A

REPORT ON INDUSTRIAL TRAINING

AT

INDIAN OIL CORPORATION LIMITED

GUWAHATI REFINERY

ON

" VIBRATION MONITORING OF EQUIPMENTS"

Submitted by:

Hrishikesh kalita

B.Sc (Plant Maintenance & Operations)

University Of Petroleum & Energy Studies

DECLARATION

I hereby declare that the project report entitled

"VIBRATION MONITORING OF EQUIPMENTS"

Submitted in the partial fulfillment of the requirement

for the degree of

B.Sc. (Plant Operation And Maintenance)

TO

University Of Petroleum & Energy Studies, Dehradun, is our original work & not submitted for the award of any other degree, diploma, fellowship or other similar title or prizes.

Hrishikesh Kalita
B.Sc. (V Sem.)
U.P.E.S, Dehradun
Uttarakhand- 248001

इंडियन ऑयल कॉर्पोरेशन लिमिटेड
रिफाइनरी प्रभाग - गुवाहाटी रिफाइनरी
नूनमाटी, गुवाहाटी - 781020 , असम

Indian Oil Corporation Limited
Refineries Division : Guwahati Refinery
Noonmati, Guwahati - 781020, Assam.
Fax : 0361-2657250, 2957251
Tel : 0361-2597777
Gram : OILREFIN
Internet Site : www.indianoilcorp.com

रिफाइनरी प्रभाग
Refineries Division

To Whom It May Concern

This is to certify that **Mr. Hrishikesh Kalita**, a student of 3rd Year (5th Semester), **B.Sc. (Plant Operations and Maintenance), University of Petroleum & Energy Studies, Dehradun,** has undergone Vocational Training at our Refinery from **4th August to 31st December 2007.**

During Training, he has been found to be punctual and diligent. After completion of the Training he has submitted a training report on **"Vibration Monitoring of Equipments".**

Wishing him all success in life.

December 31, 2007

(Kailash Chandra)
Sr. Training Manager

KAILASH CHANDRA
Sr. Training Manager
I.O.C.L.,Guwahati Refinery
Noonmati, Guwahati-781020

ACKNOWLEDGEMENT

At the very first moment, I would like to express my sincere thanks to Mr. S.K.M Barua (Senior Training & Development Officer), & Mr. K. Chandra (Senior Training & Development Manager), Guwahati Refinery, for giving an opportunity to do our project at IOC, Guwahati.

I express our gratefulness to Mr. J.C. Boro, SMLE(CDU) for his kind co-operation.

Specially, I extend our sincere and humble thanks to Mr. J. Singh, MLE(DCU) of Guwahati Refinery for his valuable contribution and guidance as our original guide during the period of our project study.

And finally, I would like to express our thanks to Mr. S. Mondal, SMLE(HGU/HDT) and all those persons who helped us directly in my project work.

HKalita

Hrishikesh Kalita
B.Sc. (V Sem.)
U.P.E.S, Dehradun
Uttarakhand- 248001

CONTENTS

1.0 INTRODUCTION

1.0 INTRODUCTION

Rotating machinery constitutes a major percentage of equipment and the continued, safe and productive operation of the same helps to increase on stream days of the refinery.

We are quite familiar with preventive maintenance is carried out after a specified no. of running hours, the same being decided either on the basis of manufacturer recommendation or on the basis of statistical records but preventive maintenance is truly effective only where the failure mechanism of component is clearly time dependent, the component being expected to fail within the life of the equipment and where the total cost of such work are substantially less than those of breakdown maintenance.

Preventive maintenance when applied to all type of equipments has the following disadvantages:

1. Since the failure mechanism is presumed to be time dependent, the frequency of maintenance so determined is not accurate.
2. As preventive maintenance does not envisage assessing the condition of the machine before undertaking maintenance, there is every chance of the equipment being under maintained or the over maintained.

In view of the above, it was realized that the condition of the equipment should be assessed for optimum maintenance the same should form the basis for scheduling maintenance.

By monitoring the condition and/or performance of an equipment, the probabilistic element in failure prediction is indeed almost eliminated, the effect of failure minimized and the equipment life maximized.

Hence, one of the major benefits of this policy is that the resulting corrective maintenance can in most cases be scheduled.

In this regard, the major causes of equipment failure can be classified as given below:

1. Wear at the interfaces between parts with relative motion.
2. Deformation or crack growth due to over stressing.
3. Over heating due to over load, loss of lubricant or coolant or the failure of insulation.
4. Corrosion and erosion.

These can be monitored by measuring mainly vibration, shock pulse, sound level, temperature, wear particles in the lubricant, properties of lubricant etc.

With developments taking place in the fields of electronics and instrumentation, the measurement of these parameters can now be made in a more precise manner and the impending failures predicted, which has paved the way for the new system of maintenance termed "predictive maintenance"

Thus condition monitoring enables us to find out impending machine failures without bringing down the equipment or opening the equipment unnecessarily.

It helps cost effectiveness and avoids expenditure by locating some incipient machinery malfunctions. Hence the objective of having fewer unexpected failures, maximizing MTBF (Mean Time Between Failures), maximizing runtime between two shutdowns can be achieved.

However, the following conditions should be fulfilled before implementing a predictive maintenance programme.

1. The existence of failures which do not occur at the regular intervals
2. These failures are either a safety hazard or incur significant loss in terms of downtime and cost of breakdown maintenance.
3. A monitoring method exists that can give sufficient advance warning of the impending failure for the maintenance/production system to act to avoid failure.
4. The monitoring and corrective maintenance costs are less than the downtime cost.
5. The monitoring method is compatible with the existing procedures in the organization.

2.0 VIBRATION

2.0 VIBRATION

2.1 DEFINATION

Vibration is simply the motion of a machine or machine part back and forth from the position of rest. A simple way to show vibration is to follow the motion of a weight suspended on the end of a spring as shown in figure which is typical example of all machines.
The cause of vibration must be a force, which is changing in either its

Direction or amount. It is the force, which causes vibration, and the resulting characteristics will be determined by the manner in which the force is generated. Hence cause of vibration has its own typical characteristics.

2.2 PARAMETERS AND UNITS

A lot can be known about machinery condition and malfunctions from the vibration characteristics. 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 as shown in figure

- The motion of the weight from its neutral position to the top limit of travel back through the neutral position to the bottom limit of travel and its return represent one cycle of motion
- The time required for completing one cycle of vibration is called a period of vibration as shown in figure
- The number of cycles repeated in a given interval of time normally minute
- Displacement at any instance of the cycle is the distance traveled by the vibrating part from the neutral position as shown in fig. Displacement is normally expressed in microns (0.001mm) in metric units and in mils (0.001 inch) in British units
- The velocity of the vibrating part is constantly changing as displacement changes as shown in figure. This is normally expressed in mm/sec in metric units and inch/sec in British units
- Acceleration is another important characteristic of vibration and peak value is measured which is normally expressed in 'g's. 'g' is the acceleration produced by the force of gravity at the surface of earth

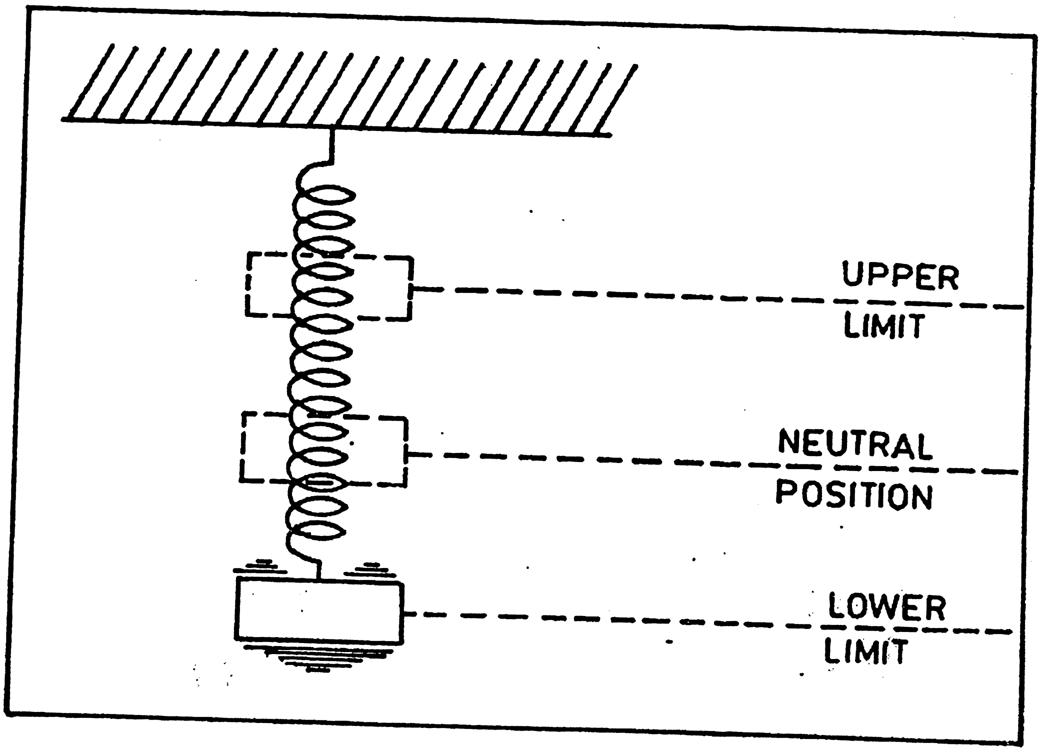


FIG.2·1 VIBRATION OF A SIMPLE SPRING MASS SYSTEM

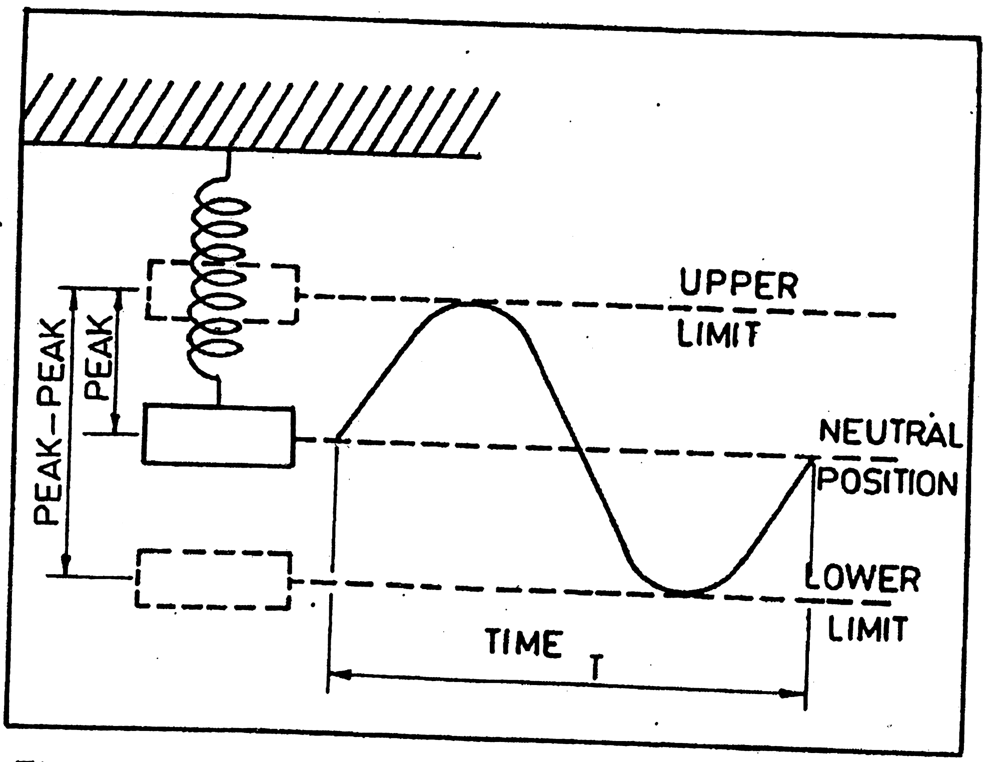


FIG.2·2 DISPLACEMENT VS TIME

(g=980.66 cm/sec square)

- Phase is another important characteristic of vibration, which is defined as the position of vibrating part at a given instance with reference to a fixed point or another vibrating part.

2.3 WHEN TO MEASURE WHAT

The displacement, velocity and acceleration of vibration are referred to as a amplitude of vibration and are measured to determine the amount of severity of vibration. Vibration velocity is directly proportional to the displacement and frequency.

V peak = 52.30 D (F/1000) x 10^{-3} ----------------------------------(1)
V peak = Vibration velocity in mm/s peak
D = 'peak-peak' displacement in microns
F = Frequency in CPM

Vibration acceleration is directly proportional to the displacement and frequency squared.
g peak = 5.6 D$(F/1000)^2$ x 1/10000 ----------------------------------(2)
g peak = vibration acceleration peak
D = 'peak-peak' displacement in microns
F = Frequency in CPM.

The forces which cause vibration are generated through the rotating motion on the machine parts and these forces change in amount and direction as the rotating part changes its position with respect to rest of the machine hence the frequency of the vibration produced would be related to the rotating speed of the part which has the trouble. It is also important to recognize that, as we would be seeing in the succeeding sections, different machinery troubles cause vibration at different frequencies, which make it possible for us to identify the nature of the problem.
Hence it is essential to know the vibration frequency to pin point the malfunction and analyzing machine's vibration.
Mechanical vibration in rotating and reciprocating machinery (ISO 2372)
In this international standard the term vibration severity defined as a comprehensive and simple characteristic unit for describing the vibratory state of machine, is used as the basis of classification and on the basis of theoretical consideration and practical experience the root mean square value

of vibration velocity has been chosen as the unit of measurement for indicating vibration severity.

2.4 FILTER OUT AND FILTER IN

The vibration of a machine may not always generate a harmonic motion as the weight suspended from the spring does. The machinery vibration is mostly complex; consisting of components at many frequencies i.e. the machinery does not vibrate at a single frequency but vibrates at many frequencies, the frequencies decided by the troubles causing vibration. The total amplitude of vibration is the vector sum of vibration at different frequencies. This is termed as 'filter out' amplitude

When the vibration is complex, we will have to analyze the vibration to know the amplitude at different frequencies of interest. For achieving this, we have vibration analysers, which incorporate a tunable filter, the details of which we would see in the following pages. By tuning the filter, vibration amplitude at different frequencies can be measured. This is termed as 'filter in' amplitude.

VIBRATION SEVERITY STANDARDS

The following severity have given in the succeeding pages and a comparative chart of their feature has been given in table 2.1

FEATURES OF SEVERITY CHARTS

S. NO	NAMEOF CHART	FREQUENCY RANGE	TYPE OF MEASUREMENT	PARAMETER	FILTERED/ UNFILTERED	REMARKS
1.	T.C Rathbone's machinery vibration severity chart	80-6000 CPM	Bearing casing / structure of machine	Displacement	Filtered	---
2	General machinery vibration severity chart	100-100000 CPM	Bearing casing / structure of machine	Displacement velocity	Filtered Unfiltered	Applicable for rigidly mounted machines or machines bolted to a fairly rigid foundation. For applying to machines mounted on resilient vibration isolators, multiply the vibration amplitude values by two except at high frequencies.
3.	M.P. Blake's vibration severity chart	100-10000 CPM	Bearing casing	Displacement velocity acceleration	Filtered Unfiltered Unfiltered	Specially meant for vibration at frequency 1xRPM
4	Vibration acceleration general severity chart	18000-600000	Bearing casing	Acceleration	Filtered	--
5.	Hydraulic institute vibration classification for horizontal and vertical non clog centrifugal pumps.	60-5200	Bearing casing for vertical pumps on top motor bearing	Displacement	Filtered	Applicable for vibration at frequency 1x RPM

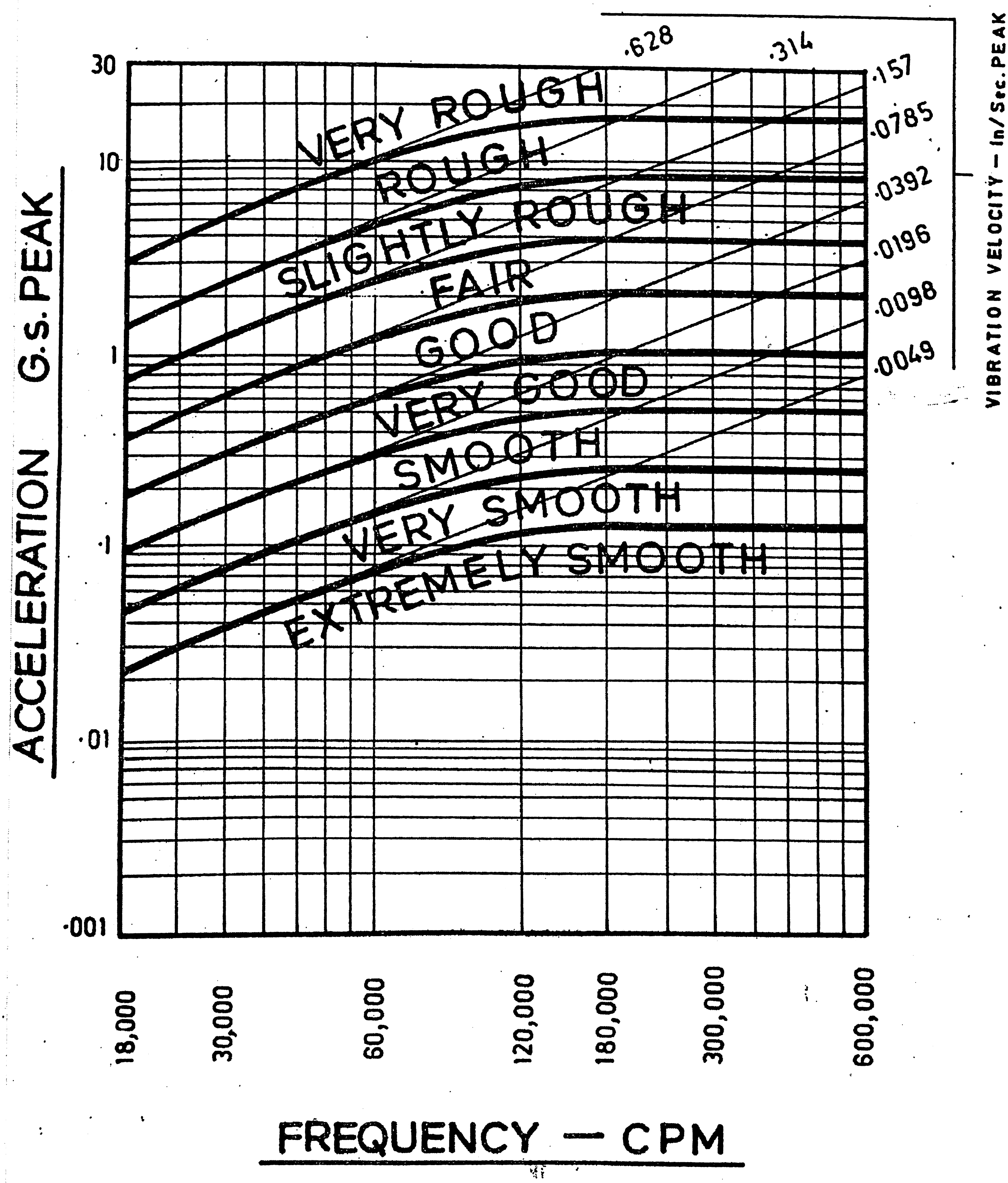


FIG 2·7 VIBRATION ACCELERATION GENERAL SEVERITY CHART

3.0 VIBRATION MEASUREMENT

3.0 VIBRATION MEASUREMENT

3.1 CLASSIFICATION OF INSTRUMENTS

Electronic instruments for measuring machinery vibration are generally classified as meters, monitors and analysers.

- A typical vibration meter is a portable device consisting of a selectable gain circuit for ranging an integrator which can be switched into transform the velocity signal from the pickup to displacement and a meter for reading the amplitude being measured. This is used for periodic vibration checks on machinery to determine the overall vibration level.
- A vibration monitor is similar in function to a vibration meter but is permanently or semi permanently installed to provide continuous protection to equipment from excessive vibration. Vibration monitors normally incorporate alarm/trip relays in conjunction with present vibration levels to warn/stop the machine when the vibration increases beyond the present level.
- A vibration analyzer includes a tunable filter for separating the individual frequencies of complex vibration. This can measure and record all vibration amplitudes at different frequencies.

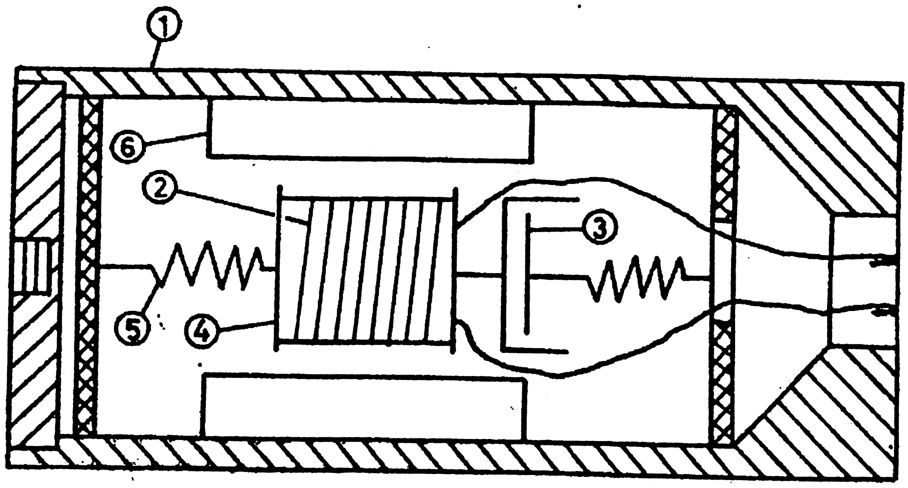
3.2 TRANSDUCERS

Regardless of which type of instrument is used to measure the vibration, the heart of the measurement system is the pickup or transducer. A transducer is simply a sensing device which converts mechanical vibration into an electrical signal. The transducers commonly used are seismic velocity pickups, accelerometers and non-contact pickups. Each pickup has distinct advantages in certain applications but they all have limitations too. Hence, it is important to select the transducer best suited for the job.

3.2.1 Seismic Velocity Pickup

Construction:

A schematic diagram of a typical seismic velocity vibration pickup and its parts is given in fig.3.1. The mass of the system consists of a coil of very



1. PICKUP CASE
2. WIRE COIL
3. DAMPER
4. MASS
5. SPRING
6. MAGNET

FIG. 3·1 BASIC CONSTRUCTION OF A SEISMIC-VELOCITY VIBRATION PICKUP.

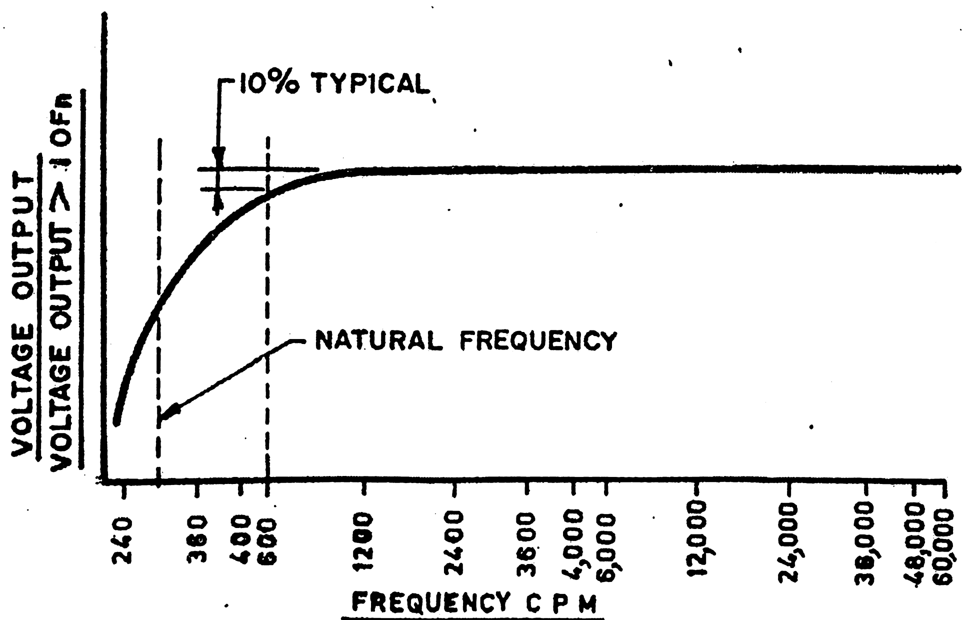


FIG.3·2 : TYPICAL VELOCITY PICKUP SENSITIVITY fine wire supported by springs. A permanent magnet is firmly attached to the case of the pickup and provides a strong magnetic field around the suspended coil.

Principle Of Operation:

When the case of the velocity pickup is attached or held against a vibrating part, the permanent magnet attached to the case follows the vibration motion. When a conductor is moved through a magnetic field or if a magnetic field is moved past a conductor, a voltage will be induced in the conductor. The amount of voltage generated in the conductor is directly proportional to the relative velocity between the conductor and the magnetic field if the length of the conductor and strength of magnetic field are fixed. Since the length of coil and strength of magnetic field are fixed, the voltage generated changes proportionate to the change in the velocity of the vibrating part.

Range of operation:

The springs that support the mass have very low stiffness and thus the spring mass system has a low natural frequency. For frequencies above the natural frequency of the spring mass system, the coil remains stationary with respect to inertial references. Below the damped natural frequency of the spring mass system, the coil no longer stimulates a fixed point in space. Hence this pickup is used for measuring vibration velocity at frequencies greater than the natural frequency of the spring mass system. velocity pickups have limitations on the measurable max and min. vibration amplitudes.
Nomographs are supplied by the manufacturer, which show the operating range of the pickup in the terms of maximum and minimum frequency as well as the max and min. amplitude values.

Sensitivity:

Sensitivity of a velocity pickup is expressed as the maximum voltage output at the point of maximum velocity i.e. millivolts peak/mm/second peak. Sensitivity can also be expressed for rms value.

The sensitivity vs frequency response curve as shown in fig.3.2 of a velocity pickup is limited at low frequencies by the critically damped first natural frequency; at high frequencies its response is limited by the amount of motion necessary to overcome the inertia of the system as well as by the presence of higher order natural frequencies.

Advantages :

- Rugged construction.
- Easy to apply and can be hand held.
- Has a high output signal level.
- Does not require any external source of power.
- By electronically integrating, vibration amplitude can be measured in terms of displacement also.
- Can be mounted in any position.
- Cable length upto 300 meters can be used, if the frequency of vibrations being measured are low and the pickup cable is well isolated from electrical noise generators.

Disadvantages:

- Alternating magnetic field present on large AC motors or generators, cables carrying large AC current, induce a voltage in the coiled conductor of the velocity pickup in the same manner as the vibration of pickup.
- Output signal decays exponentially below 600 CPM.
- Unsuitable for monitoring gear trains and roller element bearings.
- Prone to wear as a result of mechanical movement.

3.2.2 ACCELEROMETER

Since acceleration is a function of displacement and frequency squared, accelerometer is extremely sensitive to vibration occurring at high frequency squared, accelerometer is extremely sensitive to vibration occurring at high frequencies such as that originating from gears or antifriction bearings.

Construction:

An accelerometer consists of a seismically mounted mass, bearing on a force-sensing element such as a piezoelectric crystal or ferroelectric materials. Referring to fig. 3.3, the accelerometer consists of a mass rigidly attached to the piezoelectric element.

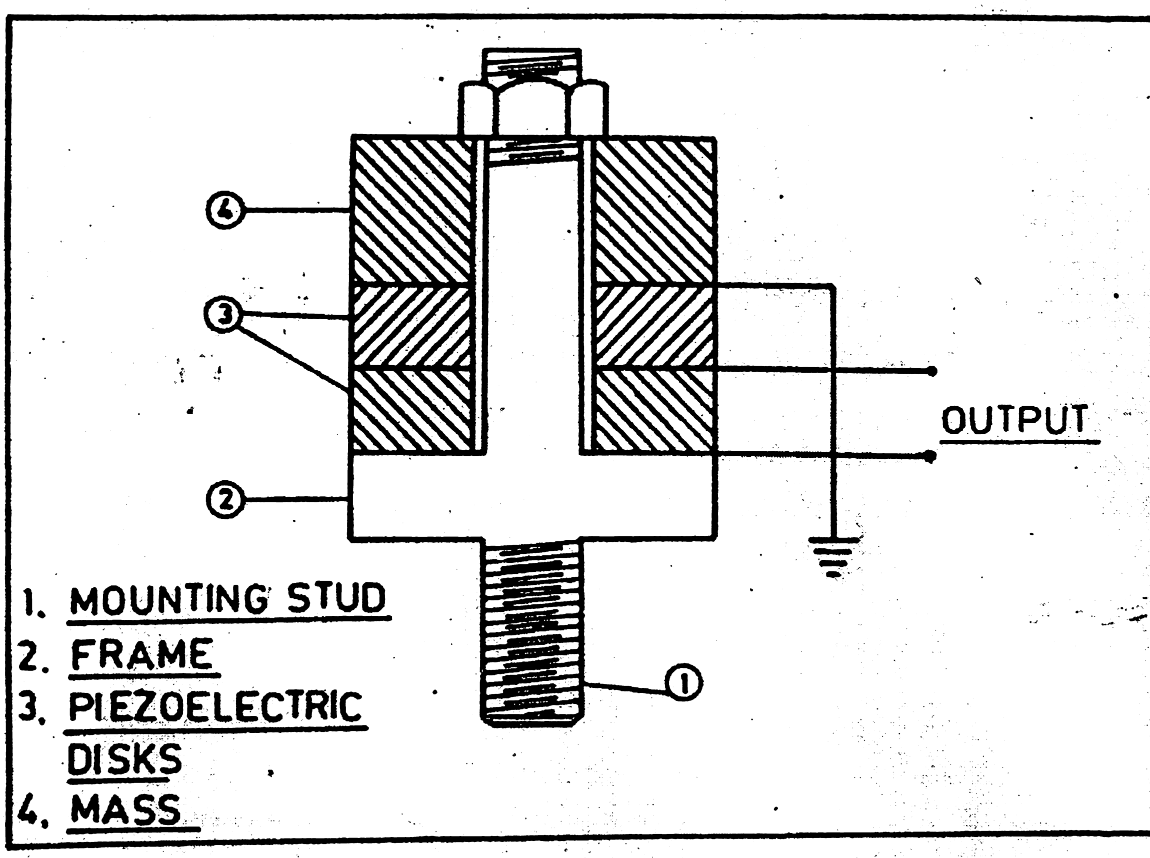


FIG. 3·3 SCHEMATIC DIAGRAM OF AN ACCELEROMETER PICKUP

Principle of operation:

When this transducer is mounted on a machine, the machine causes an inertial reaction of the mass clamped to the top of the piezoelectric disk. The force on the piezoelectric disk proportional to the acceleration of the mass, produces a corresponding charge Q between the two faces of the disk. The piezoelectric element acts as a capacitor and the charge variations appear as voltage variation across suitable high resistance sensing circuit.

Sensitivity:

Sensitivity of the accelerometer is the output voltage per unit of acceleration. This is normally expressed as millivolt per 'g'. Since the output sensitivity of accelerometers is relatively low as compared to seismic velocity pickups, an amplifier is normally used to obtain a usable signal. Accelerometer are available either with built-in amplifiers or with provision for using external amplifiers.

Range of operation:

Accelerometer function accurately for frequencies of vibration below the natural frequency of the spring mass system. There are several factors, which affect the usable frequency range of accelerometers, the natural frequency of the piezoelectric elements being the most important. The piezoelectric material used in accelerometers is very stiff and the natural frequency for commonly used piezoelectric accelerometers varies from 6,00,000 CPM upwards. Unlike the velocity pickup, an accelerometer operates below its first natural frequency. The frequency response of a typical accelerometer is about 1/4th of the natural frequency.

Advantages:

- Small, light weight and rugged.
- Operates over wide frequency range.
- Can withstand high vibration levels.
- Self-generating.
- Suitable for a wide temperature range.
  (-250degree cel. To 450degree cel with cooling).
- High resonant frequency.

Disadvantages:

- Special low noise cable and proper shield groundings are required for accelerometers with external amplifier.
- Cable length affects the maximum usable amplitude that can be measured. Cables longer than 60 meter may result in distortion of the signals in excess of 10gm depending on the frequency.
- The output is at a very high impedance to make the signal suitable for use with display, analysis or monitoring equipment.
- Incident thermal radiation can create a large frequency output.
- Base strain, caused by distortion of the accelerometer base by mounting it on an irregular surface, can significantly alter sensitivity.
- Does not give reliable results if displacement is calculated by double integration.
- Not suitable for low frequency measurement.
- Spurious signal may arise due to dirty or badly fitting cable connections.

3.2.3 NON-CONTACT PICKUP:

Many modern high speed machines such as turbines, centrifugal compressors consists of relatively light weight rotors mounted in a heavy massive cases and rigid bearing .due to the weight and stiffness of massive machine case and bearings, there is little outward evidence of rotor or shaft vibration. Hence it becomes necessary to measure shaft vibration relative to the bearing housing in this type of machinery. Non-contact pickups are used in this type of applications.

Construction:

A non-contact pickup system is transducer used for vibration and position measurement that does not require contact with the target whose position or vibration is being measured. The non-contact pickup system consists of pickup, connecting cable and signal sensor. Fig 3.4 consist of a typical non contact pick up set up for radial vibration measurement .the tip of the pickup contains a coil of wire which is connected to a signal sensor. The signal sensors consist of an electronic oscillator circuit, a filter circuit and for system an amplifier circuit. Power to operate the signal sensor is provided by the vibration monitor or read out instrument.

Principle of operation:

The non-contact pick works on eddy current principle. The pickup coil and connecting cable are part of oscillator circuit, which generates a high frequency current in the coil. This high frequency current produces a magnetic field around the pickup coil. Any conductive material such as steel brought in proximity to the coil cuts the lines of magnetic flux producing eddy currents in the coil as an impedance load change which alters the operating time of the oscillator this high frequency signal is then detected, leaving a signal that is proportional to the distance from pickup coil to the target conductive material. This distance is referred to as gap. Variations in the oscillator output due to variations in gap caused by vibration produces an output proportional to the vibration displacement. When there is no vibration the output is constant DC voltage directly proportional to gap. When there is vibration the outputs are

(i) A D.C voltage proportional to the average gap
(ii) An A.C voltage proportional to the vibration.

Sensitivity:

The slope of the curve which gives the distance between the tip of the distance between the tip of the pickup and target material and the change in signal sensor output voltage, expressed as millivolts per mil is the sensitivity of the non contact pick up system when used as a game measuring device.
The linearity of the system may be defined as the portion of the output voltage versus gap within which the slope does not vary by more than $\pm 5\%$. A typical sensitivity calibration curve has been given in fig 3.5. The set point for a non contact pickup system is selected at the mid point of the specified linear range so as to provide maximum measurement range in either direction.

Advantages:

- Can measure the motion of the shaft relative to the bearing if mounted on the bearing housing.
- Shafts orbits can be plotted or displayed on the oscilloscope.
- Can measure axial displacement of the rotor.
- Can be seismically supported for measuring absolute shaft vibration.
- Frequency response 0-300000 CPM.

- The oscillator-demodulator output can be transmitted over a distance of 300 meters.
- Unaffected by lubricating oil and most gases.

Disadvantages:

- Susceptible to changes in shaft surfaces e.g.-mechanical runout , electrical runout, coating of different conductivity (chrome plating), surface finishing etc
- Requires a separate source of power to operate
- Susceptible to induced voltage from other conductors.
- Requires recalibration, if the material of construction of the rotating part, being observed, is changed.

3.3 DIRECTION OF MEASUREMENT

Vibration readings are normally taken with the pickup in horizontal, vertical or axial direction. Vertical and horizontal readings are taken with the pickup perpendicular to the shaft centre line of the machine and are referred to as radial vibration measurements. Axial vibrations reading are those taken with the pickup parallel to the central line of the shaft. Points and directions of measurements recommended in BS 4675 part I: 1976 (ISO: 2372) is given below:
"Points of measurement should preferably be chosen where the vibration energy is transmitted to the resilient mounting or other parts of the system. For machines, which include rotating masses, the bearings and mounting points of the machine are preferred points of measurement. In individual cases it may be advisable to chose other points of measurements for e.g. at the marked points in fig. 3.6. Measurement may be made in the direction of 3 mutually perpendicular axis"
Hence, to measure the vibration we must place the pickup on or as near as possible to the machine's bearings on the machine since the vibration forces are being transmitted to the casing only through the bearings. Vibration pickup measures only the vibration occurring in the direction in which it is pointed. Hence it is necessary to identify the position the pickup for all vibration measurements.

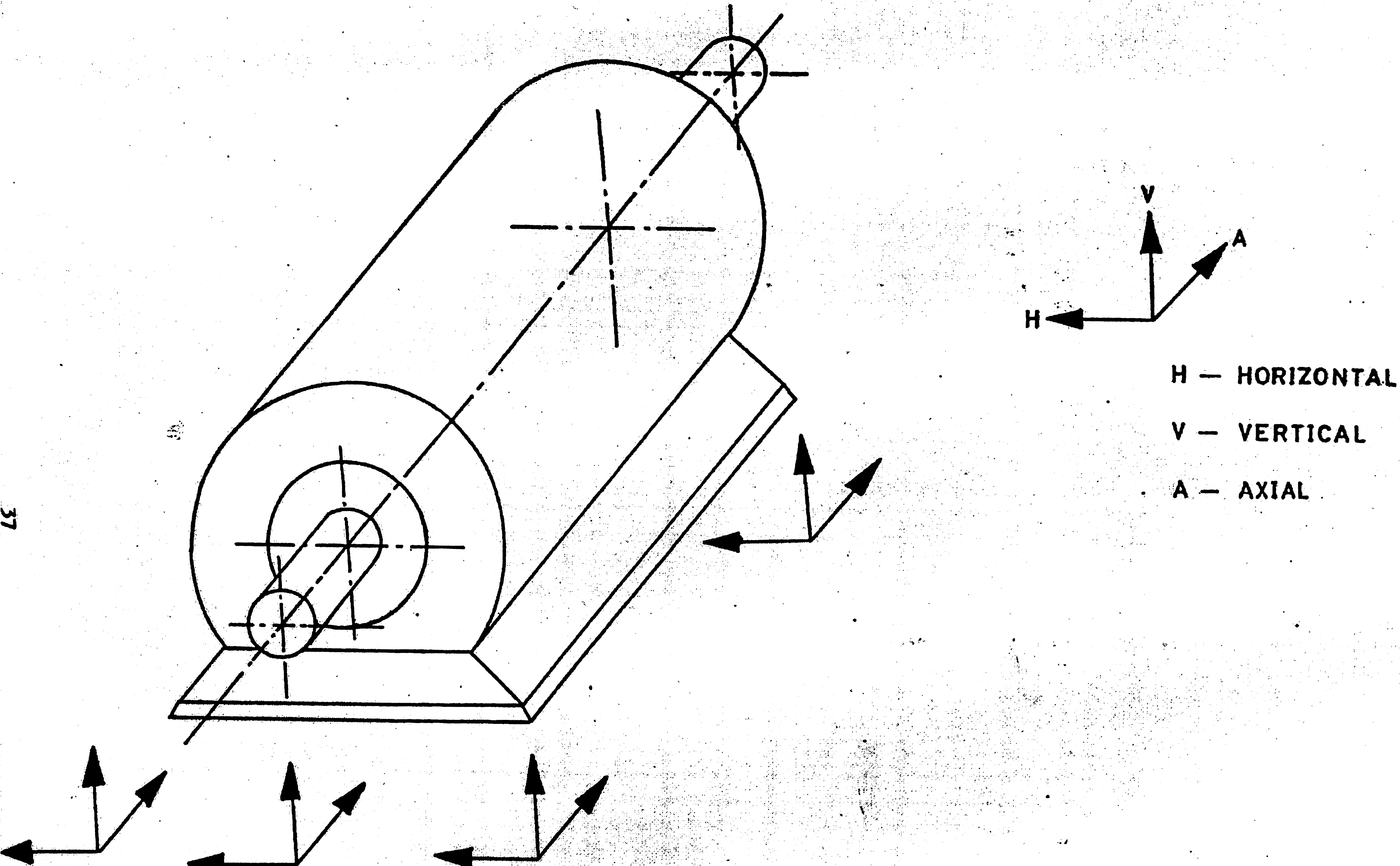


FIG.3·6 SUGGESTED MEASURING POINTS BS 4675 PART I:1976 (ISO 2372)

3.4 TYPE OF MEASUREMENT

In the preceding paragraph, we mention about placing the vibration pickup near the bearings. Since our objective of monitoring the vibration of any machinery is to detect increase in vibration level so as to get the prior warning of the impending trouble, it is quite desirable to measure that vibration which reveals the most significant level increase when problems occurs. However vibration in rotating equipment can be measured by the following 3 types:

i) **Absolute casing (bearing) vibration**
ii) **Absolute shaft vibration.**
iii) **Related shaft vibration.**

There are numbers of factors to be considered for selecting the type of vibration measurement to ensure accurate, reliable indication of machinery condition with minimum effort, time and expense.

Important factors to be considered are:

i) Type of rotor bearings
ii) Rotor speed
iii) Rotor/case mass ratio
iv) Response to specific mechanical defects
v) Instrumentation cost
vi) Ease of installation
vii) Frequency response of available transducers.
viii) Durability and resistance of available transducers to environmental factors like temperature, chemicals etc

There are some general factors regarding each type of measurement of vibration and the advantages and shortcomings of each type. Some of them are:

3.4.1 GENERAL FACTORS

Let us take the case of an equipment fitted with rolling element bearings. Characteristically, the shaft and bearing will have nearly the same vibration. The waveforms have been given in Fig.3.7.a., which appear almost identical. Relative shaft vibration with respect to the bearing would detect only the small amount of radial clearance in the bearing. Hence the choice is between the bearing vibration and absolute shaft vibration.

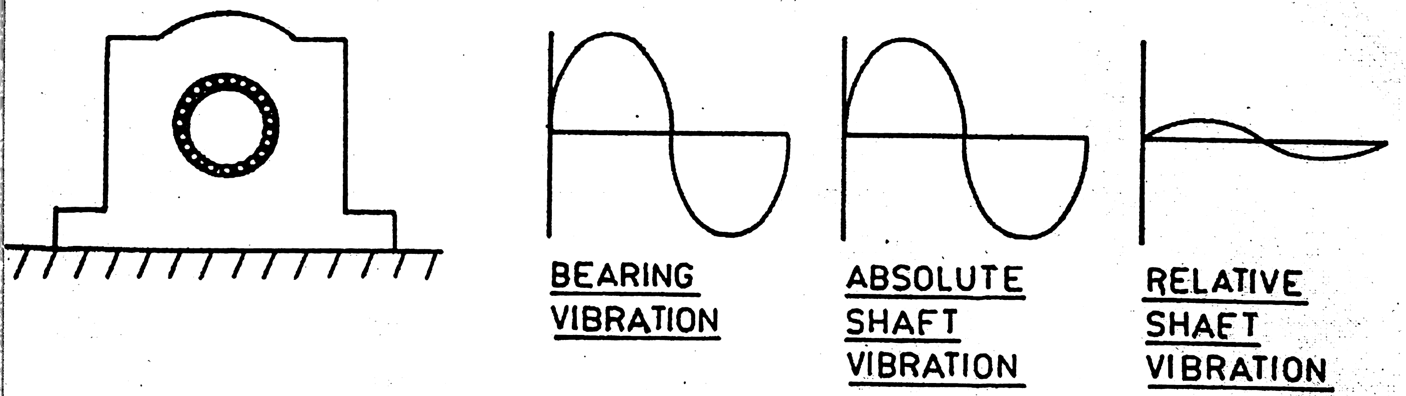


FIG. 3·7·a ROTOR ON ROLLING ELEMENT BEARING

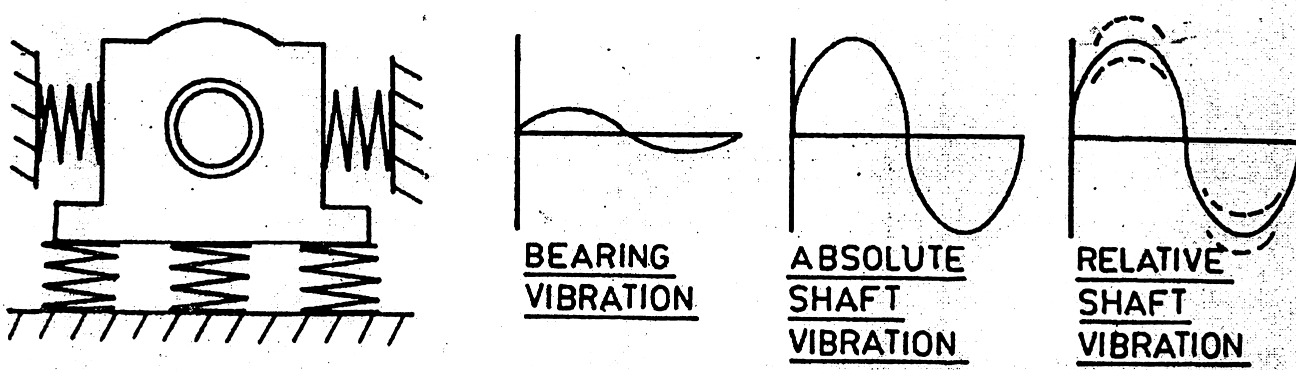


FIG. 3·7·b LIGHT ROTOR ON JOURNAL BEARING

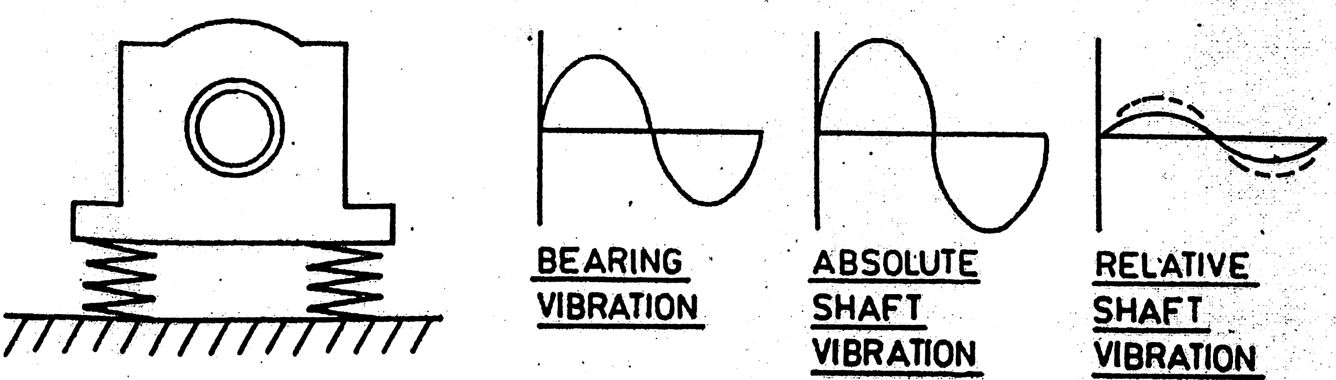


FIG. 3·7·C LOW TO MODERATE CASING STIFFNESS

Let us consider the case of lightweight rotor supported by plain journal type bearings installed in a massive and rigid casing (e.g. centrifugal compressor, turbine). Here the large massive casing remains nearly stationary even though the rotor shaft may be vibrating excessively within the clearances of the bearings. The choice would be between absolute shaft vibration or relative shaft vibration as shown in fig.3.7.b.

Let us now consider the case of a machine where the casing has low to moderate stiffness (e.g. fans, blowers, low speed pumps, machines installed on resilient vibration isolators). It can be seen from the waveforms given in fig. 3.7.c.that bearing vibration is only slightly less than absolute shaft vibration while the relative shaft vibration is comparatively low. Hence the choice would be between bearing vibration and absolute shaft vibration.

3.4.2 BEARING VIBRATION

Advantages:

i) Bearing vibration measurements are normally made using seismic velocity pickups or accelerator transducers. These transducers offer ease of installation and we can select from the wide range available.

ii) We can select any of the parameters (displacement, velocity, acceleration) for measuring bearing vibration.

Shortcomings:

i) It is not possible to rely exclusively on bearing vibration to detect significant changes in machine condition, especially in the case of machines with high mechanical impedance as shown in fig. 3.7.b

ii) Some velocity pickups used for bearing vibration measurement have an inherent phase shift, which may cover rather a broad frequency range. This phase shift can create confusion when testing machine response and critical speeds and while balancing inplace.

3.4.3 RELATIVE SHAFT VIBRATION

Advantages:

i) The most important application of relative shaft vibration measurement is for machines having high mechanical impedance where bearing vibration measurements show little or no significant increase when problems occur.

ii) Relative shaft vibration measurement using non-contact pickup is useful in measuring gap as well.
iii) Since the non-contact pickups not have an inherent phase shift like velocity pickups, there is no phase error between the actual and measured vibration.

Shortcomings:
i) Since shaft vibration is measured in terms of displacement, the same has limited sensitivity to high frequency vibration.(e.g. gear wear, blade resonance, aerodynamic and hydraulic forces.)
ii) Non-contact pickup senses the shaft vibration as well as run out of the shaft. Irregularities such as machining marks, nick or scratches on the shaft can result in measurement errors.
iii) Care should be taken to install non-contact pickups away from nodal points, which are points of minimum amplitudes in the case flexible rotors.

3.4.4 ABSOLUTE SHAFT VIBRATION

Advantages:

i) The most important application of absolute shaft vibration is on large machines with massive rotors (e.g. turbogenerators).
ii) Since absolute shaft vibration measurements are done with 'shaft riders', electrical runout problems is eliminated.

Shortcomings:

i) Measurement is affected by mechanical runouts, nicks and scratches.
ii) Measurement is generally limited to displacement
iii) Care should be taken to install shaft rider assembly away from nodal points.
iv) Since the probe tip run on shaft, surface wear is inevitable
v) The shaft rider probe's natural frequency may be on the low frequency range or on the high frequency range. Since absolute shaft vibration is measured in terms of displacement, resonant frequency on the low frequency range is troublesome.

4.0 VIBRATION ANALYSER

4.0 VIBRATION ANALYSER

4.1 INTRODUCTION

Machinery vibration consists of many frequencies all occuring at the same time. In order to pin point machinery problems, all the defining characteristics of vibration must be measured and frequencies separated from one another. This is what the vibration analyser exactly does .
Vibration transducers described earlier can be use with vibration meters, vibration monitors and vibration analysers. The transducer is connected to the analyser by the pickup cable. The vibration analyser may be connected directly to a permanently installed vibration monitor as well.

4.2 FEATURES

i) The amplitude meter displays the overall vibration amplitude (displacement, velocity, acceleration) in the 'filter out' mode and the filtered vales in the 'filter in' (broad, sharp, balance) mode. Some of the analysers incorporate an analog as well as a digital meter for this purpose.

ii) The amplitude range selector is used to set up the full scale amplitude range. The entire amplitude range of measurement of the analyser is normally divided in to number of overlapping ranges.

iii) The frequency meter is useful for tuning the instrument to a particular frequency. In the "filter out" mode, the frequency meter indicates the predominant frequency. Some of the analysers incorporate the analog as well as digital meter for this purpose.

iv) The frequency range is used for setting up the full-scale frequency range. The frequency range of measurement of the analyser is normally divided in to a number of overlapping ranges.

v) The amplitude selector/unit selector used to select the parameter to be measured so as to set up the input circuits for the type of pickup and unit of measurement.

vi) The frequency tune control manually tunes the filter to any desired value, within the range set up by frequency range selector, to facilitate analysis of the complex vibration, frequency wise. the bandwidth of a filter is usually expressed as the percentage of the difference between the upper and lower cut of frequencies vs the filter tuned center frequency. Vibration analysers normally have more one bandwidth of filters known as broad, sharp and balance with bandwidth of **±10%, ±5%,±2-1/2%** respectively.

vii) The filter selector switch/function selector is used for selecting different selector modes namely 'filter out', 'filter in' (broad, sharp). It is also useful for operating the internal oscillator to flash the strobe light bat the frequency to which the filter is tuned.

viii) Many analysers are provided with two input receeptacles for connecting two pickups which facilitates vibration analysis and balancing. By using the pickup selector switch in the analyser can be made to read the vibration amplitude from any of the pickups.

4.3 OPERATION

Details like setting up the instrument, battery checking, and test plot generation cable connections etc. are given in detail in the operating manual of each vibration analyser. Hence only some important details have been discussed.

4.3.1 Filter tuning-peak at known frequency

i) Select the parameters to be measured and the selector switch.
ii) Set the filter selector switch to the sharper bandwidth of available bandwidths.
iii) Set the frequency range selector to the range which includes the frequency of interest
iv) Tune the filter-tuning dial to the desired frequency.
v) Set the amplitude range selector after measuring the amplitude range roughly. Make finer adjustment with the filter tuning knob while observing the amplitude meter to obtain the maximum reading.

4.3.2 Filter tuning-peak at unknown frequency

i) Select the parameters to be measured.
ii) Set the filter switch to the broader bandwidth of the available bandwidths.
iii) Set the frequency range selector to the lowest frequency range available.
iv) Start scanning with the filter tuning dial from the lowest value of the frequency range selected. At this stage the frequency meter pointer would be doing any of the following.

- Moving randomly back and forth over the meter scale.
- Reading off scale.
- Reading zero.

Stop at the point when the frequency meter stops moving back and forth over the meter scale and locks on a particular frequency.

v) Make the tuning with the tuning dial so that the reading on the filter dial and on the frequency meter are same.

vi) Set the filter selector switch to the sharper bandwidth.

vii) Set the amplitude range selector.

viii) Make fine adjustment with tuning dial to get the maximum amplitude on the amplitude meter.

5.0 VIBRATION ANALYSIS

5.0 VIBRATION ANALYSIS

5.1 INTRODUCTION

The basic purpose of the vibration analysis is to identify the specific machinery problem.

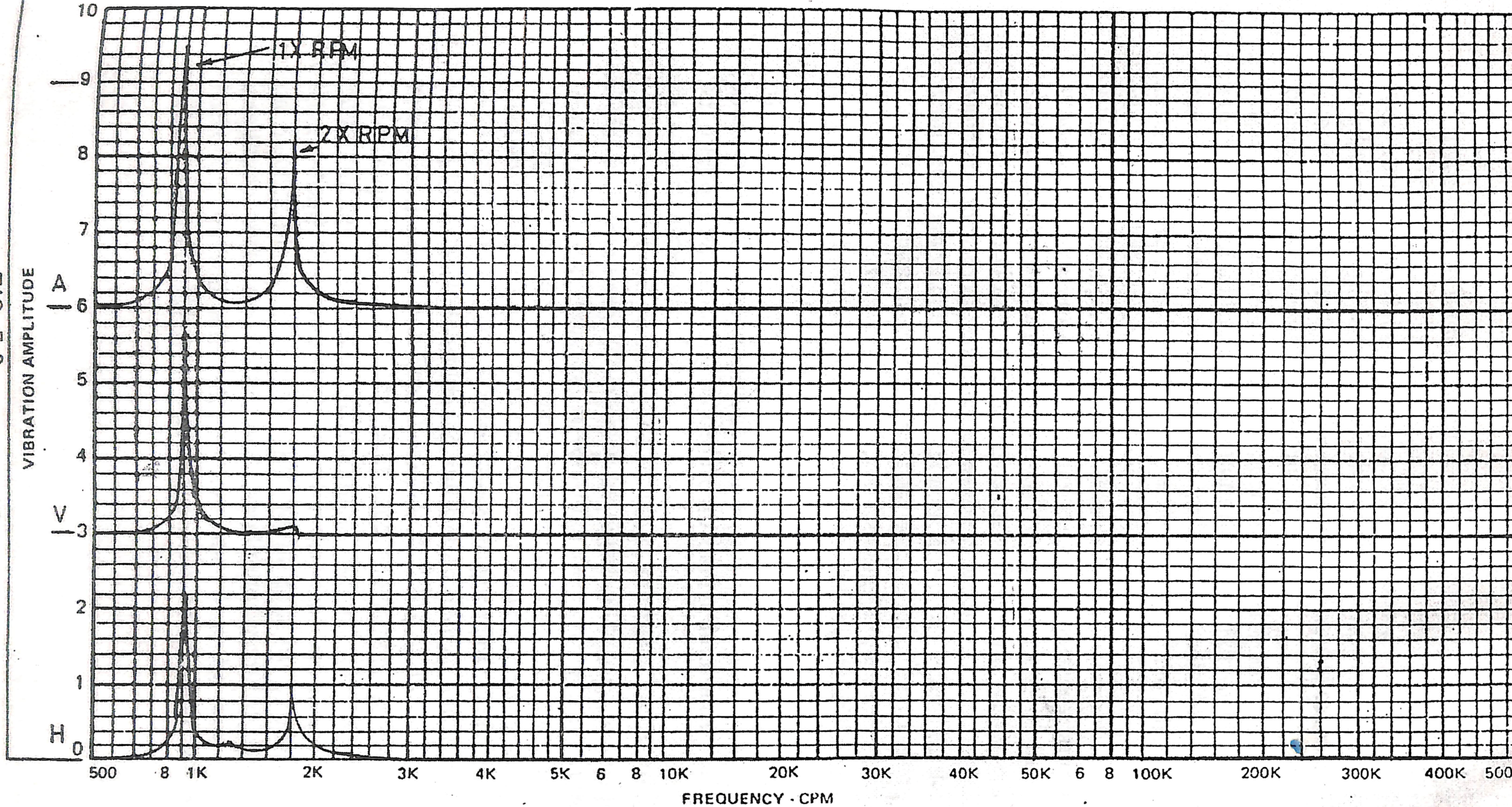
i) Vibration analysis can be done to obtain the most ideal 'baseline data' before commissioning the plant. This can be used for accepting the equipment initially as well as a reference for future analysis
ii) Vibration analysis is required to be done to obtain data with machine in good operating condition to get 'baseline data' say at the start of the predictive maintenance programme to provide the basis for comparison for future analysis.
iii) Whenever periodic checks of overall machinery vibration have revealed a significant increase, vibration analysis is required to be done to pin point the machinery problem.

The vibration analysis procedure can be divided in to two steps namely:
- Data acquisition.
- Data interpretation.

5.2 DATA ACQUISITION

Vibration data can be obtained in many ways for analysis and the most common techniques are:

i) Amplitude vs frequency.
ii) Amplitude vs time.
iii) Amplitude vs frequency vs time.
iv) Time waveform.
v) Orbits.
vi) Amplitude vs phase vs RPM.
vii) Mode shape determination.



AMPLITUDE FILTER OUT	AXIAL	MACHINE	NO. 4 F. D. FAN
AMPLITUDE RANGE (FULL SCALE)	microns ___ mils ___ mm/sec ___ in/sec ___ g	MACHINE LOCATION	
MEASURE UNITS	☐ Microns PK-PK / MILS PK-PK ☐ MM/SEC PK / IN/SEC PK ☐ g PK	OPERATING CONDITIONS (RPM, LOAD, TEMP, ETC)	900 RPM
FILTER IN	☒ SHARP ☐ BROAD		
TYPE PICKUP	☒ VEL ☐ ACCEL ☐ N/C		
MEASURE POSITION	C ☐ H ☐ V ☐ A		
DATE	BY	DATA SHEET NO.	59-G

SKETCH OF MACHINE

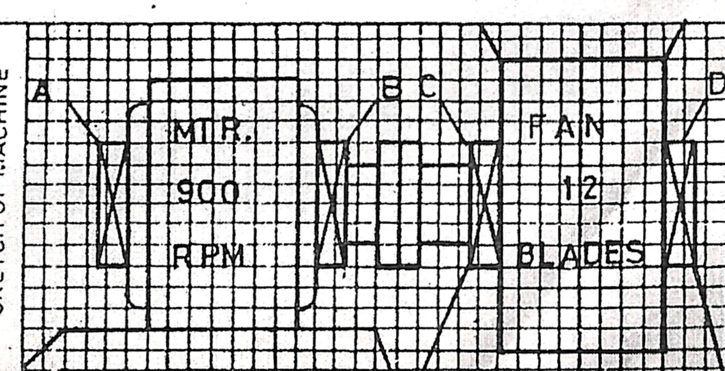


FIG. 5·2

5.2.1 Amplitude Vs frequency

The next step in data acquisition is to measure and record the 'filter out' amplitude and the predominant frequency readings in all the three direction at each measurement point. The measurement should taken on the equipment during normal operation condition.

"Operation condition such as temperature, load, speed etc., shall be specified prior to the test and actual conditions recorded. For variable speed machines, the measurements shall be made at many speeds in order to locate the resonance frequencies which occur and evaluate their effects on the measured vibration characteristics."

When the vibration is complex, the 'filter out' amplitude and frequency readings will be unsteady. In this case the unsteady amplitude readings should not be averaged but instead we should record the minimum and maximum amplitude reading. Unsteady frequency meter readings are not valid and they shall be indicated by the wavy line.

Unfiltered amplitude readings tells us about the extent of the problem and the predominant frequency reading would help us to trace the problem source quickly. However, unfiltered readings alone cannot serve as the basis for diagnosing the problems and a thorough analysis of the vibration must be done with the tunable filter.

We should start our analysis from the point and direction where the maximum unfiltered amplitude was recorded. To obtain the necessary 'filter in' readings the scanning procedure for tuning the filter is recommended and this has been described 4.3.2

As per the procedure described in 4.3.2., the filter can be tuned to the first significant frequency. There may be vibration frequencies which may show at one point (e.g. vibration at very high frequencies from faulty antifriction bearing) or at one particular component (e.g. vibration at gear meshing frequencies in the gear box) and not at other points or components. Care should be taken not to miss these frequencies. This can be done by rechecking at each point having antifriction bearing and at one point of each major component (gear box, motor, exciter etc.) of the equipment.

After acquiring the data as mentioned above, the readings shall be noted down in the Vibration Measurement Record (VMR) which has been standardized. The same is given in fig. 5.1. The data entered in VMR need to be checked for completeness. This can be done by comparing the 'filter in' amplitude with the 'filter out' amplitude recorded at various pickup locations. As a thumb rule, we can state that the sum of the 'filter in' vibration amplitudes at various frequencies should equal or exceed the 'filter

out' vibration amplitude. If the above condition is not met with at any pickup location, there may be some additional vibration information, which should be found by scanning through the frequency range at that pickup location.
The technique described so far is a manual analysis technique because of the following reason:

- Filter is to tuned manually through the frequency range.
- Peak amplitude at each frequency is to be adjusted manually.
- Recording is to be done manually in the VMR.

Other methods for obtaining the same data are:

- **Semi-automatic.**
- **Automatic.**

Semi-automatic analysis:

Some of the vibration analyzers (e.g. IRD 350) have provisions for connecting a XY recorder. One output receptacle on the analyzer provides a DC voltage proportional to the tuned frequency to drive the horizontal axis of the recorder. As the frequency range of the filter is manually scanned, the recorder automatically plots frequency in the horizontal axis and the respective amplitude in the vertical axis. These vibration amplitude Vs frequency plots called as vibration signatures are to be taken at all the points in all the directions. Standard formats for taking vibration signatures are supplied by the manufacturer of vibration analyzer, in which important data such as amplitude parameter, amplitude range, pickup location, type of pickup, machine identification, operating condition etc., are entered. A sample vibration signature has been given in fig. 5.2.
Semi-automatic analysis has the following advantages over the manual analysis.

- Human errors in observing and recording the data are eliminated.
- Chances of missing any significant vibration frequency are less.
- Time required for vibration analysis is greatly reduced.

Automatic analysis:

Some vibration analyzer (e.g. IRD 820, IRD 880) incorporate a filter which automatically tunes the filter and scans the entire frequency range of the instrument. The recording of the data is also done automatically either by a XY recorder connected to the vibration analyzer or by an inbuilt recorder in the analyzer itself.

5.2.2 TIME WAVEFORM:

Sometimes information additional to amplitude Vs frequency characteristics is required to diagnose a specific defect or to study the dynamic behaviors of a machine under specific operating conditions. In this regard, observation of the time waveform of a vibration signal on the oscilloscope has been found very useful. The vibration signal is applied to oscilloscope vertical input thereby scaling the vertical axis on the CRT in the amplitude. The horizontal axis is scaled in time.

Different troubles of the machine which have identical frequencies (yet differ considerably in dynamic behaviour) can still be identified, with the help of the characteristic waveform they produce on the oscilloscope. Some of the typical waveforms for the common troubles like unbalance, mechanical looseness, oil whirl and faulty antifriction bearing have been given in fig. 5.3.

The amplitude of transient vibration (vibration occurs only at a particular instant like caused by a broken gear tooth) would be indicated on the oscilloscope rather than by the filtered amplitude reading. When transient signals are filtered by the vibration analyzer, their amplitudes may be significantly attenuated. The waveform gives the instantaneous undamped response thus giving the true amplitude in contrast to the reading of the amplitude meters of vibration meter/analyzer which are damped to minimize erratic movement of the pointer.

The oscilloscope can be either coupled to the vibration analyzer connecting the oscilloscope output of the analyzer to the vertical of the oscilloscope or directly to the transducer. While using with the vibration analyzer the vertical gain control of the oscilloscope should be adjusted with the help of 'peak-peak' amplitude indicated by the amplitude meter. in the case of the oscilloscope being directly connected to the transducer, amplitude of the waveform shall be calculated as given below :

Sensitivity of transducer	= x millivolts/microns
The vertical gain control of the oscilloscope has been set say at	= y millivolts/divisions
'peak-peak' amplitude of the waveform on the oscilloscope	= z divisions

signal voltage = z(y) millivolts

Only a filtered vibration signal when displayed on an oscilloscope can give meaningful information. The sharper the bandwidth of the filter the more clarity the time waveform will have. If an unfiltered signal had to be displayed on the oscilloscope, the appropriate for the frequency range should be selected.

The umber of cycles displayed on the oscilloscope or the time period covered by the display can be varied to give long term waveform and short term waveform. Long term waveform displays about 50 to 100 cycles whereas short term waveform displays only a few cycles. Long term displays are useful for studying low frequency beats, low frequency amplitude modulation and for studying the behaviour of the machine when it pass through resonance.

5.2.3 ORBIT ANALYSIS

An useful technique to supplement the amplitude vs frequency vibration signatures is to display the plot of the total motion of the shaft within the bearing on a CRT of an oscilloscope. These plots are called shafts orbits or luscious patterns. For diagnosing specific machinery problems such as unbalance, misalignment, rubbing, oil whirl, resonance etc.this technique has been found to be very useful.

The prime requirement for conducting orbit analysis is two radial non-contact pickups in each bearing with the axis of the pickup separated by 90 degree. The signal from one pickup is applied to the horizontal input of an oscilloscope and the signal from the other pickup is applies to the vertical input of the oscilloscope. Where permanently installed monitors are being used, the necessary signals to drive the oscilloscope can be obtained either directly from the signal sensors or from the analyzer receptacles located on the panel of the monitor. In the absence of a monitor, a dual channel non contact accessory may have to be used. Even though it would be preferable to install the non contact pickups in the true horizontal and vertical axes to simplify interpretation of the observed pattern on CRT , for practical reasons, pickups may be mounted 45degrees away from the true axes. This will not affect the shape of the orbit, but it will cause the display to be rotated by 45degrees on the CRT. For the sake of clarity, the true horizontal and vertical axes can be marked on the face of oscilloscope.

Another important point to consider when setting up an oscilloscope to display orbits is the relationship between the direction of shaft motion and

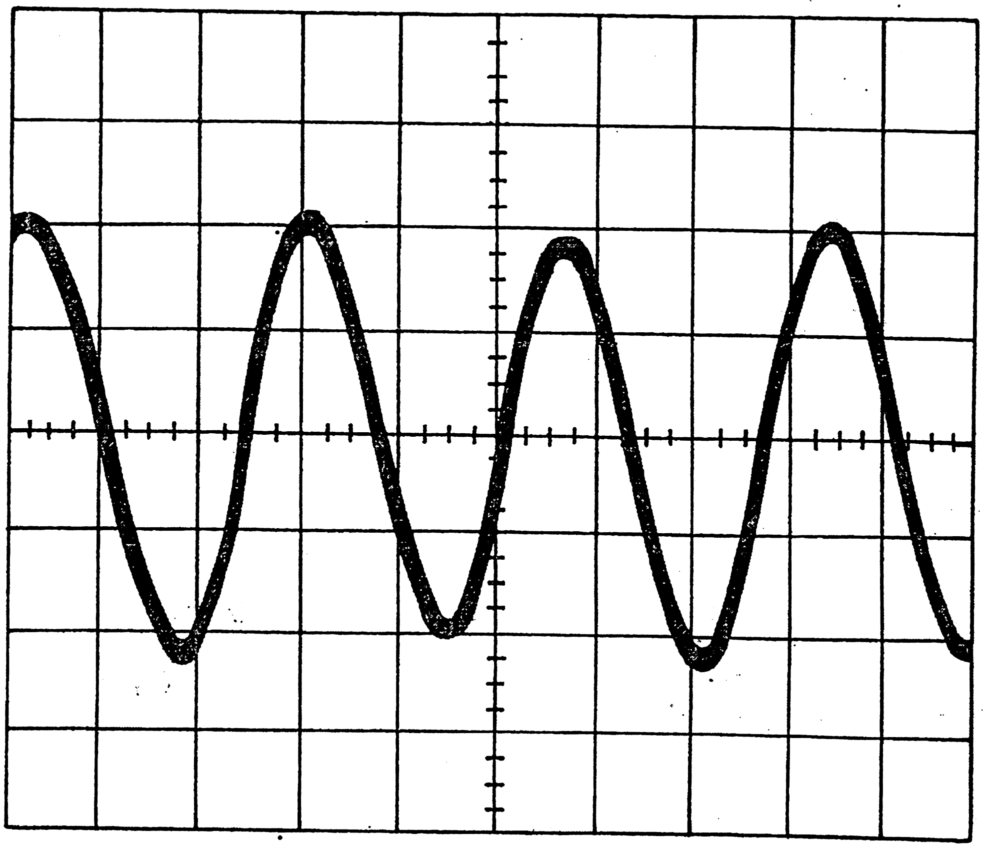


FIG. 5·3·a TYPICAL TIME WAVE FORM OF UNBALANCE OR MISALIGNMENT AT A FREQUENCY 1 x RPM

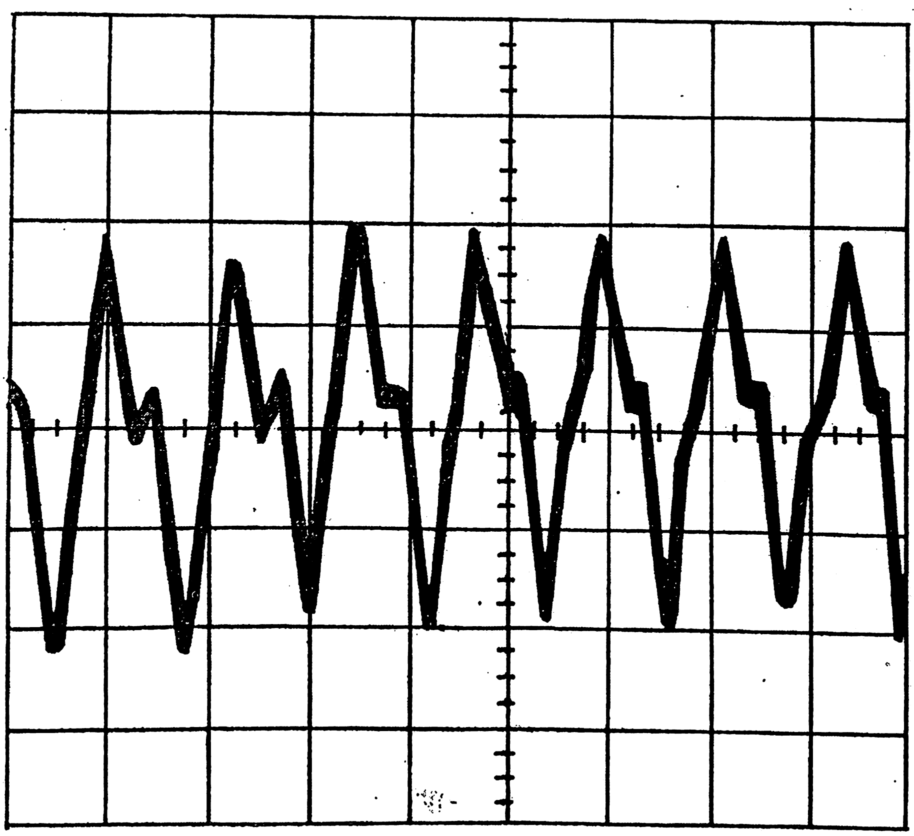


FIG. 5·3·b TYPICAL TIME WAVE FORM OF MECHANICAL LOOSENESS AT A FREQUENCY 2 x RPM

CAUSE	AMPLITUDE	FREQUENCY	PHASE	REMARKS
UNBALANCE	PROPORTIONAL TO UNBALANCE.LARGEST IN RADIAL DIRECTION	1X RPM	SINGLE REFERENCE MARK	MOST COMMON CAUSE OF VIBRATION
MISALIGNMENT COUPLINGS OR BEARINGS AND BENT SHAFT	LARGE IN AXIALDIRECTION.50% OR MORE OF RADIAL VIBRATION	1X RPM USUAL 2 & 3X RPM SOMETIMES	SINGLE DOUBLE OR TRIPLE	BEST FOUND BY APPEARANCE OF LARGE AXIAL VIBRATION .USE DIAL INDICATORS OR OTHER METHODS FOE POSITIVE DIAGNOSIS. IF SLEEVE BEARING M/C & NO COUPLING MISALIGNMENT BALANCE THE ROTOR.
BAD BEARINGS ANDANTI FRICTION TYPE	UNSTEADY. USE VELOCITY MEASUREMENT IF POSSIBLE	VERY HIGH SEVERAL TIME RPM	ERRATIC	BEARING RESPONSIBLE MOST LIKELY THE ONE NEAREST POINT OF LARGEST HIGH FREQUENCY VIBRATION
ECCENTRIC JOURNALS	USUALLY NOT LARGE	1X RPM	SINGLE MARK	IF ON GEARS LARGEST VIBRATION IN LINE WITH GEAR CENTERS. IF ON MOTOR OR GENERATOR VIBRATION DISAPPEARS WHEN POWER IS TURNED OFF IF IN PUMP OR BLOWER ATTEMPT TO BALANCE

BAD GEARS OR NOISE	LOW-USE VELOCITY MEASURE IF POSSIBLE	VERY HIGH GEAR TEETH TIMESRPM	ERRARTIC	
MECHANICAL LOOSENESS		2X RPM	TRWO REFERENCE MARKS SLIGHTLY ERRATIC	USUALLY ACCOMPANIED BY UNBALANCE AND/OR MISALOGNMENT
BAD DRIVE BELTS	ERRATIC OR PULSING	1,2,3 & 4 X RPM OF BELTS	ONE OR TWO DEPENDING ON FREQUENCY . USUALLY UNSTEADY	STROBE LIGHT BEST TOOL TO FREEZE FAULTY BELT
ELECTRICAL	DISAPPEARS WHEN POWER IS TURNED OFF	1XRPM OR 1 OR 2 X SYNCHRON OUS FREQUENCY	SINGLE OR ROTATING DOUBLE MARK	IF VIBRATION AMPLITUDE DROPS OFF INSTANTLY WHEN POWER IS TURNRD OFF, CAUSE IS ELECTRICAL
AERODYNAMI C HYDRAULIC FORCES		1X RPM OR NUMBER OF BLADES ON FAN OR IMPELLER X RPM		RARE AS A CAUSE OF TROUBLE EXCEPT IN CASES OF RESONANCE
RECIPROCATI NG FORCES		1,2 & HIGHER ORDERS X RPM		INHERENT IN RECIPROCATING MACHINES CAN ONLY BE REDUCED BY DESIGN CHANGES OR ISOLATION.

TABLE:**VIBRATION IDENTIFICATION CHART** the resultant direction of the trace on the oscilloscope. The direction in which the trace will move on the CRT will depends on the following factors:

i) The voltage polarity of the non contact pickup system.

ii) The location of the pickup i.e. above or below the shaft/to the right or to left of the shaft.

Fig. 5.4 illustration the effects of pickup position and the direction of shaft motion on the resultant scope trace when a non contact pickup system, with negative power supply and positive power supply are used.

Given below are the points to consider when using non contact pickups to observe shaft motion on an oscilloscope.

When using non contact pickups with negative polarity:

i) Vertical pickups mounted above the shaft will read shaft motion correctly on the vertical axis of the CRT.

ii) Vertical pickups mounted below the shaft should have the polarity of the input signal to oscilloscope reversed for a correct indication of the vertical shaft motion on the CRT.

iii) Horizontal pickups mounted to the right of the shaft will read shaft motion correctly on the horizontal axis of CRT.

iv) Horizontal pickups mounted to the left of the shaft should have the polarity of the input signal to the input signal to the scope reversed for a correct indication of horizontal shaft motion on CRT.

When using non contact pickups with positive polarity:

i) Vertical pickups mounted above the shaft should have the polarity of the input signal to the oscilloscope reversed for a correct indication of vertical shaft motion on the CRT.

ii) Vertical pickups mounted below the shaft will read shaft motion correctly on the vertical axis of the CRT.

iii) Horizontal pickups mounted to the right of the shaft should have the polarity of the input signal to the scope reversed for a correct indication of horizontal shaft motion on the CRT.

iv) Horizontal pickups mounted to the left of the shaft will read shaft motion correctly on the horizontal axis of CRT.

Proper calibration of horizontal and vertical gain controls of the oscilloscope is important to ensure that the shape of the observed pattern accurately reflects the true motion of the shaft so as to obtain correct amplitude data from the oscilloscope. For doing this, sensitivity of the non pickup system must be known. Therefore calibration of the scope gain settings is simply a

TYPICAL ORBITS

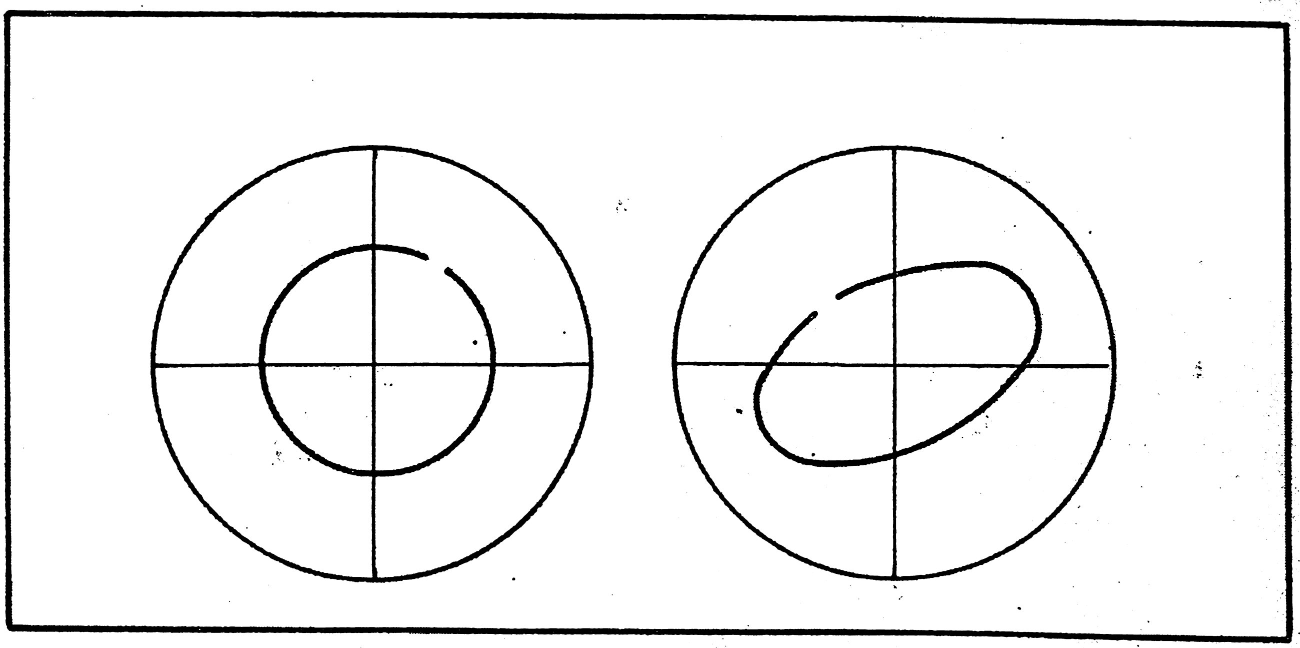


FIG. 5·5·a UNBALANCE

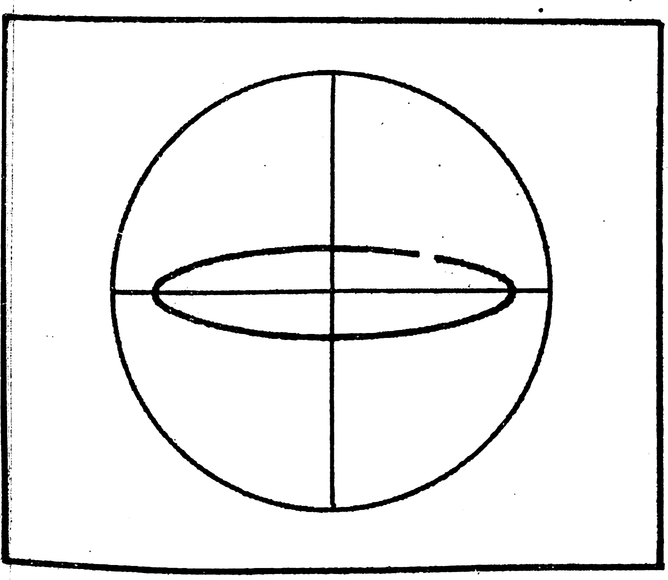


FIG 5·5·b MISALIGNMENT BEARING WEAR OR RESONANCE

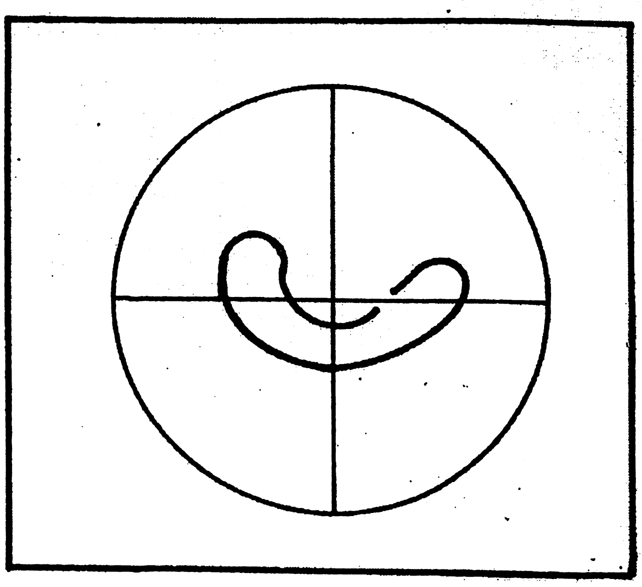


FIG 5·5·c MISALIGNMENT

matter of setting the oscilloscope gain sensitivities to match the input system sensitivity.

e.g. Input system sensitivity = 100 millivolts/micron
Oscilloscope gain sensitivity = 100 millivolts/division

Therefore one division on the oscilloscope will indicate one micron.

The horizontal and vertical gain controls must be in agreement so that one unit of movement in the horizontal and vertical direction produces the same indication on the CRT.

When observing shaft orbits on an oscilloscope, it is not possible to obtain frequency information from the display unless some frequency reference is superimposed. Normally the frequency reference is a synchronous pulse at a frequency 1 x RPM. the synchronous reference pulse is obtained by installing an electromagnetic pickup or a non contact pickup observing a protrusion or depression on the shaft. The reference pulse can be superimposed on the orbit in a couple of ways. If the oscilloscope is so equipped, the reference pulse can be applied to the Z-axis. If there is no provision for 'Z' input in the oscilloscope, the reference pulse can be connected in parallel with either the horizontal or vertical input. But the latter method is not preferred, since the reference signal may introduce unwanted signal and may be lost if the pattern is complex.

Discussed below are the characteristics of some of the typical shaft orbits associated with some of the important machinery malfunctions.

Unbalance

Unbalance will reveal a circular or slightly elliptical orbit as shown in fig. 5.5a., if no other significant vibrations are present. if the shaft orbit takes a highly elliptical shape as shown in fig. 5.5b where the ratio of major axis to minor axis is relatively high. say 8:1 to 10:1 or more, machine operating on or very near a structural resonance is suggested.

Misalignment:

A significant misalignment condition may not reveal the circular or slightly elliptical pattern typical of unbalance. Instead misalignment may reveal an orbit like banana as shown in fig. 5.5c. As the higher order frequencies take on greater significance, the orbit may appear as shown in fig. 5.5d.

Oil Whirl:

The orbit of an oil whirl condition will generally appear as a circular or elliptical pattern with an integral loop as shown in fig. 5.5e. Since the frequency of vibration in this case would be less than 50% RPM, two reference pulses are seen in the orbit. The internal loop will appear to rotate as indicated by the dotted loop.

Rubbing:

Rubbing between rotating and stationary bars can results in the several different orbits. The orbits shapes encountered will generally depend on the extent of rub. In most cases the reference pulse on the orbit will appear unsteady.
In the case of very mild rub, where the rotor only touches the stationary part once per revolution may result in theses distortion of the circular or elliptical orbit as shown in fig. 5.5F(a). Very light rubbing which causes the rotor to hit and bounce can result in sub multiple vibration generally at one half RPM. This will produce an orbit with internal loop as shown in fig. 5.5F(b). The internal loop in this case will not appear to rotate as in the case of oil whirl. As the rub become more severe, the resultant pattern may take on anyone of the number of configurations, which include harmonic frequencies, random non-synchronous frequencies and resonant frequencies of various components. Orbits, which may result from heavy or full rubbing conditions, have been shown in fig. 5.5F(c).

5.2.4 AMPLITUDE/ PHASE Vs MACHINE RPM:

Every machine and its supporting structures have resonant frequencies where very high amplitudes of vibration can result from relatively small exciting force. This condition known as resonance is a common problem. Plot of vibration amplitude and phase Vs RPM which is referred as Bode plot can tell us about the response of machine to the forces that cause vibration. Fig. 5.6 shows one such plot which has to identify the resonant frequencies, amplitude/phase Vs machine RPM would be quite useful.
Amplitude Vs RPM can be obtained using analyzer with analyzers filter in the 'filter out' position and observing the amplitude when it peak and the corresponding frequency as the machine coasts down/or brought up to speed. Since it is very difficult to observing while observing the amplitude and frequency, vibration analyzer incorporating a tracking filter utiliser reference

TYPICAL ORBITS

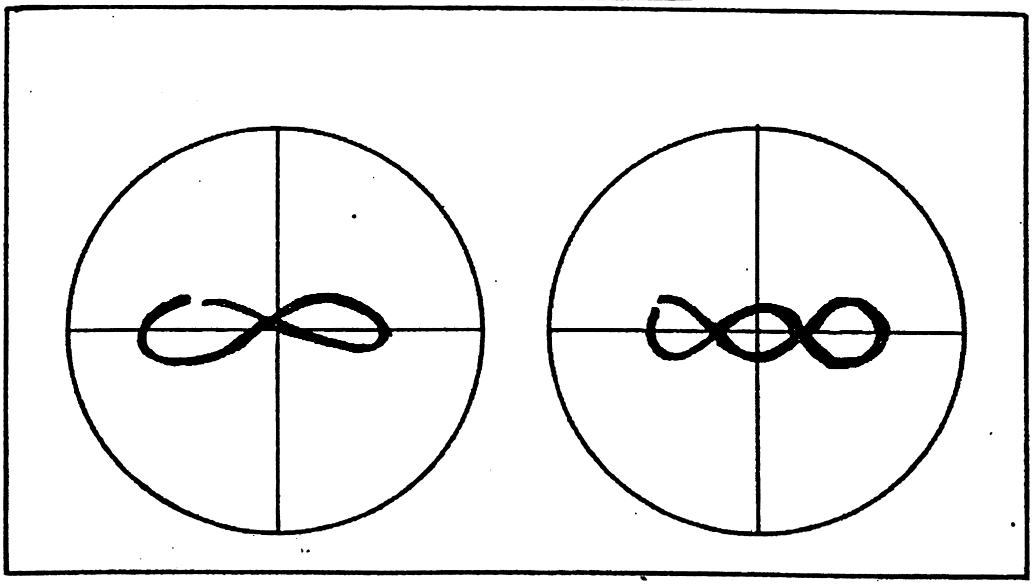


FIG. 5·5·d COUPLING MISALIGNMENT

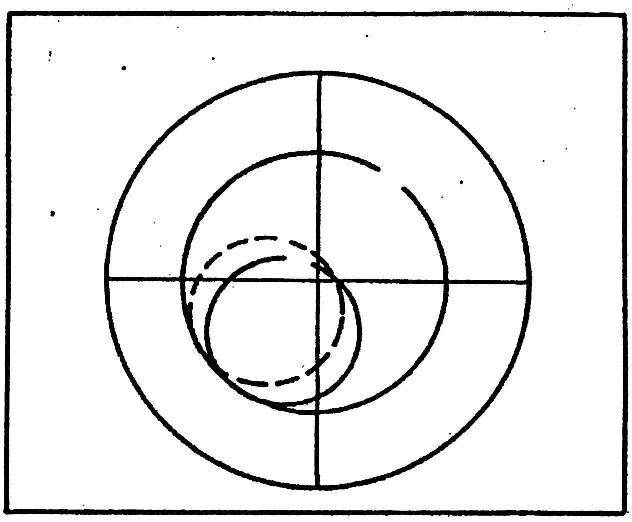


FIG. 5 5 e OIL WHIRL

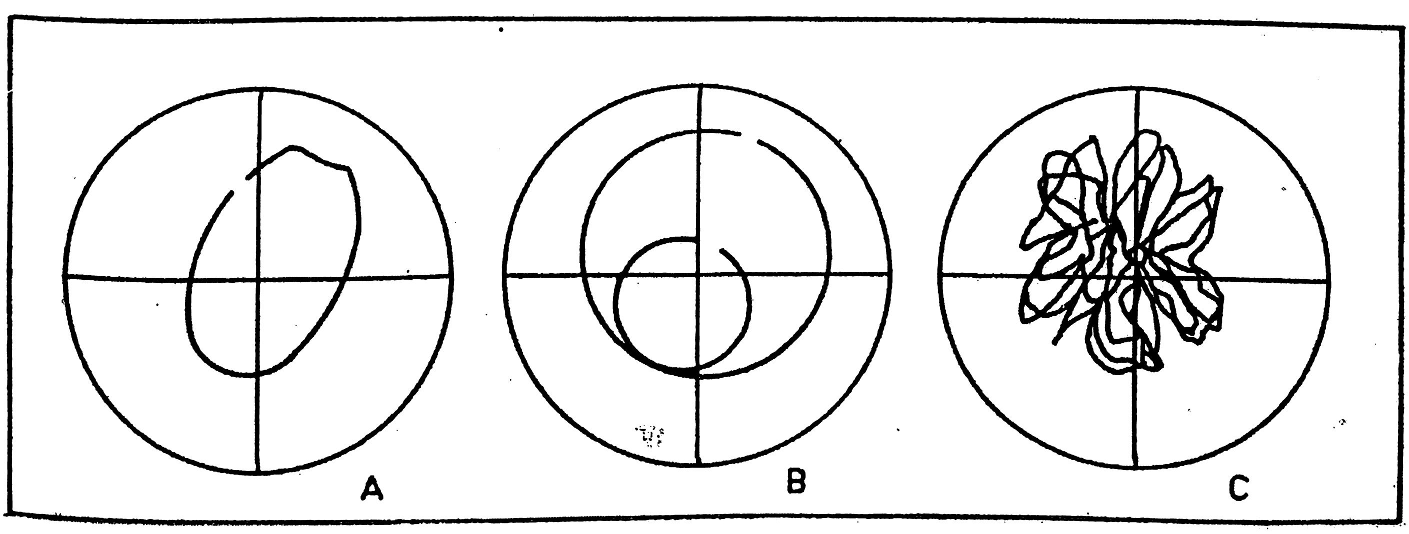


FIG. 5·5·f RUBBING

pickup at the shaft of the machine to provide a voltage pulse for each revolution of the shaft.

The **1 x RPM** signal from the reference pickup does the function given below:

i) The reference signal automatically tunes the analyzer filter to the RPM of the shaft. if the RPM of the shaft changes, the frequency of the reference signal changes accordingly.
ii) The reference signal controls a DC voltage, which is proportional to shaft RPM, is used for driving the X-axis of an XY1Y2 recorder for obtaining amplitude/phase Vs RPM plot.
iii) The reference signal acts as a fixed reference for comparison with signal from a vibration pickup resulting in DC voltage proportional to the relative phase between the two signals. This DC voltage is used for driving one of the Y-axis of XY1Y2 recorders for plotting phase Vs RPM.

Even though resonance frequency of a machine during coast down or speeding up will be indicating by a peak in amplitude and 180 degree phase shift in the amplitude/phase Vs RPM, there are some exceptions to this indication which have been discussed below.

i) Background vibration will produce a peak in the amplitude without effecting any change in the phase. This should be mistaken for resonance. There may be cases where a 180-degree shift in the phase is not accompanied by a peak in the amplitude. The phase shift suggests resonance but the following points explain the absence of amplitude peaks.

- If the exciting force at the frequency concerned is very low or if the system is heavily damped, the vibration amplitude at the resonant frequency will not be amplified much.
- When a rotor shaft or structure is excited to vibrate at resonance, it would assume one of the vibratory modes as shown in figure 5.7. Each mode has one or more nodal points where amplitude of vibration is minimum. If the vibration pickup is located at a nodal point, there would be little amplification of vibration amplitude when passing through resonance.

ii) Sometimes a peak in amplitude is accompanied by a 360-degree phase shift instead of the 180-degree phase shift accompanying resonance. This suggests that actually two systems are in resonance at or near the same frequency contributing a 180 degree phase shift each.

iii) In some cases, a 360 degree phase shift is accomplished by a dip referred as antinode in the vibration amplitude. Here two spring mass systems are in resonance at or very near the same frequency. But the vibratory forces are acting in opposite directions causing an antinode.

iv) Sometimes a dip or a peak in the amplitude is obsereved in amplitude Vs RPM plot near the critical speed but not accompained by a substantial phase shift. Amplitude an phase of shaft vibration undergo a change as the shaft passes through the critical speed. Hence vector sum of shaft vibration and run out is also undergoing a change at the critical speed. Sometimes the shaft vibration and run out vectors become momentarily out of phase and tend to cancel each other thus producing a dip and sometimes these vectors may momentarily add to each other thus producing a peak. In these cases, it is advisable to carefully measure runout amplitude before vibration measurements are taken and eliminate excessive runout physically or electronically.

v) In some cases, substantial phase shift from a higher value to zero are observed. Since the DC voltage proportional to phase is a comparison between the voltage pulse from a reference pickup and a vibration signal from the vibration pickup, in case any of these signals is lost or lacks sufficient amplitude, the phase voltage would drop to zero.

vi) Sometimes abrupt phase shift of 360 degree varying around 0 degree to 360 degree within a narrow speed range are observed in phase Vs RPM plots unaccompanied by any peak in the amplitude. This is more likely due to a phase indication which may be varying only slightly say between 1 degree to 359 degree.

5.2.5 MODE SHAPE ANALYSIS:

Determining the shape, the equipment under consideration takes while vibration known, as mode shape has been found to be very useful for confirming resonance conditions and identifying nodal points. Mode shape can be determined by measuring the vibration amplitude at various

equispaced locations. Mode shape can be plotted by marking the vibration amplitude on the vertical axis measured at different points on a sketch of the equipment drawn to scale.
If the mode shape thus drawn indicates presence of nodal points, comparative phase readings will have to be taken on both sides of the nodal point to determine whether the mode shape is like the mode shape is like the one shown in fig. 5.9a or 5.9.b. mode shapes are also useful for selecting the most appropriate plane/s for applying balance corrections.

5.3 DATA INTERPRETATION:

The next step in analysis is interpreting the data thus obtained for identifying the machinery problem. This is done by comparing the reading with the characteristic vibrations due to typical machinery troubles. The key to this comparison is mainly the relation of the frequency of vibration to the rotating speed. The vibration identification chart given in table 5.2 lists the most common causes of vibration together with its relation to the amplitude frequency and position of phase reference mark under strobe light.
Since many machine troubles have similar characteristics and several troubles may be present in a machine simultaneously, it becomes necessary to choose between several like possibilities. Hence we will have to see the relative probability of the occurance of machine faults, which have similar vibration characteristics. The vibration and noise identification chart given in table 5.3 gives a comprehensive listing of the most common machinery problems encountered and provides a relative probability rating number which provides an indication of which trouble is most likely in a given set of circumstances.

5.3.1 Vibration characteristics of malfunctions

5.3.1.1 Unbalance

Typical symptoms :

i) The vibration occurs at a frequency equal to 1 x RPM of the unbalance part.
ii) The amplitude of vibration is proportional to the amount of unbalance present.
iii) Small peaks at harmonic frequencies i.e. 2 x RPM, 3 x RPM, 4 x RPM would be observed in the vibration signature and

the amplitude of vibration at these frequencies would be 50% less than that of the vibration amplitude at the frequency **1 x RPM.**

iv) The largest amplitude of vibration occurs in the radial (horizontal or vertical) direction.

v) The ratio between the amplitude in the horizontal direction and the vertical direction normally does not go beyond 5:1

Deviation from typical symptoms:

i) In the case of overhung rotors, unbalance would result in axial vibration amplitude as high as the radial vibration or even higher.

ii) Rotor mounted between bearings and having a substantial couple unbalance may also exhibit high amplitudes of axial vibration.

Other malfunctions which produce similar symptoms:

i) If the ratio of horizontal vibration amplitude is more than 5 : 1, the problem may be more due to mechanical looseness, misalignment rather than unbalance even if other characteristic symptoms of unbalance are present.

ii) If the characteristic symptoms of unbalance are present and the amplitude of vibration of **2 x RPM** is higher than 50% the amplitude at **1x RPM,** the problem is more likely mechanical looseness rather than unbalance.

Methods to distinguish unbalance from other malfunctions which produce similar symptoms:

i) An eccentric pulley produces vibration signatures quite similar to unbalance. However, the same can be distinguished from unbalance by comparing phase measurement in the horizontal and vertical direction. A normal unbalance condition will generally reveal a difference of 90 degree between the horizontal and vertical phase reading whereas in the case of an eccentric pulley the same would be either identical or differ by 180 degree.

ii) A single chipped, broken or deformed tooth of a gear would give a vibration signature exactly similar to that of unbalance. However, the same can be distinguished from unbalance by comparing phase readings in the horizontal and vertical directions. It is quite unlikely that these phase readings would differ by 90 degree as they would otherwise in the case of unbalance. A positive way of identification is to display the time waveform in the oscilloscope.

5.3.1.2 Misalignment:

Misalignment is almost as common a problem as unbalance and the same would be caused in machinery by the coupling or the bearing. Coupling misalignment can be classified as

i) **Angular alignment**
ii) **Off set misalignment**
iii) **Combination of both.**

Since a belt shaft acts very much like angular misalignment, its vibration characteristics have also been discussed here.

Typical symptoms:

i) The amplitude of vibration generated is proportional to the misalignment.
ii) The vibration amplitude will be in both the radial and axial direction.
iii) The amplitude of axial vibration is greater than 50% of the highest radial (horizontal or vertical) vibration.
iv) Normally the vibration frequency is **1 x RPM.** When the misalignment is severe vibration frequency would be **2 x RPM.**
v) Misalignment would produce vibration at higher frequencies on machines equipped with antifriction bearings and in case of gear drives, misalignment would cause vibration at gear meshing frequency.
vi) **Bearing Misalignment:**

- Misalignment of a sleeve bearing will not result in any vibration unless there is unbalance. A radial vibration as well as an axial vibration will be present which results from the reaction of misalignment bearing to the forces due to unbalance.
- Misalignment of an antifriction bearing will result in axial vibration even when the part is balanced. The vibration may occur at a frequency of **1,2,3 x RPM** or at a frequency equal to the number of rolling elements in the bearing times shaft RPM.

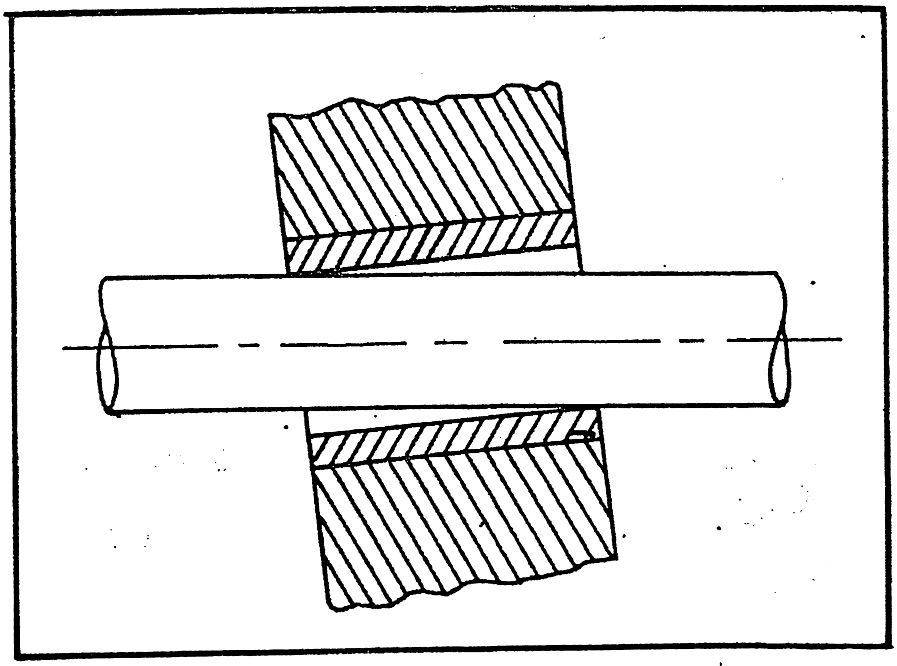


FIG. 5·11 MISALIGNMENT OF JOURNAL BEARING

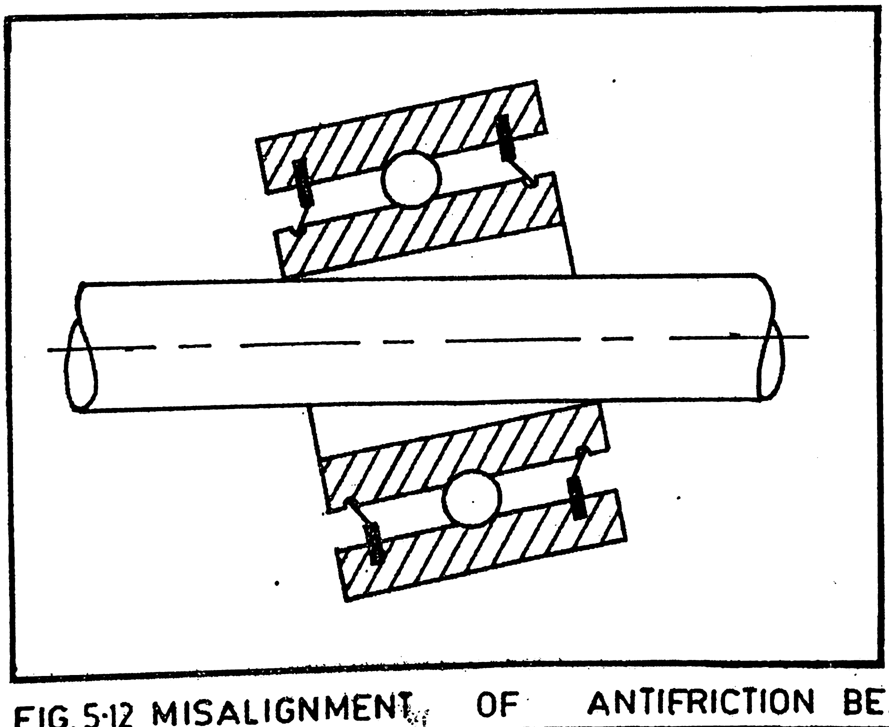


FIG. 5·12 MISALIGNMENT OF ANTIFRICTION BEARING

Other malfunctions, which produce similar symptoms:

i) Improper machine mounting, uneven base plane, where the feet of the machine rest are termed as soft foot. This produces vibration characteristics similar to misalignment. Loosening of any one of the outboard foundations bolts results in a significant change in the vibration.

ii) Underrating or improper lubrication can virtually lock up a coupling due to the driving torque between the driver and the driven unit. This condition known as torque lock produces similar vibration characteristics as that of misalignment. Since the coupling locks in a different angular position for each start or stop of the machine, amplitude and phase readings vary significantly each time. In the presence of torque lock, time waveforms obtained from axial proximity probes would appear in phase.

Misalignment vs. bent shaft

Since coupling misalignment, bearing misalignment and a bent shaft exhibit similar vibration characteristics it would be essential to identify the most likely cause before shutting down the machine.
Phase analysis, which helps us to determine the relative motion between various parts of the machine, is a valuable technique available for identifying the most likely cause. Phase analysis is discussed here since it aids since it aids in determining the type of axial movement of each bearing (i.e. whether the bearing is twisting, rocking or moving simply back and forth and the relative axial movement of bearing with respect to another.

The steps involved in conducting phase analysis are:

i) Phase measurement should be taken at four positions each 90-degree apart for each bearing as shown in fig 5.13.a.

ii) Tune the analyser filter to the RPM of the machine being examined and recorded phase measurements with the pickup in axial direction at each of the four selected positions as shown in fig. 5.13.b.

From the phase measurements recorded at a particular bearing, it can be observed that the same differ noticeably from each other or are relatively the same. if the four phase reading differ noticeably from each other at a particular bearing, twisting of that bearing is indicated as shown in fig 5.13.c. if the four phase readings are approximately same at a particular bearing, that bearing is vibrating back and forth in a planar fashion as shown in fig5.13.d.

Knowing about the movement of one particular bearing alone would not help us to identify the possible causes. Hence for diagnosing the possible causes, we should go comparing the phase measurements of one bearing with the next bearing in the transmission train in a sequence. The possible combination of phase measurements and their correlation to the cause of trouble are:

i) If large phase difference is noted between the inboard bearing of driver equipment of a direct-coupled machinery, coupling misalignment or a faulty coupling is the possible cause.
ii) If phase difference is noticed between the bearings of the same machine, the possible cause would be a bent shaft or severely misaligned bearings.
iii) If all the bearings of the system are in phase, the possible cause may be either unbalance or resonance of the foundation in the axial direction at a frequency **1 x RPM.**

5.3.1.3 Eccentricity.

Eccentricity is another common source of vibration and exists whenever the shaft rotating centerline is not concentric with the geometric centerline. Actually eccentricity is a common source of unbalance and correcting the unbalance would in some cases compensate the force causing vibration. However, eccentricity cannot always be corrected by routine balancing techniques, since eccentricity cannot always be corrected by balancing.

Eccentric gear:

In the case of an eccentric gear, the largest vibration will occur in the direction of the line through the centers of the gears at a frequency equal to **1x RPM** of the eccentric gear and at the meshing frequency of the gear. Since the gear tooth clearances are continually changing as the gears rotate, the vibration generated at gear mesh frequency at **2 x RPM.** The time waveform has been shown in fig.5.14.

Eccentric belt pulley:

In the case of an eccentric pulley, the largest vibration occurs in the direction of belt tension at a frequency equal to **1 x RPM** of the eccentric pulley.

Eccentric rotors:

Eccentricity of other rotors such as motor armatures, fan and blower rotor, pump impellers, turbine rotors, compressor rotors also cause vibration at **1x RPM** similar to unbalance. Although it may possible to compensate for this condition by balancing at a particular load condition by balancing at a particular load condition, any change in the load condition may quite likely produce an increase in the vibration.

5.3.1.4 Mechanical Looseness:

Mechanical looseness is due to loose mounting bolts, excessive bearing clearance, a crack or break in the structure of bearing pedestal, some loose rotating part on the shaft or some other loose machine part.

Typical symptoms:

i) The frequency Vs amplitude spectrum, in this case, has a peak at a frequency **2 x RPM** but sometimes at higher order frequencies such as **3,4,5,6 x RPM**.
ii) The severity of vibration at higher order frequencies is more than one half the severity of vibration occurring at rotating speed frequency.
iii) Amplitude of vibration due to mechanical looseness is often unsteady.
iv) Vibration characteristics of mechanical looseness are often unsteady.
v) Vibration characteristics of mechanical looseness will not occur unless there is some other exciting force such as unbalance or misalignment etc. Excessive looseness would result in high amplitude of vibration even when the unbalance or misalignment present is relatively small.
vi) Loose rotor bars, loose stator windings, and loose rotor windings in electrical motors and generators would cause vibration at harmonics of 'torque pulse' frequency.

5.3.1.5 Resonance

There is hardly any machine installation, in which the natural frequency of any of the components does not match with any significant exciting force generated by the machine.

Typical symptoms:

i) Amplitude and phase Vs RPM plots obtained during startup or shut down will have the characteristics peak amplitude and 180 degree shift in phase, if resonance is present.

ii) High amplitude ratios in the amplitude Vs frequency plot at a particular location (horizontal or vertical, axial).

Given below are some of the methods for determining the natural frequency.

i) Keep a vibration pickup preferably, accelerometer attached to the component whose natural frequency is to be measured and put the vibration analyser in the 'filter out' position. Strike the component with a soft object such as a block of wood with a force just sufficient to cause it to vibrate. The natural frequency of the component vibrated will be indicated by the frequency meter. Since more than one natural frequency may be excited by the above method, the frequency meter may not lock on momentarily to a particular value. In this case, viewing the time waveform in an oscilloscope would help us to identify the different frequencies.

ii) Perform a routine frequency analysis using a vibration analyser. While the machine is being bumped repeatedly, tune the analyser filter over the various frequency ranges to identify all the natural frequencies being excited.

After identifying the natural frequencies as mentioned above, we can establish whether resonance exists or not.

5.3.1.6 Rub

Rub is generally a transitory phenomenon. Rubs are said to be accompanied by a great deal of high frequency spectral activity. A flattened waveform and orbit are strong indications of a rub. While a rub and rub symptoms may be observed on start, it is highly unlikely that the same symptoms, on a machine, which has been operating for long, would be caused by rub.

5.3.1.7 Electric Discharge

Bearing failure due to a high shaft voltage discharging through bearings is caused by the buildup of static electricity on a magnetized rotor or a rotor passing through a humid atmosphere.
The static electricity discharges across the path of least resistance at the small clearances of bearings and the same produces a frosted surface on the bearings, accompanied by spark tracks which in extreme cases will lead to pitting and loss of bearing material.
This problem does not alter any of the commonly measured external characteristics and hence difficult to recognize at its initial stage. A voltage from the shaft to ground is, of course, the direct way to identify this condition. A loss of bearing material, sensed as a changing DC gap voltage from a non-contact shaft transducer, is often the first warning for this condition.
Electric discharge, under some condition can be observed in the time waveform obtained on a turbine. The irregularity of the waveform was extremely unstable, sometimes present and some times not and continued changing despite changing pickups, cables and oscillator demodulators.

5.3.2 Vibration Characteristics of Component

5.3.2.1 Antifriction bearing

Typical symptoms:

i) A defective will not generally cause a single discrete frequency of vibration but instead may cause vibration at several frequencies simultaneously.
ii) Flaws on the race ways, rolling element or cage will cause a high frequency vibration.
iii) The vibration generated would be somewhat random or unsteady. Observation of the rotating shaft with the strobe light will not show a stationary image.
iv) The vibration generated would either be at rotational frequencies or component resonant frequencies.
v) The impact of a bearing fault will excite different components to different degrees simultaneously and hence the vibration would be

different frequencies. It is unlikely that these natural frequencies would be exact multiples of RPM.

Malfunction causing defects in the antifriction bearing and the component likely to be affected in the bearing can be co-related as given below:

- If an antifriction bearing has developed a defect because of excessive unbalance, the inner race will typical be the first bearing component to show signs of deterioration.
- If the bearing has developed a defect because of background forces or misalignment, the outer race will show signs of deterioration.
- Improper lubrication, overheating or the passage of electric current through the bearing would manifest in the form of rolling element failure.

5.3.2.2 Journal bearings:

Normally, the problems associated with journal bearings are excessive bearing clearance, improper bearing load and lubrication problem which result in high levels of vibration.

A journal bearing with excessive clearance would allow a relatively minor vibratory force (e.g. unbalance, misalignment) to cause vibration at 2 times, 3 times or some higher multiple of shaft RPM.

A wiped journal bearing in several instances have been found to cause vibration whose amplitude in the vertical direction is unusually high compared to horizontal amplitudes. Moreover a wiped bearing with excessive clearance which allows the shaft to change position within the bearing, may also result in misalignment.

Oil Whirl:

Oil whirl normally occurs on machines equipped with pressure lubricated sleeve bearings and operating at speeds above the critical speed of the rotor.

Under normally operating conditions, the shaft of the machine will ride up the side of the bearing slightly as shown in fig. 5.17. The shaft operating at an eccentric position from the bearing centre draws oil into a wedge to produce a pressurized load carrying film. Since the oil molecules adjacent to the shaft tend to remain stationary, the oil film is subjected to shear. The oil film will have a tendency to rotate at a speed, which is an average of shaft speed, and bearing speed. Since the bearing is stationary, the average rotating speed of the oil film would be approximately one half shaft RPM.

However due to friction losses, the average oil film speed will be slightly less than one half shaft RPM.
The force generated by this rotating oil film is normally small when compared to other sources of static and or dynamic loading of bearing. However conditions can occur where the force of the rotating oil film becomes the predominant force in which case the oil film simply pushes the shaft around with in the integral clearance of the bearing. This is known as oil whirl.

The condition of oil whirl can be attributed to the following:

i) Improper bearing designs such that the static loading of the shaft journal against the bearing is too small.
ii) Excessive bearing wear. (As a bearing wears, the shaft will ride more and more eccentrically and become susceptible to oil whirl.)
iii) Increase in lube oil pressure or viscosity.
iv) Background vibration at the oil whirl frequency.

Whenever the vibration characteristic of oil whirl is found, a complete vibration survey of the equipment including background sources, foundation and related piping should be made to determine the cause. Several special bearing configurations are recommended to minimize the possibility of oil whirl. The three individual bearing surfaces in the lobed bearing generate pressurised oil films that act to centre the shaft within the bearing thus providing improved bearing stability against oil whirl. In the title pad bearing used on large high speed centrifugal compressors and centrifugal pumps each tilting pad provides a pressurised oil wedge which tends to centre the shaft in the bearing.

Dry whirl:

Improper lubrication of a journal bearing causes high frequency vibration, which produces the distinctive squeal normally, associated with a dry bearing. This is known as dry whirl. The frequency of vibration generated is not likely to be a direct multiple of RPM or some higher multiple of shaft.

5.3.3 VIBRATION CHARACTERISTICS OF EQUIPMENT

5.3.3.1 Centrifugal machinery

In addition to vibration component at running speed and its associated low order harmonics, most centrifugal machinery will generate prominent components at the vane passing frequency (number of impeller vanes x shaft RPM) followed by a series of harmonics.

Some unique problems other than general malfunctions discussed earlier and their characteristics vibrations symptoms pertaining to some centrifugal machinery have been discussed below:

Centrifugal pump:

Hydraulic forces

Most centrifugal pumps exhibit some vibration due to inherent hydraulic forces, which is the result of pressure pulsations within the pump created whenever an impeller vane passes a stationary diffuser. This vibration normally occurs at a frequency equal to the number of vanes on the impeller times shaft RPM. If the impeller is centrally located within the pump housing and is properly aligned with the pump diffusers, the amplitude of vibration from hydraulic forces will be minimal.

Normally hydraulic forces will be inherent in the operation of a pump and will become a source of excessive vibration if some part of the system or supporting structure happens to be in resonance at that particular frequency.

Cavitation:

Cavitation in centrifugal pumps results whenever the pump is operated at low suction pressures where the liquid starts vaporising. In this condition, highly unstable cavities of nearly perfect vacuum are formed which collapse or implode quickly on reaching the zone where the pressure is above the vapour pressure. These impacts the natural frequencies of the pump housing, impeller and other pump parts. Since these implosions may occur at random intervals at various locations within the pump or piping the resulting vibration will be random in amplitude and frequency. Hence the vibration would cover a rather broad frequency range where individual amplitudes and frequencies are constantly changing. A distinctive noise something similar to sand/gravel/rocks being pumped depending on the intensity of cavitation often accompanies cavitation.

Recirculation:

Recirculation is just the opposite of cavitation and normally occurs when a pump is operated at high suction pressures. This causes excess fluid to return from the discharge to the impeller. This reverse flow in the pump and the mixing of fluids moving in opposite directions results in random vibration similar to cavitation.

Fans and Blowers:

Aerodynamic forces

Fans and blowers will inherently have some vibration due to aerodynamic forces. This vibrations results from the fan blades striking the air and will occur at a frequency equal to the number of fan blades times RPM. Normally the amplitude of vibration resulting from aerodynamic forces will be low.

The factors, which cause vibration at aerodynamic frequency, are:

- Resonance of some part of the machine.
- Obstruction, which would disturb the smooth flow of air through the fan.
- Eccentric positioning of the fan rotor in the fan housing.
- Fan blades do not have the track or pitch.

Flow turbulence:

Flow turbulence is the result of variation in pressure or velocity of the air passing through the fan or associated duct work. Anything that disrupts the smooth flow of air through the system such as sharp right angle turn or changes in the cross sectional area of the air passage will result in flow turbulence. The vibration due to flow turbulence will generally be random and the frequencies excited may range **50 CPM upto 2000 CPM.**

5.3.3.2 Bladed machinery:

Bladed machinery where the direction of flow is axial such as stream turbines will usually generate complex vibration characteristics, particularly in the higher frequencies, compared to centrifugal equipment. As a general observation, the blade frequencies observed in stream turbine spectra will be at much lower amplitudes than the blade passing frequencies generated by axial compressors and gas turbines. Vibration amplitudes at blade passing frequencies have been found to vary significantly with changes in speed, pressure ratio and stator blade angle in this type of machinery.

5.3.3.3 Gear Drives:

Gears typically generate a complex, broad vibration spectrum, beginning with frequencies well below the rotating speed and extending to several multiples of gear mesh frequency, as shown in fig. 5.23. Gear meshing frequency is the no. Of gear teeth times the RPM of the shaft. Intermediate frequencies appear approximately midway between the rotational and gear mesh frequencies and a change in amplitude at these frequencies has been a very sensitive and primary indicator of gear failure.
The gear mesh frequency and its harmonics are generally the most prominent component in the vibration spectra and it may be surrounded by side bands spaced on either side of mesh frequency at intervals of shaft RPM. As a general rule the larger the number of teeth, lower the gear ratio, higher the quality of tooth finish and lower the load applied to the gear, the lower will be the amplitude at mesh frequency. fig.5.24. Illustrates the vibration in amplitude at mesh frequency with changing load of reduction gear unit of a gas turbine generator.
When the gears are operating under a very light load condition, the amplitude of vibration and the frequency would be erratic. Under light load conditions, the load randomly shifts back and forth from one gear to the next. The impacts which occur when the load is shifted will excite the natural frequencies of the gears, bearings and other associated machine component alongwith gear meshing frequencies and their harmonics.
An eccentric gear or gear drive which is subjected to excessive misalignment will often reveal peak at side band frequencies near the gear mesh frequency. The typical side band frequencies normally encountered are those at gear mesh frequency plus or minus rotating speed. If the predominant vibration due to misalignment is at a frequency of **2 x RPM** of the shaft, additional side band frequencies at gear mesh frequency plus or minus **2 x RPM** may also be present. As gear tooth clearances increase, the initial tooth impact may cause the gear tooth to bounce within the clearance available resulting in vibration frequencies at harmonics of gear mesh frequency (2, 3 or higher multiples of gear mesh frequency).
If a gear has only one broken, deformed or cracked tooth, peak in the vibration spectrum occurs at hunting tooth frequency which is equal to RPM of the gear divided by number of teeth on the gear. In this case viewing the time waveform on the oscilloscope would help us to distinguish spike like

signal caused by the faulty gear tooth. If more than one gear tooth is defective, frequencies of vibration would be at multiples of gear RPM. Common gear problems which cause vibrations at gear meshing frequencies are excessive gear wear, improper adjustment of backlash, gear tooth inaccuracies, faulty lubrication and eccentricity. In addition to these gear problems, other exciting forces such as misalignment or bent shaft in the machinery may also cause the similar vibration characteristics. Internal misalignment of the gear box bearings cause by the distortion of the gear box casing either from mountings on a warped base plate or due to stress relieving thermal effects or other internal forces occuring with age would also cause gear vibration. As eccentric gear, misalignment or a bent shaft results in variations of tooth clearances for each revolution of the gear, the amplitude of vibration at gear mesh frequency would appear modulated in the time waveform.
In this case of spur gears, high amplitude in the axial direction generally indicates misalignment. But in the case of helical gears, relatively high axial vibrations are quite common since the normal gear load includes an axial as well as a radial component.

5.3.3.4 Electric Motors

Vibration problems concerning motors can be grouped basically the following categories:

- **Electrically induced vibration.**
- **Mechanical vibration.**

In electric motors, often a complex vibration pattern which includes both electrical and mechanical vibration inputs and there is always some interaction between mechanical and electrically induced vibration. Mechanical problems especially unbalance and eccentricity can also lead to electrically induced vibration. Electrical problems are caused by elliptical stators; open motor bars, shorted stator turns, loose bars or stator windings etc. Modulation occurs between electrically induced and the mechanical vibration because the rotor of an induction motor turns at a slightly lower frequency than the A.C supply voltage frequency. This is termed as slip frequency. This is generally less than the line frequency by **0.3 to 0.5 Hz (18-30 CPM).** The modulation beats in the slip frequency and its multiples which is often audible. The beat frequency is **0.3 – 0.5 Hz** or twice this amount.

Fig. 5.25.a. shows the typical time waveform in the case of rotor unbalance, fig. 5.25.b. shows the typical time waveform in the case of an electrically induced vibration caused by eccentricity of the rotor within the stator. Mechanical unbalance causes vibration that can described by a vector rotating at the same speed as the rotor. However, the eccentricity of the rotor causes a non-symmetrical magnetic force that is greater at the South Pole than the north as shown in fig. 5.26. This magnetic unbalance rotates at the line frequency. The two signals therefore amplify and attenuate cyclically, beating at **18 to 30 CPM**. The maximum amplitude occurs when the two vectors are in phase and the minimum amplitude occurs when they are 180 degree out of phase.
Generally speaking, the signal tends to be more complex than this simple description because the unbalance causes electrical non-symmetric as well as mechanical vibration. Unbalance causes, the rotor to deflect so the air gap is smaller at the heavy point or approximately 180 degree opposite the heavy point if the rotor operates above its first critical speed. Hence a point of minimum air gap travelling at rotor speed passes 'n' poles per revolution (in a 'n' pole motor) and causes an unbalanced magnetic pull between the poles. Since the rotor speed lags the line frequency a maximum unbalance force with a frequency equal to twice the slip frequency is produced, introducing another modulation frequency.
Another phenomenon that occurred in all motor, but is more obvious in 2 pole induction motors, is caused by the radial magnetic force between rotor and stator cyclically distortion the stator at the line frequency times the number of poles (e.g. **100 Hz** for a 2 pole motor operating at **50Hz).**
It can be seen that a magnetic unbalance (e.g. eccentricity) occurs at **50Hz** (i.e. line frequency) where as a mechanical unbalance occurs at **50 Hz** minus **0.3 to 0.5 Hz** (i.e. slip frequency).

An effective way to distinguish between electrical and mechanical problem is to cut power while the motor is running at full load. All magnetic forces disappear when power is cut off. Whenever modulation exists, electrical or a combination of electrical and mechanical problems are indicated.
Given below are the common problems and frequency of vibration in electrical motors.

CAUSE	VIBRATION FREQUENCY
Rotor eccentricity in bore	line frequency
Eccentricity rotor	2(line frequency + slip frequency)
Elliptical stator	2(line frequency)
Open rotor bar	2(line frequency + slip frequency)
Non symmetrical bearings	2(running speed)

6.0 SHOCK PULSE MEASUREMENT

7.0 IMPLEMENTATION

6.0 SHOCK PULSE MEASUREMENT

6.1 INTRODUCTION

Several methods are currently used to monitor antifriction bearing condition like temperature, sonic and ultrasonic noise, accoustic emission measurement techniques and vibration analysis. Among these techniques, vibration analysis is the most common. The disadvantage with vibration analysis technique for this application is that by the time the vibration signals become large enough to be detected reliably, the bearing is about to fail, in many cases, giving little warning time. A method based on monitoring the mechanical impacts caused by bearing damage and operating condition problems known as Shock Pulse Method (SPM) is available now and is being used by us.

6.3 PRINCIPLE

There are certain irregularities present on the surface of the race ways and rolling element of any antifriction bearing whether new or used and these cause mechanical impacts when the bearing is in motion. These mechanical impacts generate short duration pressure pulses, which are named as shock pulses. The magnitude of the shock pulses depends on the impact velocity. The impact can be divided into two phases. In the first phase, no detectable deformation of the colliding masses takes place and in the second phase deformation takes place causing vibration. In the first phases, during the moment of impact, molecular contact occurs and a shock/compression wave develops in each mass. The molecular contact results in an infinitely large particle acceleration at the impact point. The acceleration of the material at the impact point sets up a compression wave, which propagates in all directions. Magnitude of mechanical impact is measured by detecting and measuring the resultant compression wave front. The compression wave front sets up a dampened oscillation whose peak amplitude is proportional to the impact velocity. Since the dampened transient is well defined and of constant decay rate, it is possible to electronically filter out vibration signals. The measurement and analysis of this peak amplitude is the principle behind SPM method.

6.3 TRANSDUCERS

The SPM system uses a piezo electric accelerometer to measure the mechanical impact. This transducers is mechanically and electrically tuned to a resonant frequency of **1920 K CPM.**

6.4 UNIT

The sensitivity of the SPM method is such that the shock pulses generated by a typical antifriction bearing increases upto **1000** times from when the bearing is in good condition to the condition when the same is about to fail. For covering this large range a logrithamic scale is used and the shock pulse values are expressed in decibels (db).

6.5 PICKUP POINTS

Shock pulse are generated mainly in the load zone of the bearing and spread spherically from the point of impact through the bearings, its housing and adjacent machine parts. The shock pulses are dampened when they pass an interface or are forced from their straight path. The following guidelines may be used for selection of pickup points.

i) The signal path must contain only one mechanical interface i.e. between the bearing and bearing housing
ii) The signal path between bearing and pickup joint shall be as straight and as short as possible.
iii) The pickup joint shall be in the load zone of the bearing

6.6 TERMINOLOGY

Before we go ahead with the features and use of the instrument, we discuss some of the terminologies associated with shock pulse measurement.

Initial shock value (dbi):
Even a new, properly installed and properly lubricated bearing generates shock pulses. This is known as initial shock value db. This value is primarily dependant on the rotating speed and the bore diameter.

Shock value (dBsv):
The absolute strength of the shock pulses emanating from the bearing is known as shock value dBsv.

Normalised shock value (dBn):

The increase of shock value (dBsv) above initial shock value dbi is defined as normalised shock value dBn. The normalised measuring scale is used for measuring shock pulse value in the Shock Pulse Meter. The normalised measuring scale starts from dbi and shows only that part of shock value which is directly related to the condition of the bearing being monitored.

Carpet value (dBc):

Surface roughness will cause a rapid sequence of minor shock pulses which together constitute the shock carpet of the bearing. The magnitude of the shock carpet on the normalised measuring scale is expressed by the carpet value dBc. This value helps to analyse the cause of reduced or bad operating condition.

Maximum value (dBm):

Any damage in the bearing will cause shock pulses with higher magnitude at random intervals. The highest shock pulse value measured on a bearing is a bearing is called it maximum value. This determines the operating condition of bearings.

6.7 INSTRUMENT

6.7.1 Features

The basic measuring equipment consists of the shock transducer with probe which picks up the shock pulses and the Shock Pulses Meter which measures the magnitude.

There is a black annular ring on the dial of the instrument whose one half is calibrated in terms of dbi and the other half in terms of RPM. Inside this, there is a white annular ring whose one half is calibrated in terms of dbn and the other half in terms of shaft diameter in mm. There is a thumb wheel with which we measure shock pulse value (fig.6.3). The instrument has an inbuilt microphone with provision for attaching a stethoscope.

6.7.2 **Operation**

The steps involved in using the instrument have been discussed below.
Setting up dBi:
Turn the dial of the instrument such that the shaft diameter/ bore diameter in mm and the rotating speed of the shaft align. The dbi can read in the dial as shown in fig. 6.3.
Measuring dBm:

After setting dBi in the meter, turn thumb wheel from the position where the pointer is kept at **dBn = 0.** Keep turning the thumb wheel in increments of **5dB** until the sound disappears. After waiting for few seconds, turn the thumb wheel back and forth to confirm the exact position where the sound appears after a certain time interval. The value is recorded as dBm as shown in fig. 6.4.

Measuring dBc:

The carpet value must be measured on newly installed bearings and bearings with maximum values above **20 dBn.** It is read at the instrument setting where the continuous tone breaks into a very fast sequence of sound pulses.

6.8 DATA INTERPRETATION:

An evaluation flow chart has been given in table 6.1. From the shock pulses diagram of the bearing being monitored, we can give the probable causes with the aid of this chart.

7.0 IMPLEMENTATION

7.0 IMPLEMENTATION

Predictive Maintenance Programme essentially requires two types of skill. The first type of skill is the knowledge of equipment's working, common malfunctions and methods to repair which we already possess. The second type of skill is the ability to recognize and pinpoint the malfunctions in the equipment, in the early stages of their development with the aid of parameters being monitored. This skill has been developed to a large extent in our refineries.

7.1 STEPS

Basic steps required for organising a Predictive Maintenance Programme have been discussed below.

7.1.1 Equipment list

The criticality of equipment in the refineries have been classified. A list of equipment in each refinery classifying them into three categories namely critical, semi-critical and non critical has already been prepared. The basis for this classification was arrived at in the I Rolling Plan Workshop which is given below:

"The equipment that can cause unit shut down are to be classified as critical equipment. The equipment that can cause production loss are to be classified as semi-critical equipment. Rest of the equipment are to be classified as other equipment."

7.1.2 Acceptable Level

Based on the information available in **BS 4675 Part I :**
1976(ISO 2372) and various Vibration Severity Charts, acceptable levels of vibration in terms of displacement or velocity as the case may be have been established by the refineries.

As far as shock pulse values are concerned, the manufacturer's recommendation given below can be followed.

Operating Condition	dBm
Good	0-20
Reduced	20-35
Bad	35-60

7.1.3 Equipment Condition And Normal Level

To begin with, take 'filter out' vibration velocity measurement in all the directions at all the bearings. These readings should be taken after commissioning in the case of new equipment and in the case of existing ones after overhaul. If the 'filter out' readings are low indicating that the equipment is in good operating condition, these should be entered as initial vibration level in the Baseline PM Data Sheet. While taking these readings, take care to see that the equipment operates at parameters which it would normally be operating at. If the 'filter out' readings are high indicating component defect, a complete vibration analysis is called for. After making necessary corrections based on the analysis, 'filter out' readings should be taken at each bearing to confirm that the equipment has returned to an acceptable condition which can be entered as initial vibration level.

In addition to the 'filter out' readings, signature plots (vibration velocity vs frequency) should be taken in all directions at all the bearings. These plots are called Baseline Machinery Signature which offer the following advantages:

i) Reveals the presence of very low or very high vibration frequencies which would help to decide whether vibration displacement and/or acceleration to be measured in addition to velocity at the bearing/s concerned.
ii) Identifies components in the equipment contributing vibration, on the basis of which periodic check points and frequency of monitoring can be decided.
iii) Serves as a reference for comparison at future. Vibration signature taken at the same point with the equipment operating under the same condition when compared with the Baseline Signature will help to pinpoint the defect.

For shock pulse measurement, initial shock pulse (dbi) can be found out from the shock pulse meter, if the shaft rpm and bore diameter of the bearing are known.

7.1.4 Check Points

Periodic check points should be selected taking into account the type of troubles an equipment is likely t develop so that measurements at these points represent the normal condition of the machine. Normally one measurement point should be selected for each component of the system(e.g. motor, gear box, pump). The bearing and the direction of measurement which reveals the highest level of vibration should be selected.

In the case of antifriction bearings, there should be a periodic check point at each bearing for vibration monitoring and there should be periodic measurement of shock pulse at each bearing.
Each periodic check point should be clearly marked for easy identification to ensure that the vibration/ shock pulse measurements are taken at the same location in future.

7.1.5 Frequency

While selecting the frequency of vibration monitoring, manpower required, number of equipment covered under Predictive Maintenance, number of checkpoints, critically of the equipment etc. were considered. Accordingly, it was decide in the I Rolling Plan Workshop that the following frequencies would be adhered to in the order of critically of equipment.

Critical equipment	**- Once in two weeks.**
Semi critical equipment	**- Once in four weeks.**
Other equipment	**- To be decided by the Maintenance Deptt. of the refinery concerned.**

The frequency of measuring shock pulse value depends on the magnitude of maximum shock value. The frequency recommended by the manufacturer is given below.

dBm	**Frequency**
0-20	1-3 months
20-35	1-2 times/week
35-60	Daily

7.1.6 Instrument

Required of vibration measuring instruments (Vibration Meter, Vibration Analyser) and other condition monitoring instruments for all the refineries was consolidated in the I Rolling Plan Workshop and procurement action was taken thereafter. All the instruments have been received by the refineries and a list of the various condition monitoring instruments available.

7.1.7 Data recording system

At the I Rolling Plan Workshop, the following formats were standardised:

i) Machine Vibration Log.
ii) Baseline PM Data Sheet.
iii) Vibration Measurement Record.

Machine Vibration Log is to be used by the field engineer for recording the 'filter out' vibration amplitude readings.
Baseline PM Data Sheet identifies an equipment, its rotating speed, initial vibration level and vibration limits. This has a provision for plotting the vibration trend in the form of 'trend chart'. The format has been given in fig.7.2.
Vibration Measurement Record is used for recording the 'filter out' vibration amplitude readings and 'filter in' vibration amplitudes at various frequencies. The format has been given in fig.7.3.
Shock Pulse Measurement Record in which maximum shock value and carpet value can be marked and shock pulse trend plotted is given in fig.7.4.

7.2 PREDICTIVE MAINTENANCE ON EDP:

With a view to improve the efficiency of the units, a computer based Maintenance Planning, Schedule and Monitoring System was designed in December '83 whose phase I consists of three modules which are given below:

i) Predictive Maintenance Module
ii) Preventive Maintenance Module
iii) Work Request Monitoring

The Predictive Maintenance module would generate the following output reports as shown in flow chart of Predictive Maintenance on EDP in fig.7.5.

i) Equipment due to Predictive Maintenance (C2)

This is a list of equipment due for Predictive Maintenance (vibration monitoring) in each maintenance area alongwith the criticality of the equipment concerned with their due dates. The frequency of this report will be decided by the refineries.

ii) Rotary equipment vibration reading card (C3)

This is basically a performa for recording the vibration amplitude measured in the rotary equipment, which were due for Predictive Maintenance. This output report will be generated by the computer alongwith C2.

iii) Reminder for delayed vibration (C4)

Based on the feedback received in C3 output report, this report will be generated. The frequency of this report shall match with that of C2. However, a monthly report will be generated which would go to the CMNM highlights the cases where vibration monitoring has been delayed beyond one month.

iv) Rotary equipment abnormal vibration report(C5)

This would serve more as a history giving the vibration amplitudes measured at various point in different directions in a chronological order along with the vibration limit. This report would be generated as and when the vibration level in an equipment becomes abnormal. In addition, this report will also be generated monthly/half yearly which would serve as history.

v) Equipment fault analysis report (C6)

Based on the vibration analysis done and the codified information regarding equipment status, fault, action required this report is generated. This report is generated as and when a vibration analysis is done.

vi) Planned/actual Predictive maintenance analysis report (C7)

This output report gives the percentage of vibration monitoring of rotary equipment done out of the planned quantity. This also gives the number of fault cases observed and the percentage of the total number of rotary equipment whose vibration was monitored.

8.0 BALANCING

8.0 BALANCING

8.1 INTRODUCTION

Among the various machinery malfunctions discussed earlier, unbalance has been found to be one of the common causes of vibration present to a certain extent on nearly all the rotating equipment. Unbalance is defined as the unequal distribution of the weight of a rotor about its rotating centerline. There are many reasons which causes unbalance in a rotating part and some of the common causes have been listed below:

1) Blow holes in the cast components in the rotor.
2) Eccentricity (Geometric centerline of the rotor does not coincide with the rotating centerline)
3) Lack of uniform norms for balancing of key (motor manufacturer Vs pulley manufacturer).
4) Distortion due to stress relief/ thermal distortion.
5) Accumulation of clearance/ tolerances of different mating parts in the rotor.
6) Corrosion and wear.
7) Deposit build up.

The vector sum of all the unbalance in a rotor can be considered as a concentration at a point termed 'heavy spot'. Unbalance in a rotating part is expressed as the product of unbalance weight and the radial distance from the rotating centerline i.e. gram centimeters.
Forces created by unbalance are determined to the life of the rotating equipment and cause vibration of the rotor an supporting bearings. This force is directly proportional to the square of the rotating speed and the amount of unbalance as given below:

$$F = 0.01 \times \left(\frac{RPM}{1000}\right)^2 \times \alpha$$

α = Amount of unbalance in gram centimeters.

We can infer from the above equation that more the amount of unbalance greater the force and hence greater the vibration amplitude.

Balancing is a technique for determining the amount (gms) and location (radial distance from the centerline, angle) of the heavy spot so that either an equal amount of weight added directly opposite.

8.2 TYPES OF UNBALANCE

Unbalance can be classified into four categories depending on the relationship between the central principal axis and the rotating centerline which are described below.

8.2.1 Static Unbalance

In this type of unbalance the central principal axis is displaced parallel to the centerline as shown in fig. 8.2a. A rotor with this type of unbalance when placed on parallel knife edges, will swing bringing the heavy side to the bottom. A rotor with static unbalance supported between bearings will reveal identical vibration amplitude and phase reading at the bearings. This type of unbalance can be corrected by adding/ removing weight in one plane but in the proper plane as shown in Fig. 8.2.a.

8.2.2 Couple Unbalance

In this type of unbalance the central principal axis intersects the rotating centerline at the rotor center of gravity. This condition is caused by a heavy spot at each end of the rotor as shown in fig.8.2.b. A rotor with couple unbalance supported between bearings will reveal identical vibration amplitude but phase readings which differ by 180 degree at the bearings. Couple unbalance can only be corrected by making weight corrections in two planes.

It is very rare to find a rotor having a pure static unbalance or couple unbalance. Normally rotors have a combination of both and the combination can be further classified as quasistatic unbalance and dynamic unbalnce.

8.2.3 Quasistatic Unbalance

Quasistatic unbalance is that type of unbalance where the central principal axis intersects the rotating centreline but not at the centre of gravity of rotor as shown in fig.8.2.c. A rotor with quasistatic unbalance supported between bearings will reveal higher vibration amplitude at one end and phase

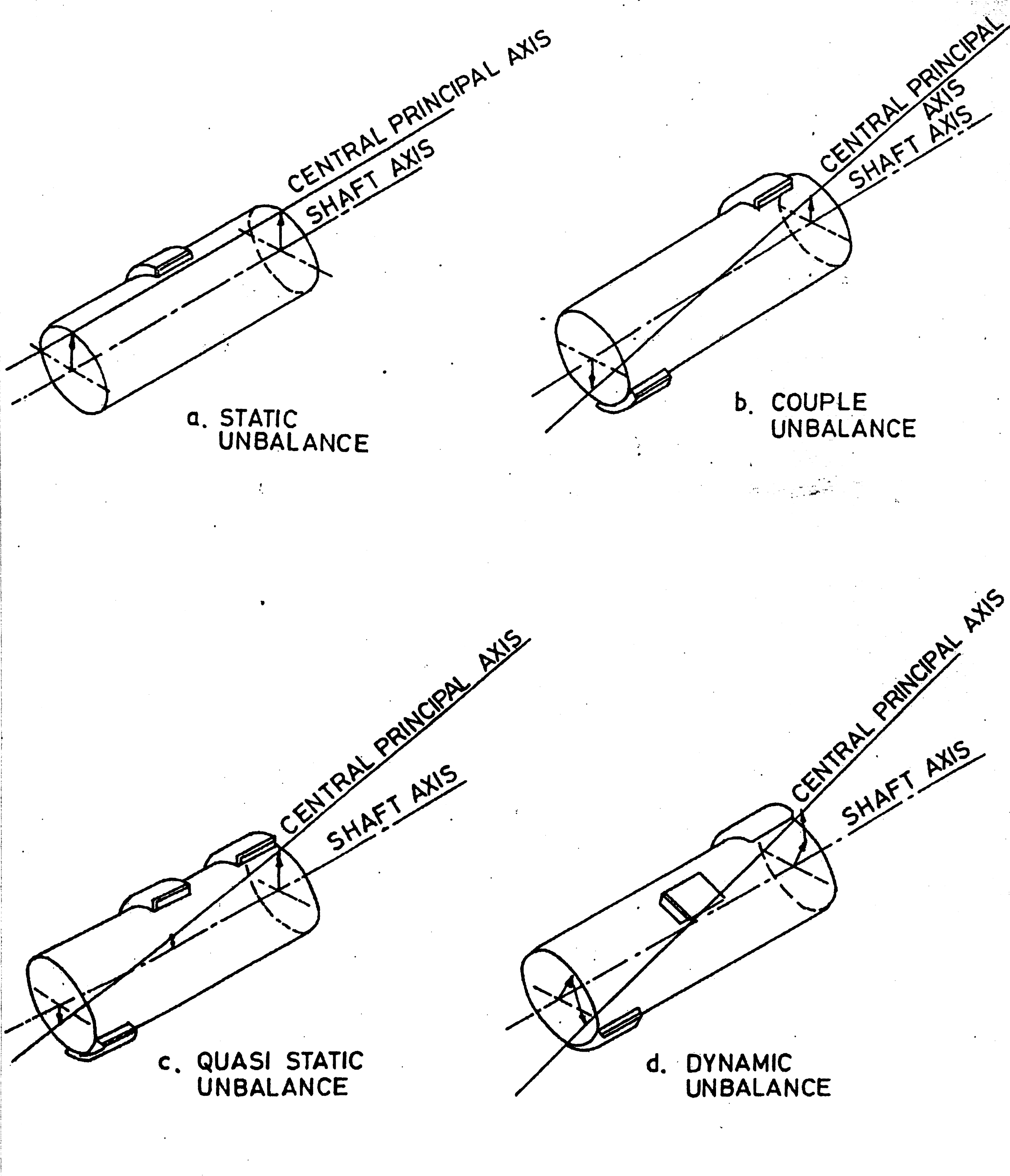


FIG. 8·2 TYPES OF UNBALANCE

readings at the bearings differing by 180 degree. Quasistatic unbalance can only be corrected by making weight corrections in at least two planes.

8.2.4 Dynamic Unbalance

Dynamic unbalance is that type of unbalance where the central principal axis and the rotating centerline do not coincide or touch. The central principal axis is both tilted and displaced from the rotating centerline. A rotor with dynamic unbalance will reveal comparative phase readings at the bearings which are neither the same nor directly opposite one another. Dynamic unbalance can only be corrected by making weight corrections in at least two planes.

8.3 TYPES OF ROTOR

The rotor can be classified as rigid rotor and flexible rotor depending on the relation of the rotating speed to the natural frequency of the rotor.

8.3 .1 Rigid Rotor

Rotors which operates below 70% of their critical speed are considered rigid. These rotors do not bend or deflect due to forces caused by unbalance. Any condition of unbalance in a rigid rotor can be compensated by making weight corrections in any two balancing planes.

8.3 .2 Flexible Rotor

Rotors which operate above 70% of their critical speed which bends or flex due to unbalance forces are called flexible rotors. A flexible rotor balanced at one operating speed may not be balanced when operating at another speed. As an example, refers the unbalanced rotor shown in fig 8.3.a. having dynamic unbalance. If this rotor was balanced at a speed below 70% of the first critical speed with correction weights added in the end planes the correction weights could compensate for all sources of unbalanced distributed throughout the rotor as long as the rotor operates at a speed 70% below the first critical speed. If the same rotor operates at a speed above 70% of the critical speed , the rotor will deflect due to the centrifugal force of the unbalance located at the central portion of the rotor as shown in fig 8.3.b.

As the rotor bends or deflects, the weight of the rotor is moved out away from the rotating centerline creating a new unbalance condition. This new unbalance can be corrected but the rotor would be out of balance at slower speeds where the rotor is rigid. The only solution to ensure smooth operation of the rotor at all speeds is to make balance corrections in the actual planes of unbalance. Hence the flexible rotor requires balancing in three planes.

Whether a flexible rotor must be balanced in more than two planes can be determined by the normal operating speeds of the rotor and the significance of the rotor deflection on the fundamental requirement of the rotating machine. However, the following general guidelines can be followed:

i) If the rotor operates at only one speed and a slight amount of deflection will not harm the machine, balancing in any two correction planes would be sufficient.
ii) If the flexible rotor operates at only one speed and rotor deflection should be very less, multiplane balancing would be required.
iii) If a rotor is required to operate at a broad range of speeds where the rotor is rigid at lower speeds but becomes flexible at higher speeds, multiplane balancing is required.

A flexible rotor can deflect in several ways depending on its operating speed and the distribution of unbalance throughout the rotor. The first. Second and third flexual modes a rotor could take at first, second and third critical speeds respectively have been given in fig 8.4.

8.4 SELECTION OF NUMBER OF BALANCING PLANE/S

Hence it is important to recognize the fact that not all balancing problems can be solved by balancing in a single correction plane. To determine whether single plane or two plane balancing is required, the ratio of the length of the rotor exclusive of supporting shaft to the diameter of the rotor can be used as a guideline. Since the unbalance of a rotor could be in any plane or planes located along the length, it is very difficult to locate the plane. Moreover, it may not always be possible to make weight corrections in just any plane. Hence the usual practice is to make corrections in the convenient planes available.

8.5 GENERAL PRINCIPLES OF BALANCING

For determining the amount of unbalance and location of unbalance, we measure the amplitude of vibration and the position of a reference mark on the rotor as observed under the strobe light respectively. While attempting to balance a rotor, two very important points to be considered are:

i) The amount of vibration is proportional to the amount of unbalance
ii) The reference mark seen under strobe light shifts in a direction opposite the shift of the heavy spot and the angle through which the reference mark shifts is equal to the angle through which the heavy spot has shifted.

At the start of a balancing exercise, the amount of heavy spot and its location are unknown and let us call the unbalance at he beginning as 'original unbalance'. The original unbalance will be identified by the vibration amplitude and phase readings known as 'original readings'. Since the amount and location of heavy spot are unknown, we add a random weight called 'trial weight'. The effect caused by 'trial weight' can be used to deduce the amount and location of unbalance. While adding 'trial weight' , We might be doing any one of the following:

i) Adding 'trial weight' right on the heavy spot.
ii) Adding 'trial weight' right opposite to the heavy spot.
iii) Adding 'trial weight' neither on the heavy spot nor opposite to it.

The effect of the above three possibilities on the vibration amplitude and the phase reference mark is given below:

Possibility	vibration amplitude	Phase reference mark
i)	Increases	No change
ii) (a) Trial weight less than the heavy spot	Decreases	No change
(b) Trial weight greater than the heavy spot	Increases	180° change
iii)	May change	Changes

In practice, the first two possibilities are rare. Third possibility is the most common.