



# Sudan University for science and technology

Faculty of Postgraduate Studies

Mechanical engineering department

# Pump inspection Using Vibration Measurements and analysis

فحص المضخات بإستخدام قياسات الإهتزازات والتحليل

Research submitted to Mechanical engineering department for the degree of Master in Mechanical engineering.

Prepared by:

Elfatíh Hassan Mohammed Malík Supervised by:

Dr. Tag Elssír HassanHussan Alí

Co-Supervised by: Dr. Hassan Abdulateef Osman Fadul

Nov2017

الإستكلال

قال تعالى:

بسماللهالرحمزالرحيم

وعِندَهُ مَفَاتِحُ الْغَيْبِ لاَ يَعْلَمُهَا إلاَّ هُوَ وَيَعْلَمُ مَا فِي الْبَرِّ وَالْبَحْر وَمَا تَسْقُطُ مِن وَرَقَةٍ إِلاَّ يَعْلَمُهَا وَلاَ حَبَّةٍ فِي ظُلُمَاتِ الأَرْضِ وَلاَ رَطْبٍ وَلاَ يَابِسِ إِلاَّ فِي كِتَابٍ مُبين. صدق الله العظيم

الاية 59 من سورة الأنعام

# Dedication

For those which their presence and encouragement has been the great help to me, and in the light of that, I would like to dedicate this work to the following:

To my parent who thought of educating me.

To my family who contributed in the same way.

To my friends and colleagues who encouraged me with all sorts of advices.

To my colleges in Khartoum Refinery Company for great help.

# Acknowledgement

If I need anyone deserves much of my thanks in making this project a reality that must definitely be my supervisor Dr.Tag ELssir Hassan Hussan Ali for his keen tireless guidance throughout this project. I acknowledge his effort in bringing this project to a successful conclusion.

I would like also to express my deepest gratitude to the cosupervisor Dr. Hassan Abdulateef Osman Fadul who helped me to reach this stage and not forgetting to extent my especial thanks to the staff of Reliability Department of Mechanical Engineering in Khartoum Refinery Company (KRC), Especially for Engineer Abu Baker Abdelmageed, for his great efforts during data collection and problem analysis.

# Abstract:

In Khartoum Refinery Company (KRC) Pump 1104B suffers of high vibration level. Many corrections had been done to find out the problem but failed. Vibration analysis techniques is used to find out the root causes of the pump vibration. New data collected from the pump, indicates high vibration in the pump drive end (DE). After collection data, was uploaded to software in which set up database of measurement points. Spectrum analysis; was used. The analysis result indicates that the angular contact bearing in the pump drive end has been failed. During this work theoretical analysis through mathematical model had been done with the assumption that good bearing has good oil film damping coefficient to restrict the rolling elements movement while the shaft is rotating the damping loss percentage illustrate the bearing status and its residual life time can be estimated through Providing the program with the bearing ball mass and diameter in addition to the equipment rotational speed in RPM to find out the maximum amplitude that will illustrate no damping coefficient x-axis in rms. The root cause of vibration was found the comparison between the theoretical investigations and the measuring data show closed similarity. The visual inspection results after bearing box dismantling is angular contact bearing defect which confirmed the theoretical &software analysis of finding vibration root causes. Bearing replacement was carried out. New data collected from pump bearing drive end and sent to vibration analyzer to verify the repair. These results can lead to apply this mathematical analysis with the equipment's if their software not available to find out the root cause of vibration because the vibration phenomenon is same.

## المستخلص

في شركة مصفاةالخرطوم (KRC) المضخة B 1104 تعاني من مستوى اهتزاز عال. أستخدمت تقنيات تحليل الاهتزازات لمعرفة الأسباب الجذرية لذلك. البيانات التي تم جمعها من المضخة تشير الى اهتزاز عالي في نهاية محرك مضخة (DE). تم تحميل البيانات التي تم جمعها على برنامج أنشئ كقاعدة بيانات لنقاط القياس وحلل الطيف. تشير نتيجة التحليل الى فشل في محمل التلامس الزاوي في نهاية محرك المضخة. من التحليل النظري خلال النموذج الرياضي مع افتراض أن أفضل محمل به قفص تزيين له أفضل معامل لتخميد الإهتزاز بعناصره المتحركة بتقييد حركة العناصر المتحركة في حالة دوران العمود بجس فقدان معامل الخمد لدى المحمل الشئ الذى يصور حالته الراهنة وعمره المتبقي. المقارنة بين التحقيقات النظرية وتحليل الطيف تشير الى تشابه قريب فى نتائج التحليل. أثبت الكشف البصري بعد التفكيك وجود عيب واضح فى محمل الاتصال الزاوي الشيئ الذى أكد نتائج التحليل النظري والبرنامج لإيجاد الأسباب الجذرية محمل الاتصال الزاوي الشيئ الذى أكد نتائج التحليل النظري والبرنامج لإيجاد الأسباب الجزية محمل الاتصال الزاوي الشيئ الذى أكد نتائج التحليل النظري والبرنامج لإيجاد الأسباب الجذرية محمل الاتصال الزاوي الشيئ الذى أكد نتائج التحليل النظري والبرنامج لإيجاد الأسباب الجذرية محمل الاتصال الزاوي الشيئ الذى أكد نتائج التحليل النظري والبرنامج لإيجاد الأسباب الجذرية محمل الاتصال الزاوي الشيئ الذى أكد نتائج التحليل النظري والبرنامج لإيجاد الأسباب الجذرية محمل الاتصال الزاوي الشيئ الذى أكد نتائج التحليل النظري والبرنامج لإيجاد الأسباب الجذرية محمل الاتصال الزاوي الشيئ الذى أكد نتائج التحليل النظري والبرنامج ليحاد لأسباب الجذرية مرمم الإهتزاز هذه النتائج تقود الى أنه يمكن تطبيق هذاالتحليل الرياضي كأداة توقع لتقييم حالة المعدات مرمم محمل الأسل كما يمكن تطبيقه لمعرفة السبب الجذري للاهتزاز مع المعدات التي لا تتوفر لها برامج تحليل لأن ظاهرة الاهتزاز هي نفسها.

# **Table of Contents**

1.1	Introduction:			
1.2	Statement of problem:			
1.3	Research objectives:			
1.4	Methodology:			
1.5	1.5 Thesis organization:			
2. Lite	rature review:			
2.1 Th	eoretical background7			
2.1.1	Maintenance philosophies7			
2.1.2	Predictive maintenance and its techniques14			
2.1.3	Vibration measuring and Transducers:			
2.1.4	State of the arts			
2.2	Previous studies:			
2.2.1	Preventive and Predictive maintenance:			
2.2.2	Vibration Monitoring:			
3.Vibr	ation analyses and data collection:			
3.1	Vibration analysis :			
3.1.1	Vibration analysis (detection mode):			
3.1.2	Vibration analysis (diagnosis mode) :			
3.1.3	Vibration analysis – benefits :			
3.1.4	Using vibration to detect machinery fault:			
3.1.5	Common problems :			
3.2	Point of measurement :			
3.3	The scope of vibration analysis :			
3.3.1	Data collection :			
3.3.2	Data acquisition :			
3.4	Actual vibration profiles :			
3.4.1	Time Domain :			

3.4.2	Frequency Domain :	69			
3.4.3	Fourier transform :	70			
3.5	Assessing the vibration severity :	73			
3.5.1	The assessment points:	73			
3.5.2	Limits and standards of vibration :	74			
3.6.	Rotating unbalance and bearing deterioration stages				
3.6.1.	Rotating unbalance	76			
3.6.2	Rolling element bearings defect stages :				
4.MATER	4.MATERIAL AND METHOD				
4.1	The mathematical analysis:	85			
4.1.1	The bearing general mathematical model:	86			
4.1.2	The case study mathematical model:	87			
4.2	Bearing general equilibrium equation:	89			
4.2.1	The free rolling element vibration calculations:	90			
4.2.2	Oil film(cage) damping effect:				
4.2.3	The optimum bearing with good damping coefficient:	92			
4.3	Experiment analysis of the case study:				
4.3.1	Overview of p – 1110/B centrifugal pump problem:				
4.3.2	Spectrum analysis and deterioration steps:				
RESULTS C	ONCLUSION AND RECOMMENDATION:	100			
5.1	Results:	100			
5.2	Conclusion:	100			
5.3	Recommendations:	101			
References					
Appendix					

# List of figures

Figure 2.1 Maintenance Philosophies(9)	
Figure 2.2 Predictive maintenance(15)	
Figure 2.3 magnet-in-coil and coil-in-magnet velocity pickups, (19)	
Figure 2.4Accelerometer	
Figure 2.5 Proximity probe principle	
Figure 2.6 internally mounted probes (29)	
Figure 2.7 External/internal mounted probes	
Figure 2.8 externally mounted probes	
Figure 2.9 Microlog Buttons and Keypad(39)	
Figure 2.10 the Main Screen Icons	
Figure 2.11: A Setup Screen(40)	
Figure 2.12 Data Display Screens	
Figure 2.13 Example of a route Hierarchy List (42)	
Figure 2.14 an Example route Instructions screen	
Figure 3.1Typical machine train(55)	
Figure 3.2 Point of measurement	
Figure 3.3 Example of a typical time-domain vibration profile for a piece of machinery	)
Figure 3.4 Discrete (harmonic) and total (non-harmonic) time-domain vibration curves	

Figure 3.5 Typical frequency-domain vibration signatures
Figure 3.6 Fourier transform
Figure 3.7 A rotor system response
Figure 3.8 Small defects in the raceways of a bearing
Figure 3.9 More obvious wear in the form of pits(81)
Figure 3.10 Wear clearly visible over the breadth of the bearing (81)
Figure 3.11 severely damaged bearing in final stage of wear
Figure 4.1 configuration of bearing and rolling element
4.2 FigureNone damped rolling element vibration spectrum(90)
Figure 4.3 oil film damping effect spectrum on rolling element ((92
Figure 4.4 Optimum bearing with good damping coefficient (93)
Figure 4.5 Status trend of the 1104B pump
Figure 4.6 the first stage of deterioration spectrum
Figure 4.7 the second stage of deterioration spectrum
Fig 4.8 the last stage of deterioration spectrum(97)

# List of table

# Nomenclature

Symbol	Description	Unit
RPM	Speed	revolutions per minute
СРМ	Speed	Cycles per minute
RCFA	Root cause failure analysis	
TPM	Total productive maintenance	
PM	Preventive maintenance	
DE	Drive end	
WPT	Wavelet packets transform	
TIR	Total indicated run out	mm
BEP	Best efficiency point	
QA	Quality assurance	
AC	Alternating current	Ampere
DC	Direct current	Ampere
RF	Radio frequency	Hertz
FFT	Fast Fourier Transform	
	Liquid Crystal Display	
ISO	International standard organization	
API	American Petroleum Institute	
BPFI	Number of Balls or Rollers	
Bd	Ball /Roller Diameter	Inch or mm
Pd	Bearing pitch diameter	Inch or mm
θ	Contact angle	degree
BPFI	Ball passes frequency – Inner	
BPFO	Ball passes frequency – outer	
FTF	Fundamental train frequency (cage)	

BEF Ball spins frequency (rolling element)		
HFD	The high-frequency detection	
NPSH	Net Positive Section Head	meter
С	Damping coefficient	
ω	Rotational	
С	Clearance	
р	Total external load	
e	Eccentricity of the shaft	
t	time	seconds
a	acceleration	$mm/s^2$
V	velocity	mm/s
d	displacement	mm

# CHAPTER ONE INTRODUCTION

# CHAPTER ONE INTRODUCTION

### **1.1 Introduction:**

All mechanical equipment generates vibration profile a corresponding to their motions which has impacts into their operation condition. This phenomenon is true regardless of speed or whether while the operation mode is one of rotation, reciprocation, or linear motion. The fundamental frequency or one times RPM can be caused by imbalance. When a rotor has one heavy point, the weight of the heavy point, distance from the shaft center line to the center of the heavy point, an angular rate of velocity or speed all combine to create a forcing function or vector at same magnitude and direction. If the rotor has more than one heavy point, the weight will be the vector sum of all the heavy points, an unbalance cause surface force. As the heavy point rotates with the rotor, a frequency at the speed of the rotor is generated. Amplitude can be maximized by looseness and resonance. It can be minimized by mass, rigidity, and damping. Amplitude can also be maximized or minimized by the most instruments measure amplitude. Harmonic content and signal distortion are other good ways to determine problem severity. All generated frequencies are equal to events, times and speed. For example, if a rotor has imbalance, the heavy spot rotates at the same speed as the rotor. When two events have took place each revolution, as with looseness, bent shaft, or two high places on a roll, the fundamental generated are second harmonic. If three events are occurring each revolution, as with misalignment, the fundamental, second, and third harmonic are generated. Vibration analysis is applicable to all mechanical equipment. With assumption it is limited to simple rotating machinery which has running speeds above 600 revolutions per minute (rpm). Vibration profile analysis is a useful tool for diagnostics and predictive maintenance in addition to many other uses.

Predictive maintenance used multi technique to monitor and analyze the critical machines and equipment's in plants. The most of them are vibration analysis, ultrasonic, thermography, tribology, process monitoring, and visual inspection. Vibration analysis is the common predictive maintenance technique that used with maintenance management programs. Predictive maintenance and vibration monitoring characteristics of rotating machinery has become synonymous to detect budding problems and to head off catastrophic failure [1].

### **1.2 Statement of problem:**

Bearing is changing due to the vender recommendation that depend on operating hours. Sometimes the bearing change with some residual life this has impact on cost, down time and unit production.

### **1.3 Research objectives:**

To develop mathematical model (using mat lab) that can predict the bearing performance and give good evaluation to the bearing lifetime.

### **1.4 Methodology:**

The research methodology is experimental that has been done through two main categories:-

#### I. Mathematical analysis:

That will be done by establishing a mathematical model which corresponding to physical model making there of prediction. And compare the mathematical results with practical results to reach accurate comparison percentage

#### II. Practical experiment that will be done into steps as follow:

- 1- Measure bearing vibration signals.
- 2- Analyze the vibration profile of the above mentioned object using vibration signals to diagnose it through SKF microlog system.
- Doing the above steps to the bearing at multi states from good to bad status.

### **1.5** Thesis organization:

- Chapter two provide literature review which contains firstly theoretical background, previous studies that consist of Preventive and Predictive maintenance and Vibration Monitoring. After that it touches over the vibration measuring and transducers which contains the types of transducers displacement, velocity, and acceleration their theories of operation, Advantages, disadvantages and applications. MICROLOG is the latest technology in vibration measurement devices it used in the experimental study for these reasons it takes more details.

- Chapter three provides the vibration analysis and data collection that consist of Vibration analysis detection mode, vibration analysis diagnose mode, the benefits of vibration analysis and how can the vibration used to detect machinery faults with the common problems that can be detected, Points of measurement. Also it provides the scope of vibration analysis by the data collection &data acquisition, vibration profiles (Time Domain & Frequency Domain), Fourier transform to convert time domain to frequency domain, assessing the severity of vibration, Limits and standards of vibration step and theoretical study for rotating unbalance and bearing deterioration stages.

- Chapter four about material and method that dividing its analysis into two main steps the first stage is experiment analysis of the case study, and the second stage is the mathematical analysis.

- Chapter five in which the research has analyze the theoretical result and compare it with the experimental results, conclusion and recommendation.

# CAPTER TWO LITERATURE REVIEW

# **CAPTER TWO LITERATURE REVIEW**

## 2. Literature review:

This chapter is discussed the related literature through two main categories theoretical background which contain maintenance philosophes with its technique & tools. Previous studies which split into (preventive &predictive maintenance) and vibration mentoring.

### 2.1 Theoretical background:

It's about predictive maintenance, its technique's and the tools used in vibration measurement.

### 2.1.1 Maintenance philosophies:

Despite of the vast variations in different process plants operation's nature. The maintenance philosophies that can be used have quite a bit of similarity. These maintenance philosophies can be divided into four different categories:

- Breakdown or run to failure maintenance
- Preventive or time-based maintenance
- Predictive or condition-based maintenance
- Proactive or prevention maintenance.

These four different maintenance categories can be clarifying in the fig (2.1). [2]

#### I. Breakdown maintenance:

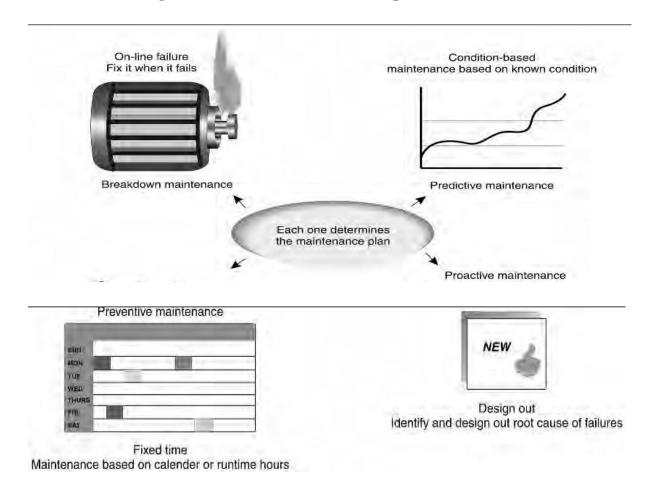
This category is to repair or replace the damaged components after allowing machinery to run to failure till the equipment reaches a complete stop.

This method is well when the equipment shutdowns has no production impact and if the labor and material costs is not matter. But the disadvantage of this approach is the maintenance department cannot be capable to plan the crisis management for continuous operation. Also it require large spare parts inventory for reacting immediately maintenance activities to cope with unexpected production interruptions. In fact, it is the most inefficient way to maintain a production facility. Because the long list of unfinished work and a set of new emergency jobs that occurred overnight, bad impact in production planning and business competition market share will take place.

Despite the many technical advances those days, Breakdown maintenance is still one of the maintenance philosophies in operating the production plants. [2]

#### **II.** Preventive or time-based maintenance

In this maintenance category the application is to schedule maintenance activities at predetermined time intervals, based on calendar days or machine's runtime hours. The repair or replacement of damaged part in equipment is carried out before obvious problems occur. This approach is good for non-continuously run equipment's, and where the personnel have enough skill, knowledge and time to perform the preventive maintenance work. But the disadvantage is the scheduled maintenance can result in performing maintenance tasks too late or too early. Equipment would be taken out for overhaul at a certain number of running hours. Also it is possible to replace components without any evidence of functional failure, and there is still has some residual life is left. It is possible to reduce the production due to unnecessary maintenance. The equipment performance can be lost in many cases through incorrect repair methods. In addition to in some cases, perfectly good machines are disassembled; their good parts removed, and new parts are installed improperly that gives troublesome results. [2]



**Figure 2.1 Maintenance Philosophies** 

#### III. Predictive or condition-based maintenance

This philosophy consists of scheduling maintenance activities only when a functional failure is detected. Scheduling for maintenance when unhealthy trends are detecting and identify troublesome parts in the machine according to Mechanical and operational conditions periodic monitoring. The machine would then be shut down at the most convenient time to replace the damaged components. If it is left these failures could result in costly secondary failures.

#### 1) The advantages of this approach are:-

1\ the maintenance events can be scheduled in an orderly fashion.

2\ allows for some lead-time to purchase parts for the necessary repair work.

3\ reduce the need for a large inventory of spares. Since maintenance work is only performed when needed, there is also a possible increase in production capacity.

#### 2) The disadvantage is:-

That maintenance work may increase actually due to an incorrect assessment of the machines deterioration. To track the unhealthy (vibration, temperature or lubrication) trends in requires

1\ specialized equipment to monitor these parameters.

2\ provide training to personnel (or hire skilled personnel).

The alternative of tracking the unhealthy trends is to outsource this task to a knowledgeable contractor to perform the machine-monitoring

duties. It is very important that the management supports the maintenance department by providing the necessary equipment with adequate personnel training. The personnel should be given enough time to collect the necessary data and be permitted to shut down the machinery when problems are identified. [2]

#### **IV.** Proactive or prevention maintenance

This philosophy is focusing on tracing all failures to their root cause. If each failure is analyzed and proactive measures are taken it allow to ensure that they are not repeated. This approach utilizes all of the predictive and preventive maintenance techniques in conjunction with root cause failure analysis (RCFA).

#### 1) The advantages of this approach are:-

1\RCFA detects the problems that cause defects.

2\ it ensures that appropriate installation and repair techniques are followed up.

3\ help to take the optimum decision if the equipment needs to redesign or modification to avoid recurrence of such problems.

4\ reduces the large spare parts inventory, because it is possible to schedule maintenance repairs on equipment in an orderly fashion as in the predictive-based program, but additional efforts are required to provide improvements to reduce or eliminate potential problems from occurring repeatedly.

 $5\$  the orderly scheduling of maintenance allows lead-time to purchase parts for the necessary repairs, because maintenance work is only performed when it is required.

#### 2) The disadvantage is:-

1\ the employees must be extremely knowledgeable in the practices of preventive, predictive and prevention/proactive maintenance. This requires special and costly training.

2\ the work may require outsourcing knowledgeable contractors who will have to work closely with the maintenance personnel in the RCFA phase.

3\Proactive maintenance also requires procurement of specialized equipment and properly trained personnel to perform all these duties.

#### V. The Appraisal of Maintenance philosophies.

Machinery maintenance in industry has developed from breakdown to time based preventive. Recently, the predictive and proactive maintenance are the most popular.

Breakdown maintenance was practiced in the early days of production technology. Equipment was allowed to run until a functional failure took place. Secondary damage was often observed with a primary failure. This reason was lead to time-based maintenance that called also preventive maintenance. In this case, equipment was be taken out of production for overhaul after completing a certain number of running hours, even if there was no evidence of a functional failure. This increase the maintenance costs by replacing machinery components even there was still some functional lifetime left in them.

An approach to schedule the maintenance or overhaul of equipment based on the condition of the equipment is required to avoid high maintenance costs when using preventive maintenance.

Predictive maintenance requires continuous equipment monitoring to detect and diagnose defects. When a defect is detected, the maintenance work is planned and executed. Till the early 1980s, justification spreadsheets were used in order to obtain approvals for condition-based maintenance programs. Today, predictive maintenance has reached an advanced industry level.

The advantages of predictive maintenance are accepted in industry today, because the clear tangible benefits in terms of early warnings about mechanical and structural problems in machinery. This method is now an essential tool for detection and diagnosis that has impact in maintenance and operational costs reduction through repair downtime and inventory control. In the continuous process industry, such as oil and gas, power generation, steel, paper, cement, petrochemicals and others, the adoption of the predictive maintenance is required because the penalties of even a small amount of downtime are immense.

Predictive maintenance has helped to improve productivity, product quality, profitability and overall effectiveness of manufacturing plants. Predictive maintenance also uses the actual operating conditions of the plant equipment's and systems to optimize the total plant operation.

12

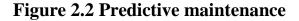
The manufacturers have been embarking upon a predictive maintenance program to become more aware of the specific equipment problems and after that try to identify the failures root causes. This led to proactive maintenance the developed kind of maintenance. [2]

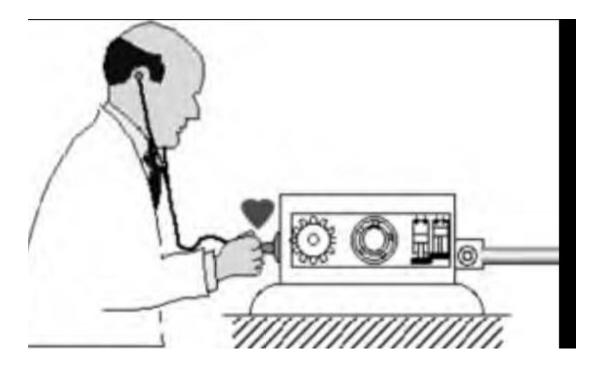
#### 2.1.2 Predictive maintenance and its techniques :

Basically predictive maintenance is a condition-driven preventive maintenance. In industrial or in-plant the average life statistics are not used to schedule maintenance activities. As figure (2.2) illustrate that predictive maintenance monitors mechanical condition, equipment efficiency and derive the approximate time of a functional failure.

A comprehensive predictive maintenance program utilizes a combination of the most cost-effective tools to obtain the actual operating conditions of the equipment and plant systems. The maintenance schedules are selected on the basis of this collected data. Predictive maintenance uses various techniques such as vibration analysis, oil and wears debris analysis, ultrasonic, thermography, performance evaluation and other techniques to assess the equipment condition. Predictive maintenance techniques actually have a very close analogy to medical diagnostic techniques. Whenever a human body has a problem, it exhibits a symptom. The first stage is detection stage the system provides the information. Furthermore, if required, more diagnostic tests are done to diagnose the problem. On this basis, suitable treatment is recommended.

Similarly if the defects take place in a machine always exhibit a symptom in the form of vibration or some other parameter. However, this may or may not be easily detected on machinery systems with human perceptions for that reason predictive maintenance techniques come to assistance.





These techniques detect symptoms of the defects that have occurred in machines and assist in diagnosing the exact defects that have occurred. Also it is possible to estimate the defects severity in many cases. The specific utilized techniques are depending on the type of plant equipment, their impact on production or other key parameters of plant operation. [2]

#### • Predictive maintenance techniques:

There are many types of predictive maintenance techniques, such as:-

#### I. Vibration monitoring:

It is the most effective technique which has been used for detecting the mechanical defects in rotating equipment.

#### **II.** Acoustic emission:

This type could be used to detect and locate the structural and pipelines Cracks to be monitor continuously.

#### III. Oil analysis:

This technique is useful for monitoring the certain microscopic particles contents that related to the bearings and gears condition through lubrication oil analysis.

#### **IV.** Particle analysis:

This technique is to collect and analysis the debris that released from worn machinery components, whether in reciprocating machinery, gearboxes or hydraulic systems, to provide vital information on these components deterioration.

#### V. Corrosion monitoring:

Ultrasonic testing is best and applicable technique to measure and track the thickness in the process facilities such as pipelines, offshore structures and other.

#### VI. Thermography:

This is non-contact diagnostic technique use in analyzing active electrical and mechanical equipment. The method can detect thermal or mechanical defects in generators, overhead lines, boilers, misaligned couplings and many other defects. It can also detect cell damage in carbon fiber structures on aircrafts.

#### VII. Performance monitoring:

This is a very effective technique to determine the operational problems in equipment. The efficiency of machines provides a good insight on their internal conditions.

Despite all these methods, it needs to be cautioned that there have been cases where predictive maintenance programs were not able to demonstrate tangible benefits for an organization. The predominant causes that lead to failure of predictive maintenance are inadequate management support, bad planning and lack of skilled and trained manpower. Upon activating a predictive maintenance program, it is very essential to decide on the specific techniques to be adopted for monitoring the plant equipment. The various methods are also dependent on type of industry, type of machinery and also to a great extent on availability of trained manpower. It is also necessary to take note of the fact those predictive maintenance techniques require technically sophisticated instruments to carry out the detection and diagnostics of plant machinery. These instruments are generally very expensive and need technically competent people to analyze their output. The cost implications, whether on sophisticated instrumentation or skilled manpower, often lead to a question mark about the plan of adopting predictive maintenance philosophy.

However, with management support, adequate investments in people and equipment, predictive maintenance can yield very good results after a short period of time. [2]

#### 2.1.3 Vibration measuring and Transducers:

Vibration can be measured or expressed in three basic quantifies: displacement, velocity and acceleration. This can be done through transducer a device that converts one type of energy, such as vibration, into a different type of energy, usually an electric current or voltage. Transducer and readout instruments are commonly used for measuring vibration with their capabilities, units of vibration and applications. Accelerometers of the voltage output type are the most commonly used with handheld data collector or data analyzers. Regularly calibration checking is recommended for QA practices, as transducers may be damaged.

On most machines, vibration is usually present at other frequencies than at the frequency corresponding to the speed of rotation (or *synchronous*, called here 1 x), making the time waveform more complex than a simple sine wave. Readout instruments perform the conversion for simple or complex vibration signals by electronic integration (i.e. Acceleration to Velocity to Displacement). By the physics laws, at low frequencies, a given vibration will have large Displacement and small Acceleration. The opposite also applies at high frequencies, high Acceleration, low Displacement. Across all frequencies, Velocity is reasonably constant that is why it is the most commonly used parameter for most machine vibration. One of the most important considerations for any application is to select the transducer. The various vibration transducers are discussed below.

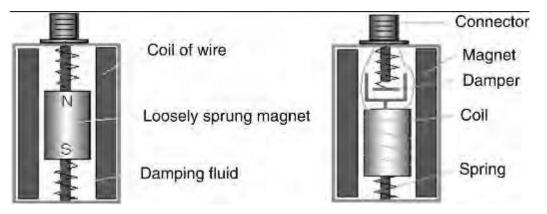
#### 1- Velocity transducers:

It is a very common used transducer for monitoring the vibration of rotating machinery. This type of vibration transducer is easily to install on most analyzers, and inexpensive compared to other sensors. For these reasons, the velocity transducer is ideal for general purpose machinemonitoring applications. Velocity pickups have been used as vibration transducers on rotating machines for a very long time, and these are still utilized for a variety of applications today.

#### • Velocity pickup theory of operation:

There are types of velocity pickup such as coil-in-magnet type and magnet-in-coil type .In coil-in-magnet type when a coil of wire is moved through a magnetic field (Figure 2.3); a voltage is induced across the end wires of the coil. The transfer of energy from the flux field of the magnet to the wire coil generates the induced voltage. As the coil is forced through the magnetic field by vibratory motion, a voltage signal correlating with the vibration is produced.

Figure 2.3 magnet-in-coil and coil-in-magnet velocity pickups



The magnet-in-coil type of sensor is made up of three components: a permanent magnet, a coil of wire and spring supports for the magnet. The pickup is filled with oil to dampen the spring action. The relative motion between the magnet and coil caused by the vibration motion induces a voltage signal. The velocity pickup is a self-generating sensor and requires no external devices to produce a voltage signal. The voltage generated by the pickup is directly proportional to the velocity of the relative motion.

Velocity transducers are manufactured differently for horizontal or vertical axis mounting referring to the gravity forces. The velocity sensor has a sensitive axis that must be considered when applying them to rotating machinery. Velocity sensors are also susceptible to cross axis vibration, which could damage a velocity sensor.

#### • Velocity pickup number of sensors:

When choosing the number of sensors to be used. The fact should be taken is all vibration sensors measure motion along their major axis. Due to the structural asymmetry of machine cases, the vibration signals in the vertical, horizontal and axial directions may differ (with respect to the shaft). Where possible, a velocity transducer should be mounted in the vertical, horizontal and axial planes to measure vibration in the three directions. For that reason three sensors will enough to provide a complete picture of the vibration signature of the machine.

#### • Mounting of velocity transducers:

To get best results the mounting location must be flat, clean and slightly larger than the velocity pickup. If it is possible, it should be clamped with a separate mounting enclosure. The surface should be drilled and tapped to accommodate the mounting screw of the sensor. Whenever a velocity pickup is exposed to hazardous environments such as high temperatures, radioactivity, water or magnetic fields, special protection measures should be taken. Magnetic interferences should be taken into account when measuring vibrations of large AC motors and generators. The alternating magnetic field that these machines produce may affect the coil conductor by inducing a voltage in the pickup that could be confused with actual vibration. Magnetic shields can be used to reduce the effect of the alternating magnetic field. A magnetic shield would be required is to hang the pickup close to the area where vibrations must be taken. If significant vibrations are observed, a magnetic shield may be required.

#### • Sensitivity of velocity transducers:

Sensitivities are normally expressed in mV/in/s or mV/mm/s. General values are in the range of 500 mV/in/s to 750 mV/in/s (20–30 mV/mm/s). The sensitivity of the velocity pickup is constant over a specified frequency range between 10 Hz and 1 kHz. At low frequencies of vibration, the sensitivity decreases because the pickup coil is no longer stationary with respect to the magnet, or vice versa. This decrease in pickup sensitivity usually starts at a frequency of approximately 10 Hz, below which the pickup output drops exponentially. The significance of this fact is that amplitude readings taken at frequencies below 10 Hz using a velocity pickup are inaccurate.

#### • Frequency response of velocity transducers:

Velocity pickups have different frequency responses depending on the manufacture. Most pickups have a linear frequency response range in the order of 10 Hz–1 kHz. This is an important consideration when selecting a velocity pickup for a rotating machine application. The pickup's frequency response must be within the expected frequency range of the machine.

#### • Calibration of velocity transducers:

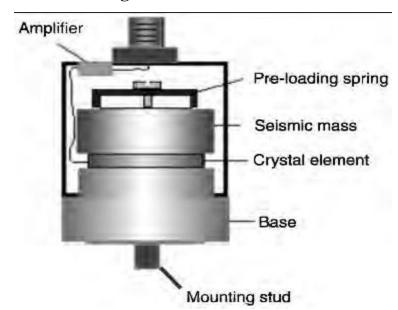
Velocity pickups should be calibrated on an annual basis. The sensor should be removed from service for calibration verification. Verification is necessary because velocity pickups are the only industrial vibration sensors with internal moving parts that are subject to fatigue failure. Verification should include a sensitivity response vs. frequency test. This test will determine if the internal springs and damping system have degraded due to heat and vibration. The test should be conducted with a shaker capable of variable amplitude and frequency testing.

#### • Advantages of velocity transducers:

- i. It is Easy to install,
- ii. It has Strong signals in mid-frequency range
- iii. It is not requiring any external power.
- Disadvantages of velocity transducers:
- i. Relatively large and heavy
- ii. Sensitive to input frequency
- iii. Narrow frequency response
- iv. Moving parts
- v. Sensitive to magnetic fields

#### **2-** Acceleration transducers:

Accelerometers are the most popular transducers used for rotating machinery applications. They are lightweight transducers with a wide frequency response range. Accelerometers are extensively used in many condition-monitoring applications. Components such as rolling element bearings or gear sets generate high vibration frequencies when defective. Machines with these components should be monitored with accelerometers. The accelerometer installation must be considered for an accurate and reliable measurement. Accelerometers are designed for mounting on machine cases. This can provide continuous or periodic sensing of case motion in terms of acceleration.



**Figure 2.4 Accelerometer** 

#### • Acceleration transducers theory of operation:

Accelerometers are inertial measurement devices that convert mechanical motion into a voltage signal. The inertial mechanical motion is relative to a mass and the signal is proportional to the vibration. Vibration accelerometer is using the piezoelectric principle.

Following to the Newton's third law of motion: body acting on another will result in an equal and opposite reaction on the first. Accelerometers consist of a piezoelectric crystal and a small mass normally enclosed in a protective metal case. When the accelerometer is subjected to vibration, the mass exerts a varying force on the piezoelectric crystal, which is directly proportional to the vibratory acceleration. The charge produced by the piezoelectric crystal is proportional to the varying vibratory force. Some sensors have an internal charge amplifier, while others have an external charge amplifier. The charge amplifier converts the charged output of the crystal to a proportional voltage output in mV/g.

#### • Mounting of acceleration transducers:

There are four primary methods used for attaching sensors to monitoring locations. These are:

- I. stud mounted,
- II. adhesive mounted,
- III. magnet (double leg or flat) mounted
- IV. Non-mounted e.g. using handheld probes or stingers.

Each method affects the high-frequency response of the accelerometer. Stud mounting provides the widest frequency response and the most secure, reliable attachment. The other three methods reduce the upper frequency range of the sensor. In these cases, the sensor does not have a very secure direct contact with the measurement point. Inserting mounting pieces, such as adhesive pads, magnets or probe tips, introduces a mounted resonance. This mounted resonance is lower than the natural resonance of the sensor and reduces the upper frequency range. A large mounting piece causes lower mounted resonance and also lowers the usable frequency range of the transducer. The mounting methods typically used for monitoring applications are discussed in more detail below.

The stud/bolt mounting method is the best method available for permanent mounting applications. This method is accomplished by screwing the sensor in a stud or a machined block. This method permits the transducer to measure vibration in the most ideal manner and should be used wherever possible. The mounting location for the accelerometer should be clean and paint-free. The mounting surface should be spot-faced to achieve a smooth surface. The spot-faced diameter should be slightly larger than the accelerometer diameter. Any irregularities in the mounting surface preparation will translate into improper measurements or damage to the accelerometer. The adhesive or glue mounting method provides a secure attachment without extensive machining. However, when the accelerometer is glued, it typically reduces the operational frequency response range or the accuracy of the measurement. This reduction is due to the damping qualities of the adhesive. Also, replacement or removal of the accelerometer is more difficult than with any other attachment method. For proper adhesive bonding, surface cleanliness is of extreme importance. The magnetic mounting method is typically used for temporary measurements with a portable data collector or analyzer. This method is not recommended for permanent monitoring. The transducer may be inadvertently moved and the multiple surfaces and materials of the magnetic may interfere with high-frequency signals.

By design, accelerometers have a natural resonance which is 3–5 times higher than the high end of the rated frequency response. The frequency response range is limited in order to provide a flat response over a given range. The rated range is achievable only through stud mounting. As mentioned before, any other mounting method adversely affects the resonance of the sensor, such as the reliable usable frequency range.

#### • Sensitivity of acceleration transducers :

Accelerometers utilized for vibration monitoring are usually designed with a sensitivity of 100 mV/g .Other types of accelerometers with a wide range of sensitivities for special applications such as structural analysis, geophysical measurement, very high frequency analysis or very low speed machines are also available.

## • Frequency range of acceleration transducers:

Accelerometers are designed to measure vibration over a given frequency range. Once the particular frequency range of interest for a machine is known, an accelerometer can be selected. Typically, an accelerometer for measuring machine vibrations will have a frequency range from 1 or 2 Hz to 8 or 10 kHz. Accelerometers with higher-frequency ranges are also available.

#### • Calibration of acceleration transducers:

Piezoelectric accelerometers cannot be recalibrated or adjusted. Unlike a velocity pickup, this transducer has no moving parts subject to fatigue. Therefore, the output sensitivity does not require periodic adjustments. However, high temperatures and shock can damage the internal components of an accelerometer.

When the reliability of an accelerometer is doubtful, a simple test of the transducer's bias voltage can be used to determine whether it should be removed from service. An accelerometer's bias voltage is the DC component of the transducer's output signal. The bias voltage is measured with a DC voltmeter across the transducer's output and common leads with the power on. At the same time, the power supply should also be checked to eliminate the possibility of improper power voltage affecting the bias voltage level of the sensor.

#### **3- Displacement transducers:**

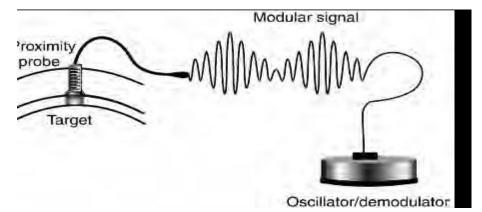
It is also known as Eddy current transducers or proximity probes they are the preferred vibration transducers for vibration monitoring on journal bearing equipped rotating machinery. Typical applications are predominantly high-speed turbo machinery. Eddy current transducers are the only transducers that provide displacement of shaft or shaft relative to the bearing vibration measurements. Several methods are usually available for the installation of Eddy current transducers, including internal, internal/external, and external mounting.

#### • Displacement transducers theory of operation

Eddy current system is a matched component system which consists of a probe, an extension cable and an oscillator/demodulator (Figure 2.5). A high-frequency radio frequency (RF) signal at 2 MHz is generated by the Oscillator/demodulator. This is sent through the extension cable and radiated from the probe tip.

Eddy currents are generated in the surface of the shaft. The oscillator/demodulator demodulates the signal and provides a modulated DC voltage, where the DC portion is directly proportional to the gap (distance) and the AC portion is directly proportional to vibration. Eddy current transducer can be used for both radial vibration and distance measurements such as the axial thrust position and shaft position.

#### **Figure 2.5 Proximity probe principle**



#### • Number of displacement transducers:

All vibration transducers measure motion in their mounted plane. In other words, shaft motion is either directed away from or towards the mounted Eddy current probe, this allow to measure the radial vibration is in this way. One Eddy current transducer system per bearing is adequate on smaller, less critical machines. The single Eddy current probe measures the shaft's vibration in that given plane. Therefore, the Eddy current probe should be mounted in the plane where the largest vibrations are expected. On larger, more critical machines, two Eddy current transducer systems are normally recommended per bearing. The probes for this type of installation are mounted 90° apart from each other. Since the probes will measure the vibration in their respective planes, the shaft's total movement within the journal bearing is measured. An 'Orbit' or Cartesian product of the two vibration signals can be constructed when both Eddy current transducers are connected to an oscilloscope.

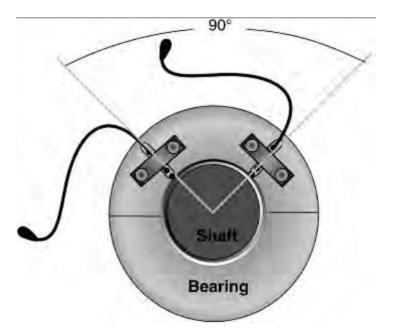
## • Mounting methods and orientation of Displacement transducers:

As most of the bearing housings on which probes are attached are horizontally split, transducers are commonly mounted at  $45^{\circ}$  on both sides of the vertical plane. If possible, the orientation of the transducers should be consistent along the length of the machine train for easier diagnostics. In all cases, the orientations should be well documented.

Care must be exercised in all installations to ensure that the Eddy current probes are mounted perpendicular to the shaft centerline. Deviation by more than  $1-2^{\circ}$  will affect the output sensitivity of the system.

#### **1- Internal mounting**

The Eddy current probes are mounted inside the machine or bearing housing with a special bracket (Figure 2.4). The transducer system is installed and gapped properly prior to the bearing cover being reinstalled. **Figure 2.6 internally mounted probes** 



# Advantages of internal mounting

- I. Less machining required for installation.
- II. True bearing-relative measurement is possible.
- III. The Eddy probe has an unconstrained view on the shaft surface.

# **Disadvantages of internal mounting**

- I. There is no access to probe while the machine is running.
- II. Cables must be tied down with extreme care, because they might break due to 'Wind age.
- III. Transducer cable exits must be provided.
- IV. The care must be taken to avoid oil leakage.

# 2- External/internal mounting

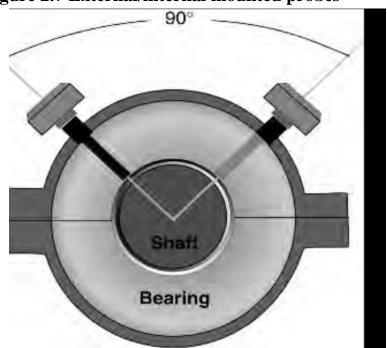
External/internal mounting is accomplished when Eddy probes are mounted with a mounting adaptor (Figure 2.6). These adaptors allow external access to the probe, but the probe tip itself is inside the machine or bearing housing. While drilling and tapping the bearing housing or cover, it is important to ensure that the Eddy probes are installed perpendicular to the shaft centerline. In some cases, due to space limitations, external/internal mounting is accomplished by drilling or making use of existing holes in the bearing itself, usually at an oil-return groove.

# Advantages of external/internal mounting

- I. Eddy probe replacement is possible while machine is running.
- II. Eddy probe has an unconstrained view on the shaft.
- III. Gap may be changed while machine is running.

# Disadvantages of external/internal mounting

- I. May not be true bearing-relative measurement.
- II. More machining required.
- III. Long probe/stinger length may cause resonance



#### Figure 2.7 External/internal mounted probes

# **3- External mounting**

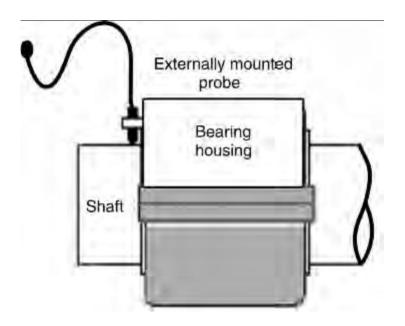
It is most old turbo machines were not equipped with radial probes. The construction of older machines may not provide ideal installation of probes. External Eddy probes are mounted on such machines (Figure 2.6). It is usually a last resort installation. The only valid reason for using this method is inadequate space available within the bearing housing for internal mounting. Special care must be given to the Eddy probe viewing area, and mechanical protection must be provided to the transducer and cable.

# Advantages of external mounting

It may record electrical and/or mechanical run out of the shaft

# **Disadvantages of external mounting**

- I. It is the most inexpensive installation.
- II. It requires mechanical protection. Provision must be made for the transducer's cable protruding from the bearing housing.



# Figure 2.8 externally mounted probes

This can be accomplished by using an existing plug or fitting, or by drilling and tapping a hole above the oil line. The RF field emitted from the probe tip of an Eddy current transducer is shaped like a cone at approximately 45° angles. Clearance must be provided on all sides of the probe tip to prevent interference with the RF field. For instance, if a hole is drilled in a bearing for probe installation, it must be counter-bored to prevent side clearance interference. It is important to ensure that collars or shoulders on the shaft do not thermally grow under the probe tip as the shaft expands due to heat. [2]

#### Mounting of vibration transducers

The more firmly a contact transducer is mounted to the machine, the more faithfully the vibration level and pattern will be measured. If handheld, response is limited to about 1000Hz, and large variations are likely between readings by different people. Marking measuring points with a shallow dimple will help repeatability. At least the measuring point should be marked. Acceptable results are however obtained with a handheld probe on rolling element bearings when measuring in the ultrasonic range, and above. Grease couplant may be needed. Magnet mounting is convenient and on a flat surface can give faithful response up to about 2000Hz. High strength types are claimed to give up to perhaps 10 000Hz response. Types with two parallel feet hold well on less-than-flat surfaces. The holding force of the magnet will not be overcome below acceleration of 4g vertical or 2g horizontal mounting. Studs, glued on or screwed in, are required for the best repeatability and highest frequency response. Some types use a quick-twist connector, requiring only one hand to mount.

#### **Transducers selection:**

Transducer selection depends upon the job. For example, defects in antifriction bearings for some speeds are best measured with a velocity transducer or an accelerometer. Lower speeds require a displacement transducer. If you want to measure how much one object is moving relative to another (commonly called relative motion measurement), a displacement transducer must be used. Some examples are: relative shaft motion in fluid film bearings and press/nip rolls; clearance in antifriction bearings; amount of misalignment; and bend in a shaft.

Much has been written on the selection and use of transducers, and most of it is inaccurate. The following rules may be helpful:

1. If you want to measure the amplitude of frequencies below 10

Hertz (Hz) or measure relative motion, a displacement transducer must be used. Some displacement transducers can be used to measure frequencies up to 1,000 Hz.

2. For measuring frequencies between 10 Hz and 2,000 Hz, a velocity transducer is the best selection in most cases.

3. For measuring all frequencies above 2,000 Hz, an accelerometer must be used. When an accelerometer is used, it must be either screwed down or glued down. If this rule is not followed, the amplitude of some frequencies can be over or understated as much as four times. Also, some higher frequencies can appear as wide-banded noise and not as discrete frequencies with sidebands. Both situations cause serious errors in analysis. Accelerometers can be used to measure frequencies below 2,000 Hz. However, the requirement for hard mounting is time consuming, and would not be the choice of an informed analyst.

4. When diagnosing problems in compressors and pumps, data from a pressure transducer is often required. These transducers are calibrated in mV/PSI and measure the pressure fluctuations in gases and liquids.

Refer to the frequency response curves in Fig. 1-27 for transducer selection. Data is often required from more than one type of transducer for accurate diagnosis.

#### • Continuous Motoring

Without question, high-speed turbo machinery and other high-speed machines should be monitored on a continuous basis. The displacement transducer is the best choice for measuring relative motion of the shaft. One of the biggest problems is the failure to monitor transducer gap voltage. In some cases, a machine can fail, and the vibration level may not increase significantly. However, the gap voltage will increase substantially. The other problem is a misunderstanding of the frequency response curves discussed in Chapter One. The result is that accelerometers are not installed to monitor high frequencies. It is essential to monitor high frequencies with an accelerometer. If this is not done, the machine can wreck, and a warning will never be observed by the displacement transducer. The major cause of such failures is normally high frequency rubs.

Diagnostic technology has developed to the point that the computer is programmed to diagnose problems. This on-line or off-line expert system is now available and is economical for most plants. [2]

#### 4- Microlog:

Microlog system is vibration data collector with host software analyzer to analyze the picked up signal from machinery through Microlog accelerometer, it could be used by machinery maintenance personnel who wish to collect and analyze vibration data from their rotating machinery. It has been used in the experimental analysis because it is the most accurate system while the experiment took place. The microlog main screen provides access to all operating modes. Available application modes are identified with icons on the main screen's display area. [3]

#### • Microlog Terminology

#### i. Hierarchy List:

A plant's list of measurement points organized by sets sub sets, and machines. A hierarchy list usually has three or four organizational levels branching from the plant ID.

**ii. Point:** Defines a machinery location at which measurement data is collected and defines the measurement type. Each reading is taken at a specific measurement point on the machine. A descriptive point ID is issued for each point.

**iii. Machine:** The final hierarchy level from which only measurement point's branch.

**iv.** Set: Equipment may be grouped into optional sets for organizational purposes. Sets are used to organize machines help to locate specific machines and points quickly.

v. Route: A point collection sequence routes are downloaded to the microlog to simplify and organize measurement point collection. The micolog displays five operating mode icons on its main screen:

**1. Route Mode**: for collecting route measurements downloaded from SKF machine analyst host software.

34

**2. Review Mode:** for reviewing measurement data stored in the microlog's memory.

**3.** Non Route Mode: for collecting measurement data for points not previously downloaded from machine analyst software.

**4. Balancing Mode**: resolves single-plane, two-plane, and static-couple balances with high precision.

5. Setup Mode: for setting microlog's system preferences. [3]

#### • The Data Collector:

The microlog data collector is a lightweight, portable, two channel, route based data acquisition and storage terminal. It collects machinery vibration, temperature, and other condition monitoring measurements. Together with visual observations, the microlog allows for detailed machine condition analyses in a harsh industrial environment. A variety of input devices may be used with microlog. Vibration measurements are collected with a handheld probe, magnetically mounted probe, permanently mounted sensors, or from an installed monitoring system. Temperature measurements are collected with a non-contact infrared sensor or with a contact probe. Values read from other indicators may be entered into the microlog by pressing the appropriate alpha/numeric keys on the microlog keypad. In addition to its function as a data collector, the microlog has all the features and performances of a powerful analyzer to capture and display high resolution spectra for detailed analysis. A Fast Fourier Transform (FFT) frequency spectrum is available for display on the color LCD (Liquid Crystal Display) screen. [3]

#### • Machine Analyst Host Software:

SKF's Machine Analyst support software works with the microlog to help machinery maintenance personnel set up measurements and manage machine condition data. Machine analyst host software helps maintenance personnel to understand the true condition of rotating machinery and to base maintenance decisions on actual machinery condition. Machine analyst host software facilitates easy measurement set up and presents collected machinery data in statistical, report, and graphic plot format to obtain useful analysis data. [3]

#### • Component Use Sequence:

For the easiest and most effective system use, you should:

#### I. Use machine analyst to:

- 1. Organize plant into database plants, sets, and machines.
- 2. Add measurement points to these database machines, configuring each point's measurement and alarm settings as it is created.
- 3. Create a collection route of measurement points.
- 4. Download the collection route to the microlog.
- 5. Upload the collected data to machine analyst.
- 6. Use machine analyst's plots and reports to analyze, trend, report, and store measurement data.

#### II. Use the microlog to:

Walk the route, collecting data for each pre-determined measurement point. [3]

#### • The Buttons using:

#### i. Enter Buttons:

The two enter buttons are located on either side of the LCD display. In any setup screen, press one of the enter buttons to select the highlighted option, or to progress through data collection screens.

#### ii. Function Buttons:

The bottom of the main screen displays up to four command words that describe the current functions of the microlog's four function buttons. Each command word on the display represents the function of the button beneath

#### iii. Arrow Buttons:

The four buttons in the center of the microlog are identified as the up, down, left, and right arrow buttons. Use of these buttons depends on your microlog activity. [4]

#### • Main Screen:

On the main screen, use arrow buttons to highlight the desired microlog operating mode option, then press an enter button to initiate the mode. In a setup screen, use the up/down arrow buttons to highlight the field whose setting you wish to change. With the desired field highlighted, use the right/left arrow buttons to open/close the field, displaying its available settings. Then use the up/down arrow buttons to highlight the field's new setting and press enter.



Figure 2.9: Microlog Buttons and Keypad.

Figure 2.10: The Main Screen Icons.

Main Menu		👻 19:38
ROUTE	REVIEW	NON ROUTE
BALANCING	BUMP TEST	
Help	АЬ	out

Setup	<mark>₽</mark> 21:56
Timeout: IrDA: Baud Rate: Flow Ctrl: Dflt Units: Date Format: Time Zone: Date/Time: Memory: Purge Cache: Route Font: Language:	Internal 15 mins Disabled 115200 Off Metric MM/DD/YYYY (GMT-08:00) Pacific Ti 21:56 04/13/2005 -seect cmd- On Small English
Help Co	nt Cont. + Appy

Figure 2.11: A Setup Screen.

# • Hierarchy List Screens

The microlog, can display one route hierarchy at a time. To select a route if multiple routes are downloaded to the microlog:

1. All route hierarchy lists start with a route hierarchy item. Use the left/right arrows to navigate up and down the hierarchy list to make the route hierarchy item active, all available routes display in a list to the right of the route hierarchy item.

2. With the route list displayed, use the up/down arrows to highlight the name of the route you wish to collect (the sub-item).

3. Use the right arrow button to select the highlighted route. A progress dialog displays as the microlog loads the route into memory. Branching from the active route hierarchy item is plant, set, machine, and point hierarchy items. Use the arrow key procedures described above to navigate these additional hierarchy items and their sub-items. [4]

#### • Data Display Screens:

In a spectral display screen, use the left/right arrow buttons to move the spectrum's cursor, and to control display expansion. Use the up/down arrow buttons to adjust the spectrum's full-scale range.

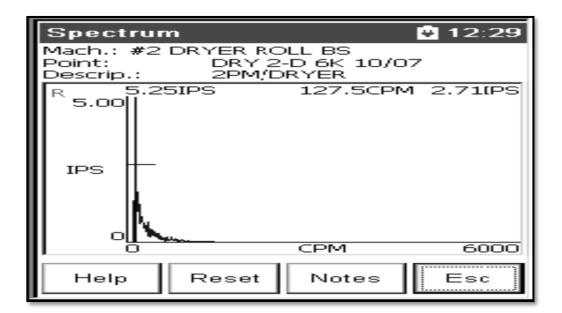


Fig 2.12 Data Display Screens

# • Route

To instruct the microlog user on how to collect route measurement data a route is a list of measurement points arranged in a sequence that facilitates the most efficient data collection path through your plant / mill. The machine analyst host software's route feature allows you to reorganize your measurement hierarchy's order into measurement collection sequences (routes) to help in perform the most efficient data collection. [4]

 Data Collection
 12:48

 Boute
 DRYER 1.

 Ground Floor RTE \*

 Ground Floor RTE \*

 Set ×>#2 DRYER ROLL BS

 Set ×= 2 DRYER ROLL BS

 Set ×=

Figure 2.13: Example of a route Hierarchy List

# A. Route Selection

Use the left/right arrows to navigate the hierarchy list to make the route hierarchy level active. When the route level is active, all available routes display in a drop down list to the right of the route hierarchy item. With the route drop down list displayed, use the up/down arrows to highlight the name of the route you wish to make active, and then use the right arrow button to select the highlighted route. A progress dialog appears as the microlog loads the route into memory, and the specified route's hierarchy displays in the route list. [4]

# **B.** How to Begin Route Data Collection

- a. From the main screen, select the route mode option. You are placed into route mode and the hierarchy list displays the downloaded route.
- b. If the Microlog contains more than one route, specify the active route.

#### C. Route Instructions

Route instructions may have been entered for the route when the route was created in Machine Analyst. If so, when the route is selected, its route instructions automatically display. Route instructions may also be accessed at any time by pressing the Help screen's Rt. Inst. function button while in route mode.

Figure 2.14: An Example route Instructions screen.

Route Instructions. 🗳 07:	:29
TO COLLECT DATA FOR THIS ROUTE: USE TWO SENSOR LOCATIONS. ONE ON THE VERTICAL PLANE OF THE ROTOR (INDICATED WITH A V IN THE POINT NAME) AND ONE ON THE HORIZONTAL PLANE OF THE ROTOR (INDICATED WITH AN H IN THE POINT NAME). TAKE TWO POINTS FOR EACH PLANE, VELOCITY AND ENVELOPED ACCELERATION.	
OK	

# 2.1.4 State of the arts:

The literature review of this research has stated through two categories theoretical background and previous studies which has been stated as follow:

- **1.** Preventive & predictive maintenance.
- **2.** Centrifugal pump vibration readings.

- **3.** Using torsional vibration analysis as synergistic method for crack detecting in rotating equipment.
- **4.** Optimal Component Replacement Decisions Using Vibration Monitoring and the Proportional-Hazards Model.
- **5.** Predictive maintenance of pump using condition monitoring.
- **6.** Using Vibration Monitoring as Predictive Maintenance Technique (Case Study): Steam Turbine Power Generator.
- 7. Appling predictive maintenance techniques to utility system.
- **8.** Using Vibration Monitoring as Predictive Maintenance Technique.
- **9.** Continuous-time predictive-maintenance scheduling for a deteriorating system.
- **10.** Statistical-based or condition-based preventive maintenance.
- **11.** Condition monitoring technique and methodology.
- **12.** Vibration monitoring of rolling element bearings by the high-frequency resonance technique.

# **2.2 Previous studies:**

## **2.2.1 Preventive and Predictive maintenance:**

Michael Walsh, in Predictive& Preventive maintenance Pumps &systems has reached that preventive maintenance of equipment, save money in the long term and can help to prevent the development of serious hazards that leading to a safety problem. Predictive maintenance can help to take future actions to optimize the operational efficiency through historical data and recordkeeping to understand trends and uncover anomalies in a process that designed for your specific system, built out of regular observation. "Predictive and preventive maintenance are different, but both are complementary and one should not be conducted exclusive of the other. Each of them can help to protect equipment and people". [5], I do agree this result because the estimated maintenance cost could be less than sudden downtime losses. In MC nally institute the research about the topic Centrifugal pump vibration readings concluded to it's necessary to be concerned about vibration because it has a major effect on the pump performance such as:

- 1- The mechanical seal is directly related to shaft movement.
- 2- Packing is sensitive to radial movement of the shaft.
- 3- Bearings are designed to handle both a radial and axial load.

4- Critical dimensions and tolerances such as wear ring clearance and impeller setting will be affected by vibration.

- 5- Pump and motor hold down bolts can become loose.
- 6- Bearing seals are very sensitive to shaft radial movement.

The common sources of vibration in rotary pumps are that mentioned above, and their solutions are different according to the source type such as Mechanical, Hydraulic and other causes of vibration. [6] Aerospace conference IEEE 6-13 March 2004 results was the torsional vibration is the method that used for monitoring and tracking small changes in crack growth of Westinghouse 93A reactor coolant pump shafts. Through characteristic changes in the natural torsional vibration frequencies that are associated with shaft crack propagation. It is the best and the most effective nondestructive method to keep tracking and predicting the failure that may take place in the rotating equipment. [7]I do agree this result because it confirms my research's assumption vibration monitoring is the best NDT method for all rotating equipment. P. J. Vlok, J. L. Coetzee, D. Banjevic, A. K. S. Jardine and V. Makis in Journal of operational research society 2002 has taken Circulating pump in a petrochemical plant as a case study that used to determine the optimal replacement policy for a critical item which is subject to vibration monitoring. The goal is to reduce the average long run cost per unit time of the pump part replacement through scientific policy. [8]To reduce the spare part consumption through equipment base monitoring is the main objective of my research because changing part while it has some residual life time has negative impact economically in the equipment operation in long term. "In order to achieve world-class performance, more and more companies are replacing their reactive, fire-fighting strategies for maintenance with proactive strategies like preventive and predictive maintenance and aggressive strategies like total productive maintenance (TPM). While these newer maintenance strategies require increased commitments to training, resources and integration, they also promise to improve performance. This paper reports

the results of a study of the relationship between maintenance strategies and performance. Based on the responses from a survey of plant managers and maintenance managers, the analysis shows strong positive relationships proactive maintenance between and aggressive strategies and performance". [9] A predictive-maintenance structure for a gradually deteriorating single-unit system. The decision model enables optimal inspection and replacement decision in order to balance the cost engaged by failure and unavailability, a mathematical model for the maintained system cost is developed using regenerative and semi-regenerative processes theory. Numerical experiments show that the s-expected maintenance cost rate on an infinite horizon can be minimized by a joint optimization of the replacement threshold and the periodic inspection times. [10]" The focus of preventive maintenance (PM) programs in industry is shifting from a pure statistical basis to online condition monitoring. Examine the short coming of statistical-based (PM) which are contributing to this shift and the potential benefits of and current research issues within condition-based (PM). Notes those statistics and quality control techniques will continue to play a critical role in this evolution". [11] Vibration monitoring of rolling element bearings by the highfrequency resonance technique is reviewed. It is shown that the procedures for obtaining the spectrum of the envelope signal are well established, but that there is an incomplete understanding of the factors which control the appearance of this spectrum. Until the envelope spectrum can be fully explained, use of the technique is limited" [12]

# 2.2.2 Vibration Monitoring:

Vibration analysis techniques is used to find out the root causes of the generator vibration. The collected data from the steam turbine and power generator indicates high vibration in the generator drive end bearing (DE). The analysis result indicates that parallel misalignment in the generator drive end bearing. Phase measurements was used as confirmatory tool to the vibration root causes, the result is confirm the spectrum analysis results parallel misalignment. Shaft coupling alignment measurements was carried out. The data verify that normal vibration spectrum amplitudes in the drive end bearing and no abnormal vibration found. [13] Condition monitoring of dynamic systems based on vibration signatures has generally relied upon Fourier-based analysis as a means of translating vibration signals in the time domain into the frequency domain. However, Fourier analysis provided a poor representation of signals well localized in time. In this case, it is difficult to detect and identify the signal pattern from the expansion coefficients because the information is diluted across the whole basis. The wavelet packet transform (WPT) is introduced as an alternative means of extracting time-frequency information from vibration signatures. The resulting WPT coefficients provide one with arbitrary time-frequency resolution of a signal. With the aid of statistical-based feature selection criteria, many of the feature components containing little discriminant information could be discarded, resulting in a feature subset having a reduced number of parameters without compromising the classification performance. The extracted reduced dimensional feature vector is then used as input to a neural network classifier. This significantly reduces the long training time that is often associated with the neural network classifier and

its generalization capability.[14]Various improves national and international standards based on German experience in the 1960s have been current until recently superseded, but arc still useful. They are based on practical experience and give a useful set of criteria that can be used to assess machines in service, as well as for specifying vibration quality for new or overhauled machines. The criteria apply for machines with speeds between 600 and 12 000 r/rain, and give the maximum vibration level in any of horizontal, vertical or axial directions on the bearing caps, measured in line with the Centre of the rotor. The unit is Velocity, mm/s rms. The assessment is broadband, with all vibration components between 10 to 1000 Hz included, each one being considered as of equal importance in severity. Below600 r/min, Displacement criteria are used. These criteria do not apply if a machine is affected by vibration transmitted from its surroundings to an extent of more than 1/3 of its own service vibration. This can be checked with the machine shut down. [15] Predictive maintenance growth led to productivity improvement of wide range machinery from petrochemical industry to food packing equipment. It has replaced breakdown maintenance as part of machinery reliability program.[16] High frequency resonance technique, used for many years in rolling element bearings for detection of rolling element bearings failure by vibration analysis, technique of synchronous averaging, has long been applied to vibration analysis of gears failure detection this two technique has been combined. By synchronizing the averaging of the envelope signal with a trigger signal that has been taken from the shaft rotation, a representation is obtained for a bearing under radial load of the influence of the load distribution and the effects of the transfer path to the vibration transducer. By synchronizing with the rotation speed of the shaft relative to the cage, an estimate is obtained of the distribution of the damage on the inner race of the bearing, and of the variations between rolling elements. The techniques are demonstrated by experiments on a laboratory test rig with a rolling bearing under radial load and multiple simulated spalls on the inner race. [17]

# **CHAPTER THREE**

# VIBRATION ANALISES AND DATA COLLECTION

# CHAPTER THREE VIBRATION ANALYSES AND DATA COLLECTION

# 3. Vibration analyses and data collection:3.1 Vibration analysis :

Vibration analysis is used to determine the equipment operating and mechanical condition. A major advantage is that vibration analysis can identify developing problems before enough time to protect equipment from unscheduled downtime. This can be achieved by conducting regular machine vibrations monitoring continuously or at scheduled intervals. Vibration can be analysis as:

# **3.1.1** Vibration analysis (detection mode):

Regular vibration monitoring can detect deteriorating or defective bearings, mechanical looseness and worn or broken gears. Vibration analysis can also detect misalignment and unbalance before these conditions result in bearing or shaft deterioration. Trending vibration levels can identify poor maintenance practices, such as improper bearing installation and replacement, inaccurate shaft alignment or imprecise rotor balancing. All rotating machines produce vibrations that are a function of the machine dynamics, such as the alignment and balance of the rotating parts. Measuring the amplitude of vibration at certain frequencies can provide valuable information about the accuracy of shaft alignment and balance, the condition of bearings or gears, and the effect on the machine due to resonance from the housings, piping and other structures. Vibration measurement is an effective, non-destructive method to test and monitor machine condition during start-ups, shutdowns and normal operation. Vibration analysis is used primarily on rotating equipment such as steam and gas turbines, pumps, motors, compressors, paper machines, rolling mills, machine tools and gearboxes. Recent advances in technology allow a limited analysis of reciprocating equipment such as large diesel engines and reciprocating compressors. These machines also need other techniques to fully monitor their operation. A vibration analysis system usually consists of four basic parts:

- 1. Signal pickup(s), also called a transducer
- 2. A signal analyzer
- 3. Analysis software
- 4. A computer for data analysis and storage.

These basic parts can be configured to form a continuous online system, a periodic analysis system using portable equipment, or a multiplexed system that samples a series of transducers at predetermined time intervals.

Hard-wired and multiplexed systems are more expensive per measurement position. The determination of which configuration would be more practical and suitable depends on the critical nature of the equipment, and also on the importance of continuous or semi continuous measurement data for that particular application. [2]

# **3.1.2** Vibration analysis (diagnosis mode) :

Operators and technicians often detect unusual noises or vibrations on the plant where they work on a daily basis. They could proceed with a vibration analysis, In order to determine if a serious problem actually exists. If a problem is indeed detected, additional spectral analyses can be

done to accurately define the problem and to estimate how long the machine can continue to run before a serious failure occurs. A vibration measurement has cost impact in (diagnosis mode) analysis particularly if budgets or manpower are limited. It's effective and strongly relies on someone detecting unusual noises or vibration levels. This approach may not be reliable for large or complex machines, or in noisy parts of a plant. Furthermore, by the time a problem is noticed, a considerable amount of deterioration or damage may have occurred. Another application for vibration analysis is as an acceptance test to verify that a machine repair was done properly. The analysis can verify whether proper maintenance was carried out on bearing or gear installation, or whether alignment or balancing was done to the required tolerances. Additional information can be obtained by monitoring machinery on a periodic basis, for example, once per month or once per quarter. Periodic analysis and trending of vibration levels can provide a more subtle indication of bearing or gear deterioration, allowing personnel to project the machine condition into the foreseeable future. The result is that equipment repairs can be planned to be done during normal machine shutdowns, rather than after a machine failure has caused unscheduled downtime. [2]

## **3.1.3** Vibration analysis – benefits :

Vibration analysis can identify improper maintenance or repair practices. These can include improper bearing installation and replacement, inaccurate shaft alignment or imprecise rotor balancing. As almost 80% of common rotating equipment problems are related to misalignment and unbalance, vibration analysis is an important tool that can be used to reduce or eliminate recurring machine problems. Trending vibration levels can also identify improper production practices, such as using equipment beyond their design specifications (higher speeds or loads). These trends can also be used to compare similar machines from different manufacturers in order to determine if design benefits or flaws are reflected in increased or decreased performance. Vibration analysis can be used as part of an overall program to significantly improve equipment reliability. This can include more precise alignment and balancing, better quality installations and repairs, and continuously lowering the average vibration levels of equipment in the plant. [2]

# **3.1.4** Using vibration to detect machinery fault:

A common machinery train is depicted in Figure 3.1. It consists of a driver or a prime mover, such as an electric motor. Other prime movers include diesel engines, gas engines, steam turbines and gas turbines. The driven equipment could be pumps, compressors, mixers, agitators, fans, blowers and others. At times when the driven equipment has to be driven at speeds other than the prime mover, a gearbox or a belt drive is used.

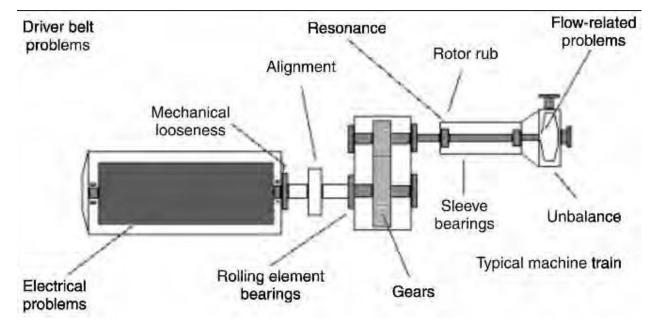


Figure 3.1Typical machine train

Each of these rotating parts is further comprised of simple components such as:

- Stator (volutes, diaphragms, diffusers, stators poles)
- Rotors (impellers, rotors, lobes, screws, vanes, fans)
- Seals
- Bearings
- Couplings
- Gears
- Belts.

When these components operate continuously at high speeds, wear and failure is easy to occur. When defects develop in these components, they give rise to higher vibration levels. [2]

# **3.1.5** Common problems :

A group of problems that is common in many machines includes imbalance, bent shaft, misalignment, soft foot, looseness, resonance, rubs, and problems that cause pulses. Each of these problems/conditions is discussed in the following sections.

#### **1- IMBALANCE**

Imbalance is a linear problem. If a rotor is out of balance, it should be out of balance by the same amount, through **360** degree of rotation. Each cycle in the time domain signal will have the same amplitude and the time signal will be sinusoidal. The first four or five harmonics of rotating speed at low levels (about 0.05 IPS) are normally present in fluid film bearings. [2]

#### 2- BENT SHAFT

A bent shaft is a form of imbalance and balancing can reduce the vibration level However, balancing cannot straighten the shaft. The bent shaft prevents adequate alignment in some cases and causes clearance problems in others, depending on where the bend occurs.

If the shaft is bent enough to cause a misalignment problem, one of the misalignment indications could be present. If a shaft is bent, the end of the shaft will rotate in an orbit. When this occurs, a force vector will be felt on the coupled shaft through 360 degrees of rotation. This force produces a high amplitude signal at the fundamental speed. Probe can be used to measure the total indicated run out (TIR) of the shaft while the machine is running.

#### **3-** SOFT FOOT

Soft foot can indicate a misalignment problem. The best way to prove soft foot is to observe the vibration level while tightening or loosening the bolts on each foot. The vibration amplitude will increase or decrease referring to applied pressure on the soft foot bolts. The frequency spectrum and time signal can indicate imbalance, bent shaft, looseness, and/or misalignment, depending on how the rotor and casing are distorted by the soft foot. When a soft foot is present, the machine may have to be removed, reset, and realigned.

#### 4- MISALIGNMENT

Misalignment can occur in several places in rotating machinery. For example, misalignment can occur along the shaft centerline between bearings. It also can occur in meshing gears. Misalignment occurs while two machines are coupled together then more frequently takes place. When misalignment occurs, the first three harmonics are generated. If coupling halves are misaligned, the first three harmonics of rotating speed are generated. If gears are misaligned, the first three harmonics of gear mesh frequency can be generated. For these reasons, solo data should be taken on all motors and turbines before they are coupled to the driven unit. To know the type of misalignment, data in the horizontal, vertical, and axial directions should be taken on the drive and off ends of the drive and driven units. If the first three harmonics are distinctive in:

1- The horizontal direction, there is vertical offset misalignment.

2- The vertical direction, there is horizontal offset misalignment

3- The axial direction, there is angular misalignment

4- All directions, there are horizontal offset, vertical offset, and angular misalignment

#### **5-** LOOSENESS

Looseness can and does take many forms. The various forms are created by the type of looseness (i.e. is the machine or rotating unit loose?), the amount of looseness, and other associated problems such as imbalance, misalignment, defective bearings, etc. The earlier a machine is diagnosed, the more precise and accurate is the diagnosis. , misalignment, bearing defects, and gear problems can be accurately diagnosed in early stages. In the later stages, only wide-banded noise may be present, and the diagnosis may be looseness. When, in fact, there are bent shaft, misalignment, and defective bearings present. This is the reason for why diagnostics of rotating machinery may not always identify all problems. Accurate diagnosis will always identify the most severe problem, which is often looseness.

#### Noise

Noise is the last stage of looseness. The noise can be wide-banded "white noise" or narrow-banded "pink noise." Noise contains all frequencies in a defined bandwidth. This machine is moving in an unpredictable manner and is generating all frequencies in the bandwidth.

#### 6- Resonance

All things have one or more resonant frequencies. These frequencies are often called modes, i.e. first mode, second mode, etc. also called natural frequencies and critical speeds. The term "critical speed" is used when a resonant/natural frequency equals the unit speed. The resonant phenomenon takes place when the forced or excitation frequency is equal to the natural frequency. If this occurs, the machine may have exceptionally high vibration levels. Natural frequencies can be determined in one of three:

1. The resonant frequency can be calculated mathematically through the stability deferential equation.

2. The resonant frequency can be determined with a "bump test."

Under this test, the machine or piece is bumped or hit, and the resonance is measured. For example, if you strike a crystal glass with a fork, it will ring or vibrate at its resonant frequency. The energy put into the glass by the fork excites the glass into vibrating at the resonant frequency.

3. Coast down and startup data .If a machine is operating above the first critical speed, coast down and startup data can be used to identify the resonant frequency, bandwidth, and amplification factors. Coast down data can be taken only from machines that will coast. Startup data can only be used to identify resonance if the machine can be started slowly.

As can be seen from the above list, all problems cannot be identified from bearing cap vibration while the machine is running. Special tests must be performed in some cases.

#### • Bearing defect

Bearing defect is divided into two types of defects:

#### i. **Bearing s Loose on the Shaft**

When a bearing is loose on the shaft, the type of signal generated depends upon how the unit is installed. If the inner race is turning on the shaft in an electric motor that is belt driving another unit, the frequency spectra may appear as imbalance with a high amplitude spectral line at motor speed. However, the time signal may be distorted in some manner. The time signal may vary in amplitude, be truncated, contain harmonics, contain a beat, or the time period may be different at each half cycle. Some of these characteristics may produce harmonics and other frequencies, such as the speed of the driven unit. On direct coupled units, if the bearing is turning on the shaft, the frequency spectra may contain a spectral line at unit speed and another spectral line a little bit lower than unit speed. The lower spectral line is the speed at which the inner race is turning. The time signal will contain a beat caused by the two frequencies going in and out of phase with each other.

#### ii. Bearings Loose in the Housing

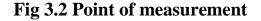
When the fourth harmonic of rotor speed is distinctive, the bearing may be loose in the housing. The fourth harmonic is distinctive, and the second and third harmonics are present. These harmonics should not be visible with prescribed calibration levels. Caution should be used in

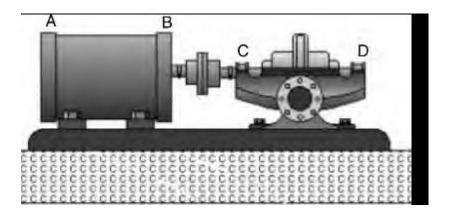
diagnosing bearings that are loose in the housing when a pump has four vanes on the impeller because the fourth harmonic may be caused by vane pass frequency. Some press rolls may also contain the fourth harmonic of speed if it has four high places. The only way to determine which problem exists is to look at the phase relationship between the fourth harmonic and the fundamental. If the fourth harmonic is out of phase or is changing phase, the bearing may be loose in the housing. If the fourth harmonic is in phase and maintains a constant phase relationship, the bearing may not be loose in the housing. Another indication that the bearing is loose on the shaft or loose in the housing is careful analysis of bearing frequencies, particularly ball pass frequencies of the outer and inner races. When calculating bearing frequencies, the rotating unit speed is used. The normal assumption is that the bearing is rotating at the same speed as the shaft, and the fixed race is not rotating. These assumptions are wrong in many cases. The bearing frequencies are actually determined by the relative speed between the inner and outer races. If either race is turning on the shaft or is loose in the housing, the bearing frequencies will be less than those calculated. Once again, care must be used because if the contact angle is increased, the ball pass frequency of the inner race will decrease. Careful analysis of bearing frequencies can identify bearings that are loose on the shaft or in the housing. The more common forms of looseness could start with an increase in the amplitude of the fundamental and the harmonic content. Many unskilled analysts try to solve these problems by balancing. Such efforts may not be successful. As the looseness increases, the harmonic content also increases, and the amplitude of the fundamental and the overall RMS value can actually decrease. The next stage of looseness could be the appearance of spectral lines at fractional shaft speed. These spectral lines could occur at 1/2, 1/3, 1/4, shaft speed, etc., depending upon the type and amount of looseness.

# **3.2** Point of measurement :

There are four points for measuring the vibration signals, they are the location when the shaft loading in the bearing for that reason they have the clearest vibration signal to be picked up as follow. [2]

- Motor non-drive end bearing A
- Motor drive end bearing B
- Pump outboard bearing (next to the coupling) C
- Pump inboard bearing (away from coupling) D





For monitoring and evaluation of severity, machine vibration measurements with contact transducers are taken at the bearings, in three directions" Horizontal and Vertical usually in line with the shaft centerline, and *Axial*". For vertical shaft pumps, the two horizontal directions should be

decided and clearly shown on a sketch: one in line with the suction pipe is usual, and others at flanges up to the top of the motor.

Routine monitoring of simple pumps may suffice with only some of these readings. For example, one radial reading at each bearing, usually horizontally, and one axial reading per shaft, requires much less time in collecting and processing than three readings per bearing. If used, tri axial accelerometers are usually mounted on top dead Centre, and therefore will not measure horizontal nor axial vibration at the Centerline.

On machines with hydrodynamic bearings, non-contact proximity displacement probes are sometimes installed permanently to sense shaft motion in X-Y directions at the bearings. For ease of access, they are usually placed at top dead Centre + 45 ~ rather than at vertical and horizontal axes. These probes are however rarely fitted to pumps.

Although temporary fitment using brackets or magnetic bases is possible, limited access to shafts minimize their use for investigations on pumps. From a diagnostic viewpoint, pump beating vibration measurements are considered to be more valuable than shaft measurements.

The outputs can be displayed on an oscilloscope or on a computer system via suitable software as displacement-time traces, or in XY mode, to give a shaft orbit, which is a magnified picture of the centerline motion. The orbit will usually be elliptical, and changes shape if vibration of other frequencies is present. A flat orbit shows that a rotor is restrained unevenly. Orbits from shafts each side of a coupling can confirm alignment condition. If vibration at higher frequencies is present, it shows as ripples on the orbit. Processing instruments can filter out frequencies other than those at l x.

# **3.3** The scope of vibration analysis :

All machines, including pumps, vibrate to some extent, but these questions often arise:

- 1- How much vibration is excessive?
- 2- If the vibration is excessive, what is its cause and how can it be solved?
- 3- What else can vibration even if not excessive? Tell about machine condition.

Vibration measurement and analysis answers these questions. It is a main method of condition monitoring applicable to rotating machines in general. Most machinery vibration is forced periodic motion excited by forces within the machine or from outside it. The amount of vibration depends on:

- 1- The exciting force,
- 2- How close the frequency of this exciting force is to structural resonances or their multiples (harmonics),
- 3- The restraints the pump structure imposes to vibration.

Noise is audible vibration transmitted through the air, and is therefore often related to vibration of machines or structures. Due to background influences, noise is less repeatable measured than vibration but new techniques may extend its use. The vibration of a pump is usually lowest when operating at best efficiency point, and can double in amplitude as flow is reduced to 25% of BEP. This is an important consideration when taking routine measurements as a range of vibration levels may occur although the pump internal condition in unchanged. If operation at BEP is not always possible, then a standard flow

may need to be chosen for routine measurements, unless a series of "normal" vibration levels at a range of flows is obtained and used as the datum. [2]

#### **3.3.1** Data collection :

Data should be collected in the load zone with due respect to flexibility. For example, imbalance is normally a radial-loaded problem. Imbalance can cause an axial load in fans. Looseness can be either radial or axial-loaded, depending on where the machine is loose and how the machine is installed. Horizontal and vertical offset misalignment cause a radial load. Angular offset misalignment causes an axial load. Monitoring of small simple machines can often be done by collecting data in the horizontal and axial directions on the drive and driven units across the coupling.

Depending on time, workload, etc., one monitoring point at 45' between horizontal and axial may suffice for monitoring. However, data must be taken in the horizontal, vertical, and axial directions when trouble-shooting and establishing baseline data for new installations, and when new equipment is added to the program. Because most machines failures occur on the drive and driven end. Data must be taken on both sides of the coupling. Data should also be taken on the "off" end of the drive and driven unit in some cases like (rotor is between two bearings). The location of where the data is taken also depends on what you want to measure. If a motor has the same bearing on both ends and one bearing is defective, often the location of the defective bearing can be identified by taking data from both ends. If the problem is a radial-loaded problem, data must be taken in the radial direction. Imbalance is a good example. If the problem is an axial-loaded problem, then data must be taken in the axial direction.

Antifriction bearings and angular misalignment are good examples of axial-loaded problems. Data should be taken in the horizontal, vertical, and axial directions on the drive and off end of each machine in the complete machine train to do perfect trouble-shooting. For routine monitoring, it may be adequate to take data in the horizontal direction on the drive and driven end of each machine. For large motors, fans, and other equipment, it may be desirable to monitor both ends.

#### **3.3.2 Data acquisition :**

The first step in the basics of vibration with data acquisition is the domain of practical vibration analysis. It includes the following main tasks in collection of machinery vibration:

- Conversion of the vibration signal to an electrical signal
- Transformation of the electrical signal to its components
- Providing information and documentation related to vibration data.

The above entails the entire hardware of the vibration analysis system or program. It includes transducers, electronic instruments that store and analyze data, the software that assists in vibration analysis, record keeping and documentation.

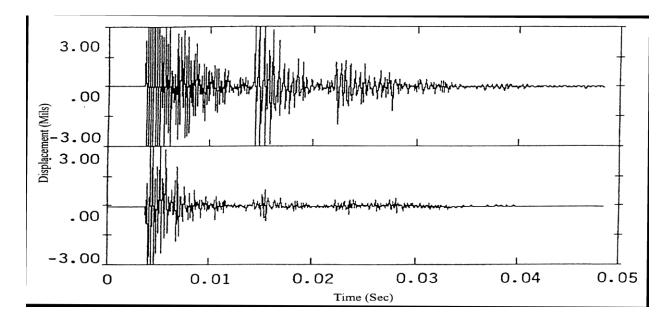
### **3.4 Actual vibration profiles :**

The process of vibration analysis requires the gathering of complex machine data, which must then be deciphered. As opposed to the simple theoretical vibration curves shown in Figures 3.2 and 3.4, the profile for a piece of equipment is extremely complex. This is true because there are usually many sources of vibration. Each source generates its own curve, but these are essentially added and displayed as a composite profile. These profiles can be displayed in two formats: time domain and frequency domain. [1]

### 3.4.1 Time Domain :

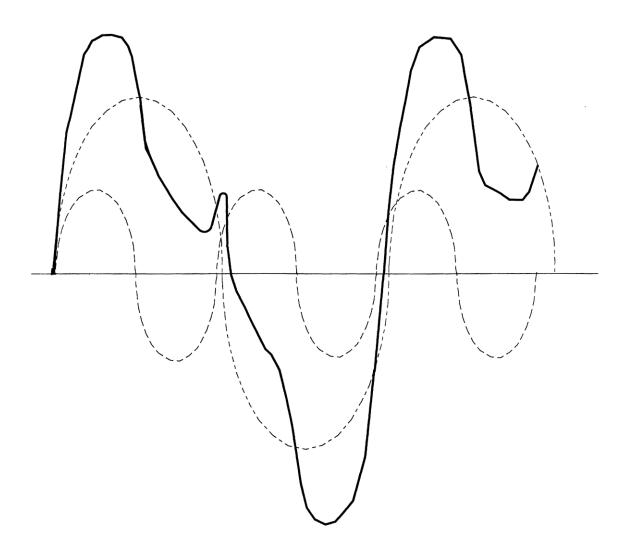
Vibration data plotted as amplitude versus time is referred to as a timedomain data profile. Some simple examples are shown in Figures 3.3&figure 3.4.Time-domain plots must be used for all linear and reciprocating motion machinery. They are useful in the overall analysis of machine-trains to study changes in operating conditions. However, time-domain data are difficult to use. Because all of the vibration data in this type of plot are added to represent the total displacement at any given time, it is difficult to determine the contribution of any particular vibration source. The French physicist and mathematician Jean Fourier determined that non- harmonic data functions such as the time-domain vibration profile are the mathematical sum of simple harmonic functions. The dashed-line curves in Figure (3.4) represent discrete harmonic components of the total, or summed, non-harmonic curve represented by the solid line. [1]

Figure 3.3 Example of a typical time-domain vibration profile for a piece of machinery.



These types of data, which are routinely taken during the life of a machine, are directly comparable to historical data taken at exactly the same running speed and load. However, this is not practical because of variations in day-to-day plant operations and changes in running speed. This significantly affects the profile and makes it impossible to compare historical data [1]

# Figure 3.4 Discrete (harmonic) and total (non-harmonic) timedomain vibration curves



#### **3.4.2** Frequency Domain :

From a practical standpoint, simple harmonic vibration functions are related to the circular frequencies of the rotating or moving components. Therefore, these frequencies are some multiple of the basic running speed of the machine-train, which is expressed in revolutions per minute (rpm) or cycles per minute (cpm).

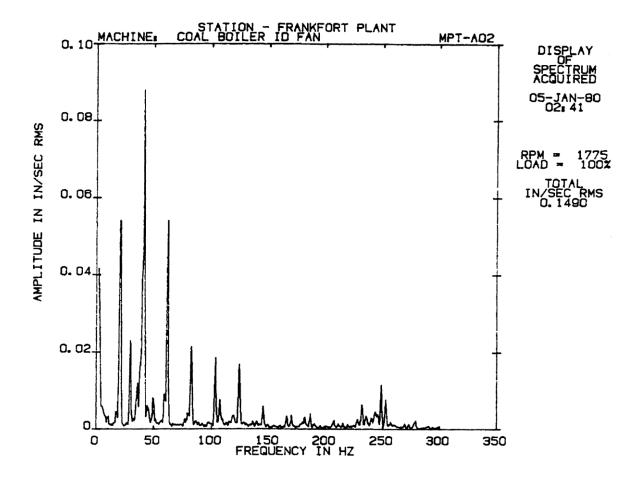


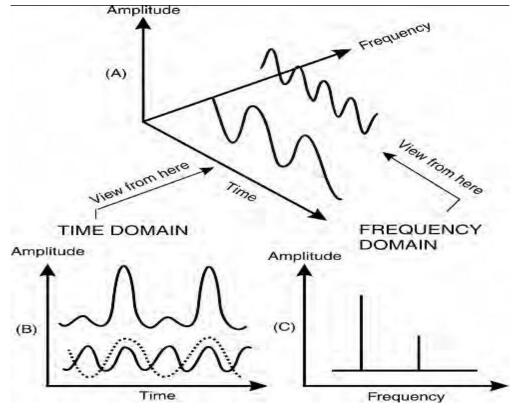
Figure 3.5 Typical frequency-domain vibration signatures.

Determining these frequencies is the first basic step in analyzing the operating condition of the machine-train. Frequency-domain data are obtained by converting time-domain data using a mathematical technique referred to as a fast Fourier transform (FFT) [1]

#### **3.4.3** Fourier transform :

A vibration or a system response can be represented by displacement, velocity and acceleration amplitudes in both time and frequency domains (Figure 3.6). Time domain consists of amplitude that varies with time. This is commonly referred to as filter-out or overall reading. Frequency domain is the domain where amplitudes are shown as series of sine and cosine waves. These waves have a magnitude and a phase, which vary with frequency.

The measured vibrations are always in analog form (time domain), and need to be transformed to the frequency domain. This is the purpose of the fast Fourier transform (FFT). The FFT is thus a calculation on a sampled signal. If FFT is a calculation on a sampled signal, the first question that arises is: how do we determine the sampling rate? [2]



**Figure 3.6 Fourier transform** 

# **3.5** Assessing the vibration severity :

Whether the vibration of a machine is acceptable or not depends on the probability of damage to the machine itself, as well as its effects on the surroundings. Measurements should be taken when a machine is at its normal steady state operating conditions. [15]

#### Vibration severity is usually expressed in:

- 1. Displacement, for low speed machines at frequencies up to 10 Hz
- 2. Velocity, for frequencies between 10-1000 Hz- most machines fit here
- 3. Acceleration, for frequencies above 1000 Hz

#### **3.5.1** The assessment points:

#### **1-** General experience with machines

Several vibration severity guidelines are available, based on general experience with a range of machines. Engineering judgment is needed if there is any conflict between them. An apt comment in Eisenmann (1997) sums up the situation: "There are no universal vibration severity limits". The advice therefore is to choose the guidelines you feel carry the highest credibility and are the most appropriate to your situation. Information is usually given with purchase of a vibration measurement instrument.

#### 2- The effect on people and buildings

Steel flame buildings usually have natural frequencies below 10 Hz-40Hz, and may suffer cracks or other failure symptoms from excessive forced vibration levels. People experience discomfort at vibration frequencies below about 40 Hz.

#### **3-** The manufacturer's advice

The advice of manufacturers is likely to be conservative. Available Standards are often used (see below), as manufacturer's engineers are often involved in their formation.

#### 4- Comparison with identical machines

Vibration measured identically on other 'identical' machines can be useful for comparison. However, such machines can have quite different vibration characteristics, yet all is in acceptable condition. [15]

#### **3.5.2** Limits and standards of vibration :

As mentioned above, vibration amplitude (displacement, velocity or acceleration) is a measure of the severity of the defect in a machine. A common dilemma for vibration analysts is to determine whether the vibrations are acceptable to allow further operation of the machine in a safe manner.

To solve this dilemma, it is important to keep in mind that the objective should be to implement regular vibration checks to detect defects at an early stage. The goal is not to determine how much vibration a machine will withstand before failure! The aim should be to obtain a trend in vibration characteristics that can warn of impending trouble, so it can be reacted upon before failure occurs.

Absolute vibration tolerances or limits for any given machine are not possible. That is, it is impossible to fix a vibration limit that will result in immediate machine failure when exceeded. The developments of mechanical failures are far too complex to establish such limits. However, it would be also impossible to effectively utilize vibrations as an indicator of machinery condition unless some guidelines are available, and the experiences of those familiar with machinery vibrations have provided us with some realistic guidelines. We have mentioned earlier that velocity is the most common parameter for vibration analysis, as most machines and their defects generate vibrations in the frequencies range of 10 Hz (600 rpm) to 1 kHz (60krpm).

#### 1- ISO 2372

The most widely used standard as an indicator of vibration severity is ISO 2372 (BS 4675).The standard can be used to determine acceptable vibration levels for various classes of machinery. Thus, to use this ISO standard, it is necessary to first classify the machine of interest. Reading across the chart we can correlate the severity of the machine condition with vibration. The standard uses the parameter of velocity-rms to indicate severity. The letters A, B, C and D as seen in (table3.1) classify the severity.

Class I Individual parts of engines and machines integrally connected with a complete machine in its normal operating condition (production electrical motors of up to 15 kW are typical examples of machines in this category).

Class II Medium-sized machines (typically electrical motors with 15–75 kW output) without special foundations, rigidly mounted engines or machines (up to 300 kW) on special foundations.

Class III Large prime movers and other large machines with rotating masses mounted on rigid and heavy foundations, which are relatively stiff in the direction of vibration.

Class IV Large prime movers and other large machines with rotating masses mounted on foundations, which are relatively soft in the direction of vibration measurement (for example – turbo generator sets, especially those with lightweight substructures).

#### Table 3.1 ISO 2372 – ISO guideline for machinery vibration

Ranges of Vibration severity		Examples of quality judgment for separate classes of machines					
Velocity – in/s – Peak	Class 1	Class II	Class III	Class IV			
0.015	0.28				1		
0.025	0.45	1			11		
0.039	0.71	1			11		
0.062	1.12		1		1		
0.099	1.8		1	and the second s			
0.154	2.8						
0.248	4.5	11					
0.392	7.1			1			
0.617	11.2						
0.993	18			1			
1.54	28				-		
2.48	45		1		1		
3.94	71_	I		Î.	Concession of the local division of the loca		

severity

A – Good	
B - Acceptable	
C - Still acceptable	
D - Not acceptable	

The shaded areas in Table 3.21 correspond to these grading's, from the smoothest level downwards:

**A: Good to excellent**- the smoothest expected from good manufacturing practice. New machines would normally fall here.

**B:** Satisfactory- readily achieved by well-designed and well-made machines, and acceptable for long-term operation.

**C:** Not Satisfactory - higher than normally expected for well-made machines, possibly pointing to a fault. Unsuitable for continuous long-term operation, this level may be acceptable if the vibration is not due to machine deterioration.

**D: Unacceptable** - there is unacceptably high probability of a severe fault or deterioration of the machine.

#### 2- American Petroleum Institute (API specification)

The American Petroleum Institute (API) has set forth a number of specifications dealing with turbo machines used in the petroleum industry. Some of the specifications that have been prepared include API-610, API-611, API-612, API-613, API-616 and API-617. These specifications mainly deal with the many aspects of machinery design, installation, performance and support systems. However, there are also specifications for rotor balance quality, rotor dynamics and vibration tolerances. API standards have developed limits for casing as well as shaft vibrations. The API specification on vibration limits for turbo machines is widely accepted and followed with apparently good results.

The API standard specifies that the maximum allowable vibration displacement of a shaft measured in ml (m-inches = 0.001 inch = 0.0254 mm) peak-peak shall not be greater than 2.0 mils or ( $12\ 000/N$ ) 1/2, where N is speed of the machine, whichever is less.

#### **3-** Speed-adjusted Calibration Standards

Another valuable utility of Table 5-1 is that problems of the same severity appear at about the same percentage of full scale for all speeds. For example, an imbalance problem on a 1200 RPM fan at 0.35 inches per second (IPS), or half scale of zero to 0.7 IPS, is just as bad as 0.05 IPS, or half scale on a 150 RPM fan. Of course, the latter fan would not require balancing. However, if a similar situation occurred with looseness or bearing defects, a fix would be required. Therefore, automatic ranging should be used only in rare cases.

SPEED IN RPM	SPEED IN FPM	ZERO-TO-PEAK -IPS
1,000 and above	2,400 and above	0 to 0.70 IPS
500 - 1,000	1,700 - 2,400	0 to 0.35 IPS
200 - 500	1,000 - 1,700	0 to 0.18 IPS
Below 200	Below 1,000	0 to 0.10 IPS

Table 3.2 Speed-adjusted Calibration Standards for rotating machine

Some frequencies at acceptable amplitudes are found in "good" machines. Some examples are imbalance, vane pass frequency, gear mesh frequency, etc. Other frequencies are not acceptable at any level within prescribed calibration standards. Some examples are the various frequencies generated by antifriction bearings, pulses, bars, or corrugations on rolls, etc. In addition, spectra taken from "good" machines should not be modulated by shaft speed or any multiple of shaft speed. However, the data from most machines contain some sidebands. The reason for this paradox is that it may not be economical to fix or repair all identified problems. Most of the problems in determining what is good or bad can be eliminated by strict adherence to calibration standards and adjusting sensitivity to speed. **[2]** 

# 3.6. Rotating unbalance and bearing deterioration stages:3.6.1. Rotating unbalance:

The assumption is the defective bearing moving elements are moving with unpredictable motion magnitude and direction . If we consider that a rotor system (Figure 3.7) has a mass M supported between two bearings. The rotor mass M is assumed as concentrated between the supported bearings; it contains an unbalance mass (*Mu*) located at a fixed radius *r* and is rotating at an angular velocity  $\omega$  Where  $\omega = 2.\pi$ .rpm/60

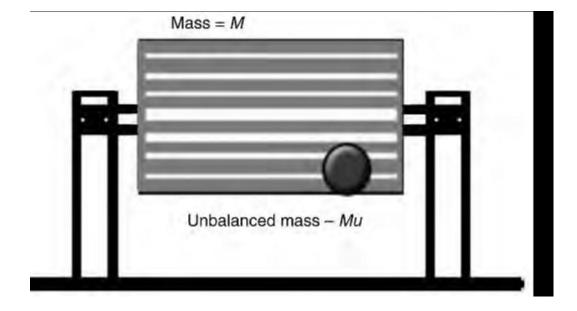


Figure 3.7 a rotor system response

The vibration force produced by the unbalance mass Mu is represented by:

F (unbalance) =  $Mu \times r \times \omega^2 \times \sin(\omega t)$  where t = time in seconds.

The restraining force generated by the three system characteristics is:

 $M \times (a) + C \times (v) + k \times (d)$  where a = acceleration; v = velocity; d = displacement. If the system is in equilibrium, the two forces are equal and the equation can be written as:

 $Mu \times r^* \omega^{2*} \times \sin(\omega t) = M \times (a) + C \times (v) + k \times (d)$ 

However, in reality the restraining forces do not work in tandem. With changing conditions, one factor may increase while the other may decrease. The net result can display a variation in the sum of these forces. This in turn varies the system's response (vibration levels) to exciting forces the unbalance that generate vibrations is illustrating the defects. Thus, the vibration caused by the unbalance will be higher if the net sum of factors on the right-hand side of the equation is less than unbalance force. In a similar way, it is possible that one may not experience any vibrations at all if the net sum of the right-hand side factors becomes much larger than the unbalance force. [2]

#### 3.6.2 Rolling element bearings defect stages :

A rolling element bearing consist of inner and outer races, a cage and rolling elements. Defects can occur in any of the parts of the bearing and will cause high-frequency vibrations. In fact, the severity of the wear keeps changing the vibration pattern. In most cases, it is possible to identify the component of the bearing that is defective due to the specific vibration frequencies that are excited. Raceways and rolling element defects are easily detected. However, the same cannot be said for the defects that rise up in bearing cages. Though there are many techniques available to detect where defects are occurring, there are no established techniques to predict when the bearing defect will turn into a functional failure.

In an earlier topic dealing with enveloping/demodulation, we saw how bearing defects generate both the bearing defect frequency and the ringing random vibrations that are the resonant frequencies of the bearing components.

Bearing defect frequencies are not integrally the harmonic to running speed. However, the following formulas are used to determine bearing defect frequencies. There is also a bearing database available in the form of commercial software that readily provides the values upon entering the requisite bearing number.

$$BPFI = \frac{Nb}{2} (1 + \frac{Bd}{Pd} \cos \theta) \times rpm$$

BPFO = 
$$\frac{Nb}{2}(1 - \frac{Bd}{Pd}\cos\theta) \times rpm$$
  
FTF=  $\frac{1}{2}(1 - \frac{Bd}{Pd}\cos\theta) \times rpm$   
BSF=  $\frac{Pd}{2Bd}[1 - (\frac{Bd}{Pd})^2(\cos\theta)^2] \times rpm$ 

It is very interesting to note that in an FFT, we find both the inner and outer race defect frequencies. Add these frequencies and then divide the result by the machine rpm [(BPFI + BPFO)/rpm]. The answer should yield the number of rolling elements. The bearing deterioration progresses through four stages. During the initial stage, it is just a high-frequency vibration, after which bearing resonance frequencies are observed. During the third stage, discrete frequencies can be seen, and in the final stage highfrequency random noise is observed, which keeps broadening and rising in average amplitude with increased fault severity. [2]

#### **1.** Stage 1 of bearing defect

The FFT spectrum for bearing defects can be split into four zones (A, B, C and D), where we will note the changes as bearing wear progresses. These zones are described as:

Zone A: machine rpm and harmonics zone

Zone B: bearing defect frequencies zone (5–30 k rpm)

Zone C: bearing component natural frequencies zone (30–120 k rpm)

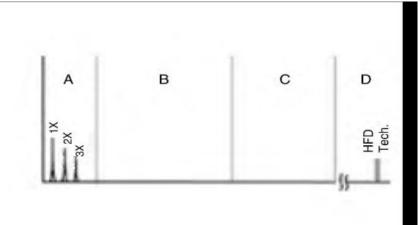
Zone D: high-frequency-detection (HFD) zone (beyond 120 k rpm).

The first indications of bearing wear show up in the ultrasonic frequency ranges from approximately 20–60 kHz (120–360 k rpm). These are frequencies that are evaluated by high-frequency detection techniques such as SE (Spike Energy), SEE, Peak Value, SPM and others.

As (Figure 3.8) shows, the raceways or rolling elements of the bearing do not have any visible defects during the first stage. The raceways may no longer have the shine of a new bearing and may appear dull gray.

Figure 3.8 Small defects in the raceways of a bearing





#### 2. Stage 2 of bearing defect

In the following stage (Figure 3.9), the fatigued raceways begin to develop minute pits. Rolling elements passing over these pits start to generate the ringing or the bearing component natural frequencies that predominantly occur in the 30-120 k rpm range. Depending on the severity, it is possible that the sideband frequencies (bearing defect frequency  $\pm$  rpm) appear above and below the natural frequency peak at the end of stage two. The high-frequency detection (HFD) techniques may double in amplitude compared to the readings during stage one.

#### **3.** Stage 3 of bearing defect

As we enter the third stage (Figure 3.10), the discrete bearing frequencies and harmonics are visible in the FFT. These may appear with a number of sidebands. Wear is usually now visible on the bearing and may expand through to the edge of the bearing raceway. The minute pits of the

earlier stage are now developing into bigger pits and their numbers also increase. When well-formed sidebands accompany any bearing defect frequency or its harmonics, the HFD components have again almost doubled compared to stage three. It is usually advised to replace the bearing at this stage. Some studies indicate that after the third stage, the remaining bearing life can be 1 h to 1% of its average life.

Figure 3.9 more obvious wear in the form of pits

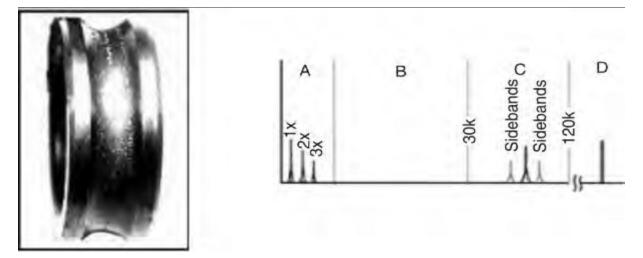
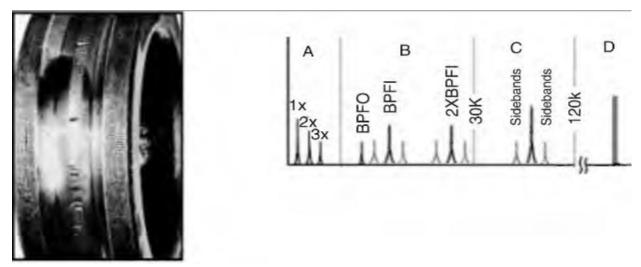


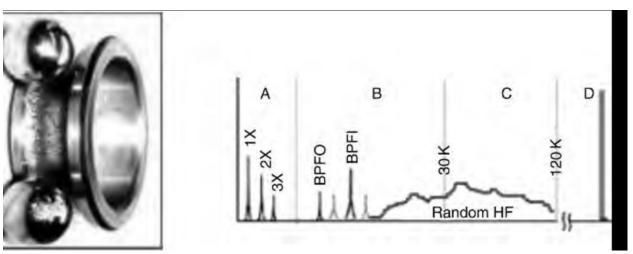
Figure 3.10 wear clearly visible over the breadth of the bearing



#### 4. Stage 4 of bearing defect

In the final phase (Figure 3.11), the pits merge with each other, creating rough tracks and spalling of the bearing raceways or/and rolling elements. The bearing is in a severely damaged condition now. Even the amplitude of the  $1 \times rpm$  component will rise. As it grows, it may also cause growth of many running speed harmonics. It can be visualized as higher clearances in the bearings allowing a higher displacement of the rotor. Discrete bearing defect frequencies and bearing component natural frequencies actually begin to merge into a random, broadband high-frequency 'noise floor'. Initially, the average amplitude of the broad noise may be large. However, it will drop and the width of the noise will increase. In the final stage, the amplitude will rise again and the span of the noise floor also increases.

Figure 3.11 severely damaged bearing in final stage of wear



However, amplitudes of the high-frequency noise floor and some of the HFD may in fact decrease (due to pits flattening to become spalls), but just prior to failure spike energy will usually grow to extreme amplitudes. By this time, the bearing will be vibrating excessively; it will be hot and making lots of noise. If it is allowed to run further, the cage will break and the rolling elements will go loose. The elements may then run into each other, twisting, turning and welded to one another, until the machine will hopefully trip on overload. In all probability, there will be serious damage to the shaft area under the bearing.

# CHAPTER FOUR MATERIAL AND METHOD

# CHAPTER FOUR MATERIAL AND METHOD

# 4. Material and method:

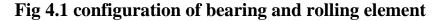
This chapter has studied the bearing deterioration steps assuming that the bearing with good situation has smooth rolling element& raceway and its cage controlling the rolling element's random movement and provide it with the optimum damping coefficient. Any bearing defect such as inner or outer race roughness or thickness loss (loose) and cage loose has vibration impact corresponding to the deterioration stage that take place in four steps that mention in detail theoretically in the previous chapter, this will be analyzed mathematically to establish a mathematical model corresponding to physical model that can be used to predict the bearing deterioration step and practical analysis to confirm the mathematical analysis.

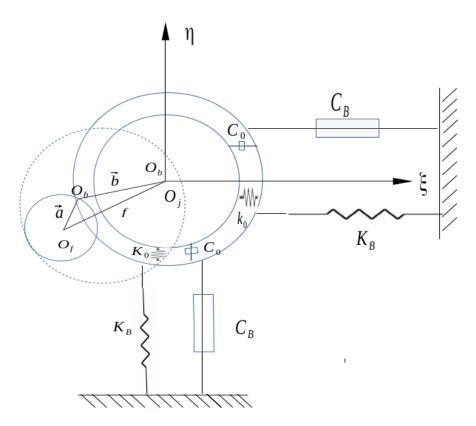
# 4.1 The mathematical analysis:

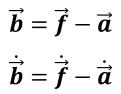
With the same assumption in the practical analysis the defective bearing moving elements are moving with unpredictable motion magnitude and direction that proportional to the moving element inertia &speed. Mathematical analysis is to study the effecting forces in the bearing and their impact in vibration reading. This can be done by substituting the bearing physical parameter into bearing general mathematical equation at specific direction.

#### 4.1.1 The bearing general mathematical model:

The following figure (4.1) shows the rolling elements in the bearing that rotate around the inner race and the relation between them to support the rotational loads in the three dimensions.







Where  $O_b$  bearing center  $O_f$  rolling element center. There are two types of coordination required when we analyzing the rotor - bearing system. The first one is an absolute coordinate reference frame that fixed the Ground. And the other is the rotating coordinates from which rotates with speed. The relations between both coordinate systems are given by the following relationship. **[18]** 

٢C	os wt	– sin ωt	0	ך0	г <i>х</i> т		ΓζΊ	l
s	in wt	cos ωt	0	0	y y		$ \eta $	
	0	Ι	0	0			ξ	(11)
	0	cos wt	– sin ωt	0	$\theta^x$	=	θ	(4.1)
	0	cos ωt sin ωt	cos ωt	0	$\theta^{y}$		$ \theta $	
L	0	0	0	IJ	$L_{\theta^z}$		L <b>ə</b> l	

# 4.1.2 The case study mathematical model:

In this chapter we are studying the maximum amplitude that comes from the bearing's rolling elements and bearing's inner race for the studied pumps bearing and compare it with the experiment horizontal direction readings. [19]

#### **4.1.2.1** The studying pump parameters.

The pump specification is as the following:-

Q= 155  $m^3$ /hr.

H= 110 m.

NPSH=4.1m

Power= 75 kW.

N= 2950 rpm.

Efficiency ( $\eta$ ) =0.69.

Weight including driving motor =600 kg.

Medium is light diesel.

#### 4.1.2.2 The studying bearing parameters.

Bearing mass	2.65 kg
Bearing diameter	83 mm

#### 4.1.2.3 The rolling ball parameters.

D=25mm.

Mass=73g.

#### 4.1.2.4 A Uniform Solid Sphere Moment of Inertia.

Denote the radius of the sphere by *R*, the constant mass density is denoted by  $\rho$ . The use of the term "Disc Method" is a reference to a method of calculation of volumes. The strategy will be to divide the sphere into infinitesimally small discs with axes in the z direction, thickness dz and radius r. accordingly, the center of each disc will be located at the coordinate from the origin and each disc will have a radius  $r = \sqrt{R^2 - Z^2}$  and an infinitesimal contribution *dI* to the moment of inertia.  $dI = \frac{1}{2}(dm)r^2 = \frac{1}{2}(\rho\pi r^2 dz)r^2$ 

$$=\frac{1}{2}\rho\pi r^{4}dz = \frac{1}{2}\rho\pi (R^{2} - z^{2})^{2}dz$$
(I)

The needed integral suggested by the last expression in

(4.2.4.1) is hardly formidable expanding the square of the term in

parentheses dI=
$$\frac{1}{2}\rho\pi(R^4 - 2R^2Z^2 + z^4)$$
dz (II)

The limits on the -integral are from z = R to R, and so

$$I = \int dI = \frac{1}{2} \rho \pi \int_{-R}^{R} (R^4 - 2R^2 Z^2 + z^4) dz$$
  
$$I = \rho \pi R^5 \left[ 1 - \frac{2}{3} + \frac{1}{5} \right] = \frac{8}{15} \rho \pi R^5$$
(III)

(Some minor algebraic steps have been skipped in the above calculation.) The mass of the sphere is the product of volume and density,

$$M = \frac{4}{3}\rho\pi R^3$$
 (IV)

And combining the results of (III) and Equation (IV), we see that  $I = \frac{2}{5} M R^2 \qquad (4.2)$ 

# 4.2 Bearing general equilibrium equation:

Assume the oil film and rolling grooves have lost their function after the last stage of failure that means they lost their damping capability of the rolling ball. [18]

In this case the equilibrium equation is:-

$$\begin{bmatrix} \cos \omega t & -\sin \omega t & 0 & 0\\ \sin \omega t & \cos \omega t & 0 & 0\\ 0 & I & 0 & 0\\ 0 & \cos \omega t & -\sin \omega t & 0\\ 0 & \sin \omega t & \cos \omega t & 0\\ 0 & 0 & 0 & I \end{bmatrix} \begin{bmatrix} \mathbf{x} \\ \mathbf{y} \\ \mathbf{z} \\ \mathbf{\theta}^{\mathbf{x}} \\ \mathbf{\theta}^{\mathbf{y}} \\ \mathbf{\theta}^{\mathbf{z}} \end{bmatrix} = \begin{bmatrix} \boldsymbol{\zeta} \\ \boldsymbol{\eta} \\ \boldsymbol{\xi} \\ \boldsymbol{\theta} \\ \boldsymbol{\theta} \\ \mathbf{\theta} \\ \mathbf{\theta} \end{bmatrix}.$$
(4.3)

The ball inertia is  $I=2/5mr^2$ .

Substituting the rolling ball parameters

 $I=2/5*73*12.5*12.5=4562.5 \text{ gmm}^2$ 

The shaft frequency is.

N= 2950 RPM or CPM

 $\omega = \frac{2\pi N}{60} = \frac{2 \times \pi \times 2950}{60} = 308.8$  rad/sec.

The substituted matrix is:

r <i>cos</i> 309t	– sin 309t	0	ך 0	[x]		۲7]	
sin 309t	cos 309t	0	0	<b>y</b>		$\eta$	
0	4562	0	0	Z	_	ξ	
0	cos 309t	– sin 309t	0	$ \boldsymbol{\theta}^{x} $	_	$ \theta $	
0	sin 309t	cos 309t	0	$ \theta^{y} $		$ \boldsymbol{\theta} $	
L 0	0	0	4562 <sup>]</sup>	<b>⊢</b> <i>θ</i> <sup>z</sup>		<b>l</b> θ]	

The assumption is the main forces that has impact in the rolling element vibration in the bearing are the raceway wear because rough surface is force the rolling elements to create motion perpendicular to raceway direction to start the shaft vibration when smooth surface lead to smooth rotating shaft and cage looseness value means the miss control in rolling elements motion that means it's also vibration source according to the assumption bearing has two main parameters affected on its status and vibration reading free rolling element vibration and Oil film damping effect.

#### **4.2.1** The free rolling element vibration calculations:

Rolling element moment of inertia with the rotational in horizontal axis (X-axis) has substituted in bearing equilibrium general equation (4.2.4.6) by neglecting the vertical &axial axis's to plot the results in (**rms**) VS the motor frequency (**CPM**) using Matlab program to find the graph in fig 4.2 that shows the free rolling element vibration spectrum.

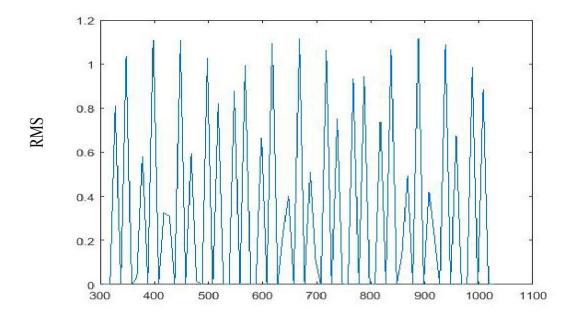


Figure 4.2 None damped rolling element vibration spectrum.

#### 4.2.2 Oil film(cage) damping effect:

The cage function is to controlling rolling elements movement between inner and outer raceways, for that reason the proper oil film is rolling element vibration damper. Rolling element moment of inertia with the rotational in horizontal axis (X-axis) has substituted the equation (4.2.4.7) that illustrates the damping coefficient proportional to external load, shaft eccentricity and shaft mass by neglecting the vertical &axial axis's to plot the results in (**rms**) VS the motor frequency (**CPM**) using Matlab program to find the graph in fig 4.8 the oil film damping effect on rolling element spectrum. [20]

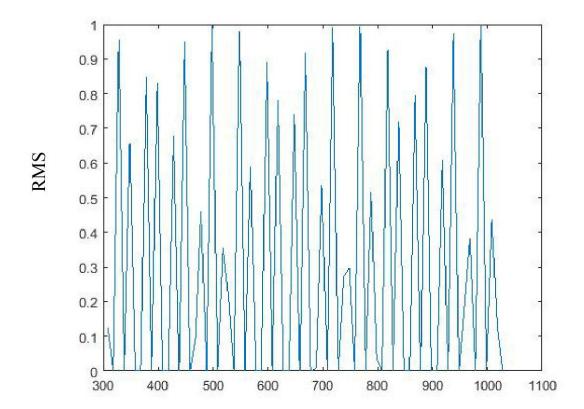
$$\frac{\ddot{x}}{c\omega} + \frac{p}{mc\omega^2} (c_{xx} \frac{\dot{x}}{c\omega} + k_{xx} \frac{x}{c} + c_{xy} \frac{\dot{y}}{c\omega} + k_{xy} \frac{y}{c}) = -\frac{e}{c} \cos \omega$$
(4.4.)

With assumption that the negligible stiffness effect and shaft mass, the above equation while it is free in x direction is:

$$X = \frac{e}{[\sin(\omega) + (0.5 \times \cos(\omega))]}$$
(4.5)

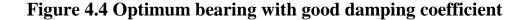
The studied bearing parameters substituted in equation (4.5) to use the results in plotting it VS the rotational using Matlab program to find the graph (Figure 4.3) that shows the oil film damping effect spectrum on rolling element.

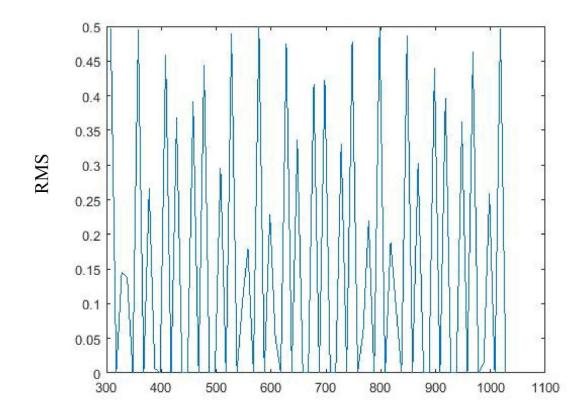




#### 4.2.3 The optimum bearing with good damping coefficient:

The bearing with good damping coefficient means the bearing is running according to motor speed and oil film damping effect. This can explain by adding the plot (4.2) to plot (4.3) and take the result in plot (4.4). The plot (4.4) amplitude is almost less than 50% of the free vibration amplitude this can illustrate the new well manufactured bearing status. From this stage the operator or the condition monitoring responsible should start to observe the condition change in vibration amplitude readings without needing to specific analyzing software for explaining the results. Only we can use the bearing manufacturing data for rolling element weight and dimensions to estimate the amplitude height to compare it with vibration measurement device reading to evaluate its status.





# 4.3 Experiment analysis of the case study:

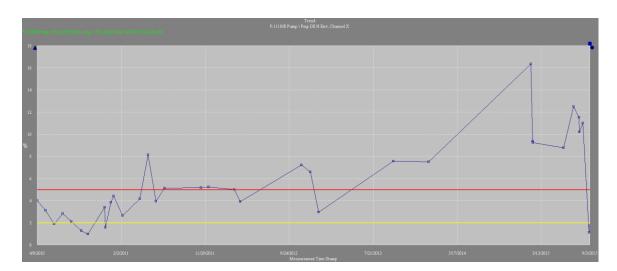
# 4.3.1 Overview of p – 1110/B centrifugal pump problem:

The table and trend are explaining the vibration readings of the above titled pump driving end horizontal direction from (the first of sep-2014) till (august -2015) by SKF unit GE (gravity enveloper) using acceleration transducer. The vibration amplitude has started to increase above the alarm zone in February 2014 and keeps increasing and decreasing over the alarm zone (see the table 4.1& fig 4.5)

POINT name	Date/Time	<u>Last</u> <u>value</u>	<u>Previous</u> <u>value</u>	<u>Units</u>	<u>% change</u>
Pump DE H Env	3-Sep-2015	1.732	1.109	GE	56.1
	31-Aug-2015	1.109	11.012	GE	-89.9
	9-Aug-2015	11.012	10.178	GE	8.19
	27-Jul-2015	10.178	11.507	GE	-11.5
	26-Jul-2015	11.507	12.497	GE	-7.92
	6-Jul-2015	12.497	8.774	GE	42.4
	1-Jun-2015	8.774	9.242	GE	-5.06
	11-Feb-2015	9.242	9.358	GE	-1.24
	10-Feb-2015	9.358	16.34	GE	-42.7
	4-Feb-2015	16.34	7.505	GE	118
	4-Feb-2014	7.505	7.551	GE	-0.61
	1-Oct-2013	7.551	2.943	GE	157

Table 4.1 vibration readings of the pump 1104B

Figure 4.5 the pump 1104 trend.



The trend shows that the vibration reading was normal less than 5 GE till the 1st of October 2013. After that from 4<sup>th</sup> of February 2014 the vibration level was start increasing above the alarm line (over 5 GE) to reach 11.012 at 9 august 2015. At this time the pump has stopped to maintenance .After dismantling the pump the observed visually is the angular contact bearing is loosed. The maintenance department has dismounted the existing bearing and install new one. Then the vibration reading become normal, the visual inspection has confirmed the vibration analysis.

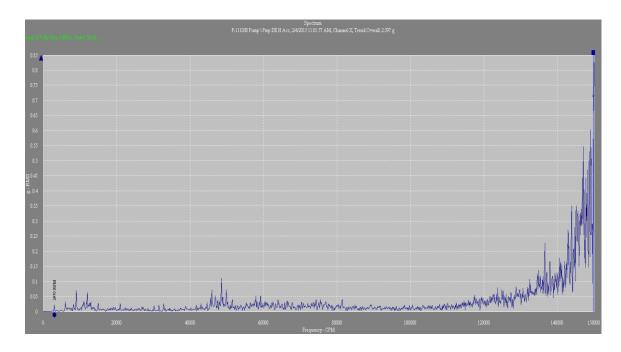
#### **4.3.2 Spectrum analysis and deterioration steps:**

#### 4.3.2.1 The first stage

In the first stage the spectrum show high vibration levels at the high frequency zone only around (150000 cpm) when the shaft speed was (3970 cpm) that less than 5000 cpm (see fig 4.6). The assumption is the good contact between the rolling elements and the bearing inner race the vibration root source at high frequency is equal to the diameters ratio between rolling element and the bearing inner race:-

Rolling element frequency = inner race frequency  $\times \frac{inner race radius}{rolling ellement radius}$ In this stage vibration amplitude rising only in the high frequency because the rolling element still controlled no surface loss in the inner race, outer race or cage and the lubricant oil penetrating easily in between the raceway and its components (cage and rolling elements).

## Fig 4.6 the first stage of deterioration spectrum.



#### 4.3.2.2 The second stage:-

In this stage there is no change in shaft zone or low frequency zone, but high vibration level appear in wide range in the high frequency zone exceeding the alarm line that (see the fig 4.7) can explained as multi balls has forgot their damping with slightly different values.

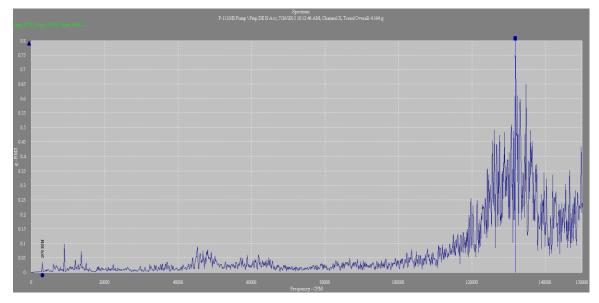


Fig 4.7 the second stage of deterioration spectrum

#### The last stage of deterioration:-

In this stage the vibration level at high frequency zone increase also the vibration readings at low frequency zone has grown up the interpretation is the rolling elements are moving freely with different frequencies between (100000cpm) to (150000 cpm) because the weight losses in moving elements contact areas in the inner race, oil film and outer race. As it is known all rotating component has impact in shaft frequency, the increasing of amplitude in low frequency evidence to unbalancing rotation in bearing's inner race. In this spectrum the vibration levels have increased in high and low frequencies which means the bearing has missed its function.

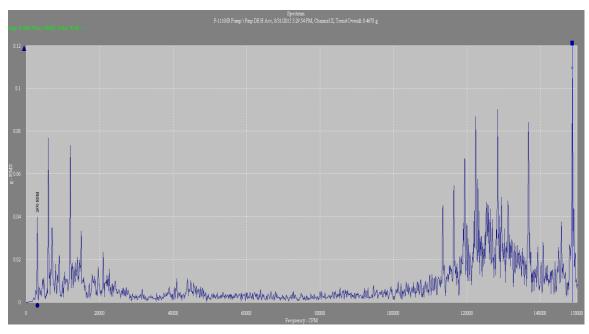


Fig 4.8 the last stage of deterioration spectrum

This is the interpretation of trend and spectrum to explain the bearings failure stages before enough time to damage. The maintenance department is prepared to repair this pump bearing, at August2015 the pump has stopped to maintain, after dismantling the visual inspection result is the bearing has been failed. This result confirms that the mathematical analysis which is similar to the practical analysis is correct.

From the previous results the research find that the main function to be checked in the bearing performance evaluation is the damping coefficient of bearing vibration that takes place due to lubricant viscosity and cage efficiency. According to the damping loss percentage the bearing status and its residual life time can be estimated through the following steps:

1- Providing the program with the bearing ball mass and diameter in addition to the equipment rotational speed in RPM to find out the maximum amplitude that will illustrate no damping coefficient x-axis in rms.

2- Checking the equipment vibration amplitude x-axis in rms at the bearing check location with any accurate accelerometer.

3- Compare the two previous values with 20% error to evaluate the bearing status putting in mind the more or less 20% value as zero value in the begging when the bearing is new.

# **CHAPTER FIVE**

# RESULT, CONCLUSION AND RECOMMENDATION

## CHAPTER FIVE RESULTS, CONCLUSION AND RECOMMENDATION

## 5. Results, conclusion and recommendation:

#### 5.1 Results:

The research calculations find that free bearings rolling elements has the maximum vibration amplitude that equal to (1rms), when the cage and lubricant oil working with the best damping coefficient the best vibration damping could be reached ,it equal to (1rms). After plotting the damping and vibration amplitude VS time in four different frequency zones the research has reached that in all zones the two plots summation give the best vibration readings that equal to (0.5 rms) after control the rolling elements random movement by the cage and oil damping . That means the good cage and compatible lubricant oil can damp 50% of the rolling elements vibration.

### 5.2 Conclusion:

After the experimental and modeling I compare the results with the calculations results that have been taken from the mathematical model and plotted using Matlap program the results was so similar with close magnitudes values and deviation almost 20%. This deviation was result to calculations and experiment& measurement tools accuracy. Following to these results we can use the mathematical model as technique to evaluate the bearing performance through damping check to take correct decision in selecting the bearing changing time because it can illustrate the actual

bearing status and its deterioration steps through functional damping coefficient losses this can lead the user to estimate the equipment bearings residual life to prepare the specific maintenance tools, spare parts, expertise technicians and the optimum down time. This can lead to:

1- Optimal Component Replacement Decisions Using Vibration Monitoring

2- The known repair date helps to have stable operation condition.

3- Good production plan which has impact in companies reliable deals and market share.

4- Optimum spare part inventory system that can help the procurement authority to save a lot of money when it pay only for consumable and predictable spare parts that can providing the best storage condition for it with minimum cost.

#### **5.3 Recommendations:**

1- I recommended that to study the pump axial and vertical direction to establish the optimum method which can help for detecting the other part residual life time and integer it with this method. This can lead us to rearranging spare part replacement in priorities to avoid the part replacement with some residual life.

2- Establish a method to do the repair in known intervals for replacing more than part in same time after finishing its maximum functional output.

#### References

- 1- Mobley, R. Keith, 1943 Root cause failure analysis. (Plant engineering maintenance series).
- 2- Paresh Girdhar BEng (Mech. Eng), Girdhar and Associates Edited by C. Scheffer PhD, MEng, and SAIMechE Series editor: Steve Mackay. Practical Machinery Vibration Analysis and Predictive Maintenance.
- 3- SKF CMTR 502-GX Introduction to the GX Series Microlog October 2010.
- 4- 4141 Ruffin Road San Diego, California 92123 USA Vibration Diagnostic Guide SKF Reliability Systems Web Site: www.skfreliability.com Copyright © 2000 by SKF Reliability Systems.
- 5- Michael Walsh, Predictive& Preventive maintenance Pumps & systems Nov 2011.
- <u>http://www.mcnallyinstitute.com/02-html/2-09.html</u>. Centrifugal pump vibration readings.
- 7- M.S.Leblod, K.Maynard, K.Reichard, M.Trethewey, D.Bireya, C.Lissenden, D.Dobbins Aerospace conference, IEEE 6-13 march 2004. Using torsional vibration analysis as synergistic method for crack detecting in rotating equipment.
- 8- P. J. Vlok, J. L. Coetzee, D. Banjevic, A. K. S. Jardine and V. Makis Journal of operational research society 2002.Optimal Component Replacement Decisions Using Vibration Monitoring and the Proportional-Hazards Model.

- 9- Laura Swanson International journal of production economics 18 April 2001, Pages 237–244 Linking maintenance strategies to performance
- A. Gral, L. Dieulle, C. Bernenguer, M.Roussignol IEEE Reliability Society, IEEE Transactions on Reliability Jun 2002, Continuous-time predictive-maintenance scheduling for a deteriorating system.
- 11- Lawrence Mann(Louisiana State University, Baton Rouge, Louisiana, USA), Anuj Saxena (Louisiana State University, Baton Rouge, Louisiana, USA)Gerald M. Knapp (Louisiana State University, Baton Rouge, Louisiana, USA) Journal of Quality in Maintenance Engineering 1995, Statistical-based or condition-based preventive maintenance.
- 12- P.D. McFadden, J.D. Smith Tribology International ,Volume
  17, Issue 1, February 1984, Vibration monitoring of rolling element
  bearings by the high-frequency resonance technique a review.
- 13- Mohamed Abduelmageed Suleiman July 2012. Nile valley university faculty of engineering, Using Vibration Monitoring as Predictive Maintenance Technique.
- 14- G. G. Yen; Sch. of Electr. & Comput. Eng., Oklahoma State Univ., Stillwater, OK, USA, Wavelet packet feature extraction for vibration monitoring 06 /2002. IEEE Industrial Electronics Society.
- 15- Ray S Beebe Elsevier Ltd .the Boulevard, Langford lane, Kidlington, Oxford OX5IGB, UK predictive maintenance of pump using condition monitoring.
- 16- A Davies (Ed) 1998 handbook of condition monitoring technique and methodology.

- P.D. MCFADDEN, M.M. TOOZHY Mechanical Systems and Signal Processing Volume 14, Issue 6, November 2000, Application of Synchronous averaging to vibration monitoring of rolling element bearings.
- 18- William T. Thomson prentice hall of India New Delhi-1100011979 Theory of vibration with application.
- 19- http://www.skf.com/sg/products/bearings/angular-contactball-bearings/ 7314 BEP.
- 20- C.R.Burrows& M.N.Sahnkaya Department of dynamic &control University of Strathclyde, Glasgow, UK October 1985 assessment of technique for estimating oil film bearing coefficients.

### Appendix

This appendix contains

1- The functions used in this study.

This program used to plotting the damped bearing spectrum.

A=10  $\omega$ =308:10:1036 e= 0.1 c=0.1  $X = \frac{e}{[\sin(\omega) + (0.5 \times \cos(\omega)]]}$ Y= Asin  $\omega$ U= Y-X Plot ( $\omega$ , subplas (u)) Grid

Axis ([300, 1100, 0, 15]):

This is the relations used to simplify the oil film damping coefficient equation.

X=Asin  $\omega$ .

 $\dot{X} = \omega A \cos \omega$ 

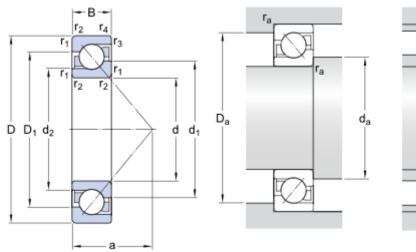
 $\ddot{X}$ = -  $\omega^2 A \sin \omega$ .

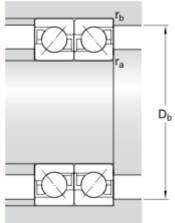
2- The studying bearing parameters.

7314 BEP

Dimensions

D		70	mm
D		150	mm
В		35	mm
d <sub>1</sub>	$\approx$	101.1	mm
d <sub>2</sub>	$\approx$	84.4	mm
<b>D</b> <sub>1</sub>	~	121.5	mm
А		64	mm
r <sub>1,2</sub>	min.	2.1	mm
r <sub>3,4</sub>	min.	1.1	mm





### **Abutment dimensions**

da	min.	82	mm
D <sub>a</sub>	Max.	138	mm
D <sub>b</sub>	Max.	143	mm
r <sub>a</sub>	Max.	2	mm
r <sub>b</sub>	Max.	1	mm

### **Calculation data**

Bearing pair arranged back-to-back or face-to-face						
Basic dynamic load rating	С	119	kN			
Basic static load rating	C <sub>0</sub>	90	kN			
Fatigue load limit	Pu	3.6	kN			
Reference speed		5300	r/min			
Limiting speed		5300	r/min			
Calculation factor	k <sub>r</sub>	0.1				
Calculation factor	ka	1.6				
Calculation factor	е	1.14				
Calculation factor	Х	0.57				
Calculation factor	Y <sub>0</sub>	0.52				
Calculation factor	Y <sub>1</sub>	0.55				
Calculation factor	<b>Y</b> <sub>2</sub>	0.93				
Single bearing or bearing pair arranged in tandem						
Calculation factor	Х	0.35				
Calculation factor	Y <sub>0</sub>	0.26				
Calculation factor	Y <sub>1</sub>	0				
Calculation factor	<b>Y</b> <sub>2</sub>	0.57				