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Vibration analysis and diagnostic techniques, with reference to the Implementation of a vibration condition monitoring programme.

Ducci, P P

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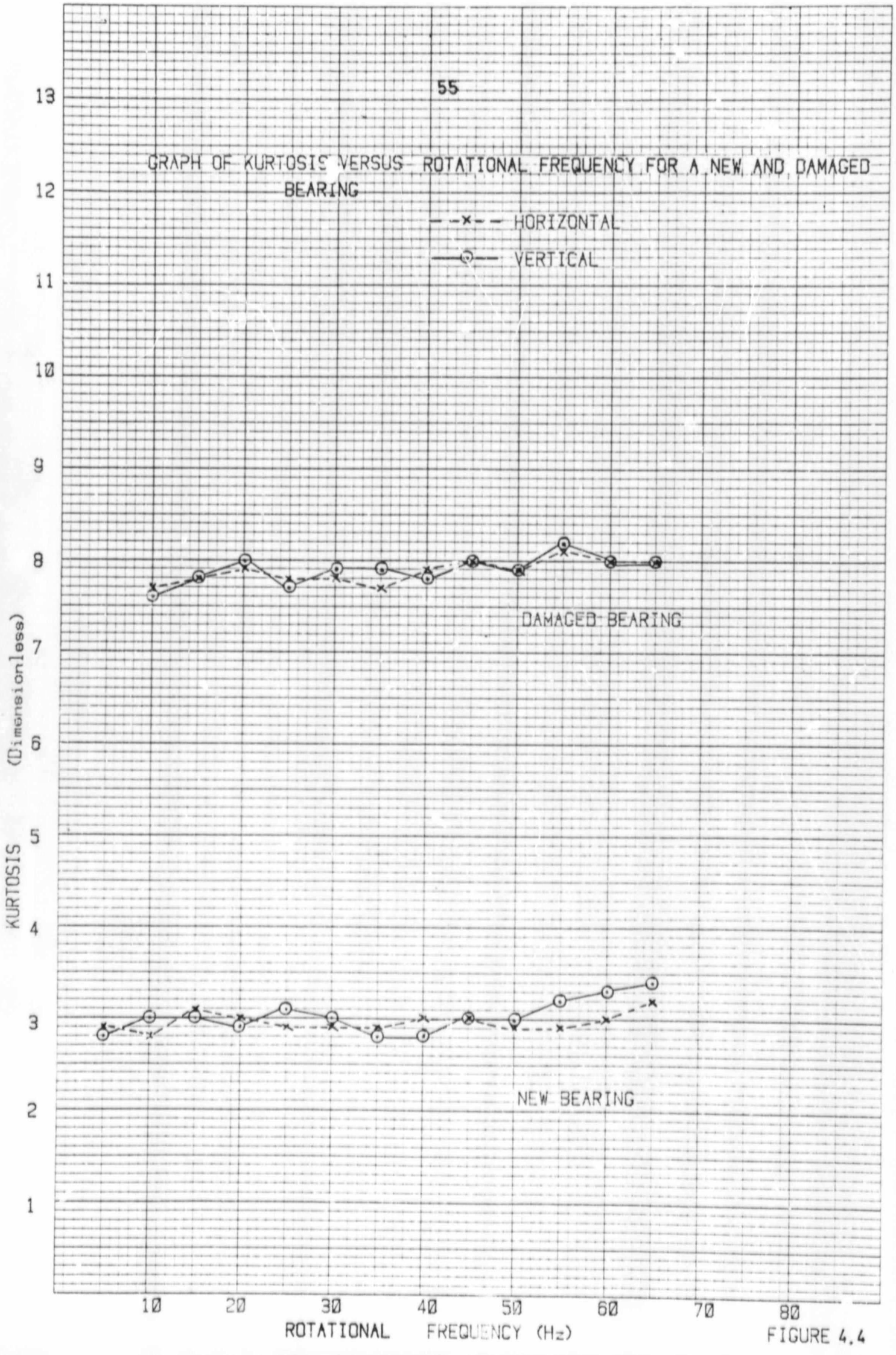

GRAPH OF KURTOSIS VERSUS ROTATIONAL FREQUENCY FOR A NEW AND DAMAGED BEARING
HORIZONTAL
VERTICAL
DAMAGED BEARING
NEW BEARING
KURTOSIS (Dimensionless)
ROTATIONAL FREQUENCY (Hz)
1
2
3
4
5
6
7
8
9
10
11
12
13
10
20
30
40
50
60
70
80

FIGURE 4.4

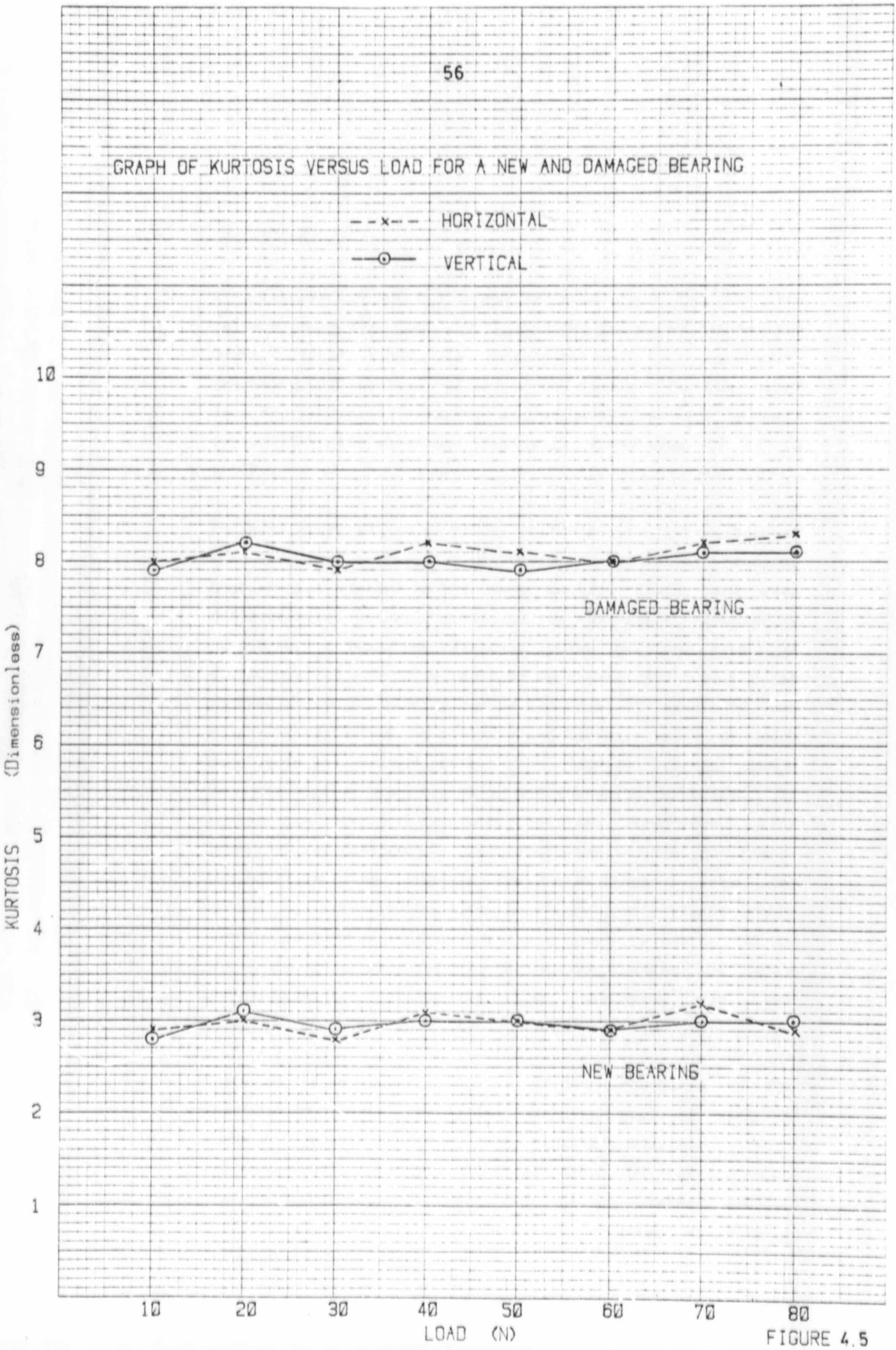

GRAPH OF KURTOSIS VERSUS LOAD FOR A NEW AND DAMAGED BEARING
HORIZONTAL
VERTICAL
DAMAGED BEARING
NEW BEARING
KURTOSIS (Dimensionless)
10
9
8
7
6
5
4
3
2
1
10
20
30
40
50
60
70
80
LOAD (N)

FIGURE 4.5

4.6 DISCUSSION

From Figures 4.4 and 4.5 it can be seen that the kurtosis value of 3, ± 10%, and 8, ± 5% obtained for a new and damaged bearing respectively, is independent of the rotational frequency and the load applied to the shaft. On the other hand the vibration severity in acceleration (g) is very dependent on the speed and load with variations of up to ± 50%.

The main disadvantage of using kurtosis is that vibration from a damaged bearing can be transmitted to other bearings that are not damaged in the same machine train, and even other machines in close proximity via the shaft, foundation or pipework. Hence the kurtosis value obtained from the abovementioned situation could be spurious. For a particular machine a damaged bearing can be isolated by examining and comparing the vibration severity and kurtosis value of each bearing in the machine train. The damaged bearing should have the highest kurtosis value and vibration severity. To isolate the vibration from other machines, synchronous time averaging was considered. Time averaging is used to reduce the level of noise. Signals that are fixed in the time record, i.e. synchronous with the trigger, will remain while non-synchronous signals eventually average to zero. Since the vibration signals from a bearing are random and hence nonsynchronous, synchronous time averaging was not successful.

Software was developed using the HP9836 interfaced to the Wavetek 100A. This statistical concept will be used at Koeberg in the near future, where they have many motors with roller and ball bearings. British Steel Corporation have been using kurtosis for monitoring rolling element bearing condition for some time and they have obtained encouraging results in the field.

The main advantage of kurtosis is that vibration measurements used for its derivation can be taken at any operating condition of the machine. This is very convenient because the operating conditions of machines in a power station vary from day to day. A periodic vibration monitoring programme operating on a stringent schedule will not be disturbed because vibration data was not obtainable, from a particular machine due to its operating condition not being the same as when the previous data was acquired.

Another advantage of this method is that it can easily be incorporated into a computer based monitoring system. Once the vibration data has been stored on disk it is relatively simple to develop software to retrieve the data and perform the necessary calculations to obtain a kurtosis value. Trending of the kurtosis values can also be achieved using the computer.

The maximum effectiveness of this method is achieved by trending the kurtosis values obtained. Trending of kurtosis values allows decisions to be made based on the history of the machine. A gradual increase in the Kurtosis value from the norm of 3 to approximagely 8 to 20 would be a clear indication that incipient failure is near.

Another important advantage of kurtosis is that it is independent of the size of the bearing, the pitch diameter, number of balls, diameter of balls and angle of contact, Rogers (22).

The Kurtosis values obtained are very dependent on the frequency range of the time window that was chosen. In this case a 3 Hz - 5 kHz and a 5 - 10 kHz frequency range was chosen with positive results for a new and damaged bearing respectively.

4.7 CONCLUSIONS

1) In laboratory tests performed, kurtosis proved to be an effective means of monitoring the condition of roller and ball bearings.

2) Load and speed has no effect on the kurtosis value obtained from a new and a damaged bearing.

3) The maximum effectiveness of kurtosis is achieved by trending the kurtosis values obtained.

4) The calculations necessary for the derivation of kurtosis values and the trending of these values can be incorporated into a computer based monitoring programme.

5) Kurtosis is independent of the size of the bearing, Rogers (22).

6) Kurtosis is very dependent on the frequency range chosen for the analysis of a kurtosis value.

5. MODULE 2
FAULT DIAGNOSIS OF GEARS USING SPECTRUM ANALYSIS

5.1 INTRODUCTION AND THEORY. LANG (28)

The operative health of a gear train is frequently indicated by the vibration it produces. For this reason the vibration signature exhibited by a geared machanism is useful for monitoring its performance and determining when it may require maintenance. The nature of the vibration signature indicates the precision of the components, the correctness of their assembly, and t' presence of any mechanical damage in the rotating compon

Geared systems can range from extremely simple two-shaft devices to highly sophisticated multi-shafted, multi-staged, epicyclic systems. Obviously, the degree of mechanical complexity determines the complexity of the resultant vibration spectrum. Fortunately, the spectrums exhibited by complex gearing may be understood as a superposition of the spectrums exhibited by simpler systems. The fundamental characteristics of the spectrum may be understood by examining the vibrations generated by a single pair of meshing gears.

The gear ratio, R, is simply the ratio of the number of teeth to the number of pinion teeth. In a system where the gear has T_g teeth and the pinion has T_p teeth, the gear ratio is:

$$R = T_g/T_p$$

The number of teeth on the gear and pinion determines not only the gear ratio, but the nature of the vibrations

exhibited by the gear pair. This spectrum will contain vibration components at discrete frequencies, dominated by sums, differences, and multiples of five basic frequencies. These are:

1) F_p = pinion rotation frequency; the frequency at which a specific tooth on the pinion enters mesh.

2) F_g = gear rotation frequency; the frequency with which a specific tooth on the gear enters mesh.

3) F_m = mesh frequency; the frequency at which teeth pairs enter the mesh.

4) F_h = hunting tooth frequency; the frequency at which specific tooth pairs, one on the gear and one on the pinion enter mesh.

5) F_a = assembly phase passage frequency; the frequency at which teeth belonging to a specific assembly phase enter mesh.

These frequencies always satisfy the inequality:

$$F_h < F_g < F_p < F_a < F_m$$

Teeth pairs engage at the mesh frequency, F_m

$$F_m = T_g F_g = T_p F_p \qquad \text{(Hz)}$$

where F_p = pinion rotation frequency in Hz
F_g = gear rotation frequency in Hz

A pair of mating gears may exhibit a series of unique assembly phases. This is a crucial point since the vibration spectrum associated with each of the assembly phases may be different. Depending upon the number of gear and pinion teeth, a specific gear pair may exhibit between 1 and T_p assembly phases. Lang (28)

The number of unique assembly phases, N, which may be exhibited by a particular pair of gears is numerically equal to the product of prime numbers common between T_p and T_g. Example, if a fifty tooth gear engages a twenty tooth pinion, then:

$$T_g = 50$$
$$T_p = 20$$

Two prime numbers 2 and 5 are common between T_p and T_g and hence N is equal to the product of the prime numbers, 10.

The number of unique assembly phases, N, determine F_a, F_h and the distribution of wear between teeth on the gear and pinion. This is caused because a single tooth on the pinion will contact exactly T_g/N gear teeth pairs, while a single tooth on the gear will contact exactly T_p/N pinion teeth pairs. In consequence, as the gear and pinion wear, they will accommodate initial faults by exhibiting T_p/N axes of radial symmetry on the pinion and T_g/N axes of symmetry on the gear. Further, a specific tooth on the pinion will contact a specific tooth on the gear once in T_pT_g/N meshes.

The numerical inter-relationships for F_a and F_g are:

$$F_h = F_pN/T_g$$
$$F_a = F_pT_p/N$$

Note that F_a and F_h are only unique for non-integer gear ratios. For R equal to any integer, $F_a = F_p$ and $F_h = F_g$. Hence, in integer ratio gearboxes, the assembly phase frequency is indistinguishable from the pinion frequency and the hunting tooth frequency is indistinguishable from the gear rotation frequency.

In a sense, integer ratio gear-pairs represent a "worst case" design situation. Integer ratio gear-pairs exhibit the maximum possible number of assembly phases, $N = T_p$. This situation provides the maximum variation in possible vibration spectra, being highly dependent upon the manner in which the two gears are initially phased. Further, because unique pairs of gear teeth always enter mesh together, integer ratios provide the minimum opportunity for gear-pairs to "wear in" to assembly dependent maximum vibration conditions. Integer ratios provide the maximum opportunity for an initial single-tooth fault to accelerate the overall degradation of the gear pair.

At the other extreme, non-integer ratios formed between gears that share no common primes between the number of teeth on the pinion and gear represent an optimum situation. Such gear sets possess only one assembly phase ($N = 1$), and therefore will always exhibit the same vibration signature regardless of the initial tooth phasing at assembly. In such a gear set, a single tooth on the gear contacts every tooth pair on the pinion. Hence, non-integer ratio gear sets, with a single assembly phase, exhibit the minimum possible hunting tooth frequency, F_g, because only one assembly phase is possible, the assembly phase passage frequency, F_a, is identical to the mesh frequency, F_m.

Between these extremes, non-integer gear ratios with more than one assembly phase are possible. In such gear sets, the hunting tooth frequency, F_a, the gear meshing frequency, F_m, the pinion rotation frequency, F_p and the gear rotation frequency, F_g, are all unique.

The total vibration signature exhibited by the gear pair will contain sums, differences, and multiples of the five fundamental frequencies defined. In general, vibration at the mesh frequency, F_m, will dominate the signature. This component is caused as the load transmitted is exchanged from tooth-pair to tooth-pair. In highly precise gear sets, properly assembled, harmonics of the mesh frequency will be at very low amplitudes. Minor clearance errors tend to generate the odd harmonics of mesh frequency while more severe mis-spacing will introduce the even harmonics. Gear sets of low precision, properly installed, typically exhibit the odd harmonics of gear mesh frequency. The generation of even harmonics of gear mesh is normally indicative of extreme wear or clearance.

Unbalance or run-out of the pinion and its shaft may produce vibrations at the pinion frequency, F_p, and its harmonics. Vibration at the gear frequency, F_g, and its harmonics are indicative of unbalance or run-out in the gear or gear shaft. Normally, simple unbalance will produce vibration at F_p or F_g. As the level of unbalance increases, the harmonics of Fp and F_g may become discernable. Normally, the presence of even harmonics of F_p and F_g only occur when an extreme amount of unbalance is present or when the bearing clearances for the pinion and gear shaft are excessive. Even harmonics may, however, be generated by modest levels of run-out in the gear or pinion.

It is very common for gearbox vibration spectra to contain sidebands spaced around the tooth meshing frequency, F_m, and its harmonics. These result from amplitude and frequency modulation of the otherwise uniform vibration signal generated at the tooth meshing rate. A perfectly circular gear, eccentrically mounted to its shaft will generate frequencies at $F_m \pm F_g$ in its vibration spectrum. Third order run-out (triangular deformation) of the pinion will introduce frequencies at $F_m + 3F_p$.

Major damage to gear pairs may be incurred when a contaminate particle passes through the mesh. Typically, such occurrences will produce damage on the two teeth in mesh at the time of ingestion. Such damage will produce abrupt changes in the normal load at the mesh at the hunting tooth frequency, F_h. The resultant vibration spectrum may contain frequency components at F_h and its even and odd harmonics. Additionally, modulation side bands of $\pm F_h$, $\pm 2F_h$, $\pm 3F_h$ etc. will be produced around the gear mesh frequency, F_m.

At initial assembly, the run-out of gear and pinion may cause significant interference. As the gear set is operated, cold working of the soft gears occurs, relieving these interferences. Hence, the characteristic vibration signature gradually reduces with time. Should the gear set be disassembled during the course of normal maintenance, it will most probably be reassembled with a different phase relationship from the initial assembly. This will cause an immediate increase in vibration levels at F_a and in sidebands of $F_m \pm F_a$. Unless the gears are marked at disassembly, the odds of reassembling the gear set in its initial phase are N! Unplanned reassembly of soft gears will almost certainly increase vibration levels at the assembly passage frequency, F_a.

5.1.1 Literature Review of Gearbox Vibration Health Monitoring

Two methods are commonly employed in diagnosing faults in gears, they are frequency spectrum analysis and cepstrum analysis. Frequency spectrum analysis can reveal useful information about the mesh, loading, eccentricity and unbalance. A defective tooth or teeth can generate and excite specific discrete frequencies. Analysis of the time signal, spectrum frequencies, shape, amplitude, and sum and difference frequencies will reveal which gears have defective teeth. The number of defective teeth on each gear, the number of gears that have defective teeth, and the location of the defective teeth with respect to some reference point can be revealed. Taylor (29).

It is very common for gearbox vibration spectra to contain sidebands spaced around the toothmeshing frequencies and their harmonics. Changes in the number and strength of such sidebands generally indicate a deterioration in condition, and the sideband spacing gives valuable diagnostic information as to the source of the modulation. Randall (30).

Cepstrum analysis can be defined in a number of different ways, but all can be considered as a spectrum of a logarithmic spectrum, that is, logarithmic amplitude, but linear frequency scale.

The cepstrum may also be useful as a data reduction technique, effectively reducing a whole pattern of sidebands into a single line in the cepstrum, and easing the problem of monitoring changes in condition. Randall (31).

Another method used for fault diagnosis of gears is that of power spectral density. Power spectral density can be described as a similarity measurement in the frequency domain, the displacement being multiplied by itself such that the coefficients of the components are exposed as the square of their magnitudes, Collacott (32).

5.2 OBJECTIVES

The purpose of the study undertaken in this module was:

1) To determine the effectiveness of using frequency spectral analysis as a means of monitoring the condition of gear trains.

2) To determine the effectiveness of using discrete frequency analysis to diagnose specific gear related problems.

5.3 APPARATUS

Recorders

TEAC seven channel tape recorder with built in amplifiers. Type MR-30.

Transducers

One B&K accelerometer type 4371

Analysers

Wavetek 100A FFT analyser

Amplifiers

One B&K charge amplifier type 2635

Test Rig

Carrier Chiller with the following specifications:

Number of teeth on the pinion = Tp = 21
Number of teeth on the gear = Tg = 69
Gear rotational frequency 2980 RPM = Fg = 49,67 Hz
Number of vanes on the impeller = Vi = 17

5.4 EXPERIMENTAL PROCEDURE

5.4.1 Data Acquisition

The B&K accelerometer was mounted on the gearbox casing. The acceleration signal from the accelerometer was integrated once using the B&K charge amplifier and then recorded on the seven channel TEAC tape recorder.

During the vibration data acquisition the chiller was maintained at a constant load of 100% Maximum Capacity Rating (MCR).

5.4.2 Data Processing

The results obtained were analysed using the Wavetek 100A and the TEAC MR-30 tape recorder. Two hundred and fifty six time samples were used to obtain an average spectrum so as to reduce the level of noise. A frequency range of 10 kHz was selected so as to accommodate the 2 x gear meshing frequency of 6 854 Hz.

5.5 OBSERVATIONS AND RESULTS

The first step when diagnosing gear faults is to calculate the discrete meshing frequencies and their harmonics.

Number of teeth on the pinion = T_p = 21
Number of teeth on the gear = T_g = 69
Gear rotational frequency = 2980 RPM = F_g = 49,67 Hz
Number of vanes on the impeller = V_i = 17

Gear ratio = $R = T_g/T_p = 69/21 = 3,2857$

Hence the pinion rotational frequency = $F_p = (R)(F_g)$

$$F_p = (3,2857)(49,67)$$

$$F_p = 163,19 \text{ Hz}$$

The gear meshing frequency = $F_m = (T_g)(F_g) = (T_p)(F_p)$

$$F_m = (69)(49,67) = (21)(163,19)$$

$$F_m = 3427 \text{ Hz}$$

The blade passing frequency = $F_b = (V_i)(F_p)$

$$F_b = (17)(163,19)$$

$$F_b = 2774 \text{ Hz}$$

The number of unique assembly phases = N = 1
Hence $F_a = F_p$

The hunting tooth frequency F_h is calculated as follows:

$$F_h = (F_p)(N)/(T_g) = (163,19)(1)/(69)$$

$$F_h = 2,37 \text{ Hz}$$

$F_m + F_g = 3477$ Hz
$F_m + 2F_g = 3526$ Hz
$F_m + 3F_g = 3576$ Hz

$F_m - F_g = 3377$ Hz
$F_m - 2F_g = 3328$ Hz
$F_m - 3F_g = 3278$ Hz

$F_m + F_h = 3429$ Hz
$Fm + 2F_h = 3432$ Hz
$F_m + 3F_h = 3434$ Hz

$F_m - F_h = 3425$ Hz
$F_m - 2F_h = 3422$ Hz
$F_m - 3F_h = 3420$ Hz

$F_m + F_p = 3590$ Hz
$F_m + 2F_p = 3753$ Hz
$F_m + 3F_p = 3917$ Hz

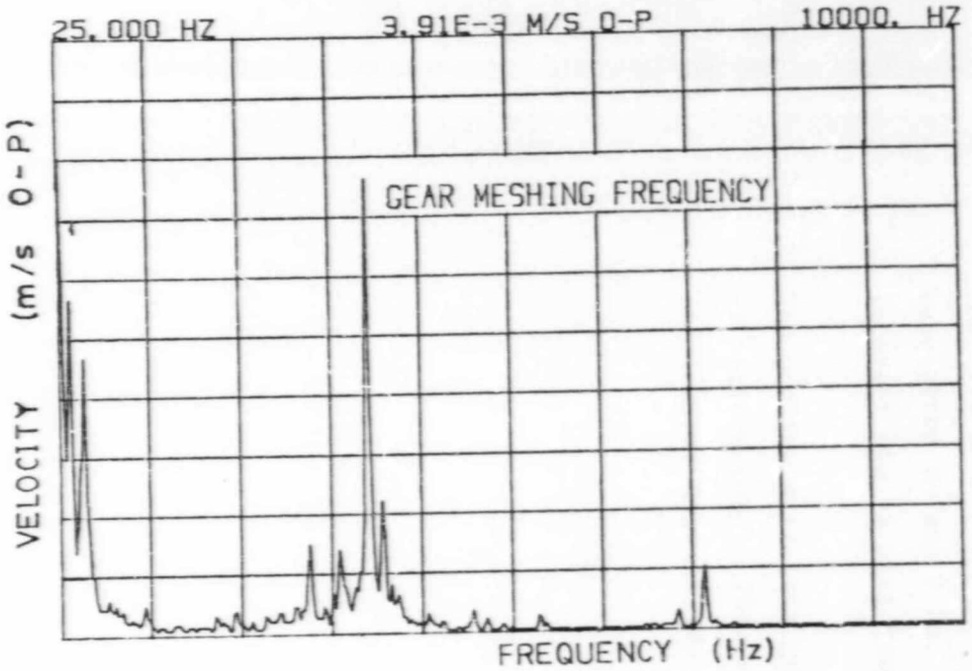
$F_m - F_p = 3264$ Hz
$F_m - 2F_p = 3101$ Hz
$F_m - 3F_p = 2937$ Hz

When analysing spectrums from gear trains it is necessary to calculate the abovementioned frequencies for diagnostic purposes. It is also convenient to calculate the 2 x gear mesning frequency as the presence of this harmonic is a clear indication of misalignment.

2 x gear meshing frequency = $2F_m = (2)(3427)$

$2F_m = 6854$ Hz

Figure 5.1 shows the frequency spectrum that was obtained from the abovementioned gearbox. The gear meshing frequency F_m at 3425 Hz is clearly visible. So too are the pinion and gear rotational frequencies, T_p and T_g at 150 Hz and 50 Hz respectively. The blade passing frequency at 2775 Hz is also evident. The $2F_m$, 2 x gear meshing frequency, at 6850 Hz is also evident. It is important to determine the extent that the background noise is contributing to the frequency spectrum obtained. Mains frequency (50 Hz) and three times mains frequency (150 Hz) components are very common background noise components in three phase motors. The contribution of the mains frequency to the 50 Hz component can be determined by examining the background noise frequency



M/S 0-P	HERTZ
3.15E-3	50.000
2.22E-3	150.01
1.83E-3	300.00
586.E-6	2775.1
551.E-6	3100.0
2.99E-3	3425.0
871.E-6	3575.0
409.E-6	7150.0

GEAR FREQUENCY ANALYSIS

FIGURE 5.1

spectrum when the motor is not operating. The contribution of the three times mains frequency component to the 150 Hz component obtained is difficult to determine, since this component only occurs when the motor is operating. When analysing frequency spectrums with machine operating speeds or harmonics of these at 50 Hz and 150 Hz care should be taken to distinguish between the background noise component and the actual vibration component from the machine.

Frequencies at $F_m + 3F_g$ and $F_m - 2F_p$ are also present at 3575 Hz and 3100 Hz. The reason for the frequencies not being exactly the same as the calculated frequencies is that the resolution of the window of 10 000 Hz containing 400 lines or cells is 10 000/400 = 25 Hz. So, only frequencies that are multiples of 25 Hz can exist. To decrease the resolution to say 5 Hz, zoom analysis around the frequency of interest can be performed.

The problem with this gear train was successfully diagnosed as uniform wear. This was found to be the case when the chiller was dismantled.

5.6 DISCUSSION

Spectrum analysis can be a very useful tool in diagnosing gear faults. Since most industries possess gear train systems on critical machines, vibration condition monitoring on such systems is essential. In power stations gear trains exist on the barring gear of the main turbine, the steam feed water pump, the electric feed pump, the mills and many more auxilliary items of plant.

Although the tests on the chillers consisted of a simple gear and pinion, complicated gear trains can be considered as a superposition of spectrums exhibited by simpler systems. Once the geometry and shaft speeds of a gear train are known discrete frequency spectrum analysis can be readily applied. Problems with specific gears and pinions can be diagnosed using spectrum analysis.

5.7 CONCLUSIONS

1) A substantial amount of diagnostic information can be obtained from the changes in the vibration signals measured externally on continuously operating gear-boxes under constant load and speed conditions.

2) The vibration signature of a simple gear set is dominated by sums, differences, and multiples of five unique frequencies. Lang (28)

3) In integer gear ratio sets, two of these frequencies are not uniquely discernable. Lang (28)

4) The nature of specific fault types in the gear pair may be identified from the increase of certain specific frequencies in the resultant vibration spectrum. Tooth pair faults, assembly phase mis-matching, gear and pinion run-out, unbalance of gear and pinion shafts, and improper spacing between pinion and gear may be detected. Lang (28) and Randall (30)

5) When disassembling gear sets with multiple phases, assembly marking is highly recommended, particularly if the gears are of unhardened material.

6) Uniform wear tends to show up as an increase of the tooth-meshing component and its harmonics, more particularly the latter. Lang (28) and Taylor (29)

7) Local faults, such as cracked teeth, spalls and localised pitting, give rise to components over a very wide frequency range, partly by modulation (for the

frequencies around and above the tooth meshing frequency) and partly as additive impulses (primarily for the frequencies below tooth-mesh). Lang (28)

8) Misalignment and eccentricity, tend to give higher level sidebands more closely grouped around the tooth-meshing harmonics. Randall (30)

9) Gear meshing frequencies occur at high frequencies and hence it is important to ensure that the transducer used can detect these frequencies. Normally accelerometers are used for this purpose.

6. MODULE 3
DISCRETE FREQUENCY ANALYSIS

6.1 INTRODUCTION

By means of appropriate frequency analysis, with the discrete frequencies for the system and components calculated, it is possible to relate peaks to particular components. Any increase in vibration energy at the peaks, indicates a problem with that component or system. The evaluation of such frequencies can be a matter of simple arithmetic or algebraic, or the evaluation can be an elaborate form of applied mathematics involving the equilibrium of forces, couples and energy under either linear or non-linear conditions.

6.2 VIBRATION DUE TO UNBALANCE

Unbalance manifests itself in the fundamental rotational frequency component. Inevitably every machine has some inherent unbalance in it's system. Hence every machine will display a one-times frequency component. Figure 6.1 shows the relationship between the unbalance mass relative to the known keyway.

6.3 VIBRATION DUE TO MISALIGNMENT

Misalignment is a common problem in rotating machinery. In spite of self-aligning bearings, flexible couplings and hydraulic couplings, it is difficult to align two shafts and their bearings so that no forces exist which will cause vibration. Figure 6.2 illustrates the three possible types of coupling misalignment.

MECHANISM OF UNBALANCE

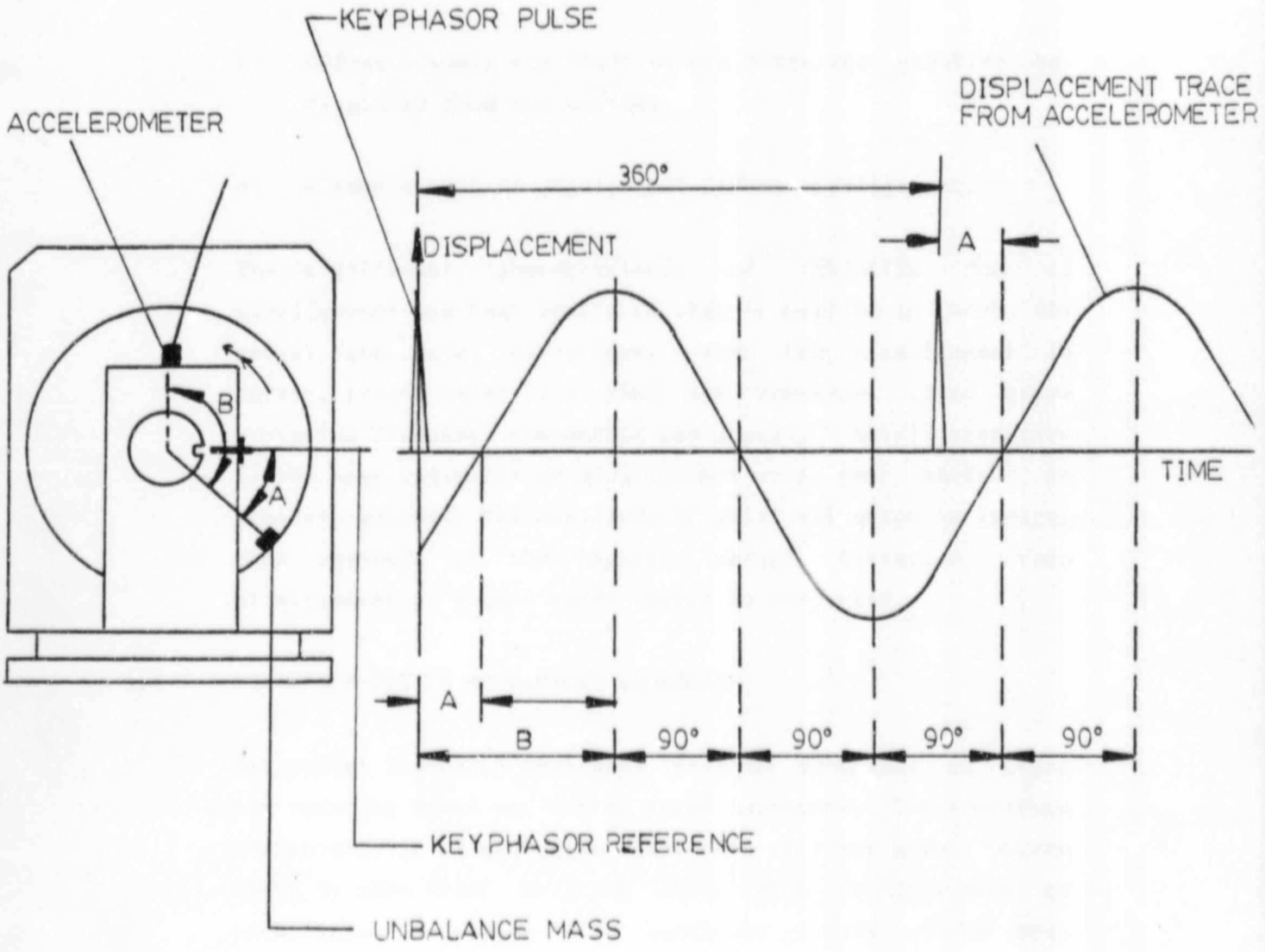


FIGURE 6.1

a) Angular - where the centre line of the two shafts meet at an angle.

b) Offset - where the shaft centre lines are parallel but displaced from one another.

c) A combination of angular and offset misalignment.

The significant characteristic of vibration due to misalignment and bent shafts is that it will be in both the radial and axial directions. When the misalignment is severe, second order (2 x RPM) and sometimes third order vibration frequency components may appear. Axial vibration is the best indicator of misalignment or a bent shaft. In general, whenever the amplitude of axial vibration is greater than one-half of the highest radial vibration, then misalignment or a bent shaft should be suspected.

6.4 VIBRATION DUE TO MECHANICAL LOOSENESS

Mechanical looseness manifests itself at frequency of twice the rotating speed and higher order harmonics. The vibration characteristic of mechanical looseness will not occur unless there is some other exciting force such as unbalance or misalignment to cause it. Looseness simply allows more vibration to occur than would otherwise appear.

The nature of mechanical looseness and the reason for the characteristic vibration at 2 x RPM can be explained by referring to the sequence of diagrams in Figure 6.3. This action of forcing the bearing down against the pedestal and then off the pedestal, produces two applied forces for each revolution of the shaft. One force is applied by the

TYPES OF MISALIGNMENT

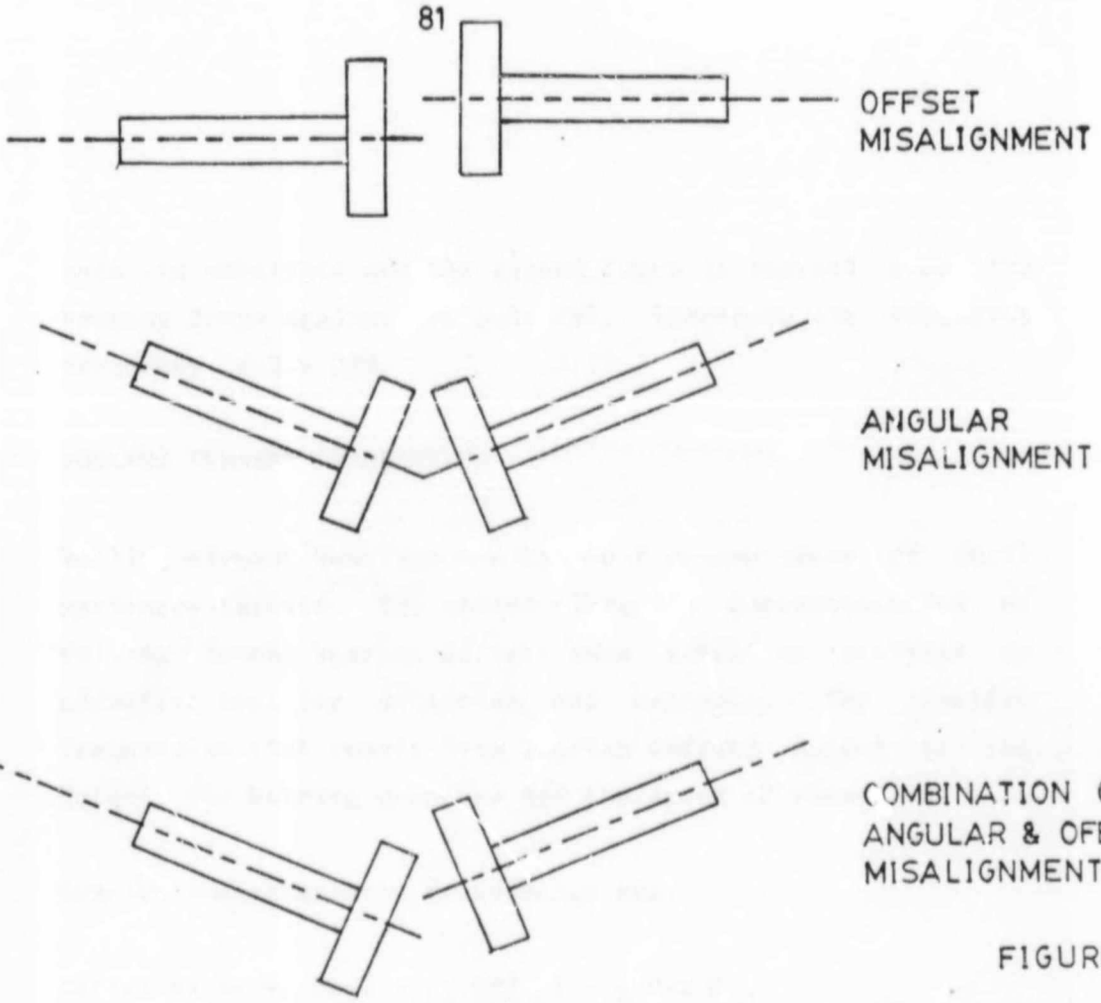
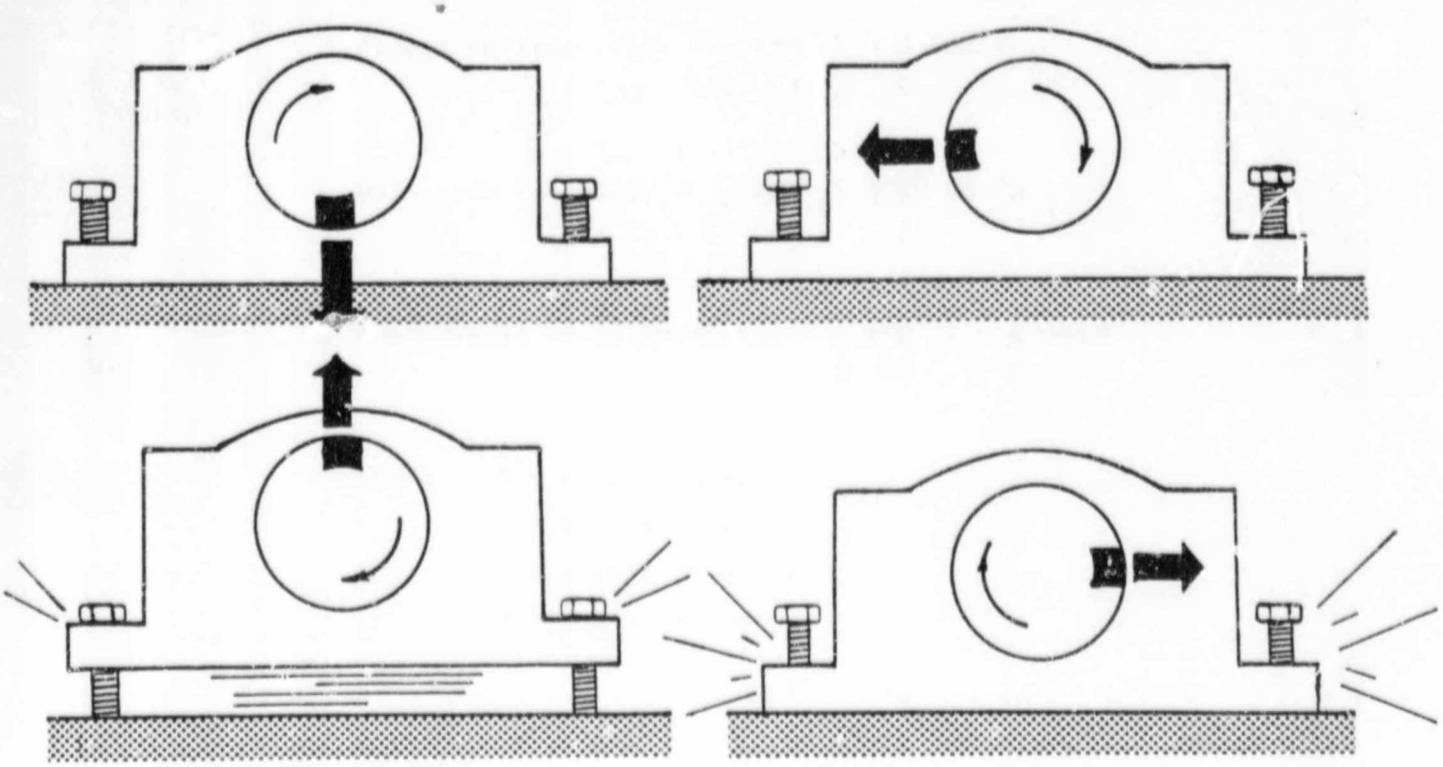


FIGURE 6.2



MECHANISM OF MECHANICAL LOOSENESS

FIGURE 6.3

rotating unbalance and the second force is applied when the bearing drops against the pedestal. Therefore the vibration frequency is 2 x RPM.

6.5 ROLLING ELEMENT FREQUENCIES

Rolling element bearings are the most common cause of small machinery failure. The unique vibration characteristics of rolling element bearing defects make vibration analysis an effective tool for detection and analysis. The specific frequencies that result from bearing defects depend on the defect, the bearing geometry and the speed of rotation.

Bearing characteristic frequencies are:

$$\text{Defect on outer race} = \left(\frac{n}{2}\right)\left(\frac{RPM}{60}\right)\left(1 - \frac{d}{D}\cos\theta\right)$$

$$\text{Defect on inner race} = \frac{n}{2}\,\frac{RPM}{60}\left(1 + \frac{d}{D}\cos\theta\right)$$

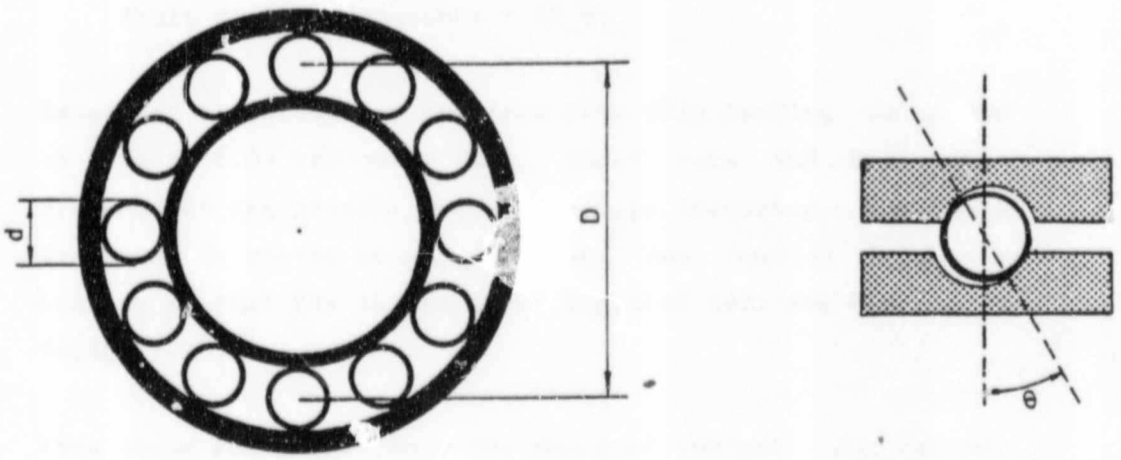
$$\text{Ball spin frequency} = \frac{D}{2d}\,\frac{RPM}{60}\left(1 - \frac{d^2}{D}\cos^2\theta\right)$$

$$\text{Fundamental train frequency} = \frac{1}{2}\,\frac{RPM}{60}\left(1 - \frac{d}{D}\cos\theta\right)$$

where D = pitch diameter
d = ball diameter
n = number of balls
θ = angle of contact

Figure 6.4 below shows the bearing geometry.

BEARING GEOMETRY



WHERE: D = PITCH DIAMETER
d = BALL DIAMETER
θ = ANGLE OF CONTACT
n = NUMBER OF BALLS

FIGURE 6.4

6.5.1 Illustrative Example

The bearing chosen for this example is the one that was used in the kurtosis tests, and has the following characteristics:

D = 34 mm
d = 8 mm
n = 8
θ = 5°

Substituting these values in the abovementioned equations results in the following discrete frequencies for a shaft speed of 2 340 RPM.

Outer race frequency = 119 Hz
Inner race frequency = 193 Hz
Ball spin frequency = 78 Hz
Train or cage frequency = 15 Hz

Examining the frequency spectrum from this bearing as shown in (Figure 6.5) the outer race, inner race and ball spin frequencies are present except the cage frequency. The cage frequency is absent because the cage was removed from the bearing so that fly ash could be injected into the bearing to damage it.

When using these equations the angle of contact depends on:

1) The shaft fit, i.e. the angle of contact will be larger in the case of an interference fit between the shaft and bearing than in the case of a loose fit.

2) The magnitude of the axial force. The greater the axial force the greater the angle of contact.

It is the author's experience that the angle of contact changes significantly with the shaft fit, the axial force and with bearing wear. Care must therefore be taken when applying these equations to bearings.

6.6 VANE AND BLADE PASSING FREQUENCIES

Problems associated with blades and vanes are usually characterised by high fundamental vibration or a large number

ROLLER BEARING FREQUENCIES.

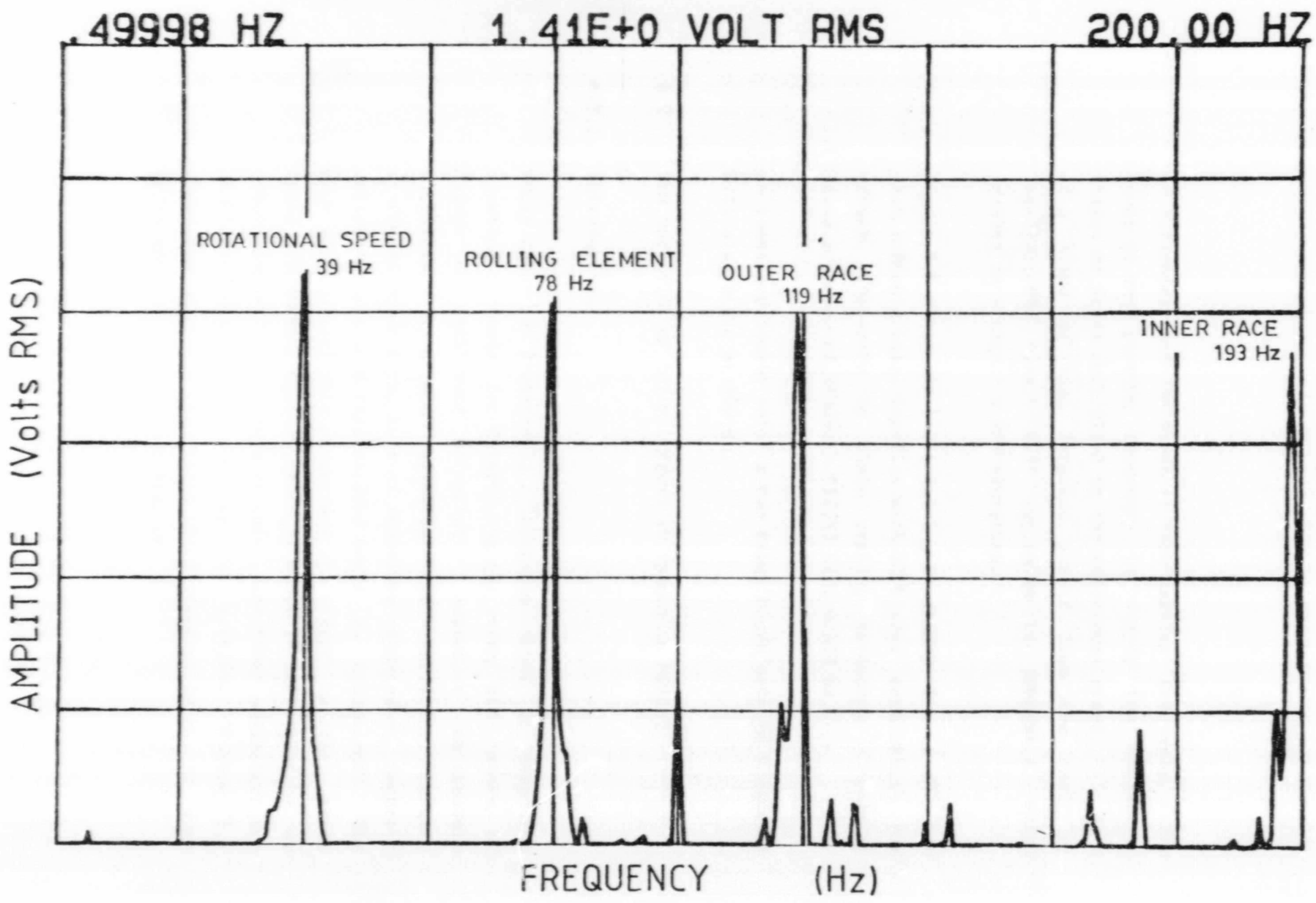


FIGURE 6.5

of harmonics near the blade or vane passing frequency. The blade or vane passing frequency is simply the number of blades or vanes multiplied by the rotational speed. If part of a blade or vane detaches itself from the shaft or impeller, the result will typically be imbalance, which results in a high 1 x RPM vibration.

Figure 6.6 shows the blade passing frequency for a FD fan with a double inlet impeller with 12 blades each which are offset. Therefore the blade passing frequency = running speed x number of blades = (12,5) (12) = 150 Hz.

As these are double inlet a two times blade passing frequency will also occur at 300 Hz.

6.7 THE DETECTION OF ROTOR DEFECTS IN INDUCTION MOTORS

6.7.1 Introduction

Where rotor bar fracture in a induction motor is suspected a method is needed for determining the extent and nature of damage to facilitate potentially expensive maintenance decisions. A method allowing differentiation between fractured rotor bars, bars of anomalous resistance, magnetic anisotropy and load fluctuations would be convenient. The most readily vailable parameters are stator current, shaft speed and frame vibration, each of which is influenced by an electromagnetically irregular rotor, and these parameters are available without having to stop the machine. Figure 6.7 shows typical rotor bar cracks which occurred in an ash pump drive motor at Duvha Power Station.

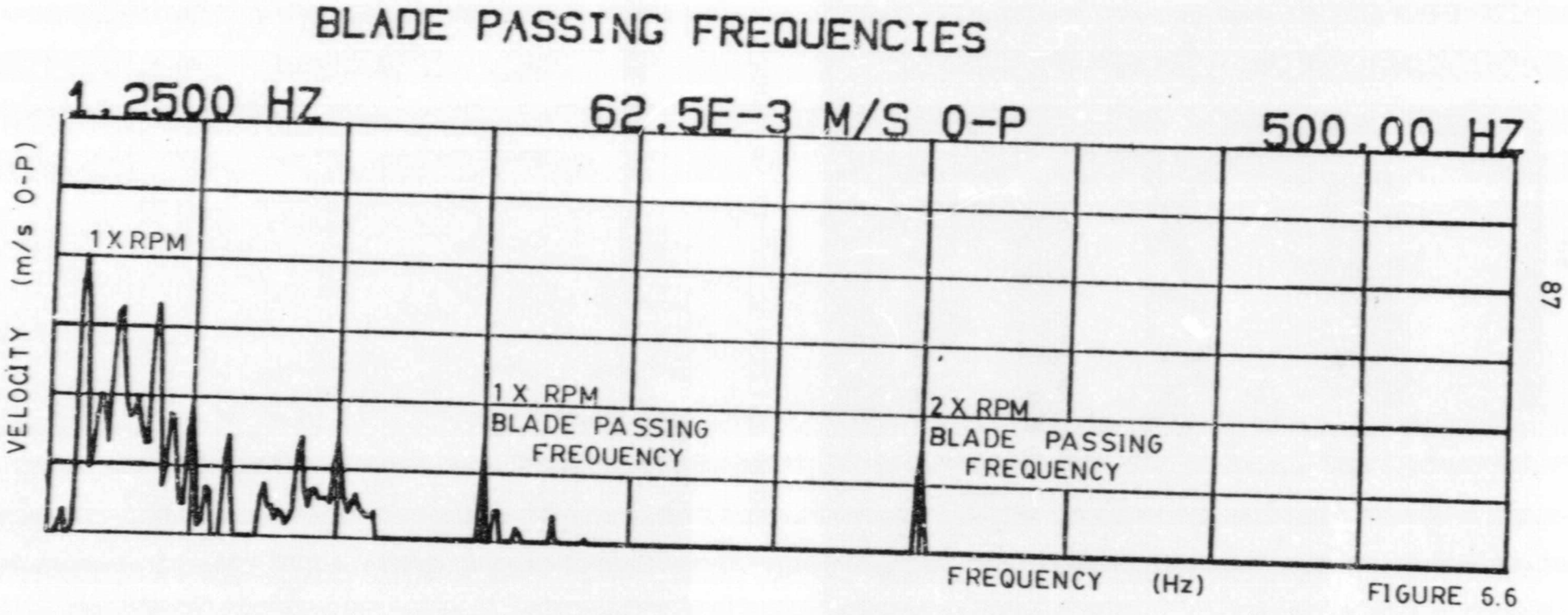

BLADE PASSING FREQUENCIES
1.2500 HZ
62.5E-3 M/S 0-P
500.00 HZ
VELOCITY (m/s 0-P)
1XRPM
1X RPM
BLADE PASSING
FREQUENCY
2X RPM
BLADE PASSING
FREQUENCY
FREQUENCY (Hz)

FIGURE 5.6

Author Ducci PP
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