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INTRODUCTION TO CONDITION-BASED MAINTENANCE*

Prepared for
CHEMICAL BRANCH OF UNIDO
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(Backst. off: M. Herce, IPCT/TP/INF)

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** TUNGSRAM Co. Ltd., Budapest, Hungary

CONTENTS

INTRODUCTION

This study is intended as a reference source for engineering to use when establishing industrial production plants and enterprices in the developing countries. Its aim is to make them familiar with the modern aspects of preventive and predictive maintenance and to demonstrate what kind of measures has to be taken for a cost-effective operation.

The study was prepared by Dr Antal Szabó director of the Tungsram Co and UNIDO-Expert since many years in co-operation with the Department of Industrial Operation and the Industrial and Technological Information Section of the United Nations Industrial Development Organization (UNIDO).

The opinions expressed are those of the expert and do not necessarily reflect the views of the secretariat of UNIDO.

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CHAPTER 1 REASONS FOR INTRODUCING PREVENTIVE MAINTENANCE

1.1 PURPOSES, OBJECTIVES AND METHODS OF FAULT DETECTION AND DIAGNOSIS

There is the task of engineers to design plant and construct machines with minimum of capital investment using a minimum of raw materials, energy and man-power, and to operate them over a long time with maximum economy, quality of output products and safety of equipment.

Chemical,oil and petrochemical plants today are characterized by

(i) complex processes
(ii) high output
(iii) long operational process trains
(iv) sophisticated and high performance equipment
(v) complex instrumentation and process control system
(vi) serious consequences for any catastrophic event.

Any malfunction of the existing plant equipment, respectively the breakdown of any machine increase the operating cost of the plant and can be very costly. Even more serious are the consequences of a serious accident, such as an explosion, because of the bad design or faulty operation.

The productivity gains due to the application of advanced technologies to the manufacturing process are offset in part by the need for sophisticated maintenance. If a failed component or equipment can be repaired while a redundant component or equipment has replaced it in service, then the overall reliability of the system is improved.

Any system or method of fault detection that permit the use of less expensive equipment, increases unit/equipment/plant availability,and/or reduces maintenance costs merits serious attention. Thus, the detection and analysis of faults (=diagnosis) in process equipment are of definite economic significance.

The degree of difficulty of fault detection and diagnosis depends very much on the nature of the fault. Complete malfunction of a part of equipment is generally relatively easy to detect, but by the time it has occured, considerable damage may have taken place. Detection of incipient or latent malfunctions is even more difficult.

Malfunction detection is very relevant from the point of view of reliability engineering. Failures of the process instruments and the key rotating machinery often can be prevented if the early signs of impending breakdown are recognized. One can distinguish the terms fault, malfunction and failure relating to equipment of any chemical plant.

We will use the terms <u>fault</u> and <u>malfunction</u> in relation to equipment as synonyms to describe the departure from an acceptable range of an observed variable or calculated parameter associated with this equipment. <u>Failure</u>, on the other hand, will be taken to mean complete inoperability of equipment or process. Most chemical processes are sufficient flexible and good organized that as soon as a fault shows up in any subsystem, the system compensates for the fault so as to continue operation. Thus, a fault will not necessary be a failure.

If more than one reason can occur, <u>fault diagnosis</u> refers to the determination of the component or the part of the equipment that is causing the fault(s). That is, diagnosis is the evaluation and the determination of which of the subsystems, or the components, or the environments is violating its given sufficient conditions for satisfying the process performance specifications. It has been found to be very difficult to isolate reasons that occur in complex systems in consequent of the existing interaction between the process components.

Diagnosis is a statistical decision making problem similar to the signal detection problem. When the signal is weak the observer must assess its degree of confidence and then make a judgment about whether to proceed as if the signal is there, or not there. When the information is obtained, other kind of information, such as probability must be taken into concideration. Diagnostic resolution (distinguishability) refers to the accuracy or precision with which a fault can be identified for the case in which mutiple faults occur. Fault diagnosis is a significant mean for effective maintenance program and vital to the success of any company.

1.2 MACHINE AND EQUIPMENT MAINTENANCE-OBJECTIVES, STRATEGIES AND ECONOMIC BENEFITS

1.2.1 Maintenance term

Maintenance is the combination of all technical and associated administrative actions intended to retain a part, an equipment, a sub-system or system in, or restore it to, a state in which it can perform its required function

Maintainability is the probability that a component or an equipment or a system that has failed will be restored to service within a given time

1.2.2 Aims

The quality of the company's maintenance program determines how productive the machine and equipment will be, how long the equipment will run and how safe they are for the people working around them. Bearing this things in your mind a properly done maintenance results in

- o extending the machine and equipment useful life
- o avoidance of the unforeseen and unscheduled machine breakdown assuring the optimum availability of installed equipment for production
- o elimination of the unnecessary overhauls
- o providing more efficient operation
- o improvement of plant quality performance

- o improvement of the plant safety and environmental protection, ensuring instant operational readiness of all equipment for emergency use
- o ensuring the safety of personnel
- o improvement of customer satisfaction.

For this reasons maintenance plays a vital role in industry and both in modern industrial and developing countries. By using advance methods, like sophisticated sensitive transducers, methods of diagnosis and the application of computer technology a significant amount of the machine maintenance costs can be reduced while at the same time both the system and the operational reliability will be improved. As a starting point for any discussion on up-to-date methods of maintenance it is useful to define what is meant by the term, and to describe how it relates to various forms of mainterance. In the interest of international standardization we will adopt the terms and definition from the British Standard BS 3811: 1984 on "Maintenance management terms in terotechnology" prepared under the direction of the Quality Management and Statistics Standards Committee. In this new revision of this standard attention has been paid to both national and international development in terminology.

1.2.3 Strategies

Maintenance measures are distinguished according to three areas of activity

- o service
- o inspection
- o repair.

The term <u>service</u> has a meaning to fulfil activities to maintain the basic condition/performance of an equipment respectively to keep it in operating condition (e.g.cleaning lubrication, replenishment of the consumables, adjustment, replacement of sealing).

Inspection is the process of measuring, examining, gauging, testing or otherwise comparing the unit with the applicable requirements.

The term repair has a meaning to restore an equipment to an acceptable condition by the renewal, replacement or mending of worn, damaged or decayed parts.

MAINTENANCE

SERVICE	**INSPECTION**	**REPAIR**
cleaning	measuring	restoring
routine	testing	renewing
attencion	examining	replacement

Figure 1-1. Maintenance measures

Of the various approaches and techniques employed to attain adequate maintenance the first and most important is maintenance prevention. Maintenance prevention has to be started at the design and engineering stages. Study of repeated failures with a view to reducing and/or eliminating their occurance through change of construction or material, improving operational conditions, training of both production and maintenance personnel are important steps towards attaining maintenance prevention after the equipment has been put into operation.

Maintenance work that is not eliminated by maintenance prevention must be planned. Several approaches and techniques can be used.

Let us now consider the different ways equipment can be maintained.

- o corrective maintenance which means the maintenance is carried out after a failure has occurred and the equipment is restored to a state in which it can perform its required function
- o preventive maintenance where the maintenance is carried out at predetermined intervals or corresponding to prescribed criteria and intended to reduce the probability of failure and the performance degradation of an equipment. Preventive maintenance can take two forms:
  - scheduled maintenance carried out to a predetermined interval of time, number of operations, mileage, etc.
  - condition-based (also called predictive) maintenance initiated as a result of knowledge of the condition of an equipment from routine or continuous monitoring.

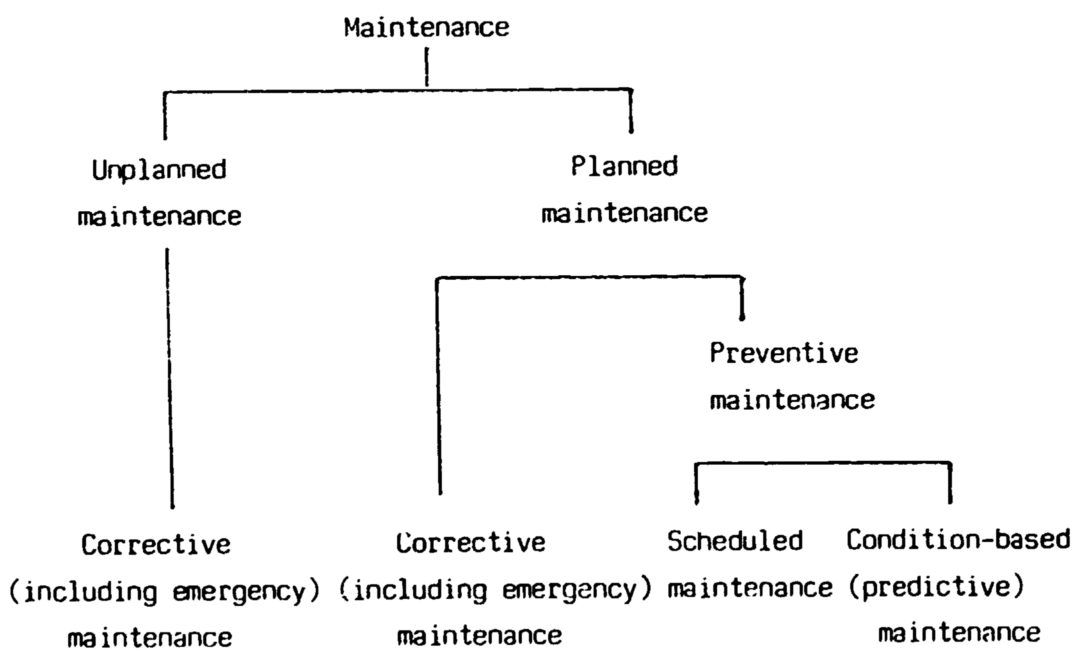


Figure 1-2. Chart showing relationship of various forms of maintenance

CORRECTIVE MAINTENANCE

In this strategy equipment, basically machines are operated until machine fails, and inefficiency or product spoilage forces a shutdown.

This concept is advisable only rarely in case of inexpensive machines and namely when the machine is duplicated or the loss of production is not important as redundant machines can take over the process.

This breakdown (also called as damage dependent) maintenance has several disadvantages. First the machine breakdown is absolutely outside the control of the operator and all machine downtime is unexpected. Secondly, machines allowed to run until failure and thus, require more extensive and expensive repaire than whould have been required if the problem had been detected and corrected early. Thirdly, some breakdowns can be catastrophic, requiring total replacement of the equipment.

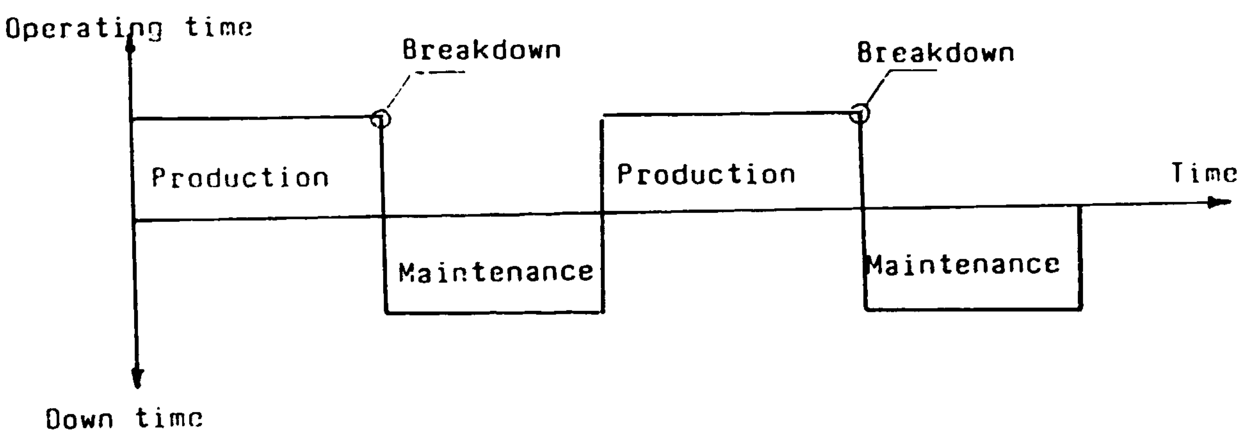


Figure 1-3. Machines histories in case of corrective maintenance

The corrective maintenance is prevalent in the domestic applications. E.g. vacuum cleaners, microwawe ovens,washing--machines, hairdriers, electric razors and air-conditioners are operated according to this conventional maintenance method.

SCHEDULED MAINTENANCE

This is a planned maintenance method organized and carried out time dependent with forethought, control and the use of records to a predetermined plan. Under this program each important and/or critical machines is shut down after a specified period of operation based on experience, and is inspected, partially or completely dismantled and replaced of its worn components - if any.

The up time is often determined statistically as the period of time from the date when the machine is in a new or fully serviced condition up to the time when the producer of the equipment expects no more than 2 % of the machine population to have failed. By servicing at this time intervals it is generally belived that 98 % of the machines should operate and survive the up time. A typical example of the scheduled maintenance is the bearing.

The scheduled maintenance is very safe, but expensive and not an optimum one. This method has disadvantages, too. First, to periodically dismantle every critical component of the equipment is expensive and time consuming. It is usually necessary to replace components which are still useable, while damaged components are replaced at to late in time. A typical example of this represent the rolling bearing. If they are replaced too soon, this incurs high costs. If the replacement comes to late, then shaft determinations, coupling breakage or other type of damage may be the costly results of the defected bearing. Secondly, as matter of fact the failure rate of many machines regularly is not improved by replacing wearing components. On the contrary, the reliability of newly-maintained machine is often reduced temporarily by human errors and interference. There is always a chance that bolts not tightened correctly, or gasket/seal will be improperly installed, or the original alignment of the machine disturbed during reassembly. Thirdly, the interval between periodic inspections is difficult to predict.

In spite of the above-mentioned this disadvantages can be reduced to an acceptable level by use of two complimentary techniques:

(i) by means of fault diagnosis
(ii) by means of continuous monitoring

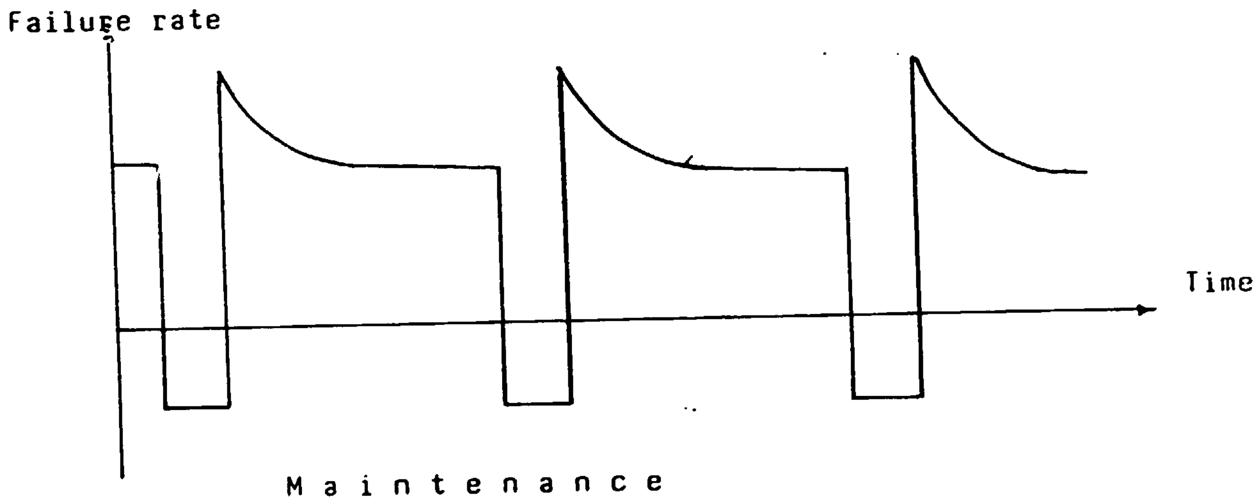


Figure 1-4. Failure rate in case of scheduled maintenance

CONDITION-BASED MAINTENANCE

It should be remembered that any industrial equipment seldom deteriorates in a few minutes, followed by a sudden failure. Generally, failure is the culmination of slow deterioration over a long period of time.

An acceptable compromise is to perform maintenance at irregular intervals, determinated by the real condition, "the health" of the equipment or its components. This method considers each equipment individually. By replacing fixed interval overhauls by fixed-interval measurement, developments in the operating condition of each individual equipment can be followed closely. That means for example we must know the actual condition of the machine at a given time and its deterioration trend over a period of time. The procedure based on this premise is known as condition-based maintenance. Condition-based (also called as predictive or condition dependent) monitoring.

This concept requires a continuous awareness of the condition of the machine. Operationally important parameters must be measured frequently, or better still continuously, evaluated and interpreted. In this connection it must not only be the actual machine condition, which is considered but the trends of the measured value will also assist in the prognosis. In this manner, the points in time when maintenance should be undertaken, can be laid down in advance and can, in turn, be brought into the operational production planning.

The fundamental difference between sc luled maintenance and condition-based maintenance is that scheduled maintenance is carried out as soon as the predetermined interval is elapsed, while condition-based maintenance requires checking at predetermined intervals with the maintenance carried out only if inspection shows that is required. Using appropriate measurements a quantitative measure of machine health can be provided, all at a low cost in relation to the scheduled maintenance, so the condition-based maintenance has a name of efficient cost-effective maintenance.

There are two main methods used for condition-based maintenance, and these are trend monitoring and condition checking.

(i) Trend monitoring is the continuous or regular measurement and interpretation data, collected during machine operation to indicate variations in the condition of the machine or its components. This involves the selection of some suitable and measurable indication of machine or component deterioration and the study of the trend in this measurement with running time to indicate when deterioration is exceeding a critical rate.

(ii) Condition checking is where a check measurement is taken with the machine running using some suitable indicator, and this is then used as a measure of the machine condition at that time.To be effective the measurement must be accurate and quantifiable, and there must be known limiting values which must not be exceeded for more than a certain number of permitted further running hours. To fix these values requires a large amount of recorded past experience for a given machine.

The condition checking method is less flexible than the trend monitoring. It can be particularly used, however in a situation, where there are several similar machines operating at the same plant as in this case comparitive checking can be done between the machine which is monitored and other machines which are known to be in good condition.

A comparison of methods of corrective and condition based maintenance is shown in Table 1.

1.2.4 Economics

The problem in planning of maintenace lies in the determination of an appropriate maintenance interval for the equipment because the operating time during which an item performs its intended function is not constant, but varies from one situation to another,due to differences in the installation and the operation condition. Fig. 1-5 shows how the maintenance costs of an equipment would be likely to vary with the life cycle of the item.

Table 1.

	PROBLEM/FAILURE DIAGNOSIS	TREND MONITORING	CONDITION CHECKING
Timing of measurements	When the problem has become manifest or after failure has occured	Reading taken at regular time intervals while the machine is running	Reading taken at one time while the machine is running
Qualitative measurements	When machine is is stopped, inspection of insponents can indicate the cause of the problem	Skilled operators can do subjective trend monitoring if they are close enough to their machines	Typical activity of an engineer when checking a machine during operation
Quantative measurements	Measurements may be analysed in considerable detail to provide guidance on possible causes of the problem	The taking of measurements and their recording and analysis gives a lead time on machine promlems	Numcrate values allow comparison with established standards or other similar machines to give knowledge of machine condition
	CO'.ECTIVE MAINTENCE	CONDITION BASED MAINTENANCE	

The most economic maintenance strategy included the method of maintenance with the maintenance measures to be carried out at the moment when the failure is just beginning attain full growth, it means that an equipment starts to departure from the acceptable condition.

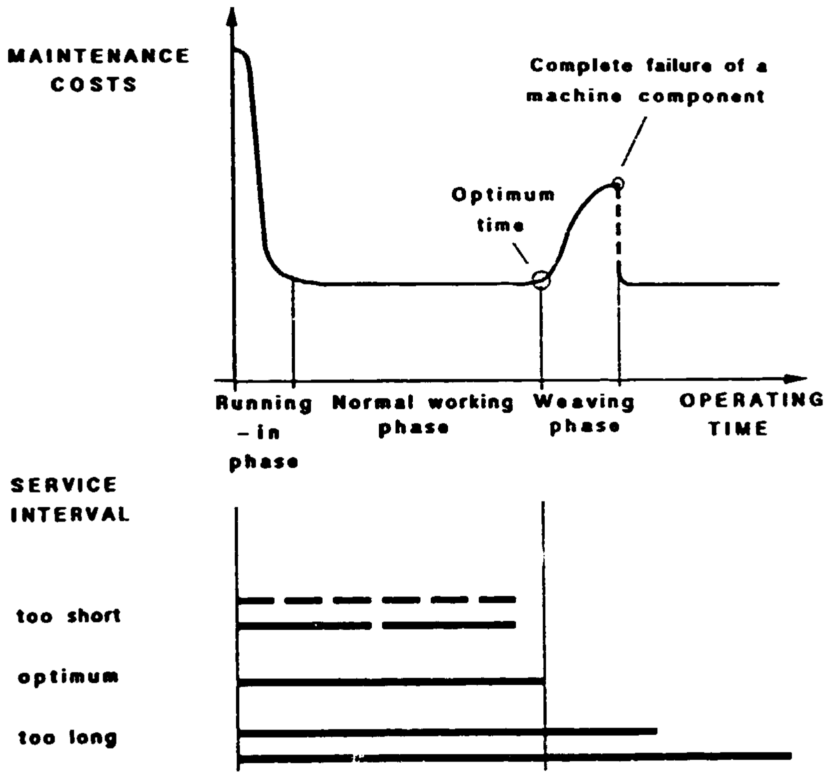


Figure 1-5. Maintenance cost function during the life cycle of an equipment

Fig.1-6 illustrates the development of life cycle costs of ownership of an equipment, taking into account all the costs of inspection, repair, maintenance and production losses in relation to the frequency of inspection.
It is generally recognised that the inspections cost in itself rises linearily with the frequency. The life cycle cost decrease as the number of inspections increases up to the optimum frequency of inspection. If the frequency of inspections oversteps that limit than the costs rise with the number of the surplus inspection.

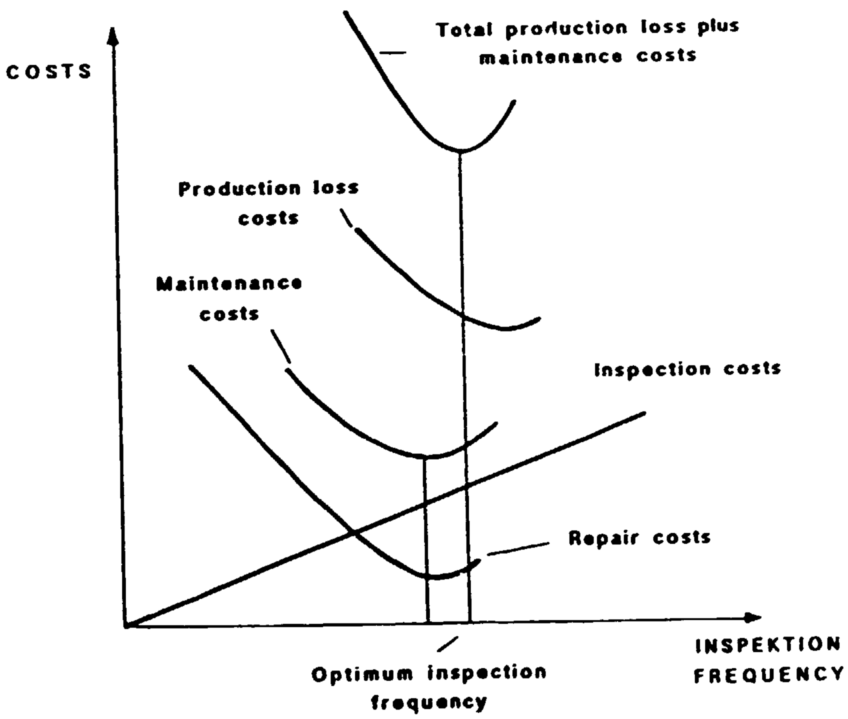


Figure 1-6. Life cycle costs in relation to inspection frequency

Figures 1-5. and 1-6. illustrate that the scheduled maintenance strategy will produce the optimum economic one only occasionally when the maintenance interval coincidens with the predictive interval of the condition-based maintenance.

1.3. THE CONDITION-BASED MAINTENANCE PROGRAM

To successfully implement a condition-based maintenance program, the following are requied:

- o appropriate methodolcgy (theoretically and practically justified methods)
- o appropriate instruments
- o adequate maintenace system including
  - uptodate maintenance organization
  - easy availability of the information
  - sophisticated informaticn processing
- o skilled personnel
  - skilled instrument operators
  - skiller diagnostic engineers
- o possibility of inspecting the component while the equipment and/or plant is running.

An effective condition-based maintenance program is a harmonic unity of the succession and logical sequences of steps of the following elements

DETECTION - INFORMATION OBTAINING

DIAGNOSIS - INFORMATION PROCESSING

CORRECTION - IMPROVING EQUIPMENT BEHAVIOR

1.3.1. Detection

The first step in the condition-based maintenance program is the CHECKING of the equipment condition, in other words the DETECTION of the problem.

It is obvious that not possible to list all types of faults and their expected occurances, that can be taken place in the industry. The reliability of process equipment such as compressors, pumps, fans, heat exchangers, separators, rectifiying columns,chemical reactors,furnaces,control valves,etc., is very difficult to collect even from historical records and predict because the failure rates and faults depend very much on the working conditions of the equipment including the fluid properties and physical parameters. Faulty operation depends not only on the steady state value of these working conditions, but even more on the dynamics of disturbances.

Deviations from process operating specifications can be determinated by particular observations and measurements, such as pressure and/or temperature deviations, level and/or flow rate increases and/or decreases, excessive and abnormal noise and/or vibration. Other measurable parameters might be: cavitation, corrosion, erosion, fluid hammer, expansion and/or contraction, fluid physico-chemical properties(density,boiling point, viscosity, chemical composition, etc), heat losses and others.

Diagnostic instruments able to check any component and detect component deterioration, that is to say the anomalous condition are essential for a successful condition-based maintenance program. The most important parameters to be measured and the instruments available are listed in Table 2. The suitable and measurable indications of wear and other component deterioration of rotating machinery are summarised in Table 3.

Many of the listed means are complementary to each other so that a correct diagnosis is often obtainable through the information processing from data acquired by different instruments.

Table 2

Parameters to be measured and instruments available
for condition-based maintenance

Type of inspection and examination	Equipment being examined: Process equipment	Electrical equipment	Rotating machinery	Measuring methods and instruments availables
Visual	o	o	o	Boroscope, stroboscope, tachometer
Temperature measurement	o	o	o	Contact and infrared thermometer
Thermal insulation and other heat losses	o			Thermography
Electrical parameters checking		o		Isolation checker, voltmeter, oscilloscope
Noise detection			o	Stethoscope, phonometer
Vibration detection			o	Vibrometer, vibration analyser
Ball and roller bearing defects			o	SPM (Shock Pulse Method) instrument, Spike energy recorder
Lubricant condition			o	Cromatography, magnetic plugs, spectroscope
Structural component defect	o			Acoustic emission, strain gauges, crack detectors
Corrosion, erosion, thickness detection	o			Non dectructive material testing (X-ray, ultrasound)
Wear	o			Radioactivity, X-ray fluorescence

Table 3

Machine components deterioration detection

DETECTION OF MACHINE HEALTH CONDITION	Total noise and vibration level Performance trends

DETECTION OF DEFECTS IN MACHINE COMPONENTS

Components		Perceptible defects	Inspection methods and parameter
Structural components			Stroboscopes Acoustic emission Resonance change Modal testing Strain gauges Ultrasonics Magnaflux
Fixed joint		Rattle, leaks, noise, fretting	Visual inspection Ultrasound measure-ment
Moving components	Seals	Wear debries, leaks, noise	
	Effi-ciency	Loss of power, friction, temperature	Vibrations Temperature
	Sur-face condi-tions	Staining Indentations Allowance changes	Vibrations SPM and Spike Energy measurements
	Wear deb-ris	Ferrous material	Ferrous chip detector
		Non-ferrous material	Filler
		Dissolved particles	Oil analysis

From practical point of view the following requirements should be taken into consideration in selecting the proper diagnostic instruments:

(i) The measuring instrument must be robust to able withstand both misuse and the industrial environment. Equipment designed for laboratory use is not suitable. Portable instruments need to be particularly robuts and should preferable be battery powerd through its own internal battery power supply.

(ii) The instrument must have sufficient accuracy and long term stability to enable it to be used for cheking machine and equipment health conditions when taking measurements over a long period of time.

(iii) The instrument should have a simple facility for checking its calibration or correct operation,so that the user confidence can be mainteined.

(iv The range of the instrument should cover all possible failure condition.

(v) The instrument should have either a built-in or connectable recorder and/or data collector to be used for evaluating the diagnostic feature of the equipment in the field and/or input data into a diagnostic maintenance system.

(v) The instrument may often have to be used in hazardous areas and an intrinsically safe version needs to be available.

(vii) Electronic instruments and their leads need to be suitably screened, as it frequently has to be used in area where high interference levels may exist.

(viii) The instrument's electronic and/or mechanic components should be easily for replacement or repair.

(ix) There are probably three levels of diagnostic instruments required:

for NEWCOMERS

very simple portable instruments with a simple un-switched indicator and/or warning lights, for use by relatively unshilled personnel. E.g.:

- o Model 808 "Troubleshooter" vibration meter (IRD Mechanalysis)
- o TAC-10 tachometer (SMP Instrument)
- o DO-301 non-contact type digital stroboscope (Shimpo)
- o Technoterm 730 thermometer (Testoterm)

for ADVANCE USERS

more comprehensive,but still portable instruments for use by more skilled or specialists/inspectors doing condition checking and trend monitoring. E.g.:

- o VIBROTEST vibration measuring instrument and/or VIBROPORT 30 vibration measuring equipment (Carl Schenck)
- o BEA-52 bearing analyzer (SPM Instrument)
- o MICROMACS analyzer (Vibro-Meter)
- o Vibration Analyzer type 2515 (Brüel & Kjaer)
- o Model 880 vibration spectrum analyzer and in-place balancer (IRD Mechanalysis)

for STATE-OF-THE-ART USERS

comprehensive special equipment for the continuous monitoring of particular plants. This can be used by experts and operated under skilled supervision E.g.:

- o Model 5915 machine monitor information center (IRD Mechanalysis)
- o SI 1220 multichannel spectrum analyzer (Schlumbergen)
- o 9000 Series ROMIS system (Bently Nevada)
- o VIBROCONTROL 2000 (Carl Schenck)

All equipment in the condition-based maintenance program should be inspected, tested or monitored, according to a prescribed schedule. Some kinds of equipment can be scheduled for checking daily,some every week or several weeks,some every month or every couple of months and,in the case of particular important and critical machines the monitoring can be done continuously.

The following are considered in setting up the frequency of information obtaining

- o criticality
- o availability of standby equipment
- o operating conditions
- o personnel safety
- o cost of repair and maintenance
- o cost of production loss
- o MTBF (Mean Time Between Failure) and MTTR (Mean Time To Repair) values.

The basic problem in planning on inspection frequency is that of determing how often a failure will take place between one cheking and other, while at the same time minimizing total life cycle costs. In spite of the fact that researchers have been suggested several formulas,but none are of pratcital use in the industry. In the chemical process industries equipment without standby in continuous processes it is rule to inspected weekly; in other manufacturing processes the frequency is once a month. When enough data are available, it will be possible to verify the adaquacy of the selected frequency and modify it if necessary.

1.3.2. Diagnosis

Once you have measured the equipment condition and trouble has been detected,the next step is to determine the nature and the purpose of the problem. This is the aim of DIAGNOSIS - to analyse and identify a specific equipment fault ("disease") by investigation of symptoms and history. In case of rotating machinery a specific machinery problem is carried out basically by identifying its unique vibration characteristics.

The techniques of fault detection and diagnosis can be listed in two general groups:

- pattern recognition
- estimation

The relationships amoung these two groups and the specific detection methods is illustrated in the Fig. 1-7.

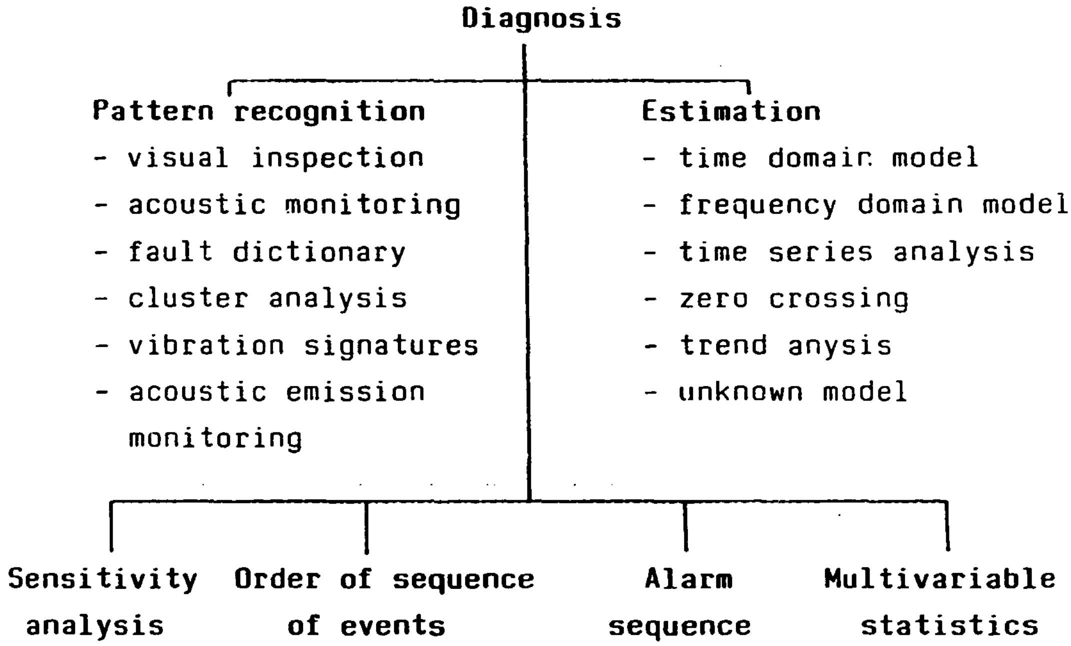


Fig. 1-7. Classification of methods for fault diagnosis

Many of the detection methods are interrelated, and could be treated as complements or extension of one another.
The most malfunctions in a plant are detected by the process operators. As the operator goes about his job, he either systematically or unsystematically carries out certain alarm scanning and fault detection. Much of this effort is usually devoted to trying to forecast production problems. Doing this he carries out a variety of checks on instruments reading using his practice and accumulated experience. He employs all human sensing facilities while checking continuously on the working of the equipment, that can

malfunction. An operator has his own set of early warming signals to eliminate the breakdown of the equipment.

The condition-based maintenance system on the other hand is based on measurements of selected equipment variables, so that calculations and deductions concerning process conditions can be made.

It is very desirable that at the time the plant is engineered, the design of the diagnostic system is also undertaken. If the only access points available for measurement are the input and output points, then the observation points can be employed so as to give the maximum possible resolution of faults.

The detection of incipient malfunction may be based on periodic equipment inspection. Trouble can be identified by comparing the readings of individual measurement either with the previous reading of the variable with healty condition or with other information about the equipment and past condition history.

Once you have diagnosed the cause of a problem, you have to prepare the equipment to seek treatment by corrective steps.

1.3.3. Correction

Once a faulty condition has been detected and the reason of the specific fault identified it is necessary to take the remedial action. When the fault is detected, a decision has to be made as to whether the fault is just temporary or it might call for a plant shutdown.

The corrective action might take different forms. It may be possible to apply a simple remedy if only a readjustment is requested. It may be that the fault has to be pinpointed so that a replaceable component can be introduced. When the incipient failure has been detected, it may often be possible to provide substitute measurements to reduce downtime. Machinery faults such as bad bearings, faulty gears, looseness or misalignment can be corrected using procedures

that are well known by maintenance people throughout the industry. However, correcting problems such as resonance or unbalance require special knowledge, skills and means. In order to improve e.g. the vibrational behaviour of machines, it may be necessary to investigate its dynamic properties to determine resonances (=natural frequencies), natural modes, dynamic stiffness and the damping factor.
Nowadays when using a supervisory control system a simply software program can prevent control action from taking place based on incorrect information by closing paths in the redundancy network which indude the faulty equipment, and diverting the information flow through a path known to be functioning correcty. In this way the plant availability can be improved even in the face of instrument or equipment failure.

1.3.4. Organizing a Condition-based Maintenance Program

Condition-based maintenance is a systematic method of monitoring and tranding plant equipment (rotating machinery, heat exchangers, rectifying column, chemical reactors, etc.) on a predeterminated scheduled basis to check and evaluate the condition of equipment subject to wear and deteriorate.
On-line detection trending and diagnostics provide an early warning and eliminates the need for periodic disassembly and inspection, and the possibility of an unexpected breakdown.
Organization and implementation of a Condition-based Maintenance Program (CMP) bases on the step-by-step approach.
IRD Mechanalysis point out the twelve essential steps in building a CMP. The flow diagram, Figure 1-8., shows the progress of these steps.

1. Plant Survey

The first step is to determine the feasibility of a CMP. Ideally this should be based on the analysis of the plant's machinery and equipment performance in terms of availability, downtime, reliability, etc. Such information is seldom available. Therefore, it is necessary to introduce a new function in the maintenance service: a systematic examination of the equipment or of the items of which the equipment composed, considering each machine as an individual with its own characteristics and life. The fundamental idea of the predictive maintenance is, that each machine is an individual and its life differs from that of other individuals, although they may look perfectly alike. The feasibility of applying this kind of maintenance is judged on the number and type of machines/epuipment plus the extensive experience of CMP personnel.

2. Machine selection

The aim here is to cover a manageable number of machines, taking into account manpower requirements, production schedules, downtime costs, etc.

3. Select Optimum Condition Monitoring Techniques

This stage is concerned with the WHAT, HOW, WHEN and WHERE of condition monitoring.

WHAT to measure? A parameter does exist which is indicative on the machine condition and failure progression?

HOW to measure? Instruments and techniques are available and capable of monitoring the parameter.

WHEN to measure? The monitoring technique must provide a useful failure detection period-that is, lead time between confirmation of a machine problem and eventual catastrophic failure. This fact will determine the frequency of monitoring.

WHERE to measure? Where to measure is of major importance in obtaining early detection of machine defects.

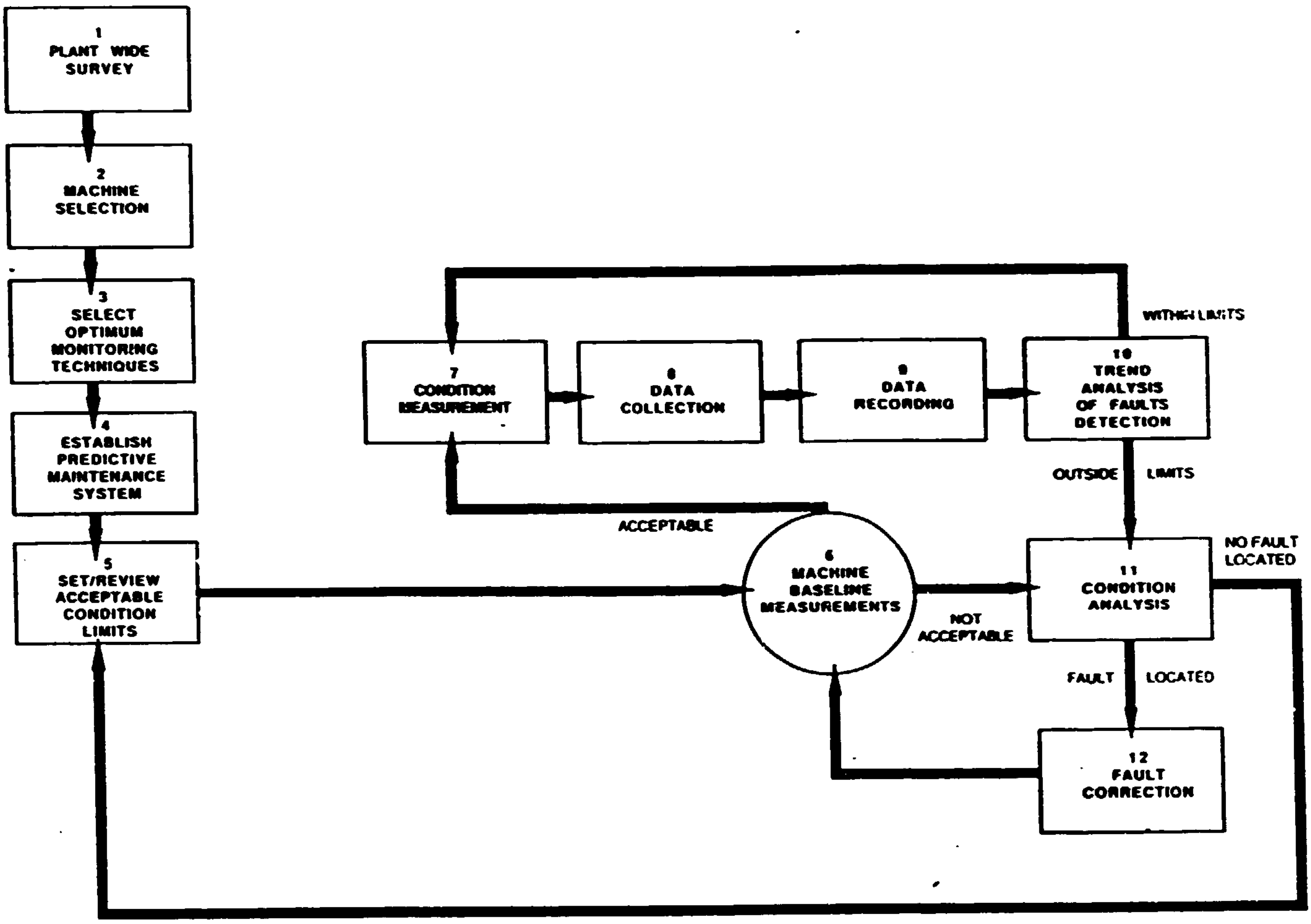


Fig. 1-8. Condition-based maintenance program flow diagram

4. Establish Predictive Maintenance System

Having established the optimum techniques for monitoring each item of the plant these are all merged to achieve a rational monitoring program encompassing:

- Establishing inspection Schedules
- Designing a simple data handling system:
  - o data collection
  - o data recording
  - o data analysis
  - o reporting

- A personnel training and education program

5. Set and Review Acceptable Condition Data and Limits

The object of this step is to establish the **"normal"** levels (**"baselines"**) of the condition monitoring parameters which represent an acceptable machine condition. This can only really be established on the basis of experience, that means by actual measurements of vibration, sound, temperature, etc. obtained when equipment is new and/or operating properly. However, in the initial stages, where no such information is available, machine manufacturers recommendations and appropriate standard general severity limits may be used to provide guidelines.

Based on these "normal" levels, action levels called as **severity limits** (alarm and emergency) are established which represent a significant change and deterioration in condition and provide a fair warning of impending failure.

The implementation phase of a CMP is a continuous evolving one. It is essential that the established severity limits are reviewed as experience and historical data dictate.

6. Machine Baseline Measurements

Since the mechanical condition of the machinery is not known initially, it is necessary to establish this through application of the selected condition monitoring techniques and comparison of observed measurements against the pre-established acceptable limits.

Where the machine condition proves acceptable, the machine passes onto the routine monitoring program. The baseline measurements serve as a "fingerprint" for comparison if a fault is detected during the lifetime of the machine.

Should the machine prove unacceptable by the established limits, this suggests a fault condition on the machine or inappropriate limits. Further analysis of the machine condition is required to locate and correct the fault or, if no fault is found, to review the established limits.

7-10. Periodic Condition Measurement

(Measurement, Collection, Recording, Trend Analysis)

These steps represent a routine monitoring program established in steps 3 and 4. The object of this program is to detect a significant deterioration in the machine condition through trend analysis of the measured data, whereupon the machine undergoes further Condition Analysis.

11. Condition Analysis

When permissible established limits are exceeded, the causes of this abnormal machine behaviour must be determined. This is an in-depth analysis(=diagnosis) of the machine condition. often involving the joint application of a number of methods and techniques. The object is to confirm whether a fault exists and to carry out a fault diagnosis and prognosis, i.e.: type of fault, location, severity, corrective action required.

12. Fault Correction

Having diagnosed the fault it is the responsibility of the Maintenance Department to schedule corrective action. At this stage it is essential to establish the cause of the fault condition and to correct it. Details of the identified fault should be fed back into the CPM to confirm the diagnosis and/or improve the diagnostic capabilities of the program.

Organizing a CMP the experimental phase is a very important one. The main problem arising is how to update information and overcome operator's suspiciousness over the new technique and its precision. During the implementation phase the correlation between diagnosis and actual conditions is verified, and the testing respectivelly checking procedure changed where necessary.

Remember that maintenance system including diagnostic equipment is only one leg of a tripod; the other two are the human recources available (the plant operator and the engineering staff)and the information system set up in the plant.

CHAPTER 2 BASICS OF VIBRATION

2.1. MACHINES AND VIBRATION

Listening to noise from rotating or reciprocating pump or compressor, or from flow in pipes and heat exchangers is an old diagnostic technique. It has been used to forecast and predict abnormalities such as:

- (i) loose bolts or other means of fastening;
- (ii) wearing away of gaskets;
- (iii) wearing away of metal;
- (iv) fatigue or cracks in the material arising in the internal structure of a vessel or in a pipe wall or joint;
- (v) incipient cavitation;
- (vi) flow blockage caused by the accumulation of material in the system, or by structural part that has broken loose;
- (vii) increasing vibration;
- (viii) instabilities or other departures from normal separation, cooling or heating processes.

Acoustic noise and vibrations represent fundamental mechanical events occuring in machinery, such as rolling and sliding or flow characteristics of the fluid and equipment. An ideal machine would produce no vibration at all because all energy would be channelled into the job of work to be done. In practice vibration occurs as a by-product of the normal transmission of cyclic forces through the mechanism. Machine elements react against each other and energy is dissipated through the structure in the form of vibration. A good design will produce low levels of inherent vibration. As the machine wears however, foundations settle, and parts deform, subtle changes in the dynamic properties of the machine begin to occur. Shafts become misaligned, parts begin to wear, rotors become unbalanced and clearances increase. All of these factors are reflected in an increase in vibration energy. This vibration energy dissipates throughout the machine, excites resonances and puts considerable extra dynamic loads on bearings. A deterioration in the machine's running condition always produces a corresponding increase of the vibration level. Cause and effect reinforce each other, and the machine progresses towards ultimate breakdown.

In the past experienced plant engineers have been able to recognise by touch and hearing whether a machine was running smoothly or whether a fault was developing. This cannot be relied upon any longer for at least two reasons:

- The personal realtionship between man and machine is not economically feasible because machines are expected to run automatically with only occasional attention from service personnel.
- Most modern machinery runs so fast that many vibrations occur at such a high frequency that instruments are needed to detect and measure them.

By monitoring vibration level it is possible to interpret in terms of the internal condition of process, the machine's condition,and also use to predict failure. Other parameters, such as temperature and oil pressure, also change as the condition of the machine deteriorates, but generally much later in the development of a fault than does the vibration level. Noise measurements can also reveal machine deteriorates, but noise is often more difficult to associate with a particular source in factories where many machines are working at the same time in the same area.

As the main sources of faults in industrial machinery have a mechanical origin, it is logical to choose a mechanical phenomenon as the representative parameter of machine condition. Mechanical vibration has proved to be one of the most reliable parameters to use in machine health monitoring to check the machine condition.

2.2 CHARACTERISTICS OF VIBRATION

2.2.1. Nature of vibration

Determination

Vibration is simply the reciprocating motion of the particles of elastic body (a machine or machine part) back and forth produced by disturbance of its equilibrium.
The simplest way to show vibration is to follow the motion of a weight suspended on the end of a spring as shown in Fig. 2-1. This is typical of all machines since they have weight and spring-like properties, too.
Until a force is applied to the weight to cause it to move, we have no vibration. By applying an upward force the weight would move upward, compressing the spring. If we released the weight, it would drop below its neutral position (equilibrium) to some bottom limit to travel, where the spring would stop the weight. The weight would then travel upward through the neutral position to the top limit of motion, and back again throuch the equilibrium position. This motion will continue in exactly the same manner as the force is reapplied. Thus, vibration is the response of a system to some internal or external excitation or force applied to the system.
This motion and thus, the vibration can be periodic - where the motion repeats itself exactly after certain periods of time - or random - where the vibration cannot be precisely predicted for any given instant of time.

2.2.2. The characteristics of vibration

The cause of vibration, regardless of the type, must be a force which is changing in either its direction or its amount; and the resulting characteristics will be determined by the manner in which the forces are generated. This is why each cause of vibration has its own individual characteristic.

The condition of a machine is determined by measuring its vibration characteristics. The more important parameters of these characteristics include:

o Frequency

o Displacement

o Velocity

o Acceleration

o Phase

Referring to the weight suspended on a spring, we can study the detailed characteristics of vibration by plotting the movement of the weight against time.

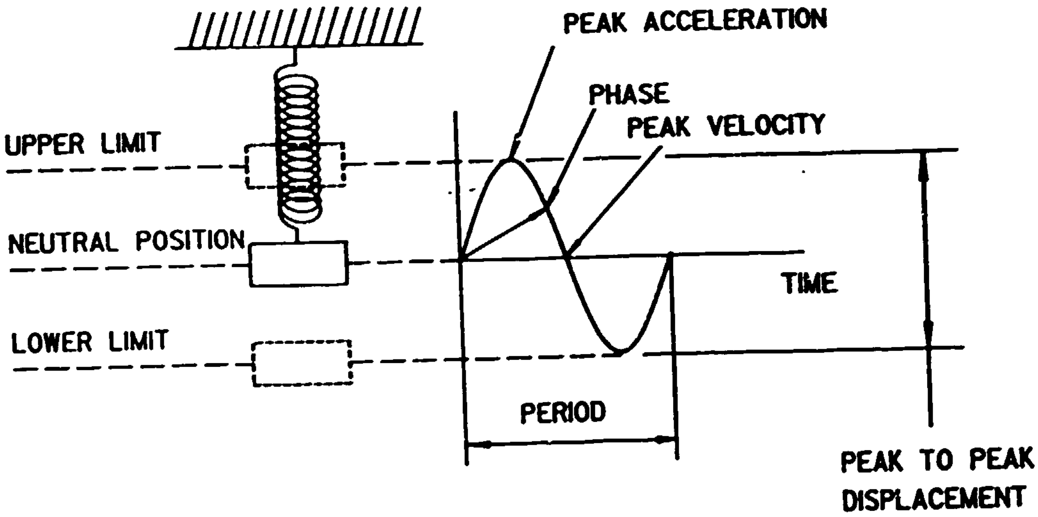


Figure 2-1. Movement of a vibrating weight plotted with time

The motion of the weight from its neutral position to the upper limit of travel back trough the neutral position to the lower limit of travel, and its return to the neutral position, represents one cycle of the motion and has all the characteristics needed to measure the vibration.

2.2.2.1 Characteristics of periodic vibration

The simplest form of periodic vibration is a simple harmonic motion which has a sinusoidal function of time (Fig. 2-2). Here T is the period of vibration, i.e. the time elapsed between two successive exactly equal positions of motion.

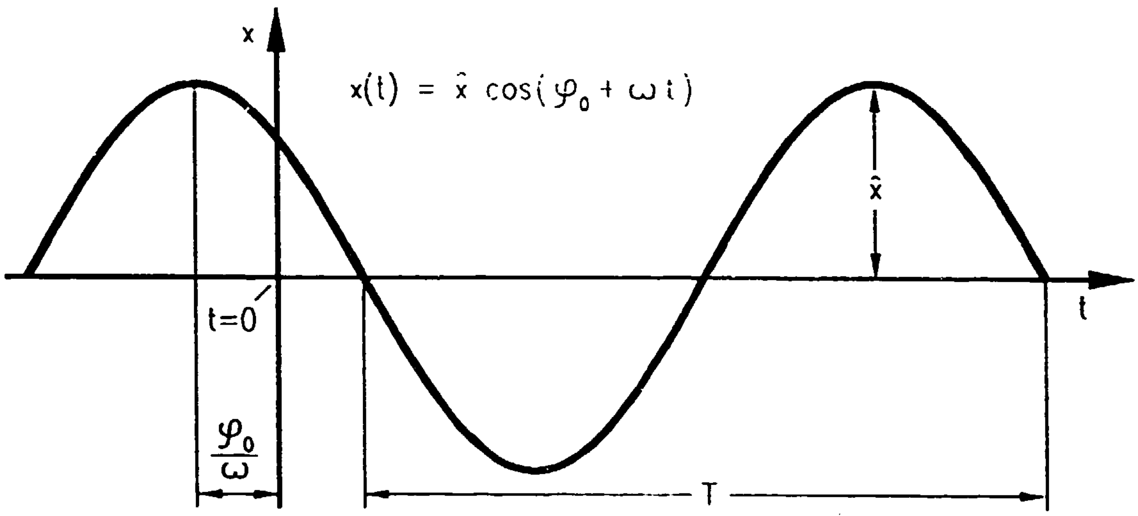


Figure 2-2. Parameters of simple harmonic vibration signal

Any periodic vibration mathematically exactly can be described by tree parameters: the amplitude, frequency and phase angle.

Thus

$$x(t) = \hat{x} \sin (\omega t + \varphi_0) \qquad (2.1)$$

where

x is the simple harmonic quantity of vibration;

x is the amplitude;

ω is the angular frequancy;

t is the independent variable, here the time;

φ_0 is the phase angle of the oscillation.

The maximum value of the simple harmonic quantity is the amplitude $\hat{x}$.

Amplitude

The actual value of the harmonic vibration can be quantified by the amplitude, which is a vector and varies sinusoidally with time. The amplitude is the measure of the vibration so this is suitable for the evaluation of the machines' vibration load.

Frequency

Vibration frequency is the measure of the number of complete cycles that occur in a specified period of time. Frequency is related to the period of a vibration pattern by

$$f = \frac{1}{\text{period}} = \frac{1}{T} \qquad (2.2)$$

The frequency of vibration is usually expressed as the number of cycles per second called the Hertz (Hz) or the number of cycles that occur each minute. This is the origin of the other term: cycles per minute, or CPM.

Specifying vibration frequency in terms of CPM makes it easy to relate this parameter to another important specification for rotating machinery: the number of revolutions per minute - RMP. So if you have a machine that rotates at 2500 RPM, you can expect certain problems to create vibration at a frequency of 2500 RPM.

Given a frequency specified in Hz, you can convert it to CPM

$$CPM = Hertz \times 60 \qquad (2.3)$$

Vibration phase

Phase is defined as "the fractional part of a period through which a sinusoidal quantity has advanced as measured from a value of the independent variable - here: time - as a reference." With other words the phase is the position of a vibrating particle at a given instance with reference to a fixed point or the stating point of the oscillating particle at time t=0.

In a practical sense, phase measurements offer a convenient way to compare one vibration motion with another,or to determine how one part is vibrating relative to another part.

Phase measurements are important when analyzing mechanical problems in machinery.

2.2.2.2 Measurement parameters of vibration

There are three quantities which are of interest in vibration studies, the vibration displacement, velocity and acceleration. These quantities relate very simple to each other. If the vibration has the form of a translational oscillation along one axis (x) only, the instantaneous **displacement** of the vibrating particle/body from the neutral position can be described by

$$x = \hat{x} \sin \left(2\pi \frac{t}{T}\right) = \hat{x} \sin (2\pi f t) = \hat{x} \sin (\omega t) \qquad (2.4)$$

where

$\omega = 2\pi f$ = angular frequency.

As the velocity of a moving particle/body is the time rate of change of the displacement, the motion can also by described in terms of **velocity** (v):

$$v = \frac{dx}{dt} = \omega \hat{x} \cos(\omega t) = v \cos(\omega t) = \hat{v} \sin (\omega t + \pi/2) \qquad (2.5)$$

Finally, the **acceleration** (a) of the vibration is the time rate of change of the velocity:

$$a = \frac{dv}{dt} = \frac{d^2x}{dt^2} = -\omega^2 x \sin (\omega t) = -\hat{a} \sin (\omega t) = \hat{a} \sin (\omega t + \pi) \qquad (2.6)$$

From these equations it can be seen that the form and period of vibration remain the same whether it is the displacement, the velocity or the acceleration that is being investigated. However the velocity leads the displacement by a phase angle of $\pi/2$ (90^0) and the acceleration leads to velocity by the same phase angle of 90^0. As characterizing values for the magnitude the maximum amplitudes called as peak values have been used, i.e. $\hat{x}$, $\hat{v}$ and $\hat{a}$.

Vibration displacement

The full distance of movement of a vibrating particle or part of a machine from the reference position is the vibration displacement Machinery vibration displacement is typically a peak-to-peak measurement of the observed motion, and is usually expressed in unit of micrometres ($1 \mu m = 10^{-3} mm$) or mils ($1 \text{ mil} = 10^{-3}$ inch).

Vibration velocity

Since the machine is moving, it must be moving at certain speed. However, the speed is constantly changing depending upon its position. The speed of velocity is greatest as the vibrating particle or part of a machine passes through its equilibrium position. The velocity of the motion is definitely a characteristic of the vibration, but since it is changing throughout the cycle, the highest peak velocity is selected for measurement. Vibration velocity is expressed in terms of millimeters per second peak (mm/s).

Vibration accelation

In discussing vibration velocity, we pointed out that the velocity of the part of a machine approaches zero at the extreme limits of travel. Of course,each time the part comes to a stop at the limit of travel,it must accelerate to reach speed as it moves towards the other extreme limit of travel. This is why the vibration acceleration is an important characteristic of vibration,too. Technically,acceleration is the time rate of change of velocity. Typical units for acceleration are expressed in g's peak,where "g" is the force of gravity at the surface of the earth ($1 g = 9{,}81 m/s^2$).

The relationship between the displacement, velocity and acceleration is given in the Table 4.

Table 4

Summary of vibration characteristics

Conversation	Displacement $x\,[\mu m]$	Velocity $v\,[mm/s]$	Acceleration $a\,[m/s^2]$
Displacement x	1	$\frac{v}{\omega}$	$\frac{a}{\omega^2}$
Velocity v	$x * \omega$	1	$\frac{a}{\omega}$
Acceleration a	$x * \omega^2$	$v * \omega$	1

2.2.2.3 Amplitude of the vibration signal

The amplitude of a vibration signal shown in Fig. 2-3 can be described by using the following measurement units: Peak-to-Peak, Peak, Avarage and RMS.

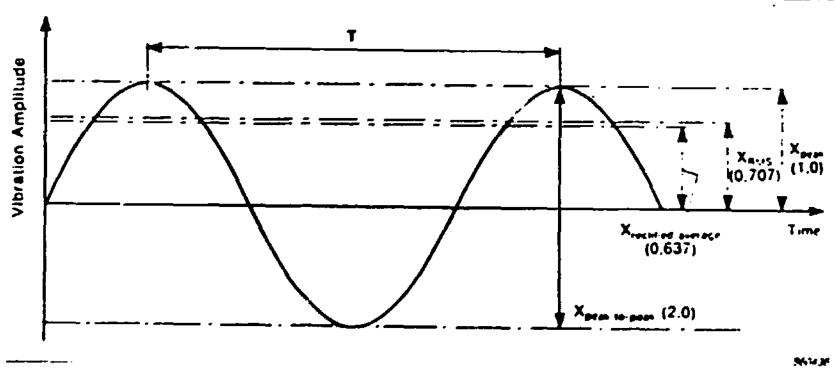


Figure 2-3. Measurement units of vibration. The amplitude referred to can be displacement, velocity or acceleration

(i) **Peak-to-Peak**

The peak-to-peak value describes the maximum excursion between the upper limit and lower limit of travel, that is the difference between the positive and negative extreem values of the dynamic motion. Vibration displacement measurements are often measured in μm, peak-to-peak.

(ii) **Peak Level**

Peak measurements are onehalf the peak-to-peak value. Peak level defines the maximum excursion which is measured and suitable in measurement of short duration shocks. However, no information is taken of the time history of the vibration.

(iii) **Average Level**

The avarage measurements take the time history of vibration into account but there is no useful relationship between the average value and the physical quantity. The average absolute value defined

$$X_{Avarage} = \frac{1}{T}\int_0^T |x| \, dt \qquad (2.7)$$

(iv) **RMS or Root-Mean-Square Level**

This value provides the most useful information about the vibration levels.

The square root of the integrated time-averaged squared function is related to the vibration energy and hence the vibration's damage potential. This term is often used when vibrations are random or consists of a number of harmonic vibrations of different frequencies. The The RMS value is basicly related to the velocity:

$$V_{RMS} = \sqrt{\frac{1}{T}\int_0^T v^2(t) \, dt} \qquad (2.8)$$

(v) **Crest Factor**

The crest factor defines the ratio of the peak value of a signal to the RMS value:

$$F_c = \frac{X_{peak}}{X_{RMS}} \qquad (2.9)$$

As the vibration becomes more impulsive or more random the crest factor increases. Hence by monitoring the growth of the crest factor, it is possible to predict a breakdown or machine part's fault.

Table 5 indicates the conversation factors of the periodic vibrations.

Table 5

Conversation of different measurement units

to → Multiplication factor from ↓	Peak-to-Peak	Peak	Average	RMS
Peak-to-Peak	1	0.5	0.32	0.35
Peak	2	1	0.64	0.71
Average	3.14	1.57	1	1.11
RMS	2.83	1.41	0.9	1

2.2.2.4 Non-harmonic vibration

Most of the vibrations encountered in daily life are not pure harmonic motions even though many of them may be characterized as periodic. Formation of a non-harmonic periodic vibration is illustrated in Fig. 2-4. This example is a combination of two sine waves. It can be seen that one sine ware has a frequency which is twice the other, and that its amplitude is much smaller. The amplitude x_1 and x_2 are summed arithmetically to obtain the combined signal (piston acceleration of a combustion engine). By determining the peak, average and RMS-value of this vibration a lot of useful information can be obtain. However, it will be practically impossible to predict all the various effects that the vibration might produce in connected structural elements.

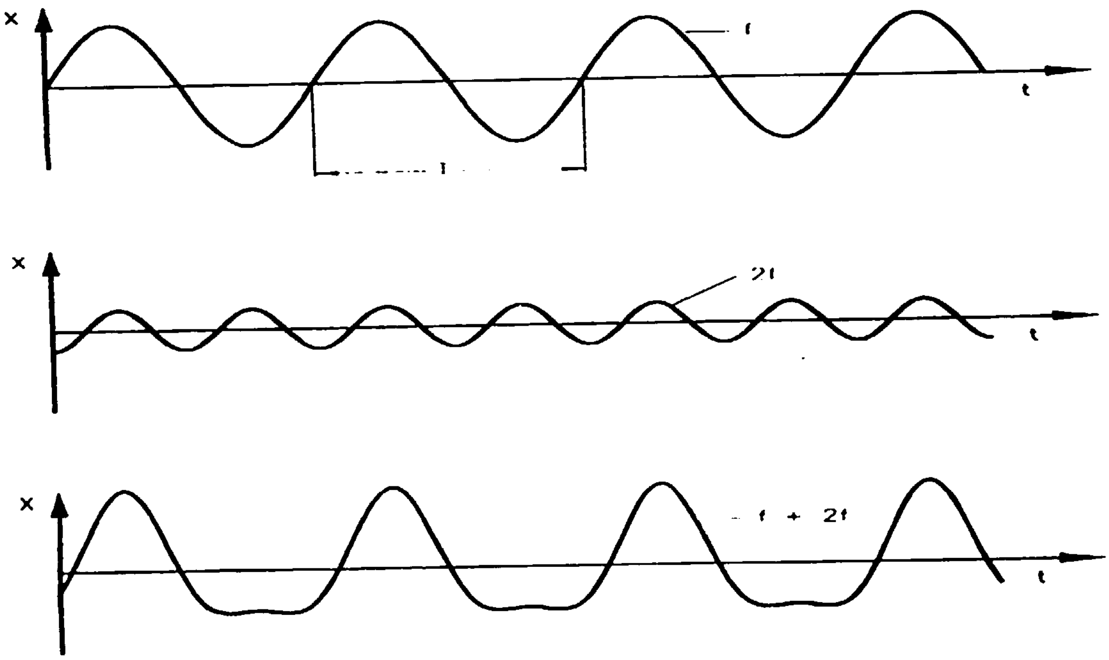


Figure 2-4. Superposition of harmonic vibration results in a non-harmonic one

One of the most powerful descriptive methods is the **method of frequency analysis**. This is based on a mathematical theorem, formulated by **FOURIER**, which states that any periodic curve, matter how complex, may be looked upon as a combination of a number of pure sinusoidal curves whith harmonically related frequencies:

$$f(t)=X_0+X_1\sin(\omega t+\varphi_1)+X_2\sin(2\omega t+\varphi_2)+X_3\sin(3\omega t+\varphi_3)+ +\ldots X_n\sin(n\omega t+\varphi_n) \qquad (2.10)$$

The number of terms required may be infinite, but in the case as the number of elements in the series is increased it becomes an increasingly better approximation to the original curve.The various elements consitute the **vibration frequency spectrum.** The frequency spectrum portraits the amplitudes and frequencies of all the different components of a complex signal. This is a plot of amplitude against frequency and a few simple examples are given in Fig. 2-5.

As can be seen from Fig.2-5. (top), the simple sine wave has a spectrum consisting of one discrete frequency (f=1/T), and is represented by a line whose height is related to the amplitude. The waveform of Fig. 2-4. has two components,each having a discrete frequency. Since one frequency is twice the other (f and 2f) the periods will also be in a ratio of two. (If the lower frequency f has a period of T seconds (f= 1/T) then the higher frequency will have a period of T/2 seconds (2f = 2/T), and viceversa.) The two frequencies each have one line on the spectrum,whose height is related to its amplitude (see middle in Fig. 2-5). In this case, the frequency f would be called the **fundamental frequency,** since it is the main one,and the frequency 2f is its second harmonic, since it is twice the frequency of the fundamental.

A "square" wave is shown in Fig.2-5. bottom which is yet another example of a periodic function. This consists of a fundamental frequency f, and odd harmonics only (i.e. 1, 3, 5, 7), which drop off in level at a known defined rate.

All periodic functions, such as those shown above can be defined by precise mathematical equations, which can make their analysis much simpler. Such functions are termed **deterministic.** A specific feature of periodic vibrations, which becomes clear by looking at Fig. 2-5 is that their frequency spectra consist of discrete lines when presented in the frequency domain. This is in contrast to random vibrations which show continuous frequency spectra.

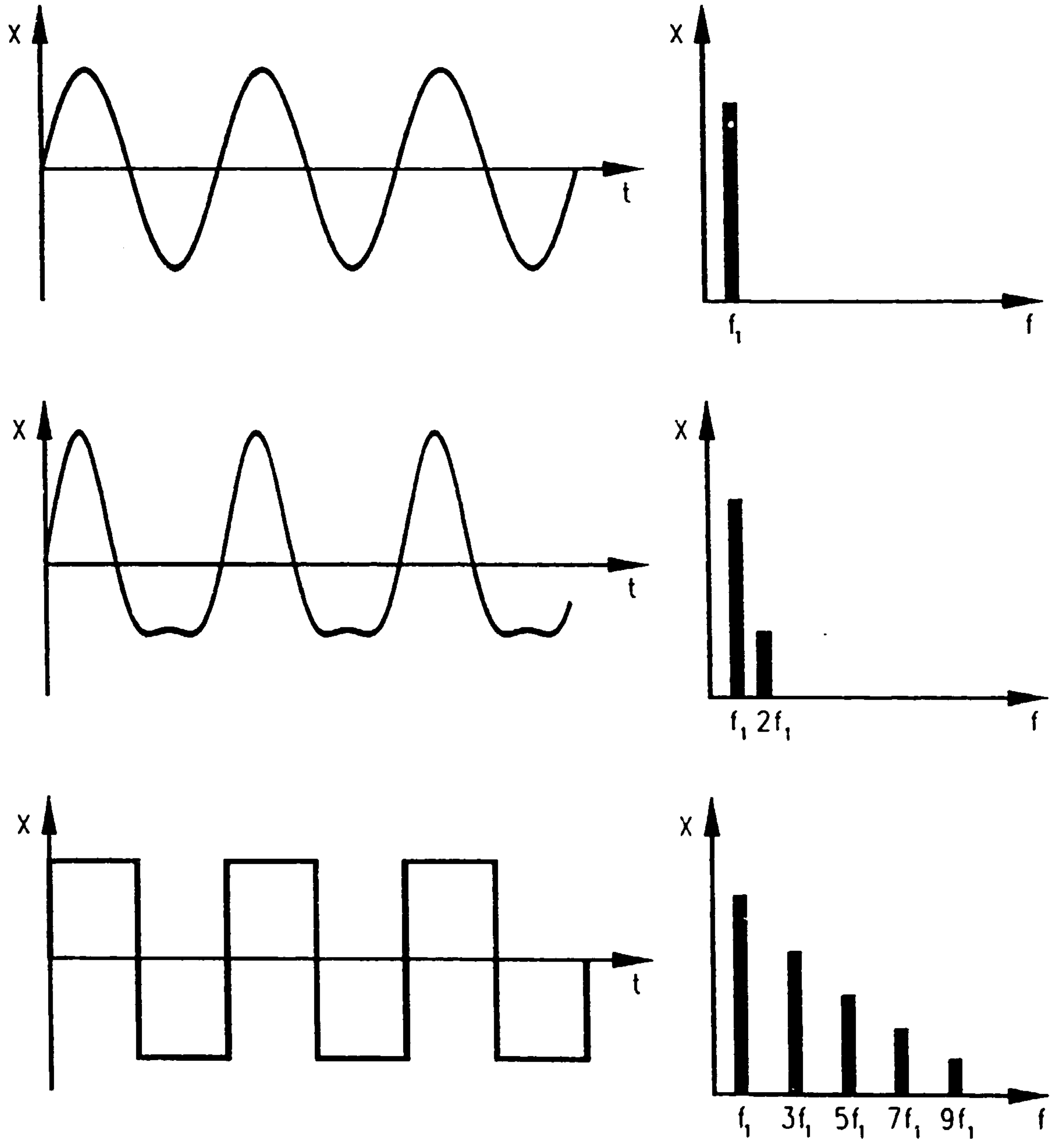


Figure 2-5. Examples of non-harmonic periodic signals and their frequency spectra

2.2.3. Information provided by vibration frequency

When analyzing a machine's vibration to investigate a particular problem, it is important to know the vibration frequency. Knowing the frequency helps to identify the reason of the problem.

The forces resulting in vibration are generated through the rotating motion of the machine's parts. These forces change in direction and amplitude according to the rotational speed of the parts. In connection with this many vibration problems will have frequencies that are closely related to the rotational speeds.

It is important to realize that different machinery problems cause different frequencies of vibration. Unbalance and eccentric journal can cause vibration frequencies equal to 1xRPM, whereas combinations of misalignment and mechanical looseness can generarate the harmonics of the fundamental frequency. Fig 2-6 shows that most machinery vibration consists of different frequencies. In a complex vibration analysis beside the fundamental frequency the other one of special importance is the dominant frequency having the largest amplitude.

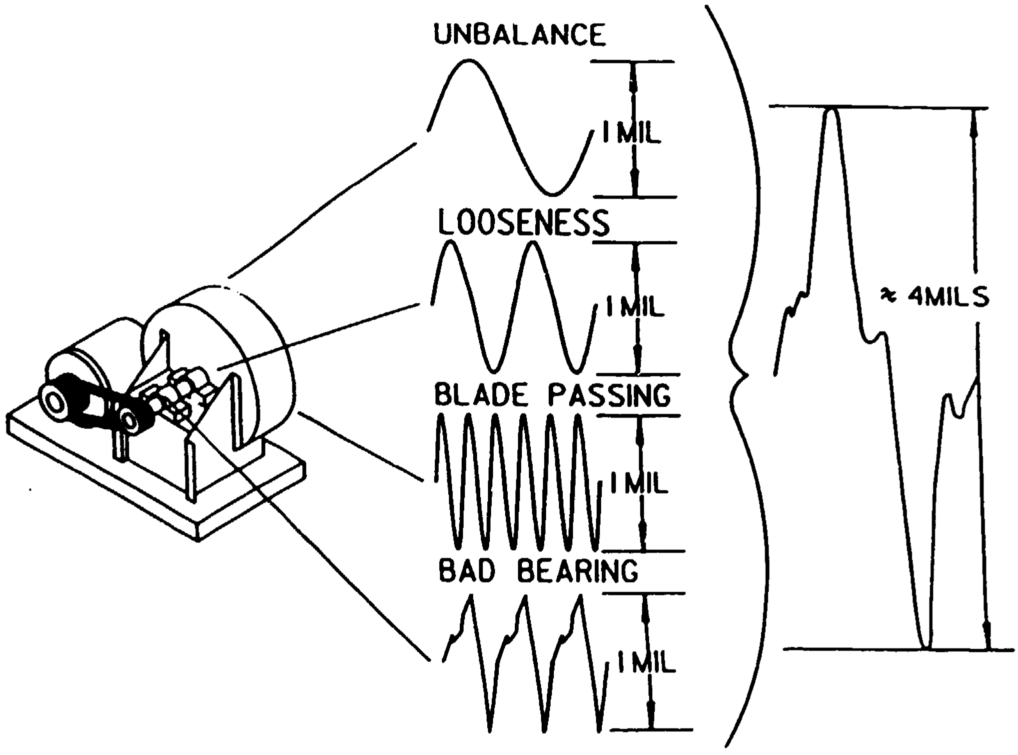


Figure 2-6. Combination of vibration frequencies

2.2.4. When to use displacement, velocity and acceleration

In terms of the operation of a machine,the vibration amplitude is the primary indicator of a machine's condition. The greater the amplitude, the more severe the vibration. However, the fact that amplitude can be measured in terms of displacement, velocity and acceleration raises the question: which should I use?

In theory it is irrelevant which of the three parameters are chosen to measure vibration.

The main significance of the relationship between these three vibration characteristics coming out from equs. (2.4) ...(2.6) is that at any given frequency one measurement can provide the other two. Differentiating the displacement will give the velocity and differentiating the velocity gives the acceleration, although these two in the practice are not normally performed. Integrating acceleration will give velocity, and integrating velocity again gives displacement, and this is the fairly standard technical procedure. So with instrumentation based on the use of an accelerometer pickup, the user can choose between acceleration, velocity and displacement as the measurement parameter. Different engineers have different preferences.

If we plot a narrow-band frequency spectrum of a vibration signal against these three parameters, you can see that they have peaks at the same frequencies, but nevertheless, they have different average slopes. Each curve gives an equally true picture of the vibration spectrum, but each of them is different from the practical point of view.

The Fig.2-7 shows a tipical vibration spectrum from a machine,expressed in terms of the three vibration parameters. It can be seen that displacement measurement give low frequency components most weight, whereas acceleration measurements are of importance at high frequency range.

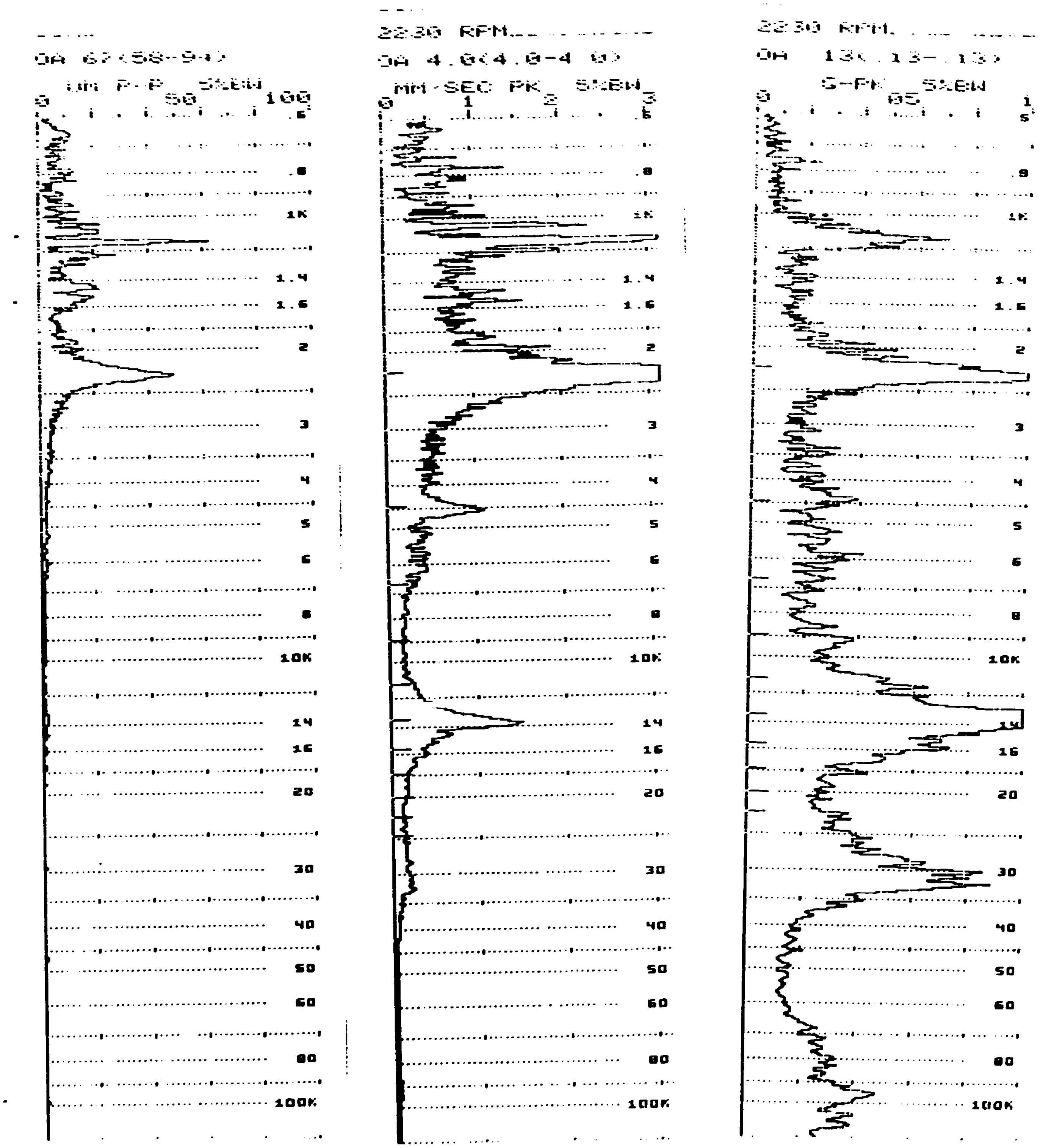


Figure 2-7. Vibration spectrums from the same machine described with displacement (left), velocity (middle) and acceleration (right)

The nature of mechanical system is such that perceptible displacements occur only at low frequencies. For example consider a slow rotating machine rotating at 60 RPM with a vibraton of .5 mm peak-to-peak displacement from rotor unbalance. In terms of vibration velocity is only .0016 mm/sec peak which would be considered as "good" for general machinery. However, keep in mind that the bearing of this machine is being deflected .5 mm. Under these conditions, failure may occur due to stress (displacement) rather than fatigue (velocity).

It is generally at low frequencies that displacement may be the best indicator of vibration severity, typically in the frequency range less than 600 CPM.

RMS velocity measurements are widely used for vibration severity. This is due to the fact that vibratory velocity is related to vibratory energy and therefore it is a measure of the destructive effect of vibration. As a rule of thumb, vibrations occuring in the 600 to 60 000 CPM (10 to 1000 Hz) frequency range are generally best measured using vibration velocity.

Acceleration is closely related to force, and relatively large forces can occur at high frequencies even though the displacement and velocity may be small. Thus, for high frequencies, typically 60 000 CPM (-1000 Hz) and up, acceleration may be the best indicator of vibration severity.

2.2.5 Information provided by phase

Phase measurements are important in vibration analysis to diagnose specific machinery faults. Comparative phase measurements are used to determine the following problems:

o Balancing - Phase is used to determine the type of unbalance (static or dynamic) and to calculate the amount and angular location of correcting weights.

- o Misalignment/Distortion - Comparative phase measurements reveal the type of misalignment (angular or offset) and the location
- o Looseness - Phase is used to detect relative movement in machine components that is due to poor grouting, broken or cracked foundations
- o Modal Studies - Comparative phase readings can reveal mode shapes in all types of machine structures.

Phase information is obtained using a stroboscopic light triggered by the vibration signal, a phase reference pickup, or an oscilloscope.

2.2.6 Vibration severity

Since vibration amplitude is a measure of the severity of the trouble in the machine, it is essential to answer to the question: How much vibration is too much ? To answer the question it is important to bear in mind that the aim of vibration checking is to detect the incipient failure and schedule an appropriate correction procedure.The real aim is to get a fair warning of impending trouble, not to determine how much vibration an equipment can withstand before is fails.

In the International Standards (see ISO 2041, ISO 2372) the term **vibration severity** defined as a comprehensive and simple characteristic unit for describing the vibratory state of an equipment.The vibration severity of a machine is defined as the **maximum root-mean-square value of the vibration velocity** measured at significant points of a machine, such as a bearing, a mounting point, etc. Acceleration, velocity and/or displacement magnitudes (a_i, v_i, s_i respectively where i=1, 2, ..., n) are determined as functions of the angular velocity ($\omega_1, \omega_2 \ldots, \omega_n$) from analyses of recorder spectra. The displacement amplitudes of the vibrations s_1,

$s_2, \ldots, s_n$ or the oscillating velocity amplitudes v_1, v_2, $\ldots v_n$ or the acceleration amplitudes $a_1, a_2, \ldots, a_n$ are know. The associated RMS-velocities characterizing the motion are given by

$$v_{RMS} = \sqrt{\left(\frac{1}{2}\right)\left[\left(\frac{a_1}{\omega_1}\right)^2 + \left(\frac{a_2}{\omega_2}\right)^2 + \cdots + \left(\frac{a_3}{\omega_3}\right)^2\right]} =$$

$$= \sqrt{\left(\frac{1}{2}\right)\left(s_1^2\omega_1^2 + s_2^2\omega_2^2 + \cdots + s_n^2\omega_n^2\right)} =$$

$$= \sqrt{\left(\frac{1}{2}\right)\left(v_1^2 + v_2^2 + \cdots + v_n^2\right)} \qquad (2.10)$$

The International Standard ISO 2372 (BS 4675/VDI 2056) defines the basis for specifying the rules to be employed in evaluating the mechanical vibration of machines in the operating range 10 to 200 rev/s and within the frequency range 10 to 1000Hz in such a way that comparison is possible with similar measurements obtained from other like machines. The vibration severity of the machine is the maximum RMS vibration level measured or calculated, using the appropriate equations (2.8) and (2.10), at the selected locations, such as bearing caps and under a specified set of operational and environmëntal condition. Recommended measurings positions are illustrated in Fig.2-8.

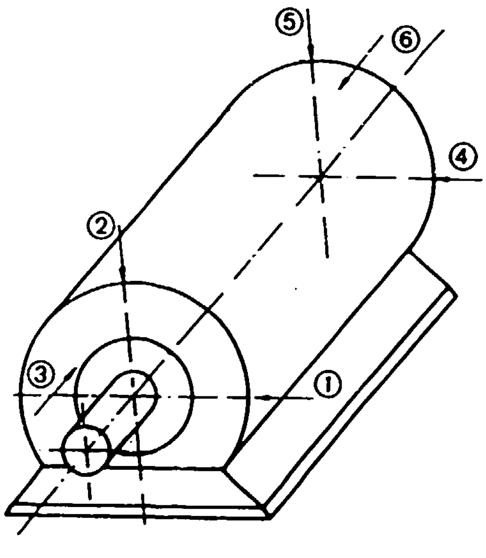


Figure 2-8. Vibration sev ity measuring positions

Based on experience, vibrations with the same RMS velocity anywhere in the frequency band 10 to 1000 Hz are generally considered to be of equal severity. Succeding ranges of the

evaluation classification should have a ratio of 1 : 1,6, giving a step of 4 dB between severity levels. A difference of an RMS ratio of 1:1,6 yields a velocity change (increase or decrease) which represents a significant change in the vibratory response in most industrial machines.
This permits the construction of a general scale similar to that of table which is independent of and not restricted to a particular group of machines. From this it follows that the term "vibration severity" may be used in such a way that it does depend on individual judgement factors; it is, in effect, an independent parameter which may be used to construct any required evaluation classification.

Table 6

Vibration severity ranges (10 to 1000 Hz)

Range	Velocity range (RMS)			
	mm/s		in/s	
	over	up to	over	up to
0,11	0,071	0,112	0,0028	0,0044
0,18	1,112	0,18	0,0044	0,0071
0,28	0,18	0,28	0,0071	0,0110
0,45	0,28	0,45	0,0110	0,0177
0,71	0,45	0,71	0,0177	0,0280
1,12	0,71	1,12	0,0280	0,0441
1,8	1,12	1,8	0,0441	0,0709
2,8	1,8	2,8	0,0709	0,1102
4,5	2,8	4,5	0,1102	0,1772
7,1	4,5	7,1	0,1772	0,2795
11,2	7,1	11,2	0,2795	0,4409
18	11,2	18	0,4409	0,7087
28	18	28	0,7087	1,1024
45	28	45	1,1024	1,7716
71	45	71	1,7716	2,7953

At present, experience suggests that the following classes of machines are appropriate for most applications:

Class I (in German practice: Group K) - Small machines
Individual parts of engines and machines, integrally connected with the complete machine (typically electrical motors up to 15 kW)
Class II (Group M) - Medium machines
Medium-sized machines (typically electrical motors with 15 to 75 kW output) without special foundation, rigidly mounted engines or machines (up to 300 kW) on special foundation
Class III (Group G) - Large machines
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 measurement
Class IV (Group T) - Turbo machines
Large prime movers and other large machines with rotating masses mounted on foundation which are relatively soft in the direction of vibration measurement (e.g. turbogenerator sets).
A suggested order of quality judgment is shown is Fig. 2-9.

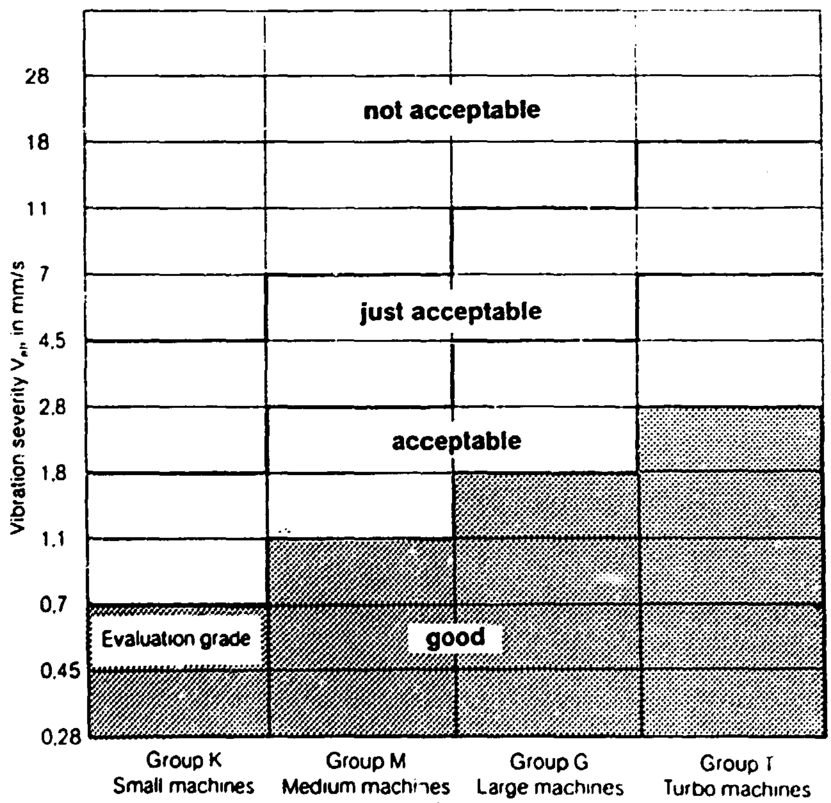


Figure 2-9. Vibration severity ranges and examples of quality judgement for separate classes of machines

The RMS-value of velocity in the 10 to 1000 Hz range is a commonly used parameter in many standards; however, in some cases, it is important to know instead the displacement amplitudes of dominant components observed in measured vibration spectra. These have been used in certain older criteria (see Fig. 2-10) and, for this purpose, it is necessary that RMS-velocity values be converted to peak displacement amplitudes.

The operation of converting vibration velocity to displacement values can be accomplished only for single-frequency harmonic conponents. If the vibration velocity of such a component is known, the peak-displacement amplitude may be computed from the relationship

$$s_f = \frac{v_f}{\omega_f}\sqrt{2} = \frac{v_f}{2\pi f}\sqrt{2} = 0{,}225\frac{v_f}{f} \qquad (2.11)$$

where s_f is the peak-displacement amplitude

v_f is the RMS-value of the vibration velocity at the frequency f

Example: A given vibration measurement has the severity of 4 mm/s, that is, the maximum RMS vibration velocity over the the range from 10 to 1000 Hz does not exceed 4 mm/s. A spectrum analysis has disclosed that the dominant frequency component occurs at 25 Hz with a RMS vibration amplitude of 2,8 mm/s. Thus, the peak amplitud is:

$$s_f = 0{,}225 \left(\frac{2{,}8}{25} \right) = 0{,}025 \text{ mm or } 25\ \mu\text{m}.$$

ISO 2372 establishes a basis for the evaluation of mechanical vibration of machines by measuring the vibration response on stationary structural members such as on the bearing caps of bearing housing. ISO 7919 provides an alternative method for evaluation of machinery shaft vibration. There are many type of machines containing flexible rotor shaft systems, for which changes in the vibration condition may be detected more decisively and more sensitively by measurements on the rotating element. Such machines are e.g. industrial steam turbines, gas turbines and turbo-compressor.

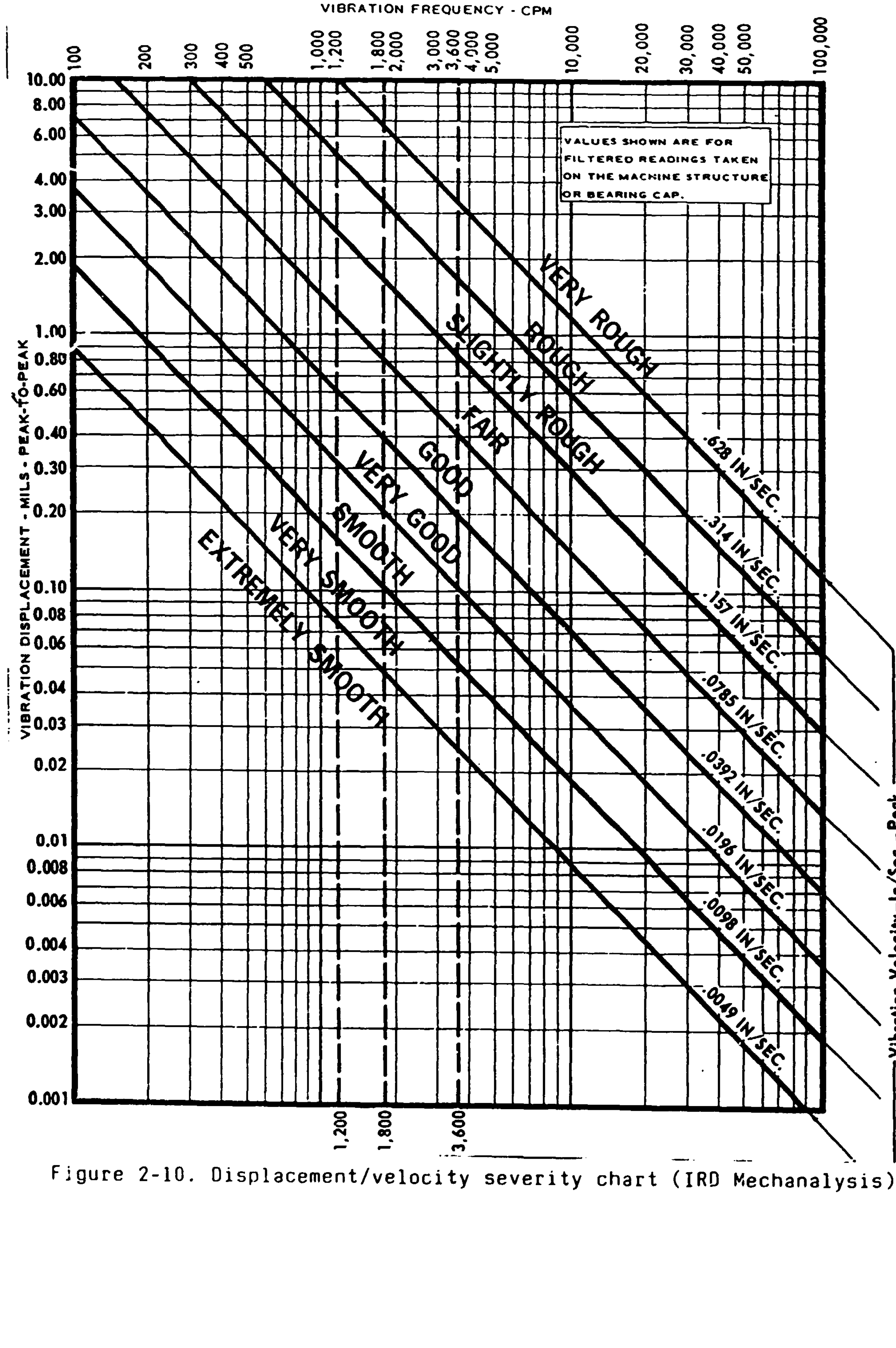


Figure 2-10. Displacement/velocity severity chart (IRD Mechanalysis)

CHAPTER 3 VIBRATION MEASURING TECHNIQUES AND INSTRUMENTS

3.1. GENERAL CONSIDERATION

Vibration measuring instruments and means are generally classified as meters, monitors and analysers.

Fig. 3-1 shows a portable, compact, battery operated instrument being used for measuring and helping eliminate vibration problems in operational machines and equipment. Vibration analyzer shown in Fig. 3-2 can perform the functions both of periodic measurements and complex fault diagnosis. Beside this is suitable for the machine off-line condition monitoring. Analyzer may also include a strobe light used for phase analysis and dynamic balancing and microprocessor-based data processing unit for spectral and harmonic analysis.

Figure 3-1. Vibration meter (Schenk)

Monitors, such as those illustrated in Fig.3-3 are vibration meters permanently employed on a specific machine and continuously surveyes its condition.

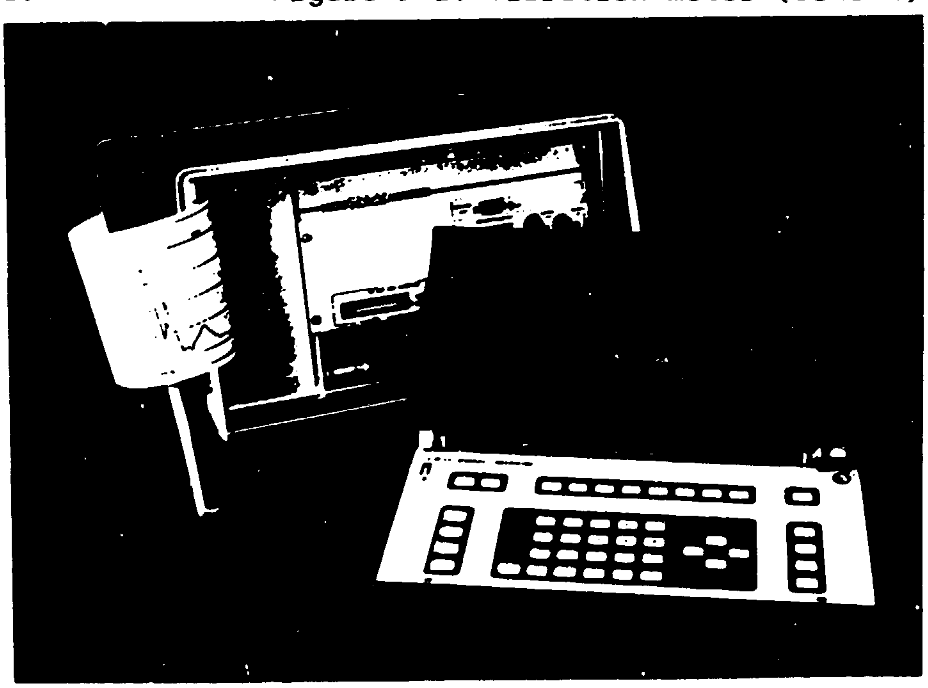


Figure 3-2. Vibration analysis (Vibro-Meter)

Figure 3-3. Multichanel vibration monitor (Schenck)

All vibration meters, monitors and analyzers use a **vibration transducer** actuated by energy from a mechanical system (strain, force, motion, etc.) converting this mechanical parameter into a proportional electrical signal. The transducer is often referred to as a vibration pickup or sensor. There is a variety of vibration transducers available. Following we will examine the principal types and determine:

- o What types of measurements are generally used.
- o How the transducers operate.
- o For what specific purpose each is used.
- o How they are mounted.

No single type of transducer can meet all the measurement requirements for up-to-date vibration detection and diagnosis. This is why the maintenance specialist must be prepared to select a transducer according to the given measurement task.

3.2. TYPES OF VIBRATION MEASUREMENT

3.2.1. Relative vibration measurements

Relative vibration motions are measured relative to a chosen stationary reference, usually a structural member (e.g. the bearing housing) of the machine. Relative vibration measurements are generally carried out with a non-contacting (proximity) transducer which senses the vibration displacement in μm between the shaft and the stationery transducer mounting.

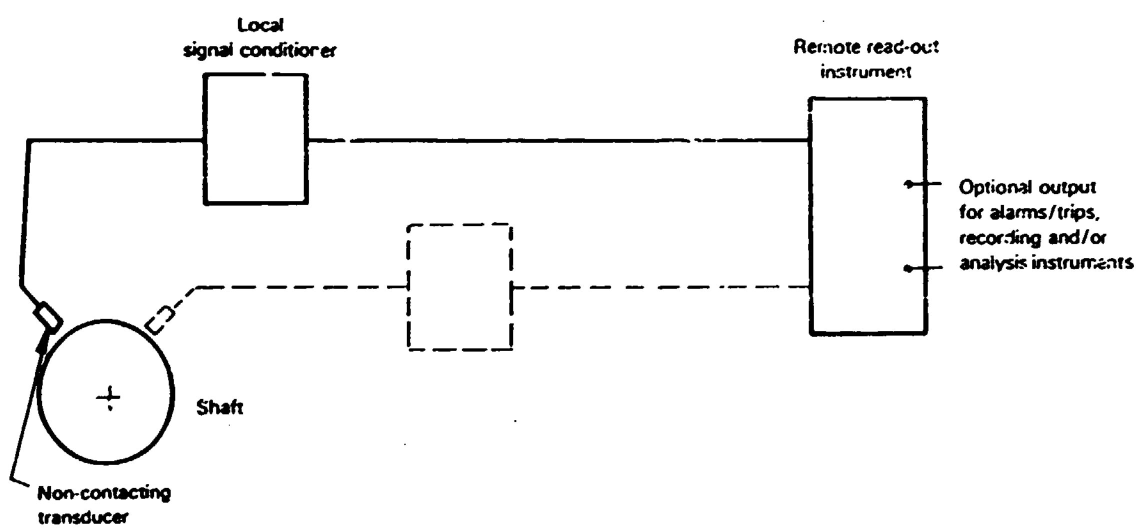


Figure 3-4. Schematic diagram of relative motion measurement system using non-contacting transducers

Relative vibration transducers of the non-contacting type are normally mounted in tapped holes in the bearing housing, or by rigid brackets adjacent to the bearing housing. Where the transducers are mounted in the bearing, they should be located so as not to interference with the lubrication pressure wedge. The surface of the shaft at the location of the pickup shall be smooth and free from any geometric discontinuities (e.g. keyways,lubrication passages,metallurgical non-homogenities and/or local residual magnetism which may cause false signals. In some circumstances an electroplated or metallized shaft surface may be acceptable but it should be noted that the calibration may different.

3.2.2. Absolute vibration measurements

Absolute vibration motions are measured relative to an inertial (fixed) reference frame,vibration with respect to a fixed point in space. Absolute vibration measurements are carried out by one of the following methods:

(i) by a <u>shaft-riding transducer</u>, on which a <u>seismic pickup</u> (velocity type or accelerometer) is mounted so that it measures absolute shaft vibration directly (Fig. 3-5); or

(ii) by a <u>non-contacting transducer</u> in combination with a seismic transducer, mounted close together so that the support structure of both transducers undergoes the same absolute motion in the direction of measurement (Fig. 3-6.). Their conditioned outputs are vectorially summed to provide a measurement of the absolute shaft motion.

The seismic transducer (velocity type or accelerometer) shall be mounted radially on the shaft-riding mechanism. The mechanism shall not chatter or bind in a manner modifying the indicated shaft vibrations.

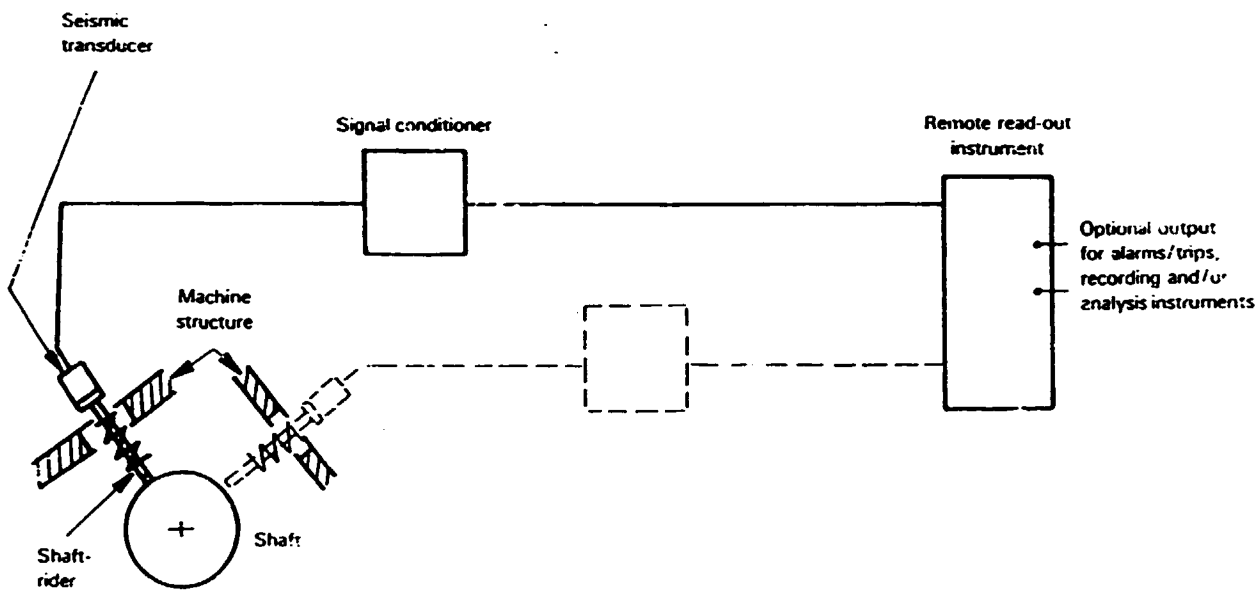


Figure 3-5. Schematic diagram of absolute motion measurement system using shaft-rider mechanism with seismic transducers

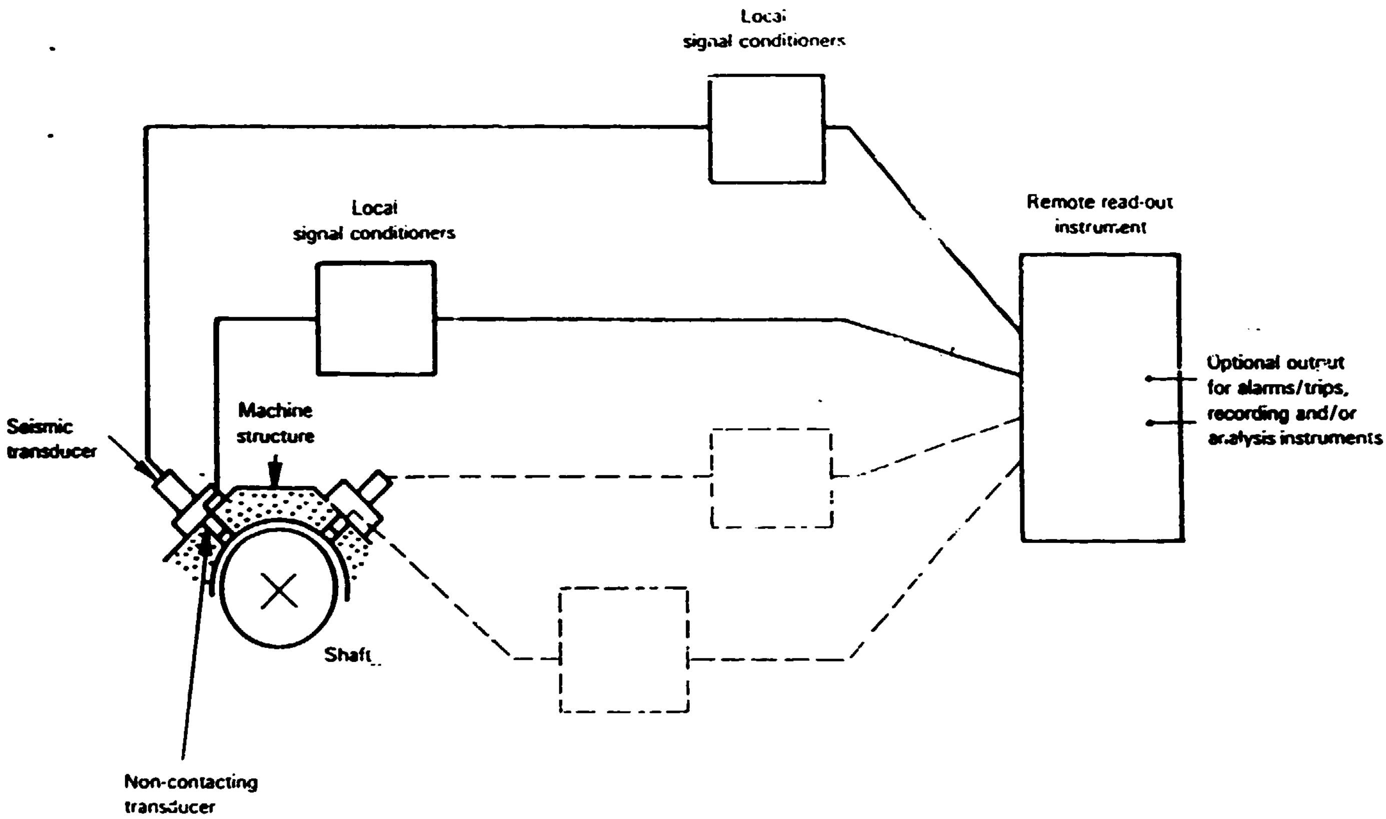


Figure 3-6. Absolute/relative motion measurement system using a combination of non-contacting and seismic transducers

3.2.3. Kinetic orbit of shaft

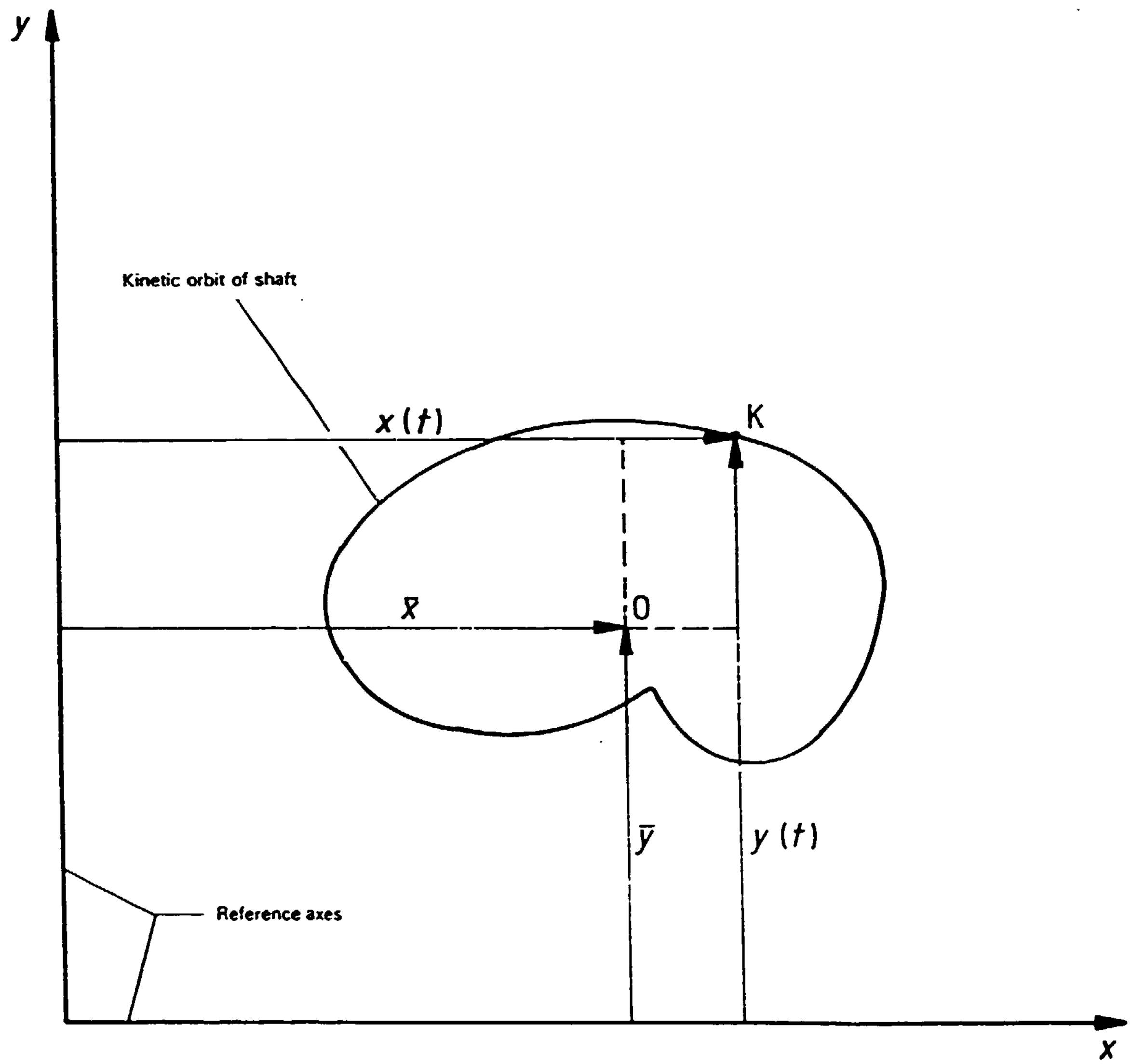
A vibration of a rotating shaft is characterized at any axial location by a **kinetic orbit**, which describes how the position of the shaft centre varies with time. Figure 3-7 shows a typical orbit. The shape of the orbit depends upon the dynamic characteristics of the shaft, the bearings and the bearing supports, the axial location on the rotor and the form of vibration excitation. One of the most important excitation forces is rotor unbalance, in which the excitation frequency is equal to the rotational frequency of the shaft. At any axial location, the orbit of the shaft can be obtained by taking measurements with two vibration transducers mounted in different radial planes, separated by 90^0.Derivation of measurement quantities is described in International Standard ISO 7919, Mechanical vibration of non - reciprocating machines - Measurement on rotating shafts and evaluation.

The mean values of the shaft displacement ($\bar{x}$, $\bar{y}$) in any two specified orthogonal directions,relative to a reference position, as shown in Fig. 3-7, are defined by integrals with respect to time by

$$\bar{x} = \frac{1}{t_2 - t_1} \int_{t_1}^{t_2} x\,(t)\,dt \qquad (3.1.)$$

$$\bar{y} = \frac{1}{t_2 - t_1} \int_{t_1}^{t_2} y\,(t)\,dt \qquad (3.2.)$$

where x(t) and y(t) are the time-dependent alternating values of displacement relative to the reference position, and $(t_2 - t_1)$ is large relative to the period of the lowest frequency vibration component. In case of absolute vibration measurements, the reference position is fixed in space. In case of relative vibration measurements these values give an indication of the mean position of the shaft relative to the non-moving parts.



0 Mean position of orbit

K Instantaneous position of shaft centre

$\bar{x}$, $\bar{y}$ } Mean values of shaft displacement

$x(t)$, $y(t)$ } Time-dependent alternating values of shaft displacement

Figure 3-7. Kinetic orbit of shaft

Changes in the values may be due to a number of factors, such as bearing, foundation movements, changes in oil film characteristics, etc., which nomally occur slowly relative to the period of the vibration components which make up the alternating values.

The instantaneous value of shaft displacement can be defined by S_1, as shown in Fig. 3-8, which is derived from the transducer measurements S_{A_1} and S_{B_1} using equation (3.3):

$$S_1^2 = S_{A_1}^2 + S_{B_2}^2 \qquad (3.3)$$

There is a point on the orbit where the displacement from the mean position is a maximum. This S_{max} value is defined by

$$S_{max} = \left[S_1(t)\right]_{max} = \sqrt{S_A(t)^{\ 2} + S_B(t)^{\ 2}}_{\ max} \qquad (3.4)$$

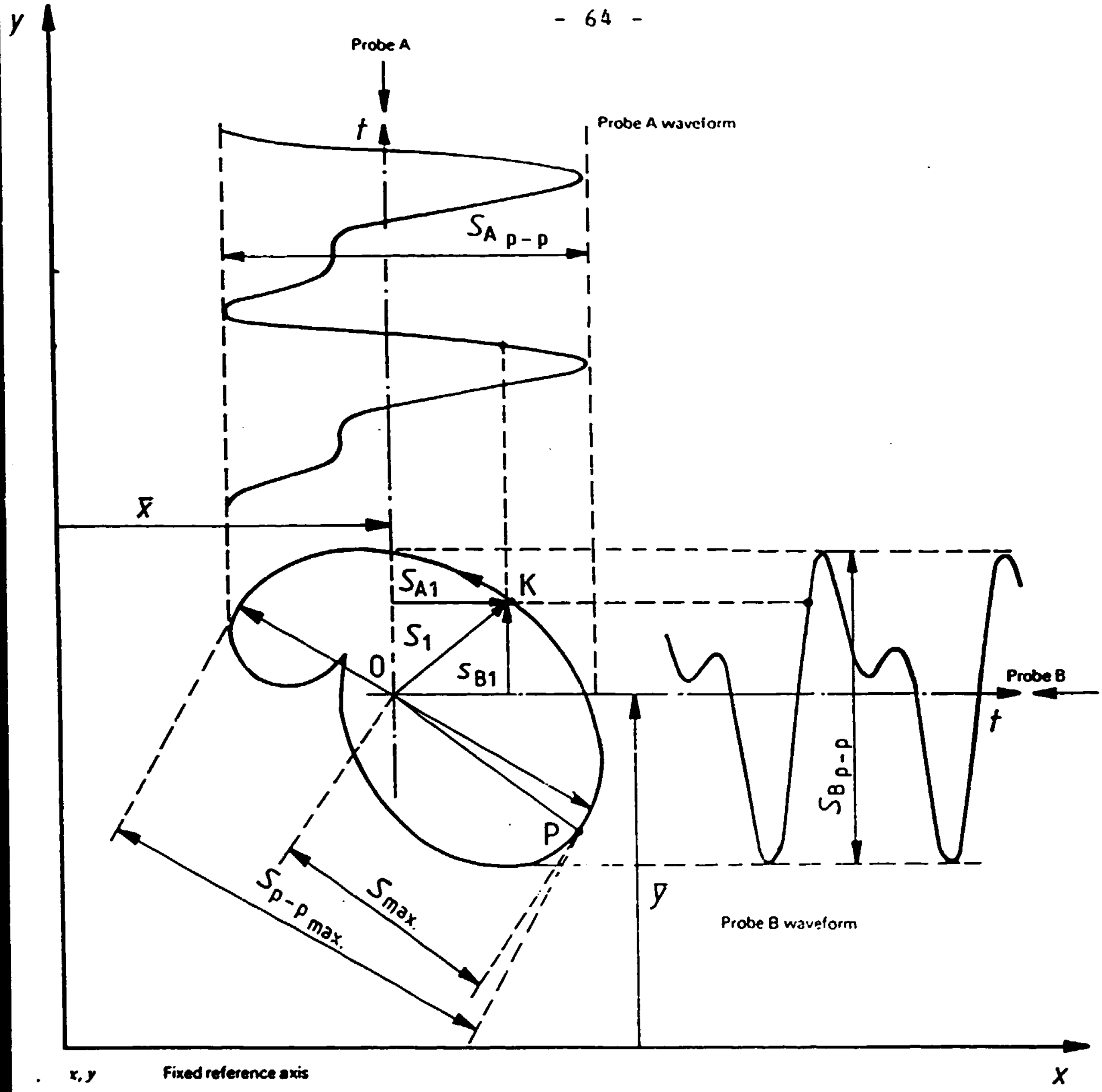
The value of the peak-to-peak displacement values can be approximated from the equation:

$$S_{p-p_{max}} = 2S_{max} \qquad (3.5)$$

Eq. (3.5.) will be correct when the two orthogonal measurements from which S_{max} is derived are of single frequency sinusoidal form.

3.3 VIBRATION TRANSDUCERS

To measure machinery or structural vibration, a transducer or vibration pickup is needed to convert mechanical motion into a proportional electrical (typically a voltage-proportional) signal. The transducers commonly used are non-contact (proximity) transducers, velocity pickups and accelerometers. Each transducer has distinct advantages in certain applications, but they all have also limitations.



x, y	Fixed reference axis
0	Time-integrated mean position of orbit
$\bar{x}, \bar{y}$	Time-integrated mean values of shaft displacement
K	Instantaneous position of shaft centre
P	Position of shaft for maximum displacement from time-integrated mean position
S_1	Instantaneous value of shaft displacement
S_{max}	Maximum value of shaft displacement from time-integrated mean position 0
S_{A1}, S_{B1}	Instantaneous values of shaft displacement in directions A a d B
$S_{p\text{-}p_{max}}$	Maximum value of peak-to-peak displacement
$S_{A_{p\text{-}p}}$, $S_{B_{p\text{-}p}}$	Peak-to-peak values of shaft displacement in directions A and B

Figure 3-8. Definition of displacement on the kinetic orbit of shaft

3.3.1. Non-contact (proximity) transducers

The most commonly used non-contact device is a promixity probe transducer operating on the **eddy-current principle.** The eddy-current displacement probe (see Fig. 3-9) contains a small coil of fine wire at its tip which is excited by a remote RF-oscillator to generate a magnetic field. As the tip of the probe is brought close to the conductive surface of the moving machine element (such as rotating shaft), eddy-current induced in the conductive material by the probe's magnetic field oppose the field and reduce the amplitude of the carrier signal by an amount proportional to the change in the distance (proximity). This distance is referred to as the gap. A demodulator, usually encapsulated in the same enclosure as the oscillator, converts the change in carrier amplitude to a low-impedance voltage output.

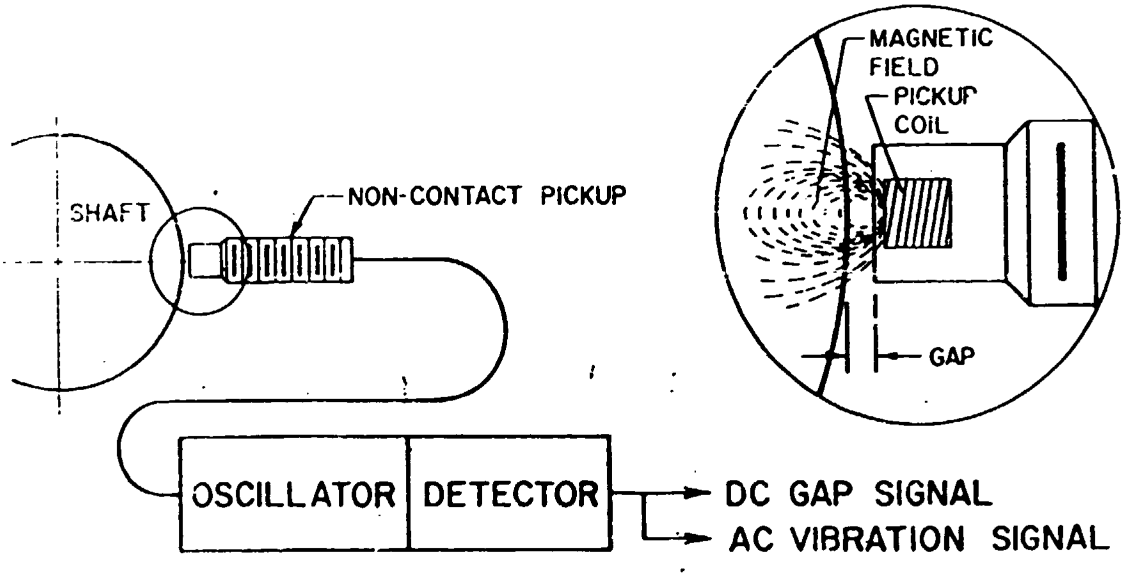


Figure 3-9. Non-contact eddy-current pickup system

An eddy-current displacement sensor and its oscillator-demodulator constitute a gap-to-voltage measuring system. The relationship of output voltage to gap approximates a straight line. The slope of this line expressed in mV/μm is the sensitivity of the non-contact pickup system. A typical linear amplitude range is 1 to 2 mm with a frequency response capability from static to more than 2000 Hz. The sensitivity changes for different target materials and with changes in cable length. A typical sensitivity of the today proximity pickups is 8 mV/μm over 2 mm gap range at 20°C if calibrated as a system (Fig. 3-10.)

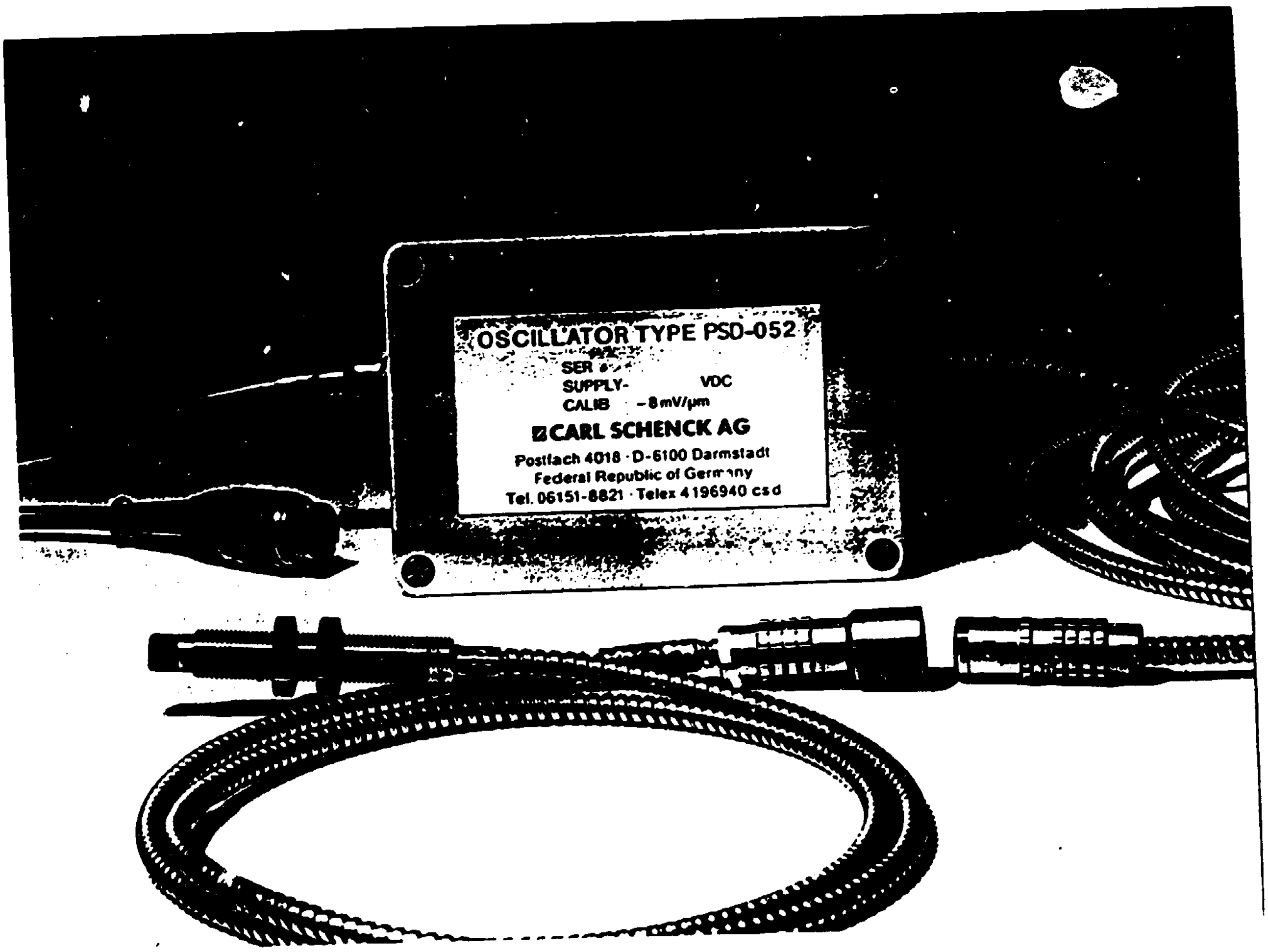


Figure 3-10. Non-contact pickup type PSD-052 (Carl Schenck AG)

3.3.2 Velocity transducers

Velocity pickups or transducers measure absolute vibration, relative to a fixed point in space. A tipical moving coil type velocity sensor (see Fig. 3-11) consists of a seismically mounted and usually critically damped, permanent magnetic core suspended in a housing rigidly attached to a vibrating surface. A wire coil attached to the housing surrounds the core. Relative motion between the magnetic core and the housing causes magnetic lines of flux to cut the coil, inducing a voltage proportional to the velocity. The voltage generated is transmitted by cable to a vibration processing device (meter, monitor or analyzer). These sensors operate above their first natural frequency between a frequency range from 10 Hz up to 2000 Hz. Care must be taken in using this transducers in strong magnetic environments.

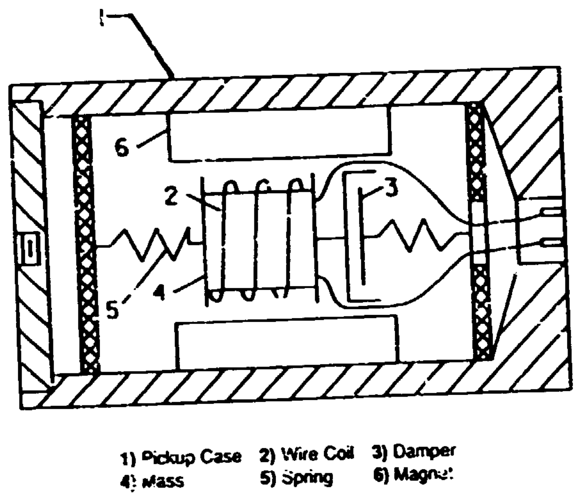


Figure 3-11 Seismic velocity transducer

Many times it is necessary to measure the vibration of a small, lightweight part or structure. The principle of operation of a direct-prod pickup shown Fig.3-12 is identical to that of a seismic velocity sensor. The thin prod extends through the endcap of the pickup and is attached directly to the moveable coil inside. Since the pickup coil

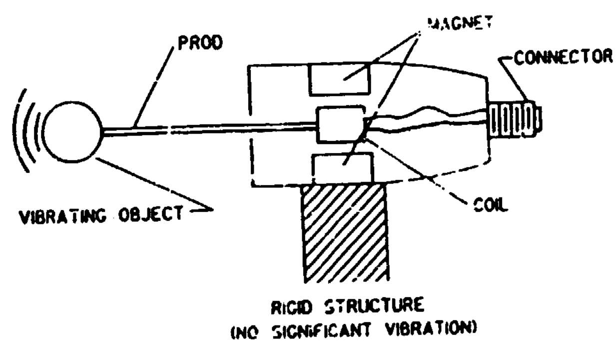


Figure 3-12 Direct-prod pickup

is moved directly by the prod, while the pickup housing remains stationary, the sensitivity of this type of pickup does not drop of at low frequencies (below 10 Hz).
Figure 3-13 shows the general specifications for some basic vibration transducers of different producers.

Model	Application	Frequency Range	Sensitivity	Temperature Range
IRD Model 544 vibration pickup	Manual and continuous monitoring	10 Hz...1000 Hz	42.5 mV/mm/s	-40^{0}C to 260^{0}C
IRD Model 546 direct prod	Vibration measurement of light-weight parts	0.3Hz..2500 Hz	42.5 mV/mm/s	-40^{0}C to 120^{0}C
IRD Model 560 seizmic pickup	General industrial applications	1 Hz...4500 Hz	42.5 mV/mm/s	-40^{0}C to 120^{0}C
Schenck VS-080	Manual measurements	1 Hz...2000 Hz	75 mV/mm/s	-40^{0}C to 100^{0}C
Schenck VS-068/069	General industrial applications	1 Hz...1000 Hz	100 mV/mm/s	-40^{0}C to 100^{0}C
Schenck VS-168/169	E Ex dIIC T6 application	1 Hz...1000 Hz	100 mV/mm/s	-15^{0}C to 65^{0}C
Bently Nevada Seismoprobe transducers	General industrial applications	1 Hz...1000 Hz	20 mV/mm/s	-29^{0}C to 240^{0}C

Figure 3-13. Specifications of industrial vibration transducers

The **seismic piezoelectric velocity transducer** uses a different principle of operation (section 3.3.3). The electrical charge from the piezoelectric element is so small that the signal must be amplified before it can be measured. Piezoelectric velocity pickups operate between a wide range of frequency down to 1 Hz to 4.5 kHz.

3.3.3. Accelerometers

The most common type of vibration sensor is the accelerometer. An accelerometer is a seismic transducer which converts acceleration motion and/or granvitational forces capable of imparting acceleration into a proportional electric signal. Both self-generating acelerometers and those requiring electrical excitation are available.

Fig. 3-14. illustrates a model of accelerometer. The active elements of the accelerometer are the piezoelectric element. These act as springs connecting the base of the accelerometer to the seismic masses via the rigid triangular centre post. When a varying motion is applied to the accelerometer, the piezoelectric element experiences a varying force excitation, causing a proportional electrical charge to be developed across it. The seismic masses are constant and consequently the elements produce a charge which is proportional to the acceleration of the seismic masses.

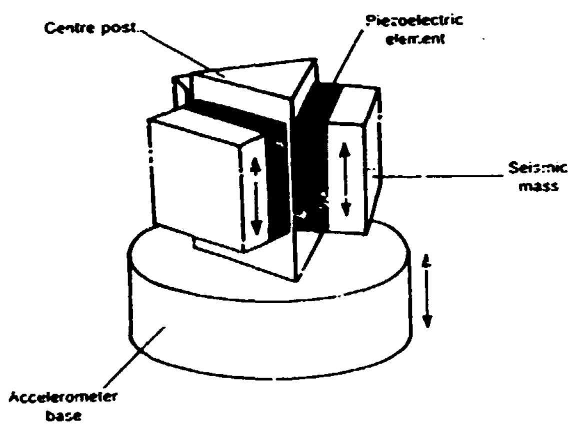


Figure 3-14 Piezoelectric accelerometer (Brüel & Kjaer, type Delta Shear)

It is an extremely linear relationship between the applied force and the developed charge, over a very wide dynamic and frequency range, which results in an excellent characteristics of the piezoelectric accelerometer.The sensitivity of a piezoelectric material is given in pC/N (in terms of picocoulombs per Newton) or pC/g. The commonly used piezoelectrical materials are lead-zirconate titanate ceramics or monocrystalline materials such as quartz.

The mechanical spring constants for the piezoelectric elements are high, and the inertial masses attached to them are small. Therefore,these accelerometers are useful to extremely high frequencies (generally up to the resonant frequency of the transducer which is in the most cases more than 25 kHz).

Because the electrical charge generated directly within a piezoelectric sensor is so small, many commercial models have a high-gain electronic preamplifier built into them.

Two principal design configurations are used for piezoelectric accelerometers (Fig. 3-15). One stresses the piezoelectric materials in **compression** while the other stresses in **shear**. In the compression accelerometer vibration varies the stress in the sensor (piezoelectric crystal) which is held in compression by the preload element. In the shear design vibration simply deforms the crystal in shear.

Accelerometers are not affected by external magnetic fields but are more susceptible to ground loops, radio frequency interference and dirty or poor contacts. Cable length effects may reduce sensitivity.

Figure 3-16 shows the general specifications for some basic industrial accelerometers of different producers.

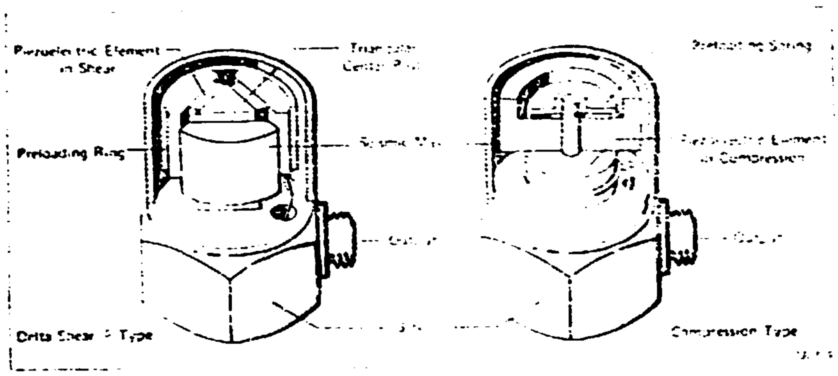


Figure 3-15. The two accelerometer configurations in common use (Brüel & Kjaer)

Model	Application	Frequency Range	Sensitivity	Temperature Range
IRD Model 960	Monitoring	2 Hz...10 000 Hz	50 mV/g	-40°C to 150°C
IRD Model 970	Monual measurements	5 Hz...10 000 Hz	50 mV/g	-40°C to 120°C
Schenck AS-011	Manual and industrial measurements	1 Hz...15 000 Hz	100 mV/g	-50° to 121°C
ASA-011	E Ex ib II C T/6/5/4			
B & K Type 4370 Delta Shear	General purpose accelerometer	0.2 Hz...6000 Hz	85 mV/g	max. 250°C
Vibro-Meter types CA...	High temperature applications	up to 20 kHz	10 pC/g	max. 650°C

Figure 3-16. Specifications of tipical industrial accelerometers

3.4 SPECIAL CONSIDERATIONS FOR VIBRATION MEASUREMENTS

As described in 3.3 considerable attention has been given to the various non-contact, velocity and accelerometer transducers used to measure machinery vibration.

The following considerations are of importance in determining wheather displacement, velocity or acceleration to be chosen for measurements:

Displacement measurements may be useful:

- o Where the amplitude of displacement is particularly important.
- o At low frequencies where corresponding velocity and acceleration measurements may yield outputs which are too small for practical use.

Velocity measurements may be useful:

- o At intermediate frequencies when displacement amplitudes are too small to measure conveniently.
- o In correlating acoustic and vibration measurements,because a vibrating component may produce sound pressure in air which is proportional to velocity.

Acceleration measurements may be useful:

- o Where forces, loads and stresses must be analyzed, since force is proportional to acceleration
- o At high frequencies, where the highest signal output usually can be obtained from such measurements.
- o Where suitable displacement or velocity sensors would be too large because of clearance requirements.

However, in addition to selecting the most suitable transducer,there are a number of additional factors to consider when measuring vibration. The most important considerations include

(i) effects of sensor mounting

(ii) electrical grounding problems

(iii) hazardous/explosion area considerations.

3.4.1 Effects of sensor mounting

The basic problem in designing a mounting is to couple the transducer to the equipment under test so that the pickup accurately follows the motion of the surface of a specimen to which it is attached. There are several methods which can be used to apply the vibration transducer, however, the method used may have an effect on the results. The main requirement is that the effective stiffness of the sensor mounting be large in the frequency range of interest; otherwise the mounting will deflect under the inertia load of the transducer mass. In order to design a transducer mounting properly, one must know the nature of its use, the frequency range and the maximum acceleration of the measurements and the mechanical specification of the specimen. Examples of the effects of various types of mountings on the frequency response of the transducer are shown in Figure 3-17.

Transducer mounting	Maximum allowable frequency
Velocity transducer Model 544 (IRD)	
Hand-held 9" probe	120 Hz
Magnetic holder	270 Hz
Hand-held without probe	1000 Hz
Threaded stud / Bolting	1500 Hz
Accelerometer Model 910 (IRD)	
Magnetic holder	2000 Hz
Hand-held without probe	7000 Hz
Treaded insulated stud	7000 Hz
Threaded stud / Bolting	10000 Hz

Figure 3-17. Frequency limits for different transducers and mounting methods

In order to effectively use standard transducers in widely differing mach'ne applications,a variaty of accessory equipment is available.Figure 3-18.shows i.e. the Bently Nevada's effective Quick Connect and Super Mag 100 mounting devices.

Quick Connect, shown with velocity transducer attached

Super Mag 100

Figure 3-18.

Transducer accessories for periodic vibration measurements (Bently Nevada)

3.4.2. Electrical grounding problems

Wiring installation should be designed to minimize the induced noise. The induced noise results when electrical energy from an electrical equipment is coupled into the measurement circuits by one or more of the following reasons:

- ground loops
- electromagnetic fields
- resistive leakage paths.

The induced noise can introduce fals signals which could be misinterpreted as vibration. Noise introduced by these effects can be limited by application of the following guideline:

(i) Observe proper machine and instrument electrical grounding practice.

(ii) Use separate signal and power current return leads to reduce ground-loop coupling between signal and power circuits.

(iii) Use twisted pair or coaxial cable to reduce inductive pickup in signal leads.

(iv) Use coaxial cable and/or shielding to reduce coupling between circuits.

(v) Use insulated mounting studs to insulate the pickup case from the specimen.

(vi) Use a "floating ground" installation.

Proper grounding of accelerometer installation to avoid problems of electrical noise susceptibility is shown in Figure 3-19.

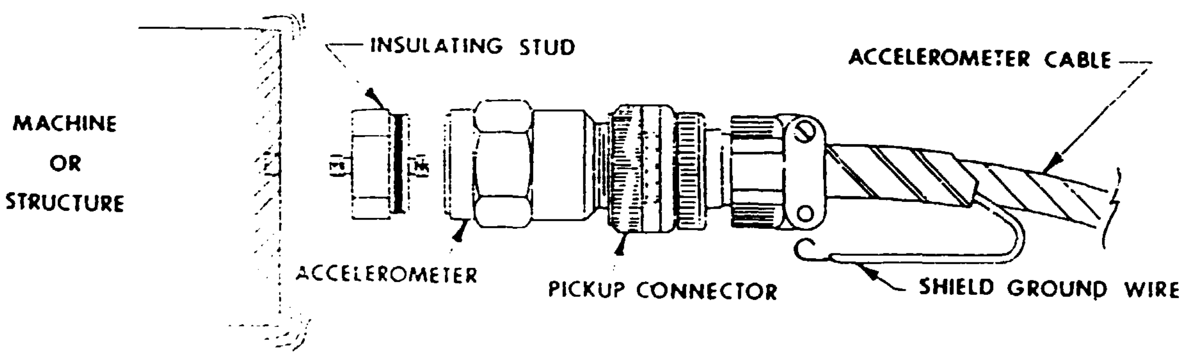


Figure 3-19. Accelerometer installation with insulated stud and shield ground wire

3.4.3 Hazardous area considerations

Chemical,petrochemical, oil and natural gas industries, etc. have strict safety regulations allowing the use of electronic instruments in the plant. Safety regulations for hazardous locations prohibit the use of any electrical equipment which may ignite hazardous materials. This should be assumed to apply to all vibration meters,signal conditioners, measuring and monitoring equipment, unless the instrument is specifically designed for hazardous service.

Whether or not an electrical/electronic instrument can be used in a designated area of a plant is usually determined by the hazardous location classification certified to international standards (National Electrical Code - NEC -, International Electrotechnical Commission, British Standards, CENELEC, etc.).

It must be emphasized, however, that although there are methods to use of electrical equipment in a hazardous environment which are generally applicable, **the final approval to permit operation of any electrical equipment in any hazardous location must be granted by the user's approving authority.**

The generally applicable methods with hazardous area are:

- o **Explosion-proof enclosure**

  The equipment is heavily armored so that if an explosion occurs, it is contained within the equipment casing. This enclosure can also be used for mechanical protection and as a weather-proof casing. Note, that all wiring must be enclosed in sealed, rigid conduit.

- o **Intrinsic safety**

  The concept of intrinsic safety is based on preventing the explosion alltogether. To meet these requirements the circuitry of the electrical equipment must be designed so that it will limit energy within the hazardous area to a level below that which could cause an explosion. The intrinsically safe installation generally use an intrinsically safe power supply unit providing a galvanically separation of the signal from the mains and an intrinsic safety barrier.

- o **Purging**

  Purging is a process of flushing the air in an electrical/electronic instrument to remove any explosive gases which may have accumulated. A supply of compressed,clean dry air or nitrogen is needed.

CHAPTER 4
VIBRATION ANALYSIS

4.1. INTRODUCTION

The measurement of the vibration severity gives information on the fact whether the vibrational behaviour of a machine exceeds the admissible values. But the measured results do not give automatically any information on the composition or the causes of the problem. This chapter describes the procedure for pinpointing the causes of vibrations.
Diagnosing machinery vibration consits of two basic scopes of tasks

o checking and evaluating of machine condition
o fault diagnosis.

There are many ways to obtain data interpretation. Some of the more common techniques include:

o Amplitude vs. measurements (overall)
o Amplitude vs. frequency measurements
o Amplitude vs. time measurements
o Amplitude vs. frequency vs. time measurements
o Phase (relative motion) analysis
o Amplitude vs. phase (Nyquist plots) measurements
o Lissajous pattern (orbits) setup
o Mode shape determination.

The main purpose of fault diagnosis is to determine the steps necessary for correcting the vibration condition to reduce the level of unwanted forces of vibration. So while studying the data,one should be mainly interested in identifying the dominant vibration magnitudes, determining their cause and correcting the problem they represent. A review of machine characteristics can be very helpful in establishing the frequencies of vibration which can be expected.This will help to determine the type of instrumentation and vibration transducers needed to do the diagnosis.

4.2. FAULT DIAGNOSIS

The more general method which can be applied to the fault diagnosis of all machines in a plant to determine the cause of a vibration once the existence of a trouble has been detected, is **frequency analysis of the vibration signal.** Here the signal is scanned over a wide range of frequencies to look for those which contain a substantial level of signal. The process is analogues to turning the tuning knob on a radio to discover where there are signals being transmitted. The conventional way of performing of a frequency analysis is to pass the measured signal through a filter to separate waves on the basis of their frequency. It introduces relatively small insertion loss to waves in one or more frequency bands and relatively large insertion loss to waves of other frequencies.
The basic concept in frequency analysis is that of the spectrum (see 2.2.2).This can be represented as a graph,plotting i.e.the amplitude or the initial phase of a vectors vs. their frequency.

4.2.1. Filters and filter characteristics

In order to analyze a signal which contains any frequencies, this signal is passed through filters which ideally transmit with zero attenuation a particular frequency range whilst rejecting all other frequencies wiht infinite attenuation. This ideal filter is shown in Figure 4-1. The lower and upper limiting frequencies are f_l and f_u respectively, so that the **bandwidth** B is defined as

$$B=f_u-f_l \qquad (4.1)$$

For convenience of reference we often use the geometric centre frequency, f_c:

$$f_c=(f_l.f_u)^{1/2} \qquad (4.2)$$

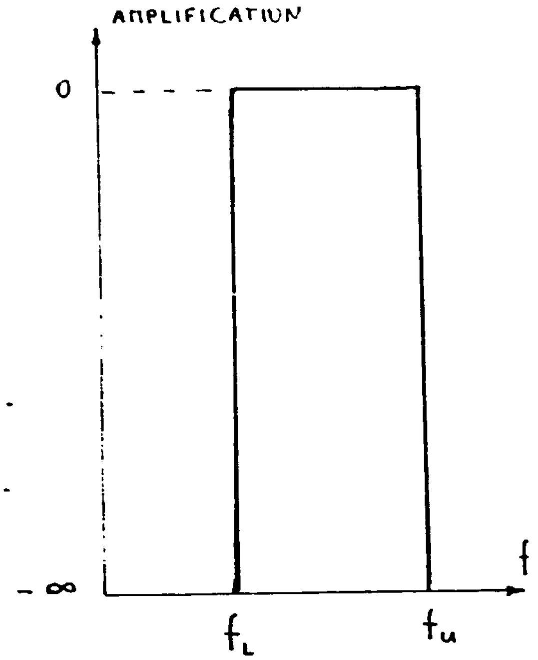


Figure 4-1. Ideal filter characteristic

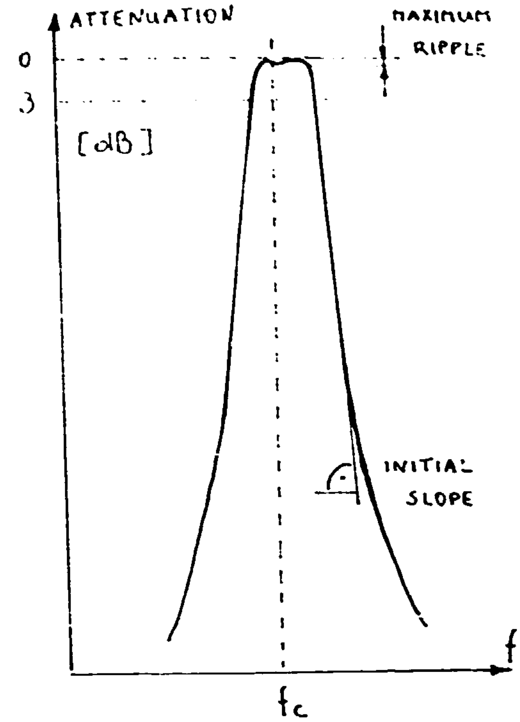


Figure 4-2. Practical filter characteristic

Unfortunately, it is not possible the design an ideal filter and so an approximation has to be made in practise.A typical practical filter response curve is illustrated in Figure 4-2. It can be seen that there is ripple at the peak and the sides are no longer vertical. The definition of the bandwidth means a spacing between frequencies at which a band-pass filter attenuates the signal by 3 dB from the nominal transmission level.

The bandwidth of a filter gives information as to its ability to separate spectrum components of approximately the same level.The selectivity indicates its ability to separate components of widely different levels.

There are two main types of filter in common use: the **constant bandwidth filter** and the **constant percentage bandwidth filter**.

At the **constant bandwidth filter** the bandwidth is constant and independent from the frequency (f_u / f_l = constant). Constant bandwidth gives uniform resolution on a linear frequency range, which provides also uniform resolution for harmonically related components. One example of this would be a 50 Hz bandwidth filter.

The **constant percentage bandwidth filter** has a bandwidth proportional to the centre frequency of the pass band, i.e.

$$B = \frac{n.f_c}{100} \qquad (4.3)$$

so that the bandwidth B is proportional to its centre frequency f_c and in the case of i.e. a 10 % bandwidth filter the bandwidth is 10 % of the centre frequency. In the practical use for vibration measurements the **Octave** and the **Third Octave filters** are important. The Octave has been inherited from the field of acoustics (=music) where it is an interval of eight notes on the diatonic musical scale. This corresponds to a doubling in frequency, and the term is used in frequency analysis to the same effect. It can be seen that the octave filter is a constant percentage bandwidth filter since

$$f_{u_i} = 2\ f_{l_i} \qquad (4.4)$$

and therefore $f_c = \sqrt{2}_{l_i}$ (4.5)

or $B = f_c/\sqrt{2} = 70{,}7\ \%$ (4.6)

The other very common used filter is the third octave band, this is defined as the frequency range bounded by the relation

$$f_{u_i} = 2^{1/3}\ f_{l_i} \qquad (4.7)$$

The bandwidth in this case is 23,1 % of the f_c.

A practical filters used in measuring instruments are made up of electrical resonance circuits, and require a certain time for the output to reach full amplitude. When using a filter of bandwidth B a record lenght of time t_R of at least 1/B is required to pass the signal without significant distortion. For practical use the following relationship has to be fullfield:

$$B\ t_R \geqslant 1 \qquad (4.8)$$

Therefore when it is required to sweep the whole frequency spectrum with filters of narrow bandwidth, we must be sure that the sweep speed is slow enough for the filter to adjust to the amplitude of each frequency component, with constant percentage bandwidth filter, however, we can sweep with logarithmically increasing speed at the disadvantage of losing resolution at high frequencies.

4.2.2. Practical frequency analysis

The procedure of obtaining and displaying the amplitudes of vibration for all the frequencies present is, perhaps, the most useful of all analysis techniques. It is estimated that over 80-90 % of the mechanical problems which occur on rotating machinery can be identified by displaying the vibration amplitude versus frequency data.
When using a simple vibration analyzer with manually tuned filter, the filter is manually tuned over the analyzer frequency ranges and the significant vibration amplitudes and corresponding frequencies identified by carefully observing the instrument amplitude and frequency meters. It can be noted that a common practice is to record the amplitude vs. frequency data measured in the horizontal (H or X), vertical (V or Y) and axial (A or X) pickup directions at each bearing of the machine beind analyzed. Data is manually tabulated on a form such as illustrated in Figure 4-3.
When the analysis data is being manually recorded, the first step is to measure and record the overall or filter-out amplitude and predominant frequency. Vibration velocity measurements are generally preferred for most analysis task. However, displacement measurements may also be taken where it is likely that vibration frequencies below 10 Hz(=600CPM) will be encountered. If very high vibration frequencies are presented (above 1500Hz) vibration acceleration measurements may also be recorded.

IRD MECHANALYSIS, INC. **DATA SHEET** DATE 6/23

☐ NOISE ☒ VIBRATION ANALYSIS OF: #2 F. D. FAN

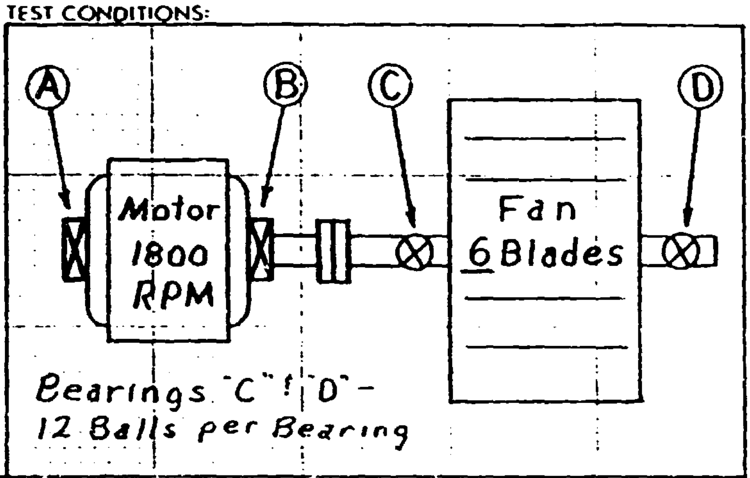
FOR: Determine cause of vibration amplitude increase

MECHANALYSIS EQUIPMENT USED: IRD Model 350 with 544 Velocity Pickup

PERFORMED BY: L. T. J.

LEGEND:
- → PICKUP POINT
- X PLAIN BEARING
- ⊗ ANTI-FRICTION BEARING
- ⊣⊢ COUPLING

TEST CONDITIONS:



PICKUP		FILTER OUT						FILTER IN					
		NOISE		DISPL.		VELOCITY		VEL.	VEL.	VEL.			
POINT	POS.	dB(C)	CPM	MILS	CPM	IN/SEC	CPM	1800	3600	10800			
A	H			4.2	1800	.54	1800	.4	.10	.06			
	V			2.3	1800	.27	1800	.21	.06	.03			
	A			.7/.9	1800	.12	1800	.09	.03	.01			
B	H			4.5	1800	.49	1800	.43	.10	.04			
	V			1.9/2.1	1800	.22	1800	.21	.09	.02			
	A			.10/.13	1800	.14	1800	.11	.04	.01			
C	H			1.9	1800	.48	~	.16	.10	.02			
	V			1.5	1800	.36	~	.11	.09	.02			
	A			.7/.9	~	.17/.22	~	.09	.05	.01			
D	H			.7/1.0	1800	.13	1800	.10	.07	.02			
	V			.8	1800	.09/.11	1800	.07	.03	.01			
	A			.3/.4	~	.04/.06	~	.04	.02	~			
	H												

Figure 4-3. Tabular vibration analysis data.

The "filter-out" amplitude and frequency readings reveal the extent of the problem and can provide a basis for comparison with filter-in amplitudes to check for completeness of the analysis data. In addition, the predominate frequency readings may quickly direct attention to the problem source.

To obtain the "filter-in" measurements each frequency range of the analyzer is carefully scanned with a tunable filter. By scanning each frequency range, all vibration frequencies of significance can be found without trying to anticipate which frequencies will be present. Tune the filter to the first significant vibration and measure/record the amplitude of vibration for this frequency in the H, V and A directions at each bearings of the machine.

After the amplitude readings have been thaken and recorded for the first vibration frequency, continue scanning with the tunable filter until the next frequency is found. Again, fine-tune for the peak amplitude and record the amplitude for this frequency at each measurement point and pickup direction.

Some analyzers are available with provisions for connecting a standard X-Y recorder for plotting amplitude versus frequency (X-Y) signatures.

Figure 4-4. Universal vibration measuring instrument with internal tracking filter and X-Y recorder (Carl Schenck AG)

Plotting vibration amplitude vs. frequency signitures has many advantages over manual/tabular analysis. This eliminates many sources of human error in observing and there is also less chance of missing significant vibration frequencies. Linear amplitude scales are by far the most common, and are familiar to most industrial personnel. There are occasions, however, when a dB (log) amplitude scale may be preferred. A dB (log) ampiitude scale has the advantage of compressing widely variing vibration or noise amplitudes into a smaller physical range, thereby making it easier to plot them grafically. Essentially a logarithmic scale provides additional emphasis to low amplitude signal (see Figure 4-5).

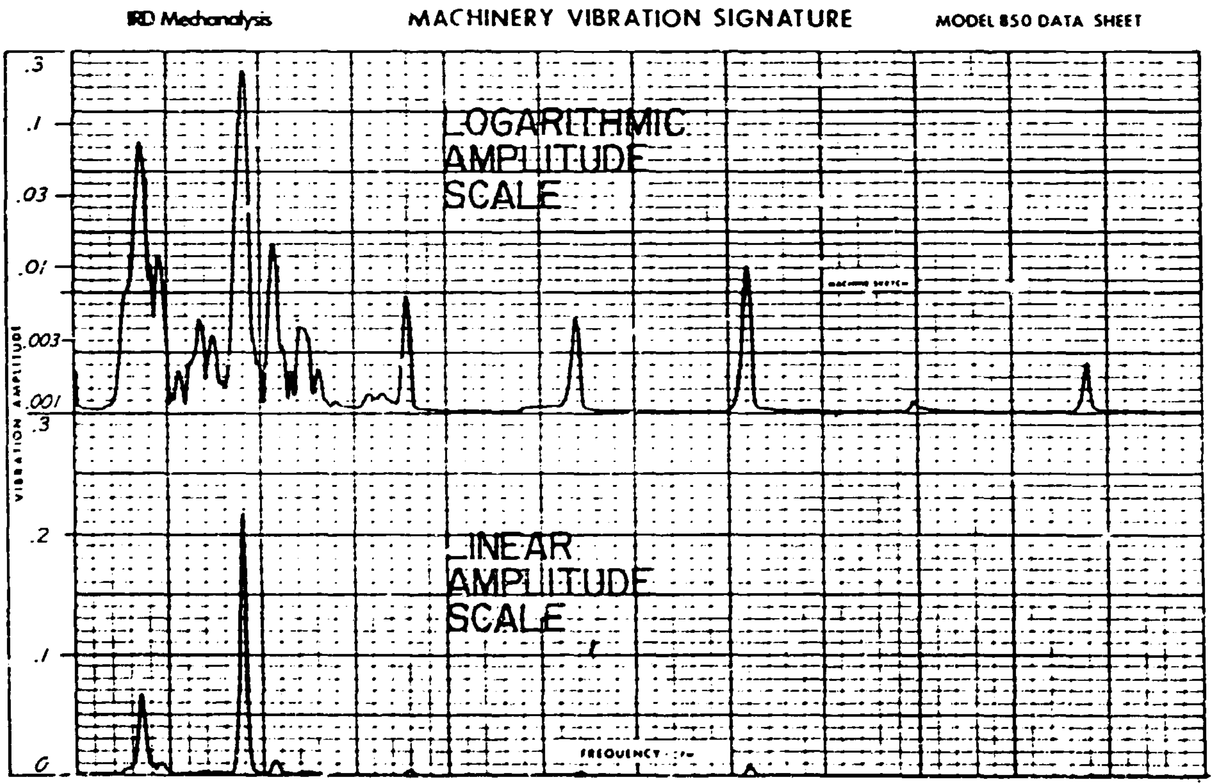


Figure 4-5. Comparison of machinery vibration signatures on logarithmic and linear amplitude scales (IRD Mechanalysis Model 850 data sheet)

4.2.3 Real time spectrum analysis and FFT analysers

Standard frequency analysis procedures are carried out with the assumption that the vibration being analyzed is reasonably steady-state and will be present for a sufficient period of time to allow the frequency ranges to be scanned with the tunable filter and the data recorded either manually or with an X-Y recorder. Unfortunately, not all analysis situations meet this ideal requirements. Some machines may operate under continually varying conditions of speed or temperature resulting in wide variation characteristics. Other machines have random vibration resulting from flow turbulence, combustion or rolling contact etc. In still other causes, the vibration may be present for such a short period of time that frequency analysis by standard procedures is impossible. For those situations where there is need to obtain frequency data very quickly, a real time spectrum analyzer can be used.

The real time spectrum analyzer (e.g. IRD Model 850/860, Brüel & Kjaer Type 2515, Solartron 1250 Series) provides the capability of analyzing complex vibration and noise hundreds of time faster than with conventional analyzers. An oscilloscope built into the front panel such as illustrated in Figure 4-6 provides an instantaneous and continually updated display of the vibration amplitude vs. frequency signature so that the analysis is essentially displayed as it occurs.

The feature of rapidly producing continuously updated spectrum is very valuable for trouble-shooting. It is found that digital filtering is the best technique for constant percentage bandwidth analysis (on a logarithmic frequency scale).

Simultaneously with the beginning of real time frequency analysis interest began to grow in the digital conversion of data between the time and frequency domain.

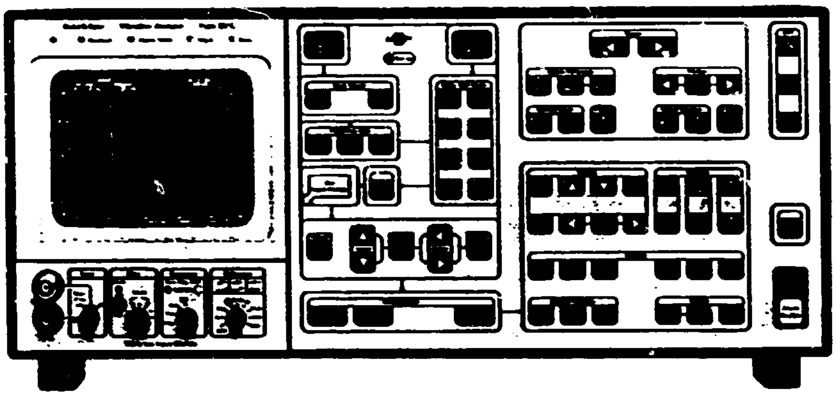


Figure 4-6.
vibration analyzer
Type 2515
(Brüel & Kjaer)

The **Fast Fourier Transform algorithm** (FFT) gives an efficient means of evaluating a Fourier Transform digitally, and digital filtering processes are gaining acceptance. As a result of the development of the large-scale integration (LSI) and the computer techniques over the past decade several system using FFT have been introduced.
Not going very deep into the mathematics we should mentioned, that the FFT procedure is a particularly efficient way of calculating the so-called Discrete Fourier Transform (or DFT). DFT may be interpreted as a finite, discrete version of the Fourier Integral Transform. The actual equation for the forward transform is:

$$G(k) = \frac{1}{N} \sum_{n=0}^{N-1} g(n) e^{-j \frac{2 \cdot kn}{N}} \qquad (4.9)$$

and for the inverse transform is

$$g(n) = \sum_{k=0}^{N-1} G(k) e^{j \frac{2 \cdot kn}{N}} \qquad (4.10)$$

where G(k) represents the spectrum value at the N discrete frequencies $k\Delta f$ from zero to the sampling frequency f_s. The frequency increment between spectrum samples is $\Delta f=f_s/N$. Likewise, g(n) represents the time signal at N discrete time points from zero to T the total record length. The time between samples $\Delta t=T/N$ and thus $n\Delta t$ represents the actual time corresponding to g(n).

Looking at equation (4.9) it will be appreciated that the N values of frequency $k\Delta f$ are equally spaced on a linear frequency scale, and it can be shown that the resolution bandwidth is a fixed proportion of the line spacing (i.e. FFT is the best technique to use for constant percentage bandwidth analysis using a linear frequency scale.)

A typical FFT analyzer has a transform size of N=1024 data samples, and in theory gives 1024 frequency values. However, since the data values are real, only the 512 positive frequency values are calculated. Not all of the 512 values can be used; to eliminate the problem of aliasing a low-pass filter is applied with a cut-off frequency. For the up-to--date analyzers it is typical to place the filter cut-off so that the first 400 lines are valid, and are displayed, while the last 112 lines are affected by the filter and are not operated on further.

Thus, e.g. for the Brüel & Kjaer Type 2031 analyzer the frequency resolution Δf is always 1/400 of the selected full-scale frequency $f_{f.s.}$ and the automatically-selected sampling frequency in 2,56 times the full-scale frequency. There is a simple inverse relationship between frequency range f and record length T given by the formula

$$T = 1/\Delta f = 400/f_{f.s.} \qquad (4.11)$$

For analysis of **stationary deterministic signals** the requirement of the averaging time is that it reduces the ripple of the sensor output to an acceptable level. This will be the case if the averaging time contains at least 3 periods of the lowest frequency to be analysed ($f.T_A \geq 3$).

For **stationary random signals** the averaging time should be chosen so as to achieve an acceptable accuracy of the result. The relative standard deviation of the error in RMS values is given by the formula

$$e = \frac{1}{2(B.T_A)^{1/2}} \qquad (4.12)$$

The FFT technique and analyzer using for efficient condition -based maintenance is a very complex one and requested high level of understanding the maintenance personnel. The frequency range, the type of the filter, the bandwidth to be used all require a certain skill from the diagnostic engineer and good practical experience. An example from the Brüel & Kjaer practice should illustrate the feasibility of the procedure: Fig. 4-7. (a) shows the vibration spectrum of a newly installed machine (a gear-box between a motor rotating at 50 Hz and a centrifugal compressor rotating at 121 Hz) The reference spectrum derived from this is illustrated in the Fig. 4-7. (b). Figures 4-7. (c) to 4-7. (e) are spectra obtained at approximately one month intervals from the machine as its condition deteriorated. Superimposed on each spectrum is a list of significant differences (6 dB) can be found by an automatic comparison program. For the first month as it's illustrated in Fig. 4-7. (c) there is no significant change. One month later (Fig. 4-7. (d) the component at the rotating speed of the high speed shaft (121 Hz) has increased significantly (14 dB) and a month later again (Fig. 4-7.(e)) a further worsening to 22 dB indicates that the situation is serious. Shortly afterwards the machine was shut down for repair. The immediate reason for the increase of the 121 Hz component was a developing misalignment, but the basic problem with the machine was an axial resonance of a disc coupling which was excited by the 4th harmonic of the shaft

speed. Note the development of this component at 484 Hz in the spectra. It should be noted that the shaft speed in Fig. 4-7. (e) is 126 Hz compared with 121 Hz for the others, but this difference has been taken care of without problems by the program so that only the significant changes are detected.
The bandwidth used in Fig.4-7. was 4% (actually 1/18 octave) which results in a spectrum comprised of 180 values over 3 decads, and gives a very high degree of certainty that changes anywhere in the spectrum will be detected. It is about the minimum which can be achieved with a 400-line FFT technique (the line spacing one decade down from full-scale frequency is 2,5 %). It is obvious, that the possibility of using a broader bandwidth at the fault detection stage should be kept in mind in the interests of economy.

4.2.4. Use the strobe light for diagnosis

One of the basic vibration diagnostic instrument is the stroboscope. A stroboscope is a light source that can be adjusted to flash at a desired rate. It may be used to illuminate the vibrating surface in a fixed reference system to be analysed. When the stroboscope flash is aimed at a revolving object or moving parts and the flash rate is set to the same frequency (RPM) or a harmonic multiple of it, the moving object will appear motionless due to persistence of vision. If the flashing frequency is slightly different from the vibration frequency, the vibratory displacement appears in slow motion.
Fig. 4-8 shows the SHIMPO DT-301 digital stroboscope incorporating a 20 W Xenon flashtube with a bright natural illumination and a high precision cristal internal oscillator. The stroboscope converts the flash rate frequency to an RPM LED digital readout.

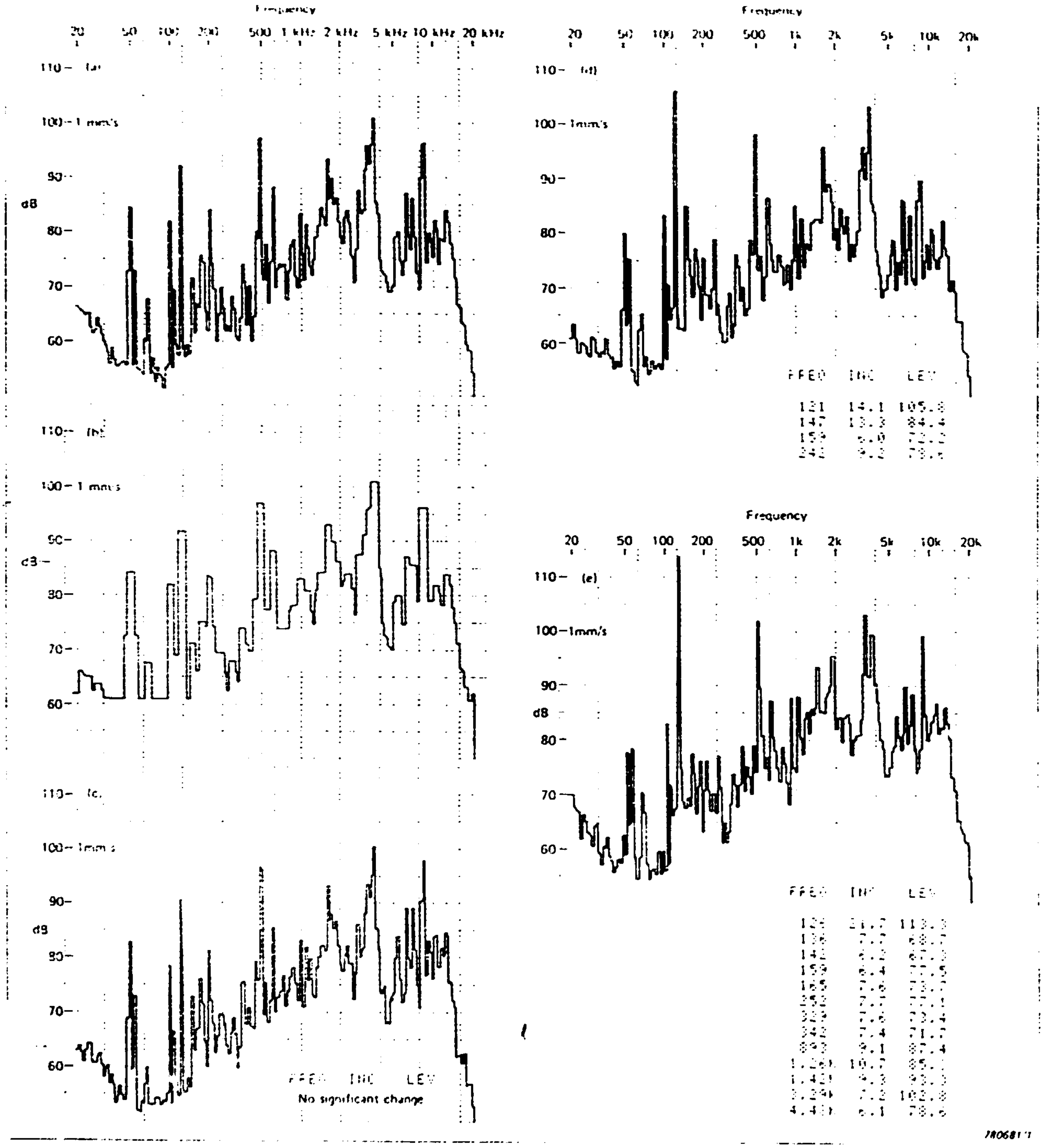


Figure 4-7. (a) Spectrum after installation
(b) Reference spectrum derived from (a)
(c) Spectrum one month later
(d) Spectrum two months later
(e) Spectrum three months later

The flash rate can be adjasted from 100 to 12 000 RPM by dial turning with an accuracy of digital measurement of $\pm$ 0.01 %

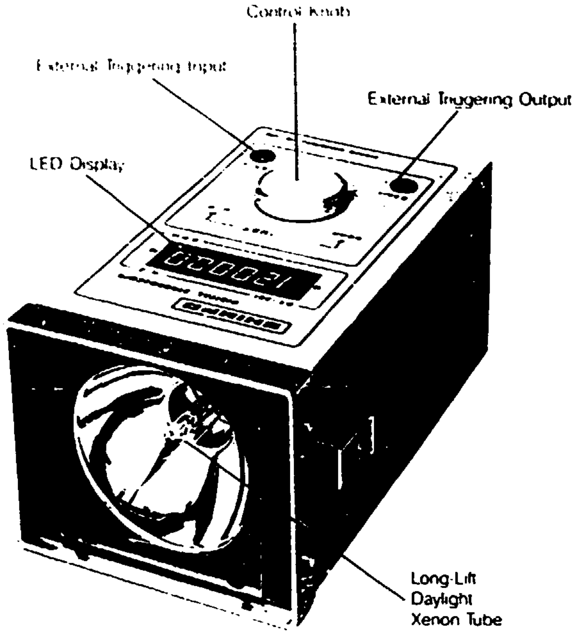


Figure 4-8
Digital stroboscope
model DT-301
(SHIMPO)

The strobe light, actually triggered by the vibration, is a valuable tool for locate the source of vibration, phase analysis and in-place dynamic balancing.

The first step in using the strobe light to measure phase is to establish a common reference point. Normally a reference mark is placed on one end of the shaft, and in a position that can be easily viewed under the strobe light. You can make the reference mark with a stroke of chalk or paint, or just use any distinguishing point like blemish, nick, rust or grease spot on the shaft.

For phase measurement there are three important elements

(i) the direction of the pickup axis

(ii) the reference mark on the revolving specimen

(iii) the angle scale put on the end of the revolving specimen.

You must make certain that phase measurements are always taken with the pickup in the same direction and position, otherwise the records for the machine's phase readings will be inconsitent.

You have seen that many kinds of machinery troubles can be diagnosed by observing changes in vibration phase, like

- o Unbalance o Misalignment o Bent Shaft
- o Looseness o Resonance o Eccentricity
- o Electrical Faults.

4.2.5. Lissajous pattern (orbit) analysis

In recent years it has become a common practice to install dual non-contacting transducers on critical high speed turbo-machines and other rigidly coupled rotating machines to provide redundant, failsafe protection and to avoid false shutdowns in the event that one of the transducers should fails.

This technique generally consists of mounting two radial non-contact pickups at each bearing with the axes of the pickups separated by 90^{0}. The signal from one pickup is applied to the horizontal input (X) of an oscilloscope while the output of the other pickup is applied to the vertical scope input (Y). The resultant display on the CRT will be a plot of the total motion of the shaft within the bearing. Such plots or displays are called **Lissajous pattern** and are also referred to as shaft **orbits**.

The Lissajous patterns are very useful to identify specific machine malfunctions such as unbalance, misalignment, oil whirl, resonance, rubbing, etc. Several type of the mechanical problems are discussed below.

UNBALANCE: Unbalance is characterised by a frequency of 1 x RPM and will reveal a circular or slightly elliptical pattern as shown in Fig. 4-9. If the pattern takes on a highly elliptical shape such as the one in Fig. 4-10, where the ratio of the major axis to minor axis is relatively high, say from 6:1 to 10:1 or more, this suggests that the machine is operating near a structural resonance.

MISALIGNMENT: Misalignment will also cause a predominante vibration at a frequency of 1 x RPM in many cases,but it will often be accompanied by harmonically related frequencies including two-three times and occasionally higher orders of rotating speed.As a result misalignment may reveal a pattern shaped like a "banana" (see Figure 4-11).

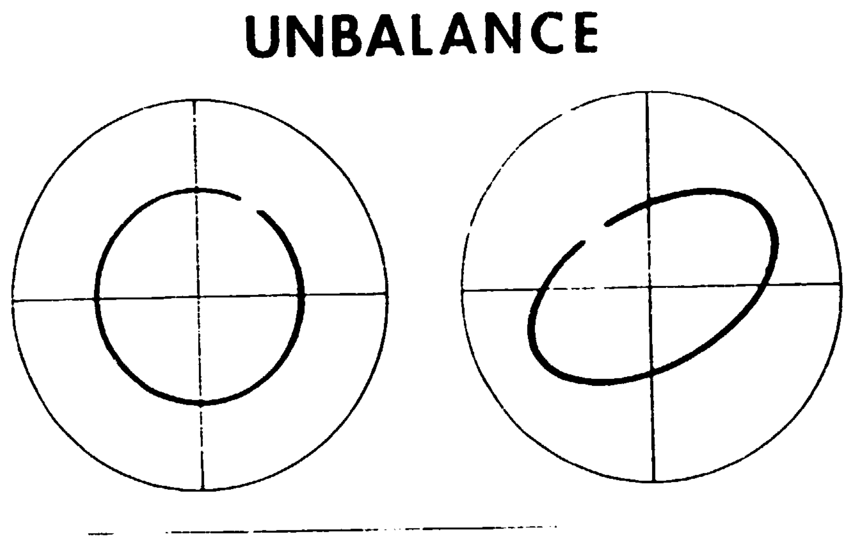


Figure 4-9. Unbalance orbit reveals a circular or elliptical pattern at 1 x RPM

As the higher order frequencies or coupling misalignment the orbit may appear as illustrated on Fig. 4-12.

OIL WHIRL: Oil whirl causes the shaft centerline dynamic motion to be forward and circular or elliptical at a frequency proportional to shaft rotating speed. The vibration frequency produced by oil whirl is a function of the average fluid circumferential velocity in the bearing or seal, but is typically 40-49 % of rotating speed frequency.

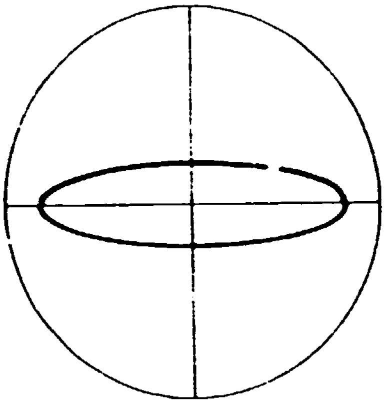


Figure 4-10. Elliptical orbit
indicating bearing wear
or possible resonance

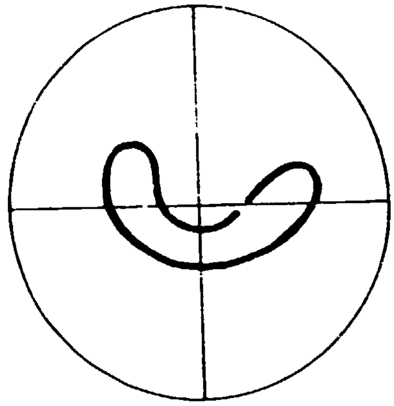


Figure 4-11. Misalignment orbit

The orbit of an oil whirl appears as a circular or elliptical pattern with an internal loop as shown in Fig. 4-13.

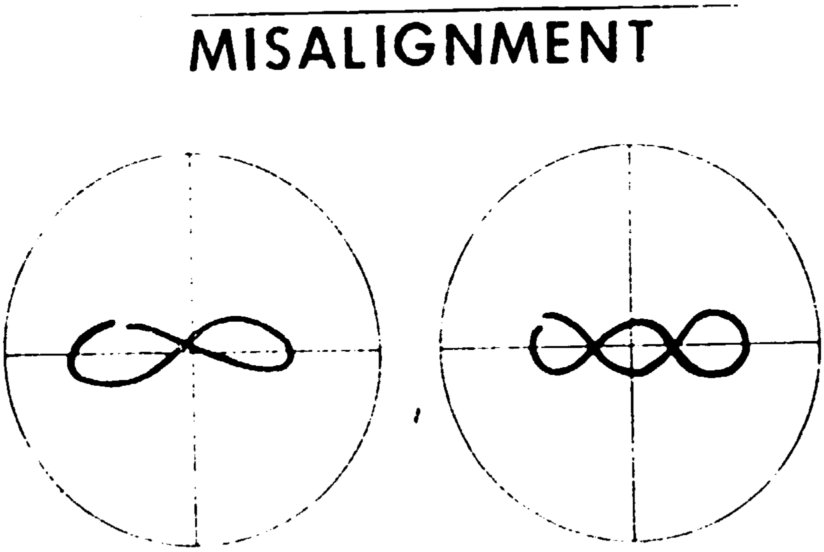


Figure 4-12. Orbit of coupling misalignment

RUBBING: Rubbing between rotating and stationary parts can result in several different patterns as shown in Fig.4-14.

The particular orbit encountered will generally depend on the extent of the rub. In most cases the reference pulse on the orbit will appear unsteady although this can also result from other problems such as mechanical looseness conditions. A very mild rub,where the rotor only touches the stationary part once per revolution may result in a slight distortion of a circular or elliptical unbalance orbit. As the rub becomes more severe, the result the result orbit may take on any one of a number of configurations which include harmonic frequencies,random non-synchronous frequencies and resonant frequencies of various components.

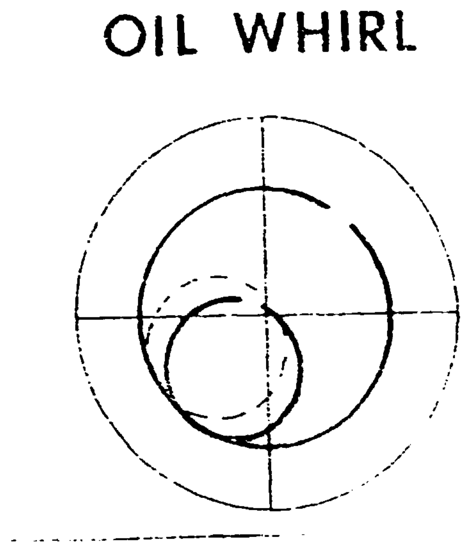


Figure 4-13. Oil whirl orbit

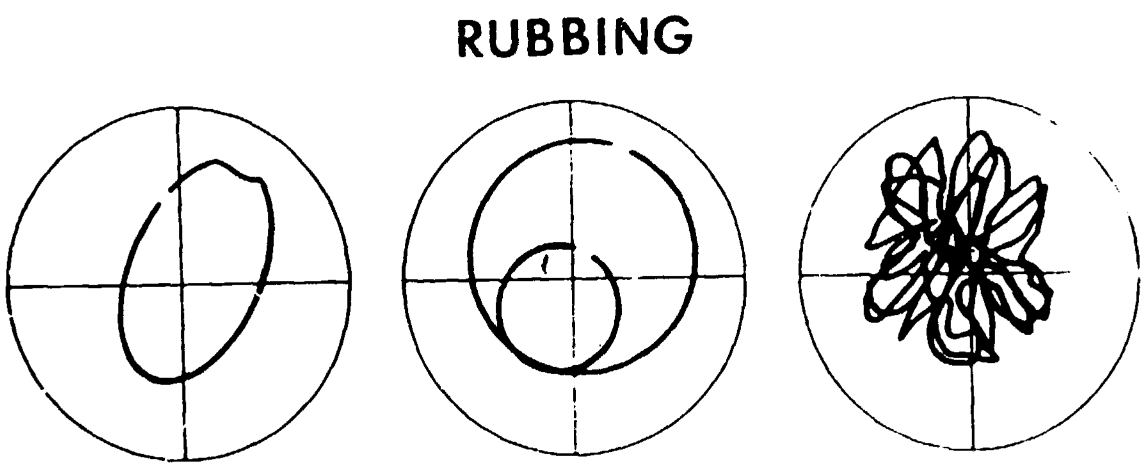


Figure 4-14. Typical orbits cased by rubbing

4.2.6 Amplitude-versus-Phase

In advanced diagnostic approach the amplitude and phase vs. frequency plots are illustrated either on a Cartesian (rectangular) format called as **Bode' plot** or in a polar format called **polar plot.**
The Bode' display represents the vibration vector response as a function of shaft rotative speed. One Y axis represents the amplitude, the other Y axis represents phase lag like on the left side of the Figure 4.15. Another technique for presenting this data is to plot the vibration vector at a specific lateral shaft location with shaft rotative speed as a parameter (see right side of the Fig.4-15).The polar plot is generated by in-phase and quadrature signals,usually during machine start-up or coastdownsuch plot is often incorrectly called a **Nyquist plot.** (According to the definition the Nyquist plot is a type of graphical presentation also in polar format, but it is used to evaluate stability of an existing real system.)

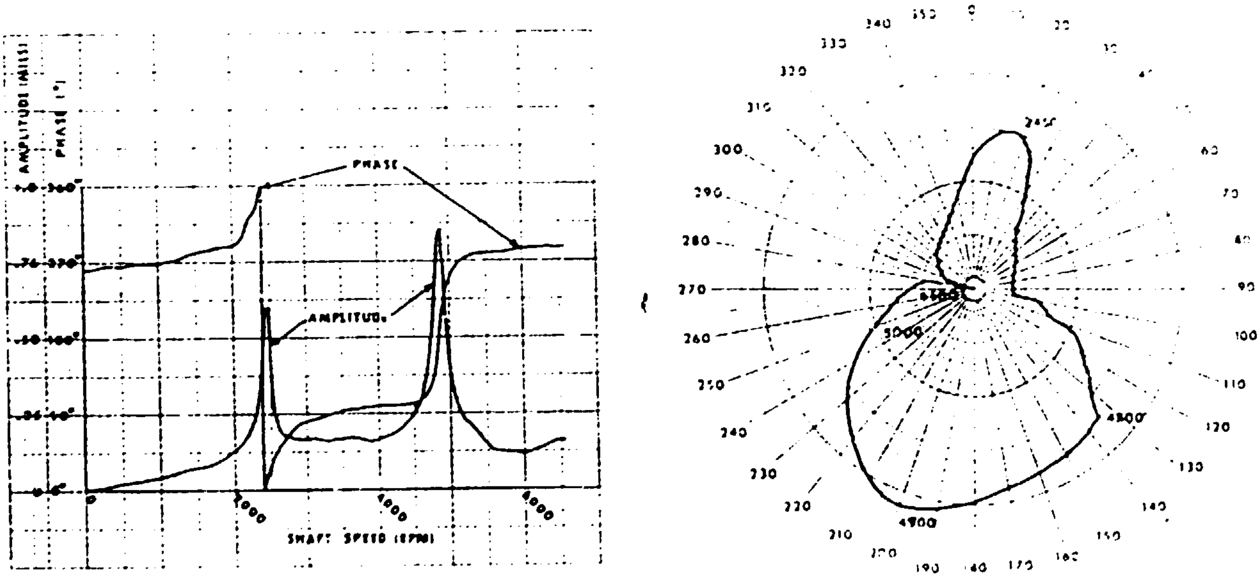


Figure 4-15. Comparing Bode' (left) and polar (right) plots

To obtain polar amplitude-vs-phase plots requires an analyser with tracking filter and polar amplitude and phase output signals for driving a standard X-Y recorder. The X-Y recorder is adjusted so that a zero amplitude signal positions the pen at the origin or the center of the polar graph paper. The amplitude control of the recorder is adjusted so that an increase in vibration amplitude will cause the pen to move outward from the center a proportionate amount. The angular direction record pen similar way will be governed by the phase of the vibration.

Advantages of the polar plots over Bode' plots include:

(i) provides an inmediate indication of the unbalance vector without the need to compare separate amplitude and phase plots;

(ii) eliminates both the confusing phase discontinuites on the Bode' plots as the phase changes from 0 to 360^{0} and the phase discontinuites when the vibration amplitude has reduced to a low level;

(iii) where only a single pen recorder is available, the polar plot allows both amplitude and phase data to be obtained during a single run-up and coast-down of the machine.

Limitations of the polar plots:

(i) while the polar plot does reveal the presence and significance of resonant conditions during run-up or coast--down, it does not provide a speed reference to indicate the RPM at which resonance occurs. As a result, it is necessary to monitor machine speed and manually record RPM values;

(ii) as far as the polar plot is a plot of the unbalance vibration and, thus, is restricted to examining only the vibration occuring at 1 x RMP. Severe resonant conditions can be excited by vibration frequencies other that 1 x RPM.

4.2.7 Tracking analysis

Despite the careful balancing in practice it may occur that machines do not run smoothly at certain operating speeds.
The most likely causes are:

o installation and foundation resonances

o rotor resonances

o partial resonances of the connecting parts, etc.

In order to detect such resonances and to define their causes the **tracking analysis** or the **order ratio analysis** is taking place.

In case of the **tracking analysis** the speed of the test specimen is changed gradually.Residual unbalances and non-homogeneity cause speed-related vibrations to influence natural vibration frequencies. The occuring vibrations are examined by vibration analysator according to amplitude and phase in a frequency selective manner. The diagrams obtained are designated amplitude and phase response. It is the objective of the tracking analysis to obtain information on the dynamic behavior of the specimen. In general the tracking curve is recorded at rotational frequency, i.e. the machine is checked as to the existence of vibrations of the 1st order. Anisotropisms and other malfunctions may set the specimens to vibrations of higher orders. By changing the speed of the rotor and by multiplying the speed one obtains the vibrations occuring at rotational frequency in the different orders. In practice testing are carryed out from the 0.5th to the 4th order. Performing tracking analyses on higher orders we call **order ratio analyses.** Figure 4-16. shows an example of an order curve.

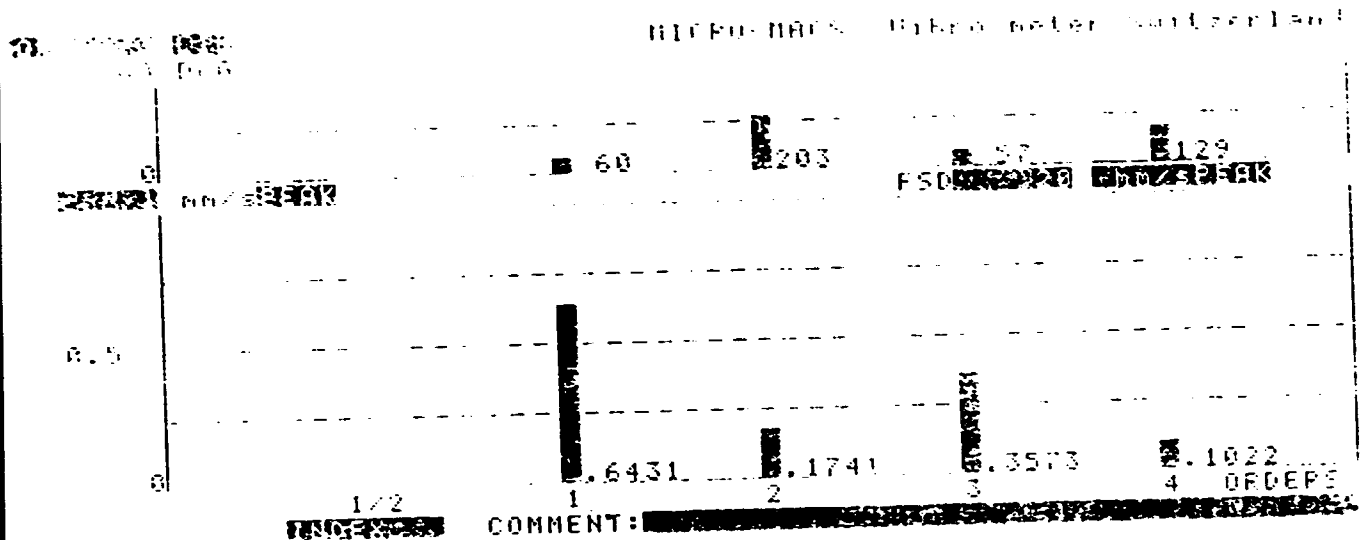


Figure 4-16 Diagrams of phase (top) and amplitude (down) responses of a specimen by order ratio analysis (Analysator model MICRO-MACS (VIBRO-METER)

4.3 DATA INTERPRETATION

After the vibration data has been obtained, the next step is to compare the readings of significance with the characteristics of vibration typical of various types of trouble. In vibration analysis, freqbency is always the key to the vibration source and to this comparison. Using frequency analysis the machine's complete vibration mixture has to be resolved into its individual components according to frequency and amplitude. A comparison by frequency is made on the basis of the rotating speed or speeds of the components/parts in the machines. The amplitude represents the magnitude of any forcing function. The phase may determine position, the sequence, the modal shape, or it may determine one of several sources which transmit similar frequencies. From the individual components of the vibration mixture it is possible to draw conclusions regarding the vibration excitations.

The Figure 4-17. lists the frequencies of vibration normally encountered in terms RPM of a part and the possible cause of the vibration. The trouble referred to will be associated with the part whose RPM is some multiple of the vibration frequency and this comparison made should indicate the part causing that trouble.

Since many machine troubles have similar characteristics and several troubles may be present in a machine simultaneously, it is often necessary to choose between several likely possibilities. One technique that has been found useful is to examine the relative probability of the occurrence of machine faults with the idea of starting with the most probable fault first. This theory was introduced by Jonh S. Shore and published first at the ASME Petroleum Mechanical Engineering Conference in Dallas in 1968.

The Vibration and Noise Identification Chart (Fig. 4-18) by IRD Mechanalysis published in 1974 provides a comprehensive listing of the most common machinery problems.

Application of the chart (Fig. 4-18) include the following steps:

(i) Calculate or determine expected vibration and noise freqencies of main rotating machine components.

(ii) Measure vibration and noise at various measuring points.

(iii) Using the tunable filter of the analyzer, carefully tune through the appropriate frequency range of the analyzer recording the amplitude and frequency at which significant components are detected.

(iv) Generate a machinery signature by making a narrow band analysis on each frequency of interest. Obtain signatures of both vibration and noise as requred.

Frequency	Most likely causes	Other possible causes and Remarks
f_{rotor}	Unbalance	1) Eccentric journals,gears or pulleys 2) Misalignment or bent shaft 3) Bad belts if f of belt 4) Resonance 5) Reciprocating forces
$2 \times f_{rotor}$	Mechanical looseness	1) Misalignment if high axial vibration 2) Resonance 3) Reciprocating forces 4) Bad belts if 2xf of belt
$3 \times f_{rotor}$	Misalignment	A combination of misalignment and excessive axial clearances (looseness) axial clearances (looseness)
$f_{synchron}$	Electrical forces	Common electrical problems include broken rotor bars, eccentric rotor, unequal air gap etc.
$2 \times f_{synchron}$	Torque pulses	Rare as a problem unless resonance is excited
n x RPM (harmoncally related frequencies)	Bad gears Aero dynamic or hydraulic forces Mechanical looseness	Gear teeth X RPM of bad gear Number of fan blades/impeller vanes x RPM Occurs at 2, 3, 4 or higher harmonics
f_{stroke} $2 \times f_{stroke}$ $4 \times f_{stroke}$	Recipro-catings forces	Higher order vibrations can only be reduced by design changes or changing the system
Less than f_{rotor}	Oil whirl (less than $1/2\ f_{rotor}$)	1) Bad drive belts 2) Background vibration 3) Subharmonic resonance

Figure 4-17. Vibration frequencies and likely causes

IRD Mechanalysis, Inc.

Vibration and Noise Identification Chart

Causes of Vibration

(RELATIVE PROBABILITY RATINGS: 1 THRU 10)

		PREDOMINANT FREQUENCIES											PREDOMINA							
													DIRECT			PRO				
		0–40%	40–50%	50–100%	1 X RPM	2 X RPM	Higher Multiples	½ RPM	¼ RPM	Lower Multiples	Odd Frequencies	Very High Freq.	Horizontal	Vertical	Axial	Rotor (Shaft)	Bearings	Casing	Foundati	F
UNBALANCE	Initial Unbalance				10								5	4	1	9	1			
	Shaft Bow–Lost Parts				10								5	4	1	9	1			
MISALIGMENT LOOSENESS AND DISTORTION	Misalignment				4	5	1						3	2	5	8	1	1		
	Mechanical Looseness					8	1				1		5	4	1		3	2	2	2
	Clearance Induced Vibration	1	8	1									5	4	1	7	1	1		
	Foundation Distortion		2		5	2					1		5	4	4	3	1	1	1	
	Case Distortion	←1→			8	½	½						5	4	1	9	1			
	Seal Rub	1	1	1	2	1	1			1	1	1	4	3	3	9	1	1		
	Rotor Rub (Axial)	←2→			3	1	1			1	1	1	4	3	3	7	1	2		
	Piping Forces				4	5	1						3	2	5	8	1	1		
BAD BEARINGS AND JOURNALS	Journal & Bearing Eccentric				8	2							5	4	1	9	1			
	Radial Brg. Damage	1→			4	2						2	4	3	3	7	2	1		
	Thrust Brg. Damage	9→										1	3	2	5	6	2	2		
	Bearing Excited Vibration	←10→											5	4	1	5	2	2	2	
	Unequal Brg. Stiff. Horiz/Vert					9	e CR						5	4	1	4	3	3		
GEARING AND COUPLINGS	Gear Inaccuracies						2				2	6	5	2	2	8	1	1		
	Coupling Inaccuracies					1	8	1					4	3	3	7	2			
CRITICALS	Critical Speed				10								5	4	1	6	4			
	Rotor & Brg. Sys. Critical				10								5	4	1	7	3			
	Coupling Critical				10								4	2	4	1	1			
	Overhang Critical				10								5	4	1	7	1			
RESONANCE	Resonant Vibration				10								4	4	2	2	1	2	3	2
	Sub–Harmonic Resonance							←10→					3	3	4	2	2	2	2	2
	Harmonic Resonance					←10→							4	4	2	2	1	1	3	3
	Casing Resonance				8	1		1					5	4	1		4	4	1	1
	Support Resonance				8	1		1					5	4	1		2	5	2	1
	Foundation Resonance				8	1		1					4	3	3		1	4	4	1
	Torsional Resonance				4	2	2				2		Torsion.			1	4	4		
MISCELLANEOUS BASIC CAUSES	Bad Drive Belts				←10 Belts→								4	3	3	5	3		2	
	Reciprocating Forces				3	5	2						3	6	1	5	3	1	1	
	Aero./Hydr. Forces				2		6					2	5	4	1	4	3	2	1	
	Friction Induced Whirl	8	1	1									5	4	1	8	2			
	Oil Whirl		10										5	4	1	8	2			
	Resonant Whirl		10										5	4	1	2	2	2	2	2
	Dry Whirl											10	4	3	3	4	2	2	1	
ELECTRICAL	Rotor Not Round				10								5	4	1	9	1			
	Rotor/Stator Misalignment				10								4	3	3	8	2			
	Elliptical Stator Bore				10								5	4	1	8	2			
	Defective Bar				10								5	4	1	9	1			
	Bent Rotor Shaft				10								3	2	5	9	1			
	Rotor Not Elect. Centered				10								3	2	5	6	4			

NOISE RADIATION

Mechanical and Electrical Defects—are noise sources which appear initially as vibration and are later transferred into airborne noise. Mechanical Noise may be associated with Fan/Motor Unbalance; Bearing Noise; Alignment; Duct and Panel Flutter–Oil Canning Effect; Flutter of dampers, blades, vanes, tubes and support as well as structural vibration. Electrical Noise—may be due to Electrical Energy transformation:
1) Magnetic Forces–A function of flux densities, number and shape of poles or slots and air gap geometry.
2) Random Electrical Noise–Brushes, electrical arcing, sparks, etc.

Aerodynamic—May be related to vort tions, windage, etc., and create bot Broad Band–a) Fan blades, vanes, c air stream. b) Mechanical Rotation–l c) Abrupt changes in direction of flc (Rumble). Differing flow velocities separations such as boundry layer e etc. Narrow Band– a) Resonances–(vibrating strings, panels, structural vortex effect– air columns excited b Rotation -Siren effects, slots, holes, parts.

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TUDE
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PREDOMINANT NOISE

...se (No of Reference Marks)	Low Freq. "Rumble"	Loud "Roar"	"Hum"	Periodic "Beat"	High Pitch "Whine"	Very High Loud "Scream"	Very High "Squeal"	Ultrasonic	REMARKS
		8	2						Most common cause of vibration whose amplitude is proportional to the amount of unbalance. May be aggravated by or may produce complications such as seal rubs, bearing failures or resonances. (Overhung rotors may show relatively high Axial Vibration).
		8	2						
: (3)		4	4	2					Misalignment appears as a large axial vibration. Use dial indicators or other methods for positive diagnosis. May produce friction or deflection forces which can be severe. Looseness creates many problems. Small amount may allow violent vibration. Looseness in bearings may be mistaken for oil whirl. Usually accompanied by unbalance and/or misalignment. Distortion causes vibration indirectly by generating misalignment, causing internal rubs or uneven bearing contact. Piping Forces & Foundation Distortion often cause resonance problems. Rubs are characterized by the presence of many frequencies all over the spectrum often ultra-sonic. Produce "Hot spots" resulting in bent shaft, bearing cavitation and resonances.
-atic	8	1	1						
-ic	6	2		2					
ic	1	5	3	1					
ric	1	7	1	1					
-:	2	5		1		1	1		
-ic	3	5		2					
(2)	3	4	3						
)		1	9						In the case of Anti-friction Bearing failures, very high frequencies will be noted with the bearing responsible being the one at the point of the largest high frequency vibration. Journal eccentricity relating to gears appears largest in line with gear centers. On motors or generators vibration disappears when power is turned off. On pumps and blowers, improvement may be accomplished by balancing. Velocity measurements are recommended when analyzing for Anti-friction bearing failures.
-ic	2	4	1		1	1	1		
-ic	8	1	1						
ic	6	1		3					
;ing		1	6	2	1				
-ic	2	1	1	2	2	1		1	Misalignment is prime cause of gearing failures. Pitting, scuffing & factures from non-uniform loading results. Couplings are susceptible to be both misalignment and torsional forces. Friction whirl/low damping also contribute.
-ic			5		5				
Chg		5	3	2					For practical purposes, the terms "Natural Frequency", "Resonance" and "Critical Speed" are synonymous. Minute unbalances cause large shaft deflections due to centrifugal force at critical speed. Differs from resonant Vibration in that the shaft does not vibrate "Back and forth" but rotates in an ever increasing bow, assuming equal radial damping. Shaft will bend rather than fail from fatigue as in the case of resonance. A critical may be improved by balancing. Resonance may be improved by internal damping.
ging		5	3	2					
ging		2	4	2	2				
ging		5	4	1					
ic	4		3	3					Resonance—Only amplifies vibrations from other sources, cannot generate vibration. Can create highly dangerous situations by amplifying normal vibration in rotating machines or from pulsations in piping. May cause rotors or bearing abnormalities such as Resonant Whirl. Torsional Vibration is not usually noticeable externally since motion is superimposed on the rotation similar to the action of a washing machine agitator. Failures may occur without warning unless gearing is involved resulting in noise; also bearing and case vibration. Special transducers usually required. Torsional Resonant frequencies coinciding with electrical frequencies can become very serious.
ing	8			2					
			4	2	4				
	2		2	6					
	2		2	6					
		1	8	1					
	1	2	2	3	1	1			
2)	1	1	3	5					Bad Belts—Strob light will freeze faulty belt. Cure is matched belt sets, equal tension & correct alignment. Recip. Forces—Inherent in reciprocating machines—can only be reduced by design changes or isolation. Aero-Hydro Forces—Occur usually at N°. of impeller blades X RPM. Random Pulses may produce related resonance. Friction Whirl—Sometimes called "Hysteresis Whirl". Rare but violent. Cause: Rotor passes thru critical; angle between unbal. & shaft "High Spot" swings 180° with friction damping also 180° out of phase. Frequency of vibration always at actual rotor critical speed. Oil Whirl—Caused by shaft being pushed around in bearing clearance by oil pressure wave. Frequency ½ shaft speed less 2%–8% due to friction effects.
ic		8	2						
r		3	2	1	2	2			
ic	6		2	2					
ic	6		1	3					
ic	6		2	2					
ic							2	8	
		8	2						*Phase at synchronous frequency. Electrical causes of vibration will show up at 60 & 120 Hz (1 & 2 X line frequency) and disappear quickly when power is turned off. A "Slip-beat" vibration may occur at slip speed times number of poles. "Beat Frequency" relates to more than one machine operating at nearly the same speed. Mechanical defects may be detected with conventional indicating methods. Defective Bar-Break bar connection, energize one phase with low voltage and turn rotor by hand. Current surge will indicate broken bar. Check air gaps.
ng		5	2	3					
e		6	2	2					
		2	6	2					
		8	2						
		3	3	4					

...ressure pulse-
...row band noise.
...upports in the
...lts, slots, etc.
...ction of ducts
...eams-flow
...ssion effects,
...effects,
b) Sharp edge
...e). c)Mechanical
...es on rotating

Impactive—Created by the forceful contact of one body or element with another such as the noise produced by a dropped hammer a thunder clap, sonic boom, etc. Tooth impact in gearing may be audible as well as the slap of faulty drive belts. Impact noises may occur so rapidly that special high speed recording techniques must be used to distinguish the periodic impact from the unpredictable transient. Areas with many impact generators will have a steady state "Drone" resulting from the accumulation of many impact "Peaks"

Fig. 4-18.

(v) Study the machinery signature and examine each significant vibration component (amplitude vs frequency).

(vi) Relate all significant vibration components to rotational speeds of machines main components.

(vii) Follow the appropriate vertical columns on the chart looking for machine troubles that high probability ratings.

(viii) Perform further detailed analysis as required to complete the identification of all sources of vibration in the machine in question.

CHAPTER 5 ROLLING BEARING DEFECTS DIAGNOSIS

5.1. PIONEERS OF ROLLING BEARING DIAGNOSIS

A great number of rotating machine use rolling element bearing and many of them are designed for high-speed and high-performance operation. They most often perform over long periods of time and under severe conditions; and when the bearings in the machines fail, downtime can be very costly. The measurement and evaluation of the rolling element bearings occupies a special position within the broad field of maintenance using machine diagnostics. Vibration problems associated with anti-friction bearings on critical machines can be difficult to understand and analyze. Defects in these bearings cause high frequency vibrations of comparatively small amplitude. These vibration components can lie well about 1000 Hz,so that the are not picked up by bearing vibration measurement according to ISO. The purpose of this chapter is to describe rolling element bearing defects in some detail. The theory and the practice detecting rolling element bearing base on the work and long experience of two very progressive and advanced companies, namely

- SPM INSTRUMENT AB (Strängräs, Sweden) working out its famous **Shock Pulse Method** and
- IRD MECHANALYSIS (Columbus, Ohio, USA respectivelly Chester, UK) developing the theory of **Vibration Spike Energy**.

The further explanating will base on the articles of these two companies.

5.2 BEARING FAILURES AND THEIR CAUSES

The measurement and analysis of the condition of rolling element bearings occupies a special position within the broad field of predictive maintenance. Rolling element bearings are manufactured under one of the most sophisticated quality control requirements. As a result they are one of the most precisely made devices available. Notwithstanding that under ideal operation conditions bearing can work through many years of continuous use,time dependent maintenance for bearings is inefficient due to the fact that the service life time for individual bearings cannot be predicted.
The life of a rolling element bearing depends on the conditions under which it is manufactured, the conditions of storing and handling it, installation practices, the load foreces and the general operating environment.
Only a small percentage of bearings fail because the natural fatigue limit of the steel has been reached. For the large majority material fatigue starts early because rolling elements and raceways are not properly separated by a protective lubricant film. The cause for this is mostly insufficient or incorrect lubrication. Improper mounting, e.g. too much preload, is another reason. Damage is also caused by dirt in the bearing,excessive force when mounting, electric current, machine vibration. Table 7 lists some specific types of bearing troubles and their causes.

Table 7

Bearing troubles und their causes

CAUSE	EFFECT
Excessive load	Fractures Overheating Cold flow of metal
Unbalanced load	Damage to raceways
Misalignment	Gouged grooves in raceways Fractures Retainer Failure
Defective shaft seats and housing bores	Spalling
Faulty mounting	Spalling Assembly damage
Improper fit	Spalling Abrasion Fretting Assembly damage
Inadequate or improper lubrication	Surface fatigue Smearing Scoring Debris denting Overheating
Poor sealing	Abrasion Atmospheric corrosion Scoring Debris denting
Background vibration, shock loads, poor mounting, bad transportation	True brinelling False brinelling
Electric current	Electrical damage

5.3. DETECTING BEARING FAILURES

It is very difficult to detect the defects in the bearings since they cause high frequency signals with comparatively very small amplitude. This vibration components can lie well above 1000 Hz, so that they can not be picked up by bearing vibration measurements according to ISO/VDI vibration severity criteria. It is even more difficult to reliably interprete the obtained signal in terms of bearing condition.
Using high frequency vibration analysis, it is possible to detect rolling element bearing damage, but the instrumentation costs for this are considerable. Therefore a special vibration measurement method was developed, in order to be able to obtain an unambiguous information on the condition of such rolling element bearings in a simply way.
This method requires the use of a piezo-electric accelerometer pickup having a high natural frequency (generally above 20 kHz).

5.3.1. Principle of measurement

The principle of measurement of rol ng element bearings is based on the **Shock Pulse Method (SPM)** patented by SPM Instrument AB or on a similar method called as **Spike Energy Measurement**, developed by IRD Mechanalysis. Shock pulse analysis technique is illustrated in Figure 5-1.
In spite of the fact that bearings are one of the most precisely made devices available, the bearing raceway consists of millions of inperfections. A defect on the raceways or rolling elements causes intermittent impacts between the bearing components. At the moment of impact between two colliding bodies, a pressure wave spreads through the material of both bodies. When the wave front hits the bearing mounted accelerometer, this shock pulse excites the sensor into vibration. This vibration has a feature of a dampened

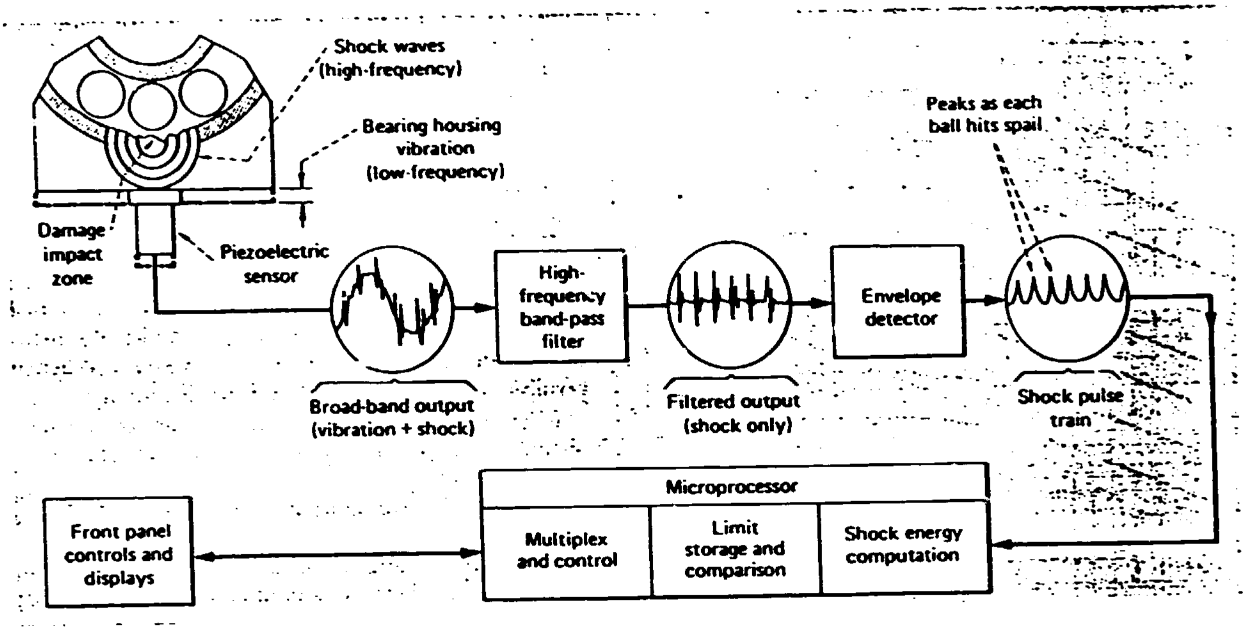


Figure 5-1. Shock pulse/Spike energy analysis technique

oscillation on its reference mass. The peak amplitude is a function of the impact velocity at the moment of impact. During the next phase of the collision both bodies will start to vibrate. The frequency of this vibration is a function of the mass of the colliding bodies. The initial pressure wave is transient and it quickly dampens out. It causes the reference mass of the shock pulse pickup to vibrate at its own resonance frequency in practice of 32 kHz.

Spike Energy has beend defined as

"vibration energy generated by short-duration, metal-to-metal impacts and random vibration that propagate through the structure."

Compared with other vibrations, shock pulses are very weak signals which will hardly affect the general vibration spectrum of a machine.The shock pulse or spike energy transducer, however, reacts with a large amplitude vibration because it is excited at its own resonance frequency. The resonance frequency of the bearing failures transducer is higher than that of the vibrating machine parts. It is therefore possible to filter out all vibration frequencies except the resonance frequency of the transducer.

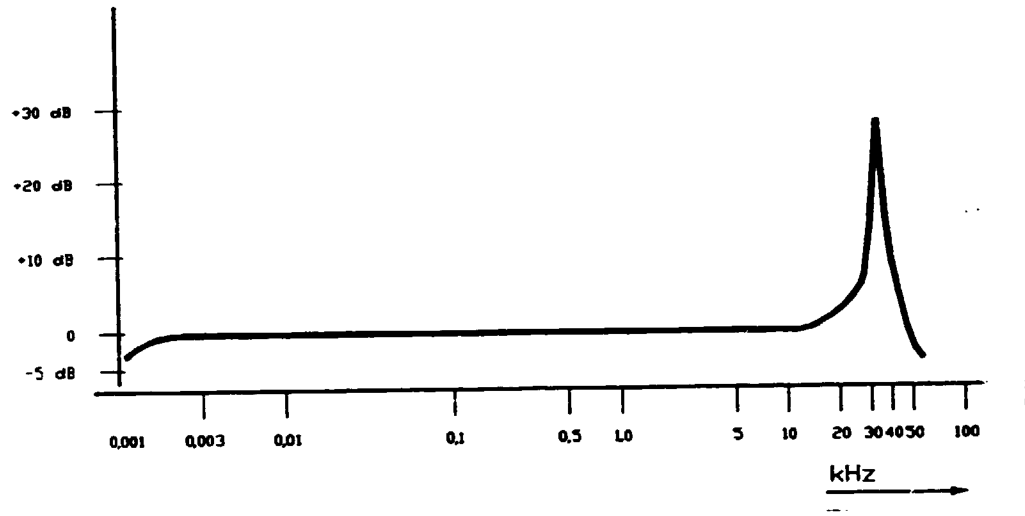


Figure 5-2. Response characteristics of AS-011 accelerometer (SCHENCK)

Figure 5-2 shows the typical response characteristics of the SCHENCK accelerometer. An IRD Model 970 accelerometer pickup has a mounted natural frequency of about 27 kHz. The frequency of both accelerometers is well above the vibration frequencies generated by mechanical problems in the machine -unbalance, misalignment, aerodynamic or hydraulic forces, electrical problems, etc.

As a result, the only sources of vibration that excite the natural frequency of this accelerometer are the impact, or spike, forces generated by a defective bearing.

The broad-band output of the piezoelectric pickup as shown in Figure 5-1 consists of vibration signal from the machine. The transients at the resonance frequency, caused by shock pulses, are superimposed on the vibration signal. The sensor output is conducted to a high-frequency pass filter that excluded any remaining frequency components below 5000 Hz. Consequently, increases in vibration due to problems such as machinery inbalance or misalignment do not cause any increase in the shock value reading. In other words the filtered output consists only of the shock pulses. The amplitudes depend on the energy of the shock pulses. The transients are then converted into analogue pulses with different amplitudes.

5.3.2 Instruments available

The Bearing Analyzer BEA-52 of SPM Instrument is the result of the development of the in earlier years well-known Shock Pulse Meter 43 A.The newest Bearing Analyzer BAS 10 consists of a complete bearing analysis system, which measures bearing damage, analyzes lubrication condition, stores input data and readings, provides graphic trend displays,downloads readings to printer and interfaces through RS-232 and Centronic with IBM compatible PC.
Shock pulse meters like the BEA-52 and the BAS-10 measure shock pulse strength in dBsv (Decibel Shock Value) at two levels:

HR = High Rate of occurence, the level at which 1000 shocks per second can be counted,i.e. a measure for the weaker shock pulses.

LH = Low Rate of occurance, the level at which 50 shocks per second can be counted,i.e.a measure for the strong pulses in the pattern.

HR, LR and the difference between them (delta dBsv) are used to evaluated the signal and determine a bearing's operating condition.

The measured signal is evaluated by a microprocessor-based electronics. It needs input data defining bearing type and rolling speed. Rolling speed is calculated from RPM and mean diameter D_m, and input as the NORM number. Bearing type, defining function (radial or thrust bearing) and shape (ball, roller, needle, single or double row) is input as the SPM type number 1-9.

The bearing condition is described by a number code:

- The LUBRICATION NUMBER is a measure for absolute oil film thickness. LUB 3 for ball bearings, 6 for roller bearings is needed to reach the full rated life. If a bearing has this or better values, lubrication condition is good.

- The CONDITION NUMBER is indicates the degree of surface damage.

The IRD model 810IS and 811 Vibration/Spike Energy Detector as well as the Model 820 Vibration/Spike Energy Signal Analyzer show the measured spike energy in g-SE units. The Model 810IS portable vibration meter is BASEEFA and HSE(M) approved, so it is suitable for measurements from machines located in a hazardous environment in industries such as oil and gas, chemical, petrochemical, mining, marine. Figure 5-3 shows an example of a Spike Energy pattern vs frequency due to a defective bearing.

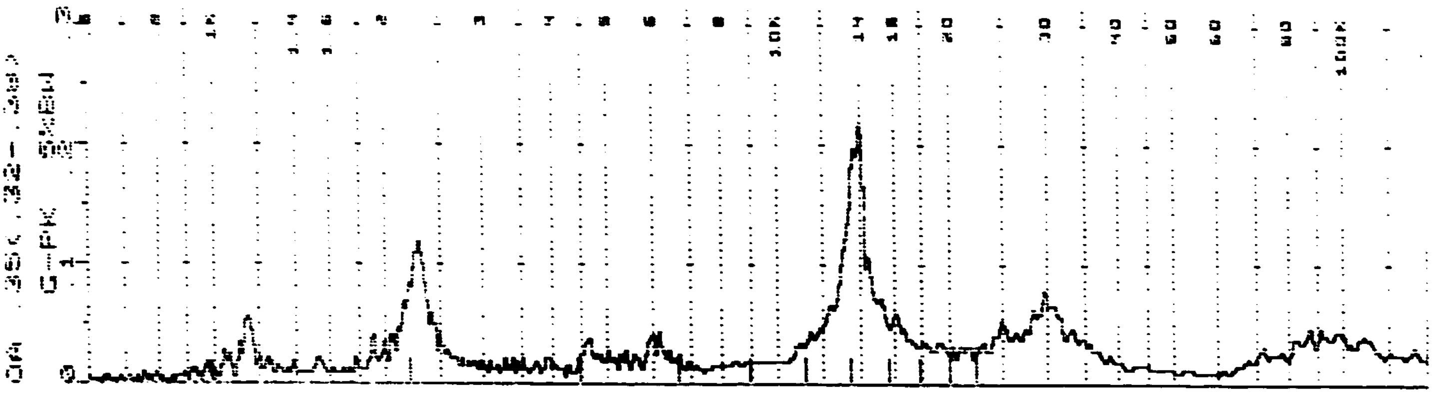


Figure 5-3. Typical Spike Energy signal (IRD)

5.3.3. Establishing severity criteria

Evaluating the condition of a rolling element bearing there are no general guidelines available, which enable one to judge,whether the measured value exceeds permissible limits. Apart from rotor speed, the geometrical dimensions of the components and the bearing itself, the shape of the rolling elements and some other parameters affect the measured results.Because these variations can occur,however, the recommended approach to the determination of bearing condition is the use of **comparison and trending methods**, rather than relying on absolute values of severity charts.

Establishing bearing condition severity criteria by comparison involves measuring the Shock Pulse/Spike Energy levels of similar bearings on a number of similar machines.

The reference scale can be determined in the running-in phase. In the operating phase for good bearings the bearing condition will generally fall within a limited level range. The bearings with significantly higher levels should be singled out for further analysis. If the bearing condition value exceeds the as-new value by a factor of 3 or 4, it is likely that bearing damage will occur in the near future.

Figure 5-4 shows a variation in the so-called **Bearing Condition Unit** (BCU) introduced by SCHENCK for bearings.

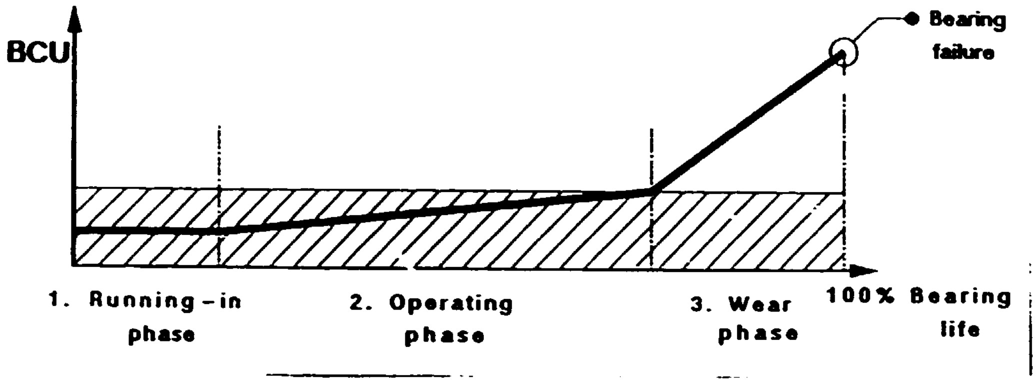


Figure 5-4. Experimental evaluation of the bearing condition

CHAPTER 6 CONTINUOUS MACHINE MONITORING

6.1. WHY USE CONTINUOUS MACHINE MONITORING?

All the machine-vibration measurements discussed in the previous chapters have been based on periodic condition checks. When using time dependent machine maintenance with fixed intervalls between inspection, there always exists a risk during the operation life of the machine,that unpredicted failures will occur. Therefore,permanent (continuous) vibration monitoring plays a significant role in today modern plant management. As the name shows, this type of vibration monitoring system is permanently employed on a specific machine and continuously surveys its condition. Permanent monitoring is employed primarily to give immediate warning of sudden changes in the condition of expensive non-duplicated machinery whose breakdown would cause high cost.Such machines are for example:

- o turbines and generators
- o compressors, fans and ventilators
- o centrifuges

o separators

o cooling pumps in power stations, etc.

Fault conditions are detected immediately, or within seconds/minutes of occurance,and trigger alert or alarm signals in the plant control room so that appropriate measures can be taken before catastrophic failure occurs. The involving of the machine monitoring into the process controlling presupposes an extremely high reliability and working security of the devices.

Monitoring devices are machine protection means of low investment cost which reliably protect high installation values.These systems are widely used in power-generation, chemical and petro-chemical industries on feed pumps, turbines, gas compressors etc.

6.2. WHICH PARAMETER SHOULD BE MONITORED?

In the design of a fault detection systems, one wants to have a system that responds rapidly when a fault occurs. The first step in setting up a fault detection system is to know the process. All sorts of special factors that influence detection system design are associated with a particular process.

Because of the different types of machines,it is in general not possible to say what kind of parameters should be used for monitoring purpoces. Mechanical vibrations are the most informative indicators of machine condition. Figure 6-1 illustrates those parameters which are normally monitored for different machine types. The choice of parameters to be monitored is,of course not limited to those given in the figure. Depending on requirement,one can also use pressure, flow rate, load and many others.

Measurement parameter / Machine type	Absolute bearing vibration	Relative shaft vibration	Axial rotor shift	Speed	Temperature	Casing and shaft expansion
Fans, Ventilators	■					
Rotary compressors	□	■	■			
Centryfuges, Separators	■					
Pumps	□	■	■			
Generators	■	□			□	
Gear boxes	■	□		□		
Electric motors	■	□			■	
Steam turbines	□	■	■	■	□	■
Water turbines	□	■	■	■		
Gas turbines	□	■	■	■		

■ **Standard measurement parameter**

□ **Additional or alternative measurement parameter**

Figure 6-1. Machine types and frequently used monitoring parameters

A second step, one particularly useful for diagnostic, is to establish the subsystem boundaries of parameters.

The third step is directed to determine the hardware required for the detection system. It usually modifies the process design to include the capability of detecting faults, compensating for them by control algorithms, activating backup equipment, etc. A prime requirement of hardware is an extremely high operational reliability and long-term stability.

One of the main problems with any system is in deciding whether the information given by various transducers are correct. There is very important to evaluate the readings either by and active check on the performance of the instrument or by passive check. These systems typically included an automatic test function so that the plant operator can immediately check whether the instrumentation is fuctioning correctly in the event of an alarm.

Figure 6-2 illustrates the examples of error protection circuits. These internal protective circuits prevent limit value indications, which are not caused by dangerous or "not permissible" conditions of the machine, but rather by external disturbing effects.

Protective circuit	Effect
Mains return surge protection	Surpresses transients in the electronics
Supply failure bridging	Immobilises the limit value relais during short duration supply failures
Internal monitoring	Indicates defective pickups and supply failures
OK trip defect	Blocks defective measuring channels
Alarm delay	Surpresses indication, when limit value is exceeded for short duration only
Trip multiplier	Raises limit values for transiting resonance

Figure 6-2. Protective circuits and their effects

6.3 MONITORING EQUIPMENT

6.3.1 Which is the correct machine monitoring installation?

Machine monitoring installations are available in different design and constructions.

Single channel monitors are best applied, when only a small number of measuring of positions is available on individual machines. These instruments are generally an individual or universal monitors in compact form designed monitoring

- o absolute bearing vibrations
- o relative shaft displacements and/or
- o axial displacements.

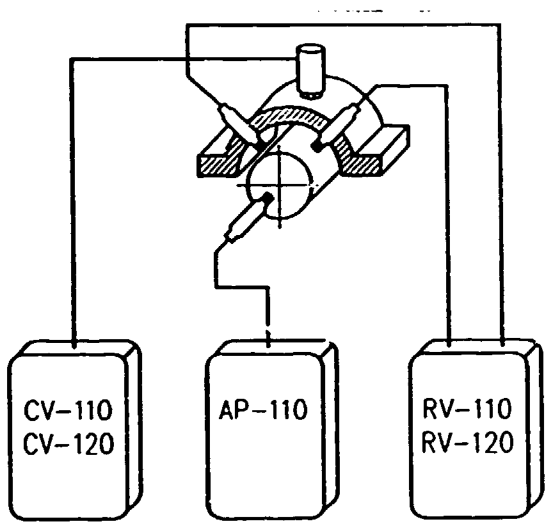


Figure 6-3. VIBROCONTROL 1000 single channel monitor system (SCHENCK)
CV-110/120 - absolute bearing vibration monitor
AP-110 - axial displacement monitor
RV-110/120 - relative shaft vibration monitor

If it is necessary to monitor a multitude of measuring positions and parameters on individual machines or groups of machines, then the use of modular monitoring systems is necessary and economically justified. The modular conception allows the realization of small or large multi-channel installations according to individual requiremenst. The individual modules can be adapted to the particular monitoring task.

6.3.2 Single channel monitors

In the case of one group of macnines, eg small pumps,motors, fans,centrifuges, separators, etc., monitoring vibration velocity at only one bearing position in itself provides reliable machine protection. For this purpose single channel channel monitors are the correct solution.
In the market various monitors are available for single channel use. Figure 6-4 gives a summary of the types and models most called for.

Company	Model/Typ
BENTLY NEVADA	o VAM-CB 100 Velocity Alarm Modul o 27473 Low Frequency Monitoring Unit
BRÜEL & KJAER	2505 Multipurpose Monitor
CARL SCHENCK	VIBROCONTROL 1000 Single Channel Machine Monitoring Units
IRD MECHANALYSIS	o 1229 Single Channel Vibration Monitor o 5802 Machine Monitor
VIBRO-METER	o VMS 850 VIBROMETER o VMS 855 Universal Single Channel Monitor

Figure 6-4. Single channel monitors

Although the technical solutions and the components used to realize the instruments are different, their principle of operation is rather similar. A simplified block diagram of the monitor is illustrated in Figure 6-5.

In the standard system the acceleration signal from the transducer is passed through a differential charge amplifier into an integrator, determining the monitoring parameter.The signal is then passed through a low pass filter and into an amplifier and attenuator. From there the signal is rectified and passed to the meter unit and to the detector stages of the warning system. The meter module will display a continuous indication of the vibration level.
The monitor consists of a 3 level alarm and trip system to indicate when the vibration levels depart from the normal values. The Minimum Level indicator shows when the vibration falls below a preset level,i.e.,indicating a loss of signal. The Alarm Level indicator shows when the vibration exceeds a preset level and finally a Trip Level indicates when a signal exceeds an even higher level. All three level indicators have relay outputs to operate a remote warning system.

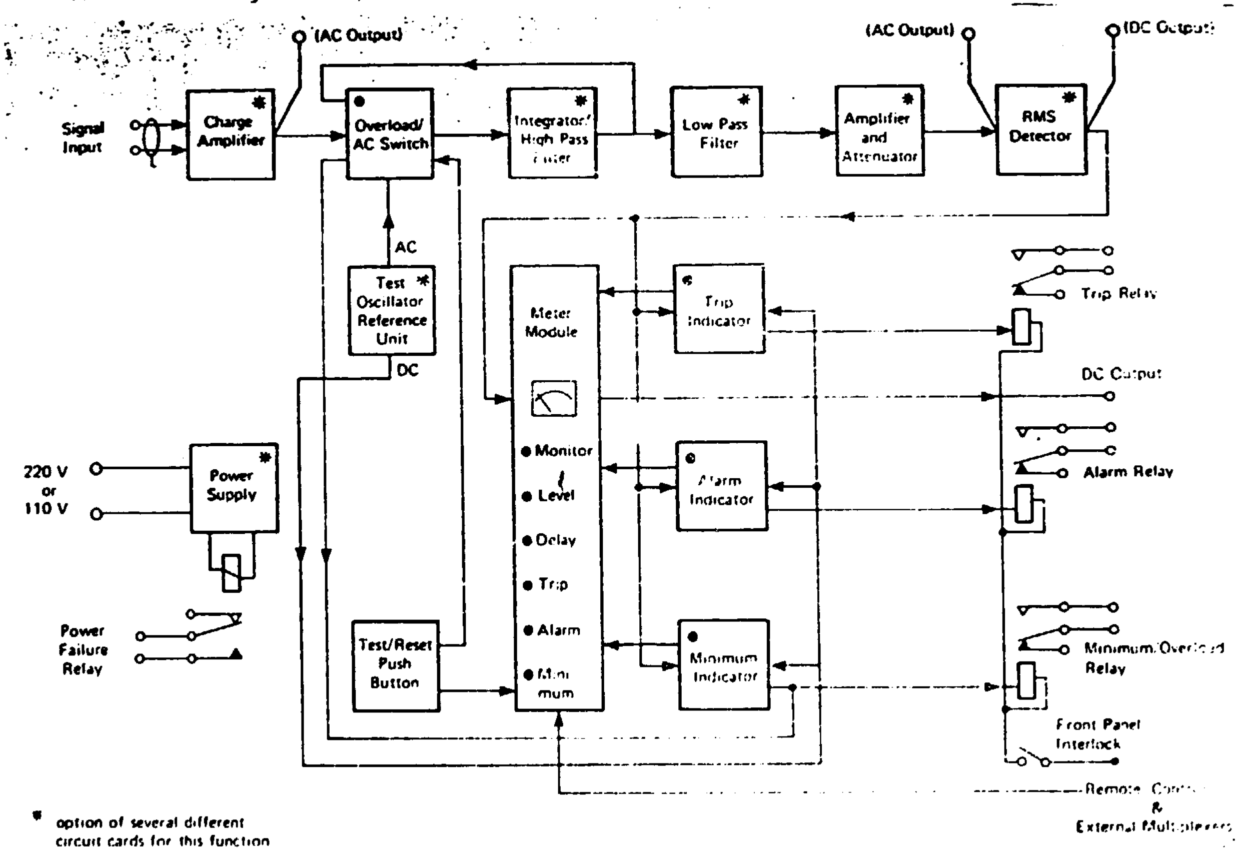


Figure 6-5. Block diagram of the 2505 Multipurpose Monitor (Brüel & Kjaer)

6.3.3. Modular monitoring systems

When critical machinery is involved, the modular, multi-channel monitoring system is the most economical solution. It is universally applicable, regardless of whether many different parameters or a multitude of the same parameter are to be monitored. Suitable monitoring modules are available for every parameter which is relevant to monitoring.
Figure 6-6 gives up a summary of the modular systems available of world-wide interest. Figure 6-7 shows an example of the VIBROCONTROL 2000 monitoring installation.

Company	System
BENTLY NEVADA	o 3300 System o 7200 Monitoring System o 9000 Monitoring System
CARL SCHENCK	VIBROCONTROL 2000 Machinery Monitoring System
IRD MECHANALYSIS	8800 Series Unicel II System
SPM INSTRUMENT	BMS Bearing Monitoring System
VIBRO-METER	Multi-channel monitoring installations by using PIEZO and/or VIBRAX systems

Figure 6-6. Modular, multi-channel monitor systems

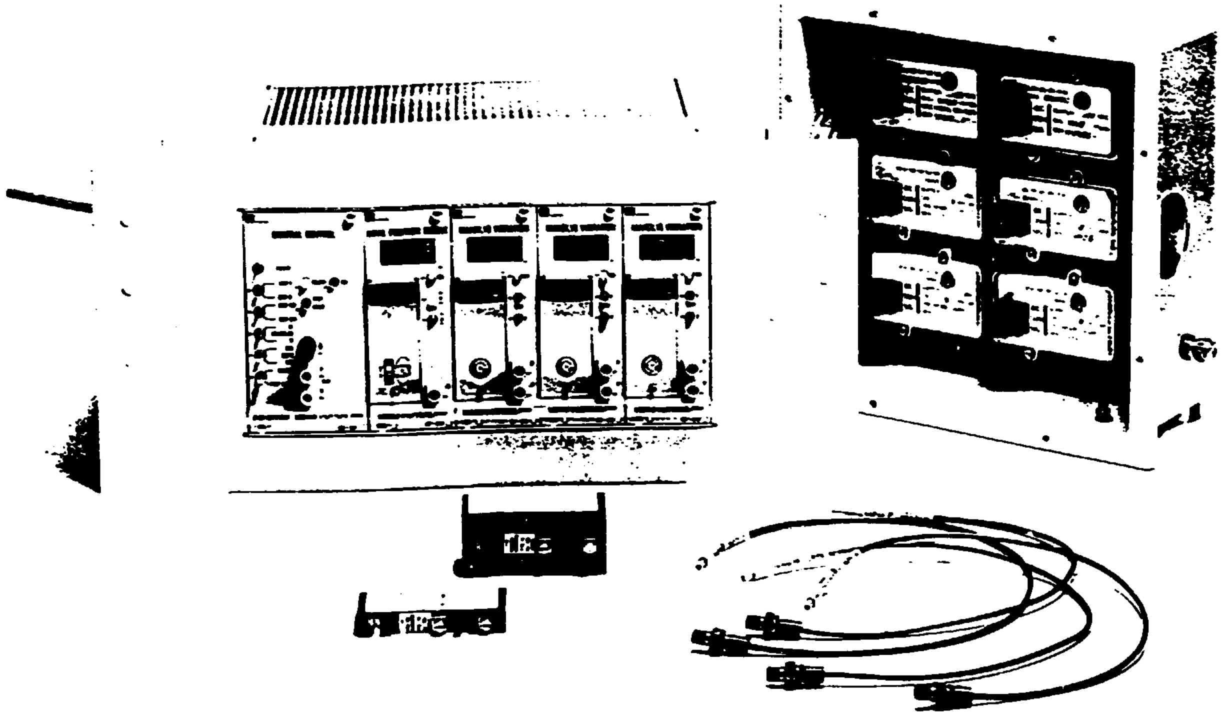


Figure 6-7. Modular machine monitoring installation with VIBROCONTROL 2000 (Schenck)

CHAPTER 7 COMPUTER AIDED PREDICTIVE MAINTENANCE

The principles of an effective predictive maintenance program are built around an organized schedule of periodic checks to measure and monitor any changing condition. When changes are detected at an early stage, maintenance action can be planned to prevent costly unscheduled downtime and lost production.

Computer aided predictive maintenance programs, utilizing a microprocessor based **data collector** and a commercial **personal computer** with commercial printer provide a new dimension in the organization and operation of an efficient system. The time required to obtain information, process it and maintain a recordkeeping system has been drastically reduced. The increased productivity justifies the use of the computer aided machinery maintenance comparing it with the manual method of obtaining and recording measurement data even in a small scale plant with a number of checking points over 100.

Computer aided predictive maintenance enable

- o cost effective recording of a practically unlimited number of measuring points on site by means of a portable measuring and data processing device called data collector;
- o modern vibration measurement methods;
- o automatic evaluation of all measuring results for early detection of machine damage;
- o powerful machine diagnosis and predictive monitoring of the machine condition.

The most sophisticated and state-of-the-art computer aided predictive maintenance systems are listed in Figure 7-1.

Company	System	Data Collector
BENTLY NEVADA	o SNAPSHOT	SNAPSHOT 2 Data Aquisition Instrument
	o TRENDMASTER	Trendmaster Field Instrument
CARL SCHENCK	VIBROCAM 1000	VIBROSTORE 41 Data Collector
IRD MECHANALYSIS	7090 Predictive Maintenance System	890 Machinery Maintenance Analyzer/Data Collector

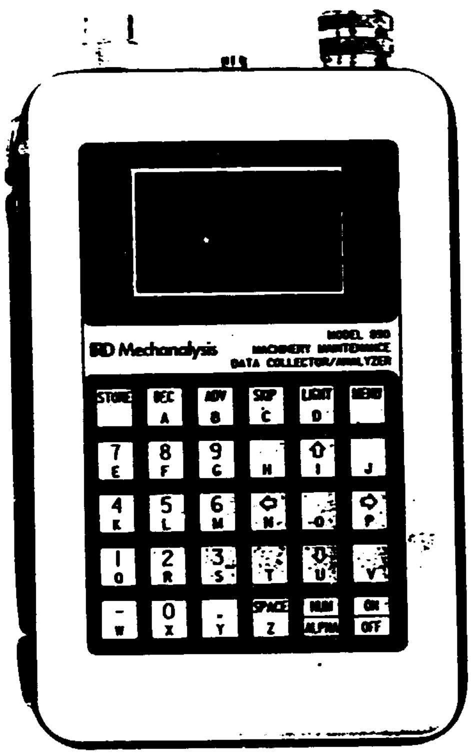
Figure 7-1. State-of-the-art computer aided predictive maintenance system.

Figure 7-2 shows the means of the VIBROCAM 1000 computer aided maintenance system developed by Carl Schenck.

Figure 7-2. A VIBROCAM 1000 system (Schenck)

7.1 THE DATA COLLECTOR

The data collector is a portable, battery operated device to provide the collection and storage of the information for predictive maintenance programs.



The Model 890 analyzer/ data collector illustrated in the Figure 7-3 effectively combines the field analysis and the predictive maintenance data collection into one easy to use system. As an analyzer, the 890 is capable of providing complete onboard FFT spectral analysis or of displaying a time waveform.

Figure 7-3. Model 890 Analyzer/Data Collector (IRD)

The frequency band amplitudes will be calculated from an independent 100 line FFT and will be the root sum of the squares of the FFT lines, within the user defined minimum and maximum frequency setpoints.

The data collector is capable of measuring displacement, velocity and acceleration from a single transducer. The data collector in general also has a separate detection circuit that operates in the ultrasonic frequency range that is specifically designed to detect the incipient defects in bearings and gears. Generally there is possible to connect the direct input of D.C. voltages that are proportional to other physical parameters.The data collector has a front panel and key-in the operating and process parameters. The unit shall have the capability to store a minimum 1000 sets of data in the memory for downloading to a personal computer for analysis and trending purposes.

7.2 THE SOFTWARE PACKAGE

It is the task of the software to

- (i) identify checking points and sequence
- (ii) collect the information from each stored measuring run
- (iii) load the accumulated data from the data collector
- (iv) archive the data
- (v) produce reports for inspection of machinery condition
- (vi) produce analysis reports for machine diagnostics
- (vii) produce a prognosis for the condition behaviour to be expected.

All the above mentioned routines are called up on the personal computer. In order to guide the operator the software is presented in a format that is easy to use and requires only minimal computer training to implement an effective predictive maintenance program.
The details and the performance of the software are described on the base of the IRD 7090 PMP software system respectively the Schenck CM-120 software package.
Simple MENU displays are presented to guide the operator through the various modes of operation. HELP selections are available on each MENU to provide additional detailed instructions.

Preparation

In accordance with the hierarchical breakdown, plant, machines and measuring position the operator can describe each checking position and lay down the parameters to be measured at each position.

The description of the measuring position should include, among other points:

- o plant name
- o machine train
- o machine description and its identification number
- o measuring direction
- o parameter
- o measuring value
- o alarm limits
- o frequency range of the spectrum
- o periodicity of check inspection.

Designing the application software the measuring points should be configurated and loaded manualy into the computer which build the Data Base of the system. Figure 7-4 illustrates an example of the configuration input.

```
POINT 00001        PLANT: PAINT ROOM                                   Record No.

        *     TRAIN  VENTILATION FAN1   1  ALARM TYPE
        *   MACHINE  MTR 15 HP ABC 10      0=Off  1=Above Alarm
        *  POINT ID  MTR-AV BT4571123      2=In Window  3=Out of Window
  DESC MOTOR VIBRATION-IPS(POS:A DIR:V) 1  FILTER OPTION
                RPM   1600                 See Help
   MEASUREMENT TYPE    1                4  MAXIMUM FREQUENCY (Hz)
           See Help                        2=500  4=2000  6=10000
    AMPLITUDE UNITS  0                  4  NUMBER OF SPECTRAL LINES
  0=English 1=Metric                       0=25  1=50  2=100
FULL SCALE AMPLITUDE  3                    3=200  4=400
           See Help                     4  NUMBER OF AVERAGES (0-31)
 CALIBRATION (mv/eu)  100.              2  AUTO SPECTRUM COLLECT
SIGNAL DETECTION TYPE  0                   0=No  1=On Alarm  2=Always
0=Peak 1=Pk-Pk 2=RMS                    0  AUTO STORE SPECTRA TO ARCHIVE
     ALARM NUMBER 1       0.60000          0=No  1=Yes
     ALARM NUMBER 2       0.80000
```

Figure 7-4. Screen display of the parameter input for one measuring position

Schedules

The schedule reports are used to determine when routes (Group of machines or equipment) are due to be measured. For each plant it is possible to define several routes with several data collectors. Check intervals can be established for each route depending on how critical the machines are to continued plant operation. All routes can be printed out for documentation purposes.

Data acquisition

By the term data aquisition is meant the communication process between the data collector and the computer.
On the one part complete routes and/or individual machines are loaded into the data collector with all of the measurement point details to guide the operator through a route of periodic checks. On the other measurements are off loaded to the host computer for storage and data processing.
The procedures required for this communication are controlled interactively.

Reports and evaluation of the measuring data

This series of reports deals with overall report about,among other things

- o all points on a routes
- o all points that exceeded the alarm setpoint on a selected route
- o the most recent data for all measurement points on a selected machine
- o a statistical report of projected amplitudes of any measurements to the end of the next check interval or 30 days

and so on.
Once problems are identified,further detailed information is available evaluating the measuring data. The more extended and intensive the evaluation is made,the more exactly the residual operating life of the machines will be determined.

Figure 7-5 illustrates the evaluation methods suitable for the purposes of predictive maintenance.

EVALUATION METHODS

Machine evaluation
Measured value list
Trend diagram

Limit value comparison
Alarm list and trend
diagram with limit value
Frequency spectrum with
limit spectrum

Prognosis
Trend diagram with limit
value and extrapolation

Fault diagnosis
Frequency spectrum

Early fault diagnosis
Cascade/Waterfall
diagram
Selective trend

Figure 7-5. Evaluation methods

For evaluating the condition of a machine you commonly examine its mechanical vibrations. In order to deepen the judgement on the machine condition partly the temperature, pressure,power consumption, and/or other parameters are examined additionally. The representation of the results can be indicated as a measured value report or as a graphic. The visual quality of the graphics can be increased by using a color monitor and a colour plotter. In order to achieve a better visual quality you should prefere the graphical representation.
A change of the measured values points out an alteration of the machine condition. It is easier to recognize it looking to a trend diagram than looking to a measured value list.

Increasing amplitude values allow the suspicion on beginning machine damages.
Figure 7-6 shows an example of the trend of the vibration severity of all measuring points of one machine.

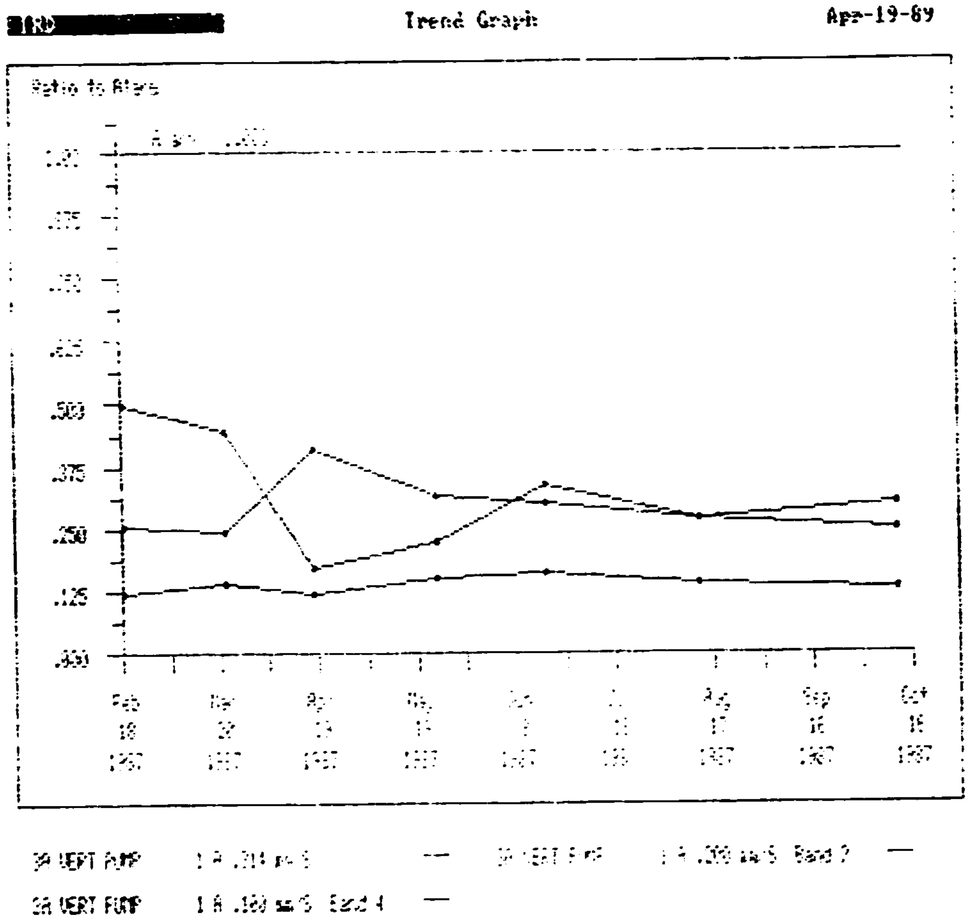


Figure 7-6. Trend diagram with alarm limit value

In order to evaluate the potential critical machine condition the measured values are compared with the predetermined limit values. The overlay of limit values is automatically announced. The limit values can be automatically transferred to the trend diagram.

Machines have to be shut down when the current condition exceeds predetermined limit values. For the determination of the left residual operating time,respectively of the optimal maintenance time, the software package automatically extends the trend course (extrapolation) so far till it crosses the limit value. At this point of intersection the optimal time for repair work is reached. The residual operating time is the time between the instant of examination till the reaching of the limit value.

Whereas broad band vibration parameters can be used to evaluate the vibration behaviour of a machine, the frequency spectrum, however, goes beyond this in providing information for targetting the maintenance work. Every line in the spectrum allows one to draw conclusions about possible causes of the vibrations.Observation of the trend of individual lines makes it possible to recognize incipient damage even when the vibrational behaviour of the machine is still good. In order to be able to evaluate the high information content of the frequency spectrum as fully as possible,the following spectrum reports are available:

- o Single spectrum display
  Detailed display of the latest spectrum for a selected measurement point.
- o Baseline spectral comparison.
  Displays the current spectrum, baseline spectrum and a differences spectrum for a selected measurement points to highlight any significant changes in the spectral data.
- o Selected spectral comparison.
- o Superposition of several spectra in a cascad, in other terms waterfall diagram (Figure 7-7).
- o Presentation of trends of spectra lines.

7.3 COMPUTER HARDWARE

An IBM XT, AT or PS/2 compatible computer, fulfilling the following specification, is suitable as the working tool:

- o at least 640 Kbyte user memory;
- o at least 40 Mbyte hard disc as auxillary storage;
- o IBM Asynchronous communications adapter (RS 232C serial port);
- o color graphics adapter and color graphics - display with high resolution performance (CGA or EGA, respectively VGA for PS/2);
- o IBM or compatible Dot addressable graphics printer with cable and printer adapter;
- o IBM PC-DOS/MS-DOS Ver 3.00 or higher operating system.

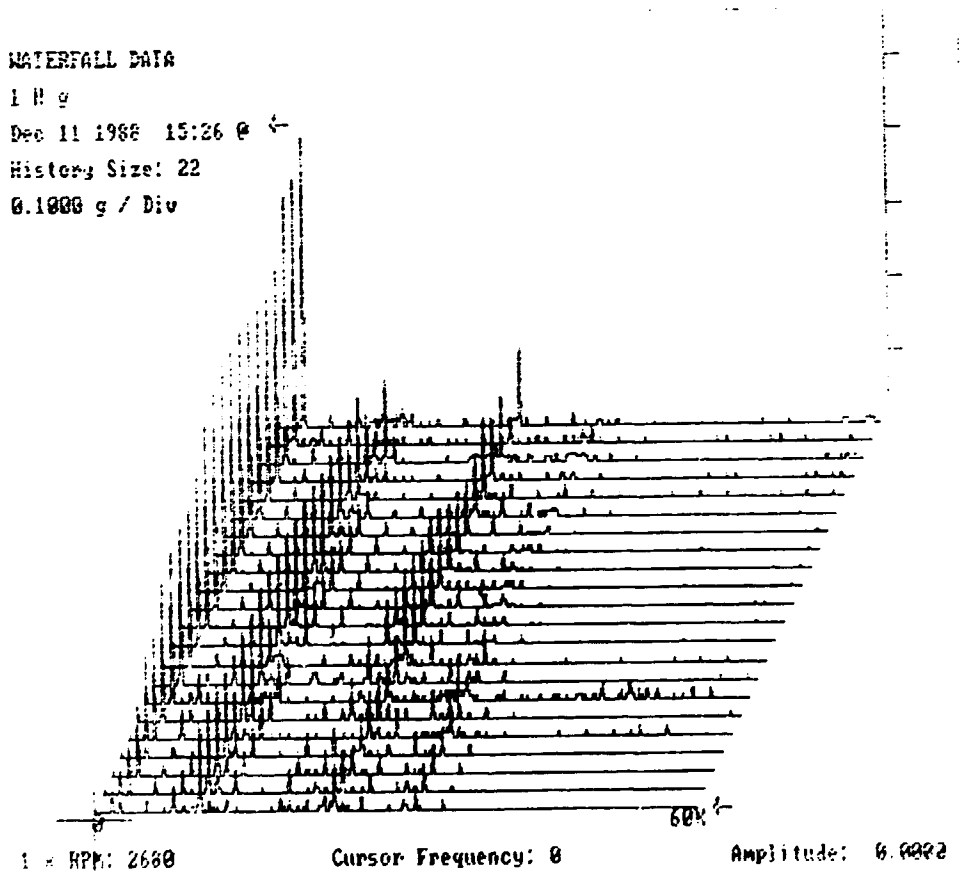


Figure 7-7. Cascade/Waterfall diagram

APPENDIX

GUIDELINES AND STANDARDS FOR MACHINE MONITORING

(i) General

ISO 2041:1975	Vibration and shock - Vocabulary
ISO 4865:1983	Vibration and shock - Methods for analysis and presentation data

(ii) Requirements for measuring methods and instruments

ISO 2954:1975	Mechanical vibration of rotating and reciprocating machinery. Requirements for instruments for measuring vibration severity
ISO 7919/1:1986	Mechanical vibration of non-reciprocating machines.Measurements on rotating shafts and evaluation. Part 1: General Guideliness
ISO 5347:1987	Methods for the calibration of vibration and shock pickups
ISO 5348:1987	Mechanical vibration and shock - Mechanical mounting of accelerometers
API 670	Vibration, axial position and bearing temperature monitoring systems
API 678	Accelerometer based vibration monitoring systems

(iii) Limit values for permissible vibration during acceptance tests

API 610	Centrigufal pumps for general refinery services
API 611	General purpose steam turbines for refinery services

API 612	Special purpose steam turbines for refinery services
API 616	Type H industrial combustion gas turbines for refinery services
API 617	Centrifugal compressors for general refinery services
API 619	Rotary type positive displacement compressors for general refinery services
API 672	Packaged, integrally geared, centrifugal plant and instrument air compressors for general refinery services
API 673	Special purpose centrifugal fans for general refinery services

(iv) **Limit values for permissible vibration during operation**

ISO 2372:1974	Mechanical vibration of machines with operating speed from 10 to 200 rev/s - Basis for for specifying evaluation standards
ISO 2373:1974	Mechanical vibration of certain rotating electrical machines with shaft heights between 80 and 400 mm - Measurement and evaluation of the vibration
ISO 3945:1985	Mechanical vibration of large rotating machines with speed range from 10 to 200 rev/s - Measurement and evaluation of vibration severity in situ