ	2 000 CPM	4 000 CPM	6 500 CPM	8 000 CPM	10 000 CPM	12 000 CPM
BAND ALARM Speed A:	90% OA	50% OA	50% OA	40% OA	35% OA	20% OA
Speed B:	30% OA	90% OA				
DESCRIPTION OF BAND COVERAGE	Speed A: 120 - 2 000 CPM Speed B: 2 000 - 4 000 CPM					

APPENDIX F

SPECTRAL ALARMS

(i) GENERAL ROLLING ELEMENT BEARING MACHINES WITHOUT ROTATING VANES

ITEM	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
BAND LOWER FREQ	1% Fmax	1.2 X RPM	2.2 X RPM	3.2 X RPM	12.2 X RPM	50% F max
BAND HIGHER FREQ	1.2 X RPM	2.2 X RPM	3.2 X RPM	12.2 X RPM	50% Fmax	100% Fmax
BAND ALARM	90% OA	50% OA	40% OA	30% OA	25% OA	20% OA
DESCRIPTION OF BAND COVERAGE	Sub-synchronous thru 1 X RPM	1.5 - 2.0 X RPM	2.5 - 3.0 X RPM	Funda. Bearing Defect Frequencies	Lower Harmonic Bearing Frequencies	Higher Harmonic Bearing Freqs & Brg. Natl. Freqs.

(ii) GENERAL SLEEVE-BEARING MACHINES WITHOUT VANES

ITEM	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
BAND LOWER FREQ	1% Fmax	0.8 X RPM	1.8 X RPM	2.8 X RPM	3.8 X RPM	10.2 X RPM
BAND HIGHER FREQ	0.8 X RPM	1.8 X RPM	2.8 X RPM	3.8 X RPM	10.2 X RPM	100% Fmax
BAND ALARM	30% OA	90% OA	50% OA	40% OA	35% OA	20% OA
DESCRIPTION OF BAND COVERAGE	Sub-synchronous Band	1X - 1.5 X RPM	2X - 2.5 X RPM	3X - 3.5 X RPM	4X - 10 X RPM	10.5 X RPM - Fmax

(iii) MOTOR ELECTRICAL ROTOR BAR PASS FREQUENCY POINT (MEASURE AT MOTOR OUTBOARD BEARING HORIZONTAL)

ITEM	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
BAND LOWER FREQ	30 000 CPM	75 000 CPM	120 KCPM	165 KCPM	210 000 CPM	255 000 CPM
BAND HIGHER FREQ	75 000 CPM	120 000 CPM	165 KCPM	210 KCPM	255 000 CPM	300 000 CPM
BAND ALARM	2.03 mm/s	1.52 mm/s	1.52 mm/s	1.52 mm/s	1.52 mm/s	1.52 mm/s
DESCRIPTION OF BAND COVERAGE	30K - 75K	75K - 120K	120K - 165K	165K - 210K	210K - 255K	255K - 300K

(iv) MOTOR ELECTRICAL 12 000 CPM MEASUREMENT POINT AT MOTOR INBOARD BEARING HORIZONTAL

ITEM	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
BAND LOWER FREQ	240 CPM	2 000 CPM	4 000 CPM	6 500 CPM	8 000CPM	10 000 CPM
BAND HIGHER FREQ	2 000 CPM	4 000 CPM	6 500 CPM	8 000 CPM	10 000 CPM	12 000 CPM
BAND ALARM Speed A:	90% OA	50% OA	50% OA	40% OA	35% OA	20% OA
Speed B:	30% OA	90% OA				
DESCRIPTION OF BAND COVERAGE	Speed A: 120 - 2 000 CPM Speed B: 2 000 - 4 000 CPM					

(v) SPECIAL MACHINE TYPES

TYPE 1 - CENTRIFUGAL MACHINES WITH AN UNKNOWN NUMBER OF VANES AND ROLLING ELEMENT BEARINGS

ITEM	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
BAND LOWER FREQ	240 CPM	2 000 CPM	4 000 CPM	6 500 CPM	8 000CPM	10 000 CPM
BAND HIGHER FREQ	2 000 CPM	4 000 CPM	6 500 CPM	8 000 CPM	10 000 CPM	12 000 CPM
BAND ALARM	30% OA	90% OA	50% OA	40% OA	35% OA	20% OA
DESCRIPTION OF BAND COVERAGE	Sub-synchronous Band thru 1X RPM	1.5 - 2.0 X RPM	2.5X - Funda. bearing Defect Freq.	BPF +/- 1X Sidebands	Lower Harmonic Bearing Freqs. & BPF Harmonics	Higher Harmonic Bearing Freqs. & Brg. Natl. Freqs.

TYPE 2 - CENTRIFUGAL MACHINES WITH AN UNKNOWN NUMBER OF VANES AND ROLLING ELEMENT BEARINGS

ITEM	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
BAND LOWER FREQ	1% Fmax	1.2 X RPM	2.2 X RPM	3.2 X RPM	6.8 X RPM	50% Fmax
BAND HIGHER FREQ	1.2 X RPM	2.2 X RPM	3.2 X RPM	6.8 X RPM	50% Fmax	100% Fmax
BAND ALARM	90% OA	50% OA	40% OA	60% OA	35% OA	20% OA
DESCRIPTION OF BAND COVERAGE	Sub-synchronous Band thru 1X RPM	1.5 - 2.0 X RPM	2.5 - 3.0 RPM	Possible BPF for pumps	Lower Harmonic Bearing Freqs. & BPF Harmonics	Higher Harmonic Bearing Freqs. & Brg. Natl. Freqs.

TYPE 3 - CENTRIFUGAL MACHINES WITH AN UNKNOWN NUMBER OF VANES AND ROLLING ELEMENT BEARINGS

ITEM	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
BAND LOWER FREQ	1% Fmax	0.8 X RPM	1.8 X RPM	3.8 X RPM	BPF - 1.2 X RPM	BPF + 1.2 X RPM
BAND HIGHER FREQ	0.8 X RPM	1.8 X RPM	3.8 X RPM	BPF - 1.2 X RPM	BPF + 1.2 X RPM	100% Fmax
BAND ALARM	30% OA	90% OA	50% OA	30% OA	70% OA	30% OA
DESCRIPTION OF BAND COVERAGE	Sub-synchronous Band	1X - 1.5 X RPM	2X - 3.5 X RPM	4X - Lower RPM Harmonics	BPF +/- 1X RPM	Higher RPM Harmonics and BPF Harmonics

TYPE 4 - CENTRIFUGAL MACHINES WITH AN UNKNOWN NUMBER OF VANES AND ROLLING ELEMENT BEARINGS

ITEM	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
BAND LOWER FREQ	1% Fmax	0.8 X RPM	1.8 X RPM	3.8 X RPM	6.8 X RPM	9.8X RPM
BAND HIGHER FREQ	0.8 X RPM	1.8 X RPM	3.8 X RPM	6.8 X RPM	9.8 X RPM	100% Fmax
BAND ALARM	30% OA	90% OA	50% OA	70% OA	25% OA	30% OA
DESCRIPTION OF BAND COVERAGE	Sub-synchronous Band	1X - 1.5 X RPM	2X - 3.5 X RPM	4X - 6.5 X RPM	7X - 9.5 X RPM	10X RPM - Fmax

APPENDIX G

FAULT DETECTION AND COST SAVINGS ACHIEVED

Period Jan '93 - Jan' 95

#	WA NO.	FAULT DETECT	CORRECTIVE ACTION TAKEN	ESTIMATED COST -SAVING
1	WA 174M	Electrical	-	-
2	WA 175M	Bearings	-	-
3	WA 177M	Misalignment & bearings	-	-
4	WA 202C	Bearings		-
5	WA 83C	Cavitation		
6	WA 328M	Misalignment	-	-
7	WA 329M	Unbalance	-	-
8	WA 330M	Bearing problems	Replace bearings	1 500
9	WA 331M	Misalignment & bearings	Re-align/Replace bearings	1 500
10	WA 210C	Bearing problem	Replace bearings	1 500
11	WA 212C	Unbalance/Bearings	-	-
12	WA 216C	Bearing problems	Replace bearings	1 500
13	WA 282M	Bearings	-	-
14	WA 97M	Bearings	-	-
15	WA 248M	Loose bearing & unbalance	Repair housing/Balance rotor/Re-align	8 000
16	WA 1059M	Frame distortion	-	-
17	WA 625M	Unbalance	Balance rotor	2 000
18	WA 326C	Looseness & bearings	Overhaul pump	1 500
19	WA 683C	Looseness & unbalance	Repair	1 500
20	WA 271C	Unbalance	-	-
21	WA 272C	Looseness & misalignment	Re-align	4 500
22	WA 551C	Mechanical looseness		-
23	WA 319M	Misalignment & soft foot	Re-align/correct softfoot	1 500
24	WA 320M	Misalignment & soft foot	Re-align/correct softfoot	1 500
25	WA 321M	Misalignment/ Bearings	Re-align/Replace bearings	1 500
26	WA 322M	Unbalance	-	-
27	WA 323M	Misalignment/softfoot	Re-align/correct softfoot	1 000
28	WA 324M	Misalignment/Bearings	Re-align	2 000
29	WA 195C	Misalignment	Re-align	1 000
30	WA 193C	Misalignment	Re-align	1 000
31	WA 194C	Misalignment	Re-align	1 000
32	WA 196C	Misalignment	Re-align	1 000
33	WA 511M	Misalignment	-	-
34	WA 512M	Misalignment, bearings & loose bearing in housing	Re-align/Replace bearings/Re-sleeve bearing housing	5 000
35	WA 307M	Loose bearing in housing/Unbalance	Re-sleeve bearing housing/Balance rotor	4 500
36	WA 51C	Looseness/Misalignment/ Softfoot	-	-
37	WA 52C	Mechanical looseness & misalignment	-	2 000
38	WA 224M	Defective coupling/ unbalance	Replace	2 500
39	WA 231M	Misalignment & bearings	Re-align/Replace bearings	2 500
40	WA 427M	Misalignment	Re-align	2 000
41	WA 428M	Misalignment & loose baseframe	Re-align/Tightened baseframe bolts	2 500

42	WA 163 M	Bearings	Replace	1 500
43	WA 165M	Bearings	Replace	1 500
44	WA 235C	Resonance	-	-
45	WA 236C	Misalignment	Re-align	1 000
46	WA 555C	Bearings	Replace	2 500
47	WA 825C	Impeller & bearings	Repair	2 000
48	WA 827C	Impeller & bearings	Replace bearings	1 500
49	WA 656M	Looseness/Bearings	-	-
50	WA 660M	Misalignment & bearings	Re-align/Replace bearings	2 500
51	WA 661M	Bearings	-	-
52	WA 657M	Loose bearing in housing	Re-sleeve bearing housing	2 500-
53	WA 680C	Misalignment/Softfoot/ Bearing problem	Replace bearings/Re-align	10 000
54	WA 681C	Resonance	Correct	5 000
55	WA 678C	Mechanical looseness	-	-
56	WA 89M	Misalignment & bearings	-	-
57	WA 157M	Misalignment	-	-
58	WA 158M	Misalignment	-	-
59	WA 515M	Bearing defects	-	-
60	WA 135C	Unbalance, looseness & cavitation	-	-
61	WA 136C	Misalignment/Bearing problem/Looseness	-	-
62	WA 141C	Bearing problem	Replace bearings	2 500
63	WA 561M	Bearings	Replace	2 500
64	WA 562M	Bearings	-	-
65	WA 39M	Bearings	Replace	2 500
66	WA 65C	Possible bent shaft	-	-
67	WA 69C	Unbalance	-	-
68	WA 44C	Unbalance	-	
69	WA 106M	Misalignment	Re-align	2 500
70	WA 160C	Looseness & unbalance	-	-
71	WA 68C	Loosness, unbalance & bent shaft	Repair pump	200 000
72	WA 643C	Bearing problem	Replace bearings	5 000
73	WA 244C	Bearing problem/Cavitation	Replace bearing	5 000
74	WA 260C	Cavitation	-	
75	WA 444M	Misalignment/Soft foot	Re-align	2 500
76	WA 887M	Bearing defects	-	-
77	WA 888M	Looseness & bearings	-	-
78	WA 433M	Electrical	-	-
	WA 437M	Looseness/Softfoot/ Bearing problem	Replace bearings	2 500
79	WA 433C	Mechanical looseness	-	-
80	WA 445C	Impeller problem	Repair	5 000
81	WA 445M	Misalignment & bearings	-	-
			TOTAL SAVINGS	**315 000**

APPENDIX H

MACHINE IMPROVEMENT REPORT

CASE 6

Plant: Ruacana Clear Water
Train: Purification
Machine: WA248M/272C
Equipment Type: Motor/Pump

PROBLEM: Detect high vibration on motor and pump. Did vibration analysis and found misalignment and mechanical looseness (Bearing turning in housing).

Repair Savings = Historic Cost - Actual Cost

Repair savings = N$16 346

Electrical Motor (WA248M)

	ACTUAL COST	HISTORIC COST
Parts:	2 000	7 846
Labor:	500	1 500
Total:	2 500	9 346

Repair Savings: N$ 6 846

Pump (272C)

	ACTUAL COST	HISTORIC COST
Parts:	0. 000	8 000
Labor:	500	1 500
Total:	500	9 500

Repair Savings: N$9 500

Reported by: E Smith
Diagnosed by: E Smith
Assisted by: P Kirsten/K Leischering

Fault Detection Date: 06/07/94
Repair Completion Date: 10/01/95

Justification: Prevented further bearing wear and possible failures on motor and pump.

APPENDIX 1

MACHINE HISTORIES

CASE 6

Machine histories taken from Maintenance Records

Motor (WA248M):

08/05/89 **Motor failed. Overhauled motor. N$7846**

No further history available standby for a long period

14/10/94 **Detected mechanical looseness. Vibration analysis indicating a possible loose bearing. Submitted a request to Maintenance to open motor covers for inspection. It was found that the outboard bearing was turning into the casing.**

Pump (WA272C):

28/10/91 **Overhauled pump. N$ 6 100**

6 months later

24/03/92 **Bearings became hot. Overhauled pump. N$8 000**

13 months later

14/10/94 **Vibration analysis detected severe misalignment.**

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GLOSSARY

Acceleration: The time rate of change of velocity. Typical units are mm/sec², and g's. Acceleration measurements are usually made with accelerometers.

Accelerometer: Transducer whose output is directly proportional to acceleration. Most people use piezoelectric crystals to produce output.

Aliasing: A phenomenon which can occur whenever a signal is not sampled at greater than twice the maximum frequency component. Causes high-frequency signals to appear at low frequencies. Aliasing is avoided by filtering out signals greater than ½ the sample rate.

Alignment: A condition whereby the axes of machine components are either coincidental, parallel, or perpendicular, according to design requirements.

Amount of Unbalance: The weight required in one or more planes to effect coincidence of an object's mass centre line and rotational centre line. Usually given as the product of the weight and the radius at which it must be located.

Amplitude: The magnitude of dynamic motion or vibration. Amplitude is expressed in terms of "peak-peak", "zero-to-peak", or rms. For pure sine waves only, these are related as follows: rms = 0.707 times "zero-to-peak"; peak-to-peak = 2 times "zero-to-peak".

Asynchronous: Vibration components that are not related to rotating speed.

Availability: Actual time that plant machinery and systems are available for production. One measurement of maintenance effectiveness.

Averaging: Digitally averaging several vibration measurements to improve accuracy or to reduce the level of asynchronous components. Refer to the definitions of Root-Mean-Square.

Axial: In the same direction as the shaft centre line.

Balance: The condition which exists in a part so that no vibratory force is transferred to the supporting bearings of that part due to centrifugal force. Occurs when the mass centre line of a rotor is coincidental.

Band-Pass Filter: A filter with a single transmission band extending from lower to upper cut-off frequencies. The width of the band is determined by the separation of frequencies at which amplitude is attenuated by 3 dB (0.707).

Band width: The spacing between frequencies at which a band-pass filter attenuates the signal by 3 dB. Window factors are 1 for uniform, 1.5 for Hanning, and 3.63 for flat top.

Baseline Spectrum: A vibration spectrum taken when a machine is in good operating condition; used as a reference for monitoring and analysis.

Blade Pass Frequency: A potential vibration frequency on any bladed machine. It is presented by the number of blades times the shaft rotating speed (running speed).

BPIR: Ball pass inner race. The relative speed between the balls or rollers in a rolling element bearing and the inner race.

BPOR: Ball pass outer race. The relative speed between the balls or rollers in a rolling element bearing and the outer race.

Broadband Trending: Vibration analysis technique that plots the change in the overall or broadband of a machine-train.

BSF: Ball spin frequency. The frequency created by the balls or rollers in a rolling - element bearing. The actual turning speed of each individual ball or roller in a bearing.

Cavitation: A condition which can occur in liquid-handling machinery (e.g., centrifugal pumps) when system pressure decreases in the suction line and the pump inlet lowers fluid pressure and vaporization occurs. The result is a mixed flow which may produce vibration.

CPM: Cycles Per Minute. Favoured by many in machine vibration analysis, because the vibration caused by unbalance shows up at a frequency in CPM equal to the RPM of the shaft. 60 cycles per minute (CPM) is equivalent to 1 cycle per second which equals 1 hertz.

Critical Machinery: Machines which are critical to a major part of the plant process.

Decibels (dB): A logarithmic representation of amplitude ratio, defined as 20 times the base-ten logarithm of the ratio of the measured amplitude to a reference.

Displacement: The actual change in distance or position of an object relative to a reference.

Displacement Transducer: A transducer whose output is proportional to the distance between it and the measured object.

Dynamic Range: A measurement of the ability of a data acquisition system (i.e., transducer, signal conditioner, recording instrument) to capture measurable amplitudes of vibration frequency components. The ability to separate meaningful, low-level frequency components from the noise floor within a machine's signature. Also the ability to simultaneously capture high-amplitude and low-amplitude frequency components within a machine's signature. In normal practice, a minimum of 50 dB dynamic range is required for vibration analysis.

Eccentricity, Mechanical: The variation of the outer diameter of a shaft surface when referenced to the true geometric centre line of the shaft.

Fast Fourier Transforms (FFT): A computer (or microprocessor) procedure for calculating discrete frequency components from sampled time data. A special case of the discrete Fourier transform is where the number of samples is constrained to a power of 2.

Forced Vibration: The oscillation of a system under the action of a forcing function. Typically, forced vibration occurs at the frequency of the exciting force.

Free Vibration: Vibration of a mechanical system following an initial force - typically, at one or more natural frequencies.

Frequency: The repetition rate of a periodic event, usually expressed in cycles per second (Hz), revolutions per minute (rpm), or multiples of rotational speed (orders). Orders are commonly referred to as 1x for rotational speed, 2x for twice rotational speed, etc.

Frequency Response: The amplitude and phase response characteristics of a system.

Fundamental Frequency: The first frequency in a series of harmonic frequencies. For example, the orders of shaft turning speed occur at harmonics (integer multiples) of shaft running speed.

G (or g): A unit of acceleration, commonly used with the English system of units; 1 g represents the acceleration due to gravity at sea level and is approximately equal to 9.806 m/s^2.

Harmonic: An integer multiple of **a Fundamental Frequency**.

Hertz (Hz): The unit of frequency represented by cycles per second.

High-Pass-Filter: A filter with a transmission band starting at a lower cut-off frequency and extending to (theoretically) infinite frequency.

Journal: Specific portions of the shaft surface from which rotor-applied loads are transmitted to bearing supports.

Key phasor: A signal used in rotating machinery measurements, generated by a transducer observing a once-per-revolution event. The key phasor signal is used in phase measurements for analysis and balancing.

Low-Pass Filter: A filter whose transmission band extends from dc to an upper cut-off frequency.

Natural Frequency: The natural frequency of free vibration of a system. The frequency at which an undamped system with a single degree of freedom will oscillate upon momentary displacement from its rest position.

Period: The time required for a complete oscillation or for a single cycle of events. The reciprocal of frequency.

Phase: A measurement of the timing relationship between two or more signals, or between a specific vibration component and a once-per-revolution event.

Phase Angle: The angular measurement from the leading edge of the vibration component to the following positive peak of the 1x vibration component.

Piezo-electric: Any material which provides a convergence of mechanical and electrical energy. If mechanical stresses are applied on two opposite faces of a piezoelectric crystal, electrical charges appear on some other pair of faces.

Probe Orientation: The angular relationship of a transducer with respect to bearing housing, shaft etc. Normally given as a polar co-ordinate (e.g., 0 = top, vertical; 90 = right, horizontal; 180 = bottom, vertical; 270 = left, horizontal).

Radial: A direction on a machine which is perpendicular to the shaft centre line. Usually refers to the direction of shaft or casing motion or measurement.

Resolution: The smallest change in applied stimulus that will produce a detectable change in the instrument output.

Resonance: The condition of vibration amplitude response caused by a corresponding system sensitivity to a particular forcing frequency.

Root-Mean-Square: Square root of the arithmetical average of a set of squared instantaneous values. RMS is the closest approximation of total energy of a vibration spectrum.

RS232: A serial, asynchronous communications standard; a type designated for cables which are used to connect communications ports on a computer with other digital devices such as digital analyzers, printers and modems.

Side band: A frequency component which represents the effect of modulation on a signal. If a modulated signal has more than one component, each component will show side bands. A side band is spaced off from the frequency of the modulated signal by an amount equal to the modulating frequency. If the modulating signal has multiple components or if there are frequency modulations, the side band pattern may be very complicated including sum and difference frequencies between the side band component frequencies (intermodulation effects).

Signal Conditioner: A device placed between a signal source (e.g. transducer) and a readout instrument to change the signal. Examples: attenuators, preamplifiers, signal convertors and filters.

Signature (FFT): Term usually applied to the vibration frequency spectrum unique to a particular machine or machine component, system, or subsystem at a specific location and point of time.

Sleeve Bearings: A bearing which supports the shaft on a thin film of oil. The fluid-film layer may be generated by journal rotation (hydrodynamic bearing) or by externally applied pressure (hydrostatic bearing).

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Spectrum: The frequency domain representation of a signal. In practical measurements, the spectrum is usually displayed as a plot of magnitude against frequency over a limited frequency range.

Spectrum Plot: An XY plot in which the X-axis represents vibration frequency and the Y-axis represents vibration amplitude.

Subharmonic: Sinusoidal quantity of a frequency which is an integral submultiple of a fundamental frequency.

Subsynchronous: Component of a vibration signal that is less than the fundamental (1x) running speed of the shaft.

Supersynchronous: Component of a vibration signal that has a frequency greater than the fundamental (1x) shaft running speed.

Synchronous: Vibration frequency components that vary in direct proportion to changes in the fundamental (1x) rotative speed.

Threshold: The smallest change in the measured variable that will result in a measurable change in an output signal. See **Dynamic Range**.

Transducer: A device for transmitting the magnitude of one quantity into another quantity. The second quantity often has dimensions different from the first and serves as the source of a useful signal. Vibration transducers convert mechanical motion into a proportional electronic signal.

Vibration: The oscillary motion of a mechanical system about a mean position.

Wear Particle Analysis: A predictive maintenance technique that uses the metallic particles in lubricating oil to determine machine condition.

Zoom: A frequency analysis at higher resolution than the base band spectrum over a limited frequency span in order to see more detail.