RELATIVE DOMINANCE OF THE FORCE EQUATION COMPONENTS

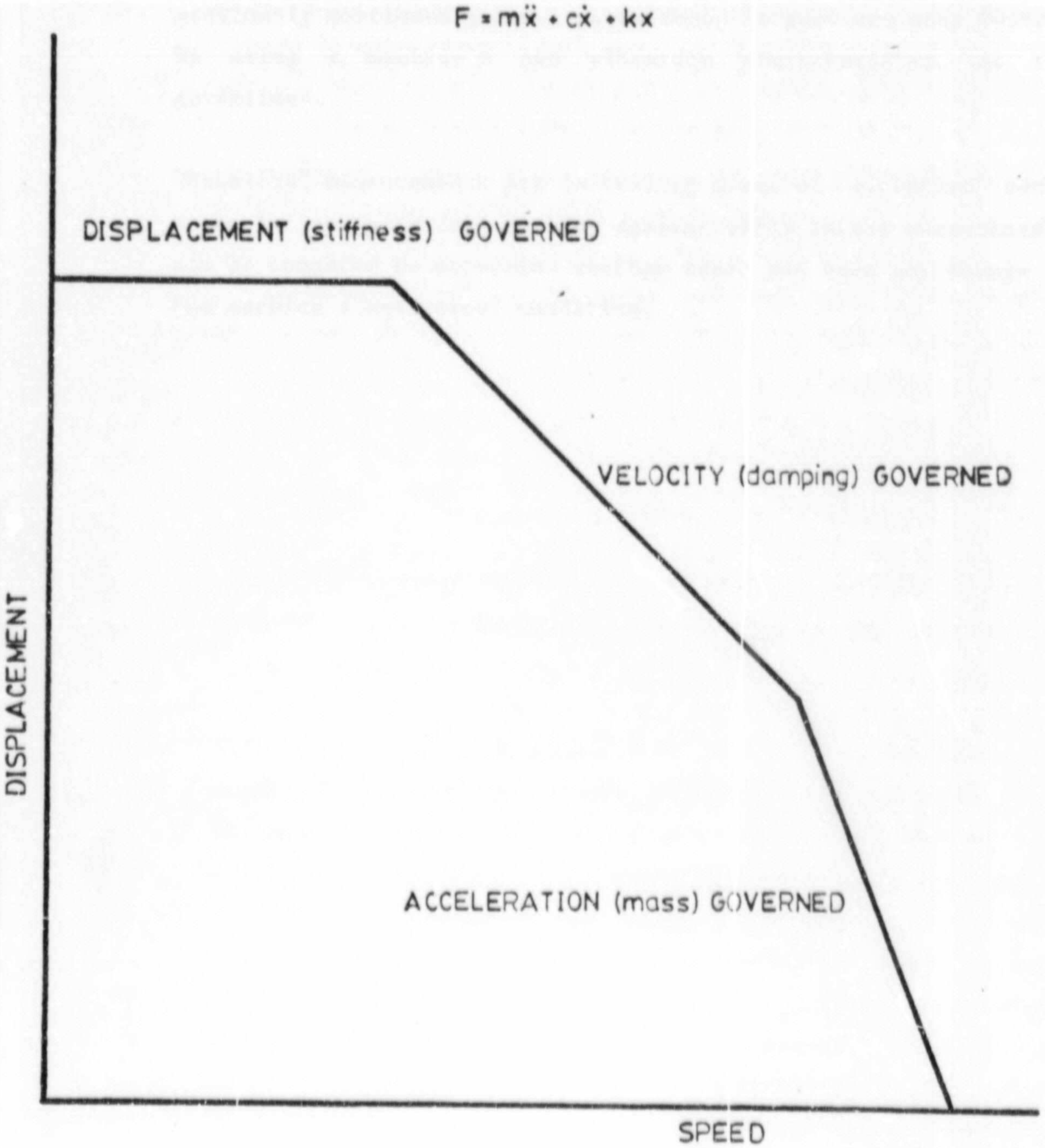


FIGURE 1.7

more precisely pinpoint the cause. The technique, described by Catlin (13), which proved to be most effective has been called the "baseline" measurement technique. Although similar in approach to standard vibration measurements which use the previously mentioned general guidelines, it goes one step further by using a machine's own vibration characteristics as the guidelines.

"Baseline" measurements are in reality a set of vibration bench marks for an individual machine against which future measurements can be compared to determine whether there has been any change in the machine's mechanical condition.

1.2.3 Economic Aspects of a Vibration Condition Monitoring Programme

Michael Neale and Associates (7) found that on average British companies lose about four days production each year due to plant downtime, and there is no reason to believe that the figures for other countries are appreciably different. British industry has a daily added value output of the order of £200 000 million so the cost of downtime is about £800 000 million a year.

In South Africa, it is estimated that more than "R15 billion" is spent on maintenance each year. This constitutes approximately twice as much as spent on all South African construction projects. This figure excludes indirect costs such as poor availability, high reject rates and poor recoveries. (14)

ESCOM's maintenance costs amounted to R282 million for the year 1983 (15) and increased to R307 million in 1984 (16). These figures exclude replacement costs.

A forced outage involving a 600 MW unit costs R96 700 per weekday consisting of 24 hours, R676 900 per week and R2,9 million per month (17). On average a unit is off stream for one hundred and fifty hours during such a forced outage which costs R604 375 in replacement costs. These replacement costs are a direct result of having to run older and less efficient plant to account for the loss in power, during a forced outage. If a vibration condition based monitoring programme were to be introduced and one forced outage could be averted or the downtime reduced, by knowing what was wrong with the machine before it was brought out of service, the capital cost for the vibration equipment would be amortised in a relatively short space of time.

Papamarcos (18) believes that the greatest expected benefit of a vibration monitoring system is the increased plant availability, through a reduction in machine damaging failures. This financial benefit can be calculated as a differential between costs of generated and purchased power. He quotes New Haven Habor Station as an example. The daily cost of plant outage ranges from \$20 000 to \$80 000. The plant carries a swing load at certain times, so its full 465 MW capacity is not always needed. A conservative figure of \$50 000 for the cost of a one day plant outage is used. If it were possible to increase the total maintenance cycle on the turbo-generator from four to five years, the annual savings would average about \$500 000. He also concludes that if one forced outage is avoided in the systems lifetime, condition monitoring is economically beneficial.

Martin (19) reports that downtime on paper machines at Bowater's Kemsley Mill in Kent averaged 21 hours/month in 1981 as a result of mechanical breakdown. This year it should be down to around 6,5 hours/month representing a saving of at least £180 000. Behind these impressive figures is a condition monitoring programme introduced by the company, in response to the consistently low levels of machine availability. Vibration analysis must have made a considerable contribution to this figure. As an example, cost of parts for pump rotating elements from one supplier fell from £8781 in 1981 to £3040 in 1983.

Lorio (20) states that excessive maintenance costs can be significantly reduced if problems are quickly detected, analysed and corrected using vibration condition monitoring. This approach to maintenance has contributed significantly to greatly reducing maintenance costs, improving productivity and reliability, and preventing catastrophic failure.

Thomas (21) carried out an analysis of all outages exceeding 28 days on 660 MW and 500 MW plant over a ten year period, for the whole of the CEGB. The costs due to loss of generation for those cases due to major damage of rotating parts, excluding rotor cracks, were £15 M. If 25% of these extensive damage cases could have been avoided by vibration monitoring the annual national benefit would be £1,88 M. Thomas also states that the implementation of a vibration condition monitoring programme gives a benefit-to-cost ratio of greater than 10:1, which rises to 17:1 if crack detection is included as a benefit.

A United Kingdom Department of Industry survey (7), based on experience of existing users in the UK, suggests that for general purposes an initial investment in condition monitoring of about 1% of the capital value of the plant is generally reasonable. If there is a special safety hazard, users generally have been prepared to invest up to 5% of the plant's capital cost. On top of this initial outlay must be added the salary of the personnel employed full time to supervise and interpret the condition monitoring system. If Duvha Power Station (3600 MW capacity) is used as an example, then an initial investment in condition monitoring of 1% of the capital cost of Duvha amounts to R180 million!

1.3 PURPOSE

The purpose of this study is:-

a) To become familiar with the various vibration monitoring and diagnostic techniques used for a vibration condition monitoring programme.

b) To determine the effect of load variations on the vibration severity for draught group fans and feed water pumps.

c) To determine the effectiveness of using Kurtosis as a means of monitoring the condition of roller and ball bearings.

d) To determine the feasibility of using the Prohl lumped spring mass method and the transfer matrix method to ascertain shaft critical speeds.

e) To provide recommendations for the implementation of a vibration condition monitoring programme at ESCOM's new power stations.

2. PRESENT STATUS OF VIBRATION CONDITION MONITORING IN ESCOM

2.1 MAINTENANCE CONCEPTS

Maintenance strategies commonly employed in industry can be classified roughly in ascending order of cost effectiveness as follows:

2.1.1 Breakdown Maintenance

In industries running many inexpensive machines and having each important process duplicated, machines are usually run until they breakdown. Loss of production is not significant because spare machines can usually take over. Breakdown maintenance, also called "Run to Breakdown" or "Fire Brigade" maintenance is work necessitated by unforeseen breakdown or damage. In this situation there is often no economic or safety advantage in knowing when machines will breakdown and if the machines themselves indicate well enough where they have failed, vibration condition monitoring will be of no advantage.

2.1.2 Preventive Maintenance

Where important machines are not fully duplicated or where unscheduled production stops can result in large losses, maintenance operations are often performed at fixed intervals. Compulsory statutory inspection of machines also regulates the interval between maintenance operations.

After a specified elapsed operating time, machines are shut down regardless of their condition, they are inspected and components are replaced if necessary. Sometimes this maintenance strategy is referred to as "Distrubance" maintenance because often machines in "good health" are stopped for inspection, and often this inspection does more harm than good.

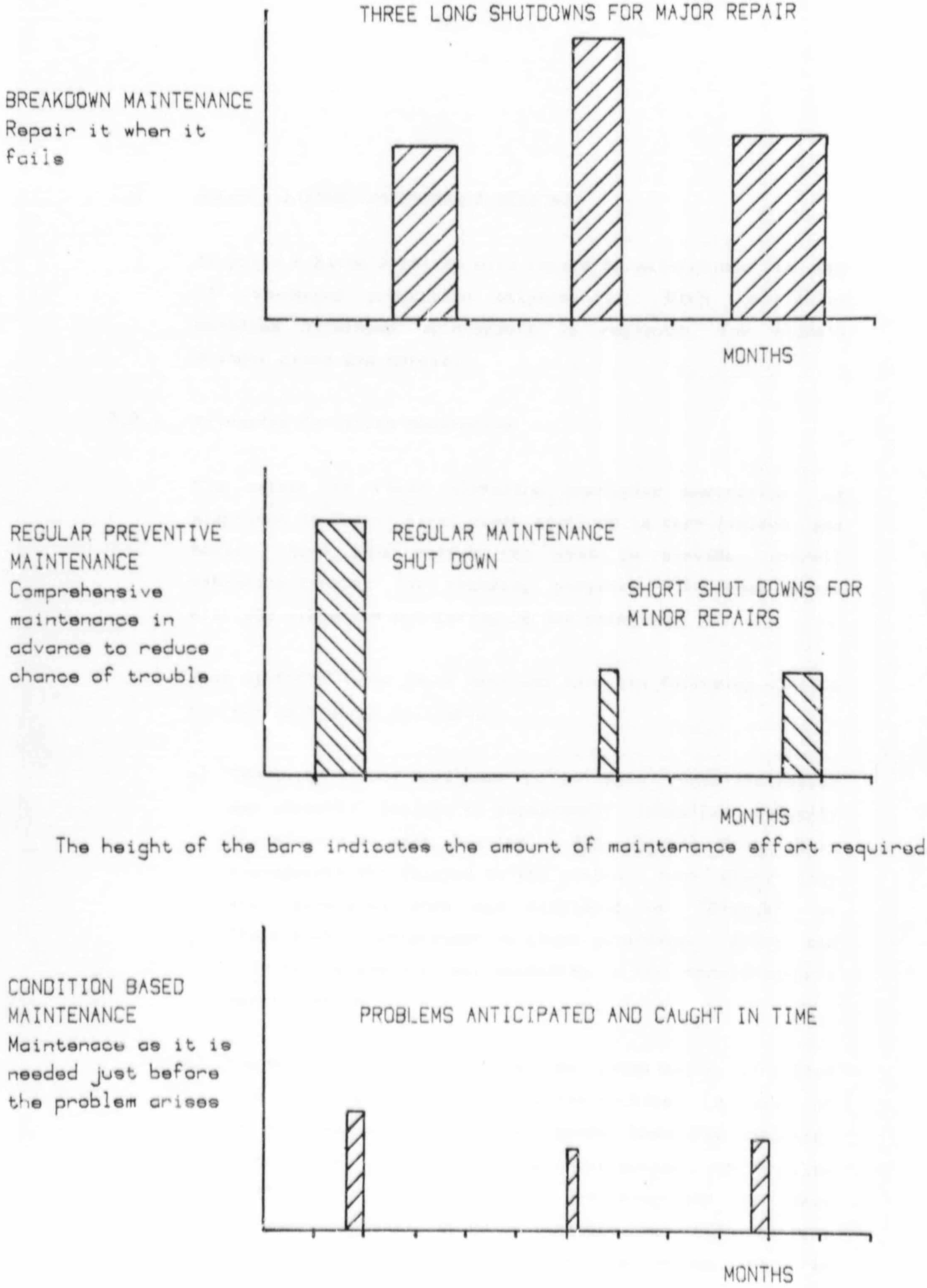
2.1.3 Condition Based Maintenance

Condition based maintenance replaces fixed interval overhauls by fixed interval measurements of the machine's running condition. The condition of all important components is monitored, watching for trends, so as to obtain warning of incipient problems, in time for effective action with least inconvenience.

Servicing of machinery is only permitted when measurements show it to be necessary. This agrees with most engineer's intuition that one should not interfere with smoothly running machines.

Being able to anticipate failure allows the plant engineer to obtain the necessary spare parts well ahead of time, and thereby avoid a large stock of spare parts. Furthermore, the maintenance personnel are better prepared and can be expected to effect a more reliable repair in a shorter time. Condition based maintenance thus takes planned maintenance a stage further towards rationality and effectiveness.

Figure 2.1 shows the three abovementioned maintenance strategies, as a function of time and the amount of maintenance effort required for each strategy.



<u>RESULTS OF DIFFERENT MAINTENANCE STRATEGIES</u>

FIGURE 2.1

2.2 ESCOM's PRESENT MAINTENANCE STRATEGY

At present Escom's policy with regard to maintenance is one of time-based preventive maintenance. With particular machines breakdown maintenance is employed, for example draught group fan motors.

2.2.1 Vibration Condition Monitoring

The extent to which vibration condition monitoring is practised at fossil fired power stations is very limited and basic. Hand held meters are used to provide overall vibration levels, for trending purposes. At some power stations vibration monitoring is non existent.

Most of ESCOM's new power stations have the following on-line protection systems available:-

a) The turbo-visory equipment on the main turbo-alternator set normally consists of permanently installed velocity transducers on each bearing. The signals from these transducers are relayed to the control room where they are integrated once and displayed as "0-peak" or "Peak-Peak" displacement on chart recorders. Alarm and trip levels are also set according to the manufacturer's specification.

b) The main feed pump has one or two permanently installed proximity probes at $\pm 45^{o}$ from the vertical at each of the two bearings. Some feed pumps have two velocity transducers. The signals from these probes are relayed to the control room where they are displayed on chart recorders or digitally or a cathode ray tube screen. Alarm and trip levels are set according to the manufacturer's specification.

c) The draught group fans have two permanently installed velocity transducers mounted horizontally on the drive-end and non drive-end bearing housing of the fan. The signals from these transducers are relayed to the equipment room where they are displayed on a vibration level meter or digitally as RMS velocity in mm/s, or peak velocity in mm/s.

The older generation stations have turbo-visory equipment, but no protection systems on the feed water pumps and the draught group fans. No facility at the power station exists to store and analyse transient vibration data. Also, no phase analysis is performed.

The periodic vibration monitoring activity at fossil fired power stations involves the acquisition and trending of overall RMS vibration levels, using a hand held vibration meter. At some power stations periodic vibration monitoring is non existent.

Two specialist groups within ESCOM are engaged in vibration activities, namely Dynamics & Noise, and Central Maintenance Services (CMS) Balancing Department. These two specialist groups are responsible for solving all vibration related problems that arise within ESCOM.

Recently a vibration committee has been established in ESCOM whose objectives are:

1) To increase plant availability and reliability and decrease maintenance costs by changing progressively from an exclusively time based maintenance programme to a condition based maintenance programme.

2) To increase safe operation of plant by pre-emting catastrophic failure.

This vibration committee will formulate ESCOM's philosophy towards vibration condition monitoring.

2.3 BENEFITS OF VIBRATION CONDITION MONITORING PROGRAMME - SUMMARY

1) Increased plant availability leading to greater output and reduced loss of production.

2) More efficient plant operation and more consistent quality obtained by matching the rate of output to the plant condition.

3) Reduced system downtime with the detection and ratification of potential or incipient failure.

4) Reduced maintenance costs and possible reduction in personnel.

5) Increased safety and reduction in human error.

6) Better energy utilisation and reduced operation costs.

7) Advanced downtime planning leading to improved deliveries and better customer relationships.

8) Effective negotiations with plant manufacturers based on better-known measures of plant condition.

9) Improved plant and machinery specification in the design and purchase of future plants.

10) Reduced spares requirements.

11) Reduction in the level of standby plant resulting in reduced capital cost.

3. TRANSDUCER SELECTION

Transducers play a vital role in the acquisition of vibration data. Different transducers have different applications when it comes to machinery health monitoring. It would therefore be useful to list their principle of operation and characteristics.

3.1 ACCELEROMETERS

Accelerometers operate on the principle that force is proportional to mass times acceleration. In the construction of an accelerometer a vibrating mass applies a force to a piezoelectric crystal that produces a current proportional to the force and thus to acceleration. The output from an accelerometer is a low-level, high impedance signal that requires signal conditioning. This is achieved by using a charge amplifier. Figure 3.1 shows a cross-sectional view of an accelerometer.

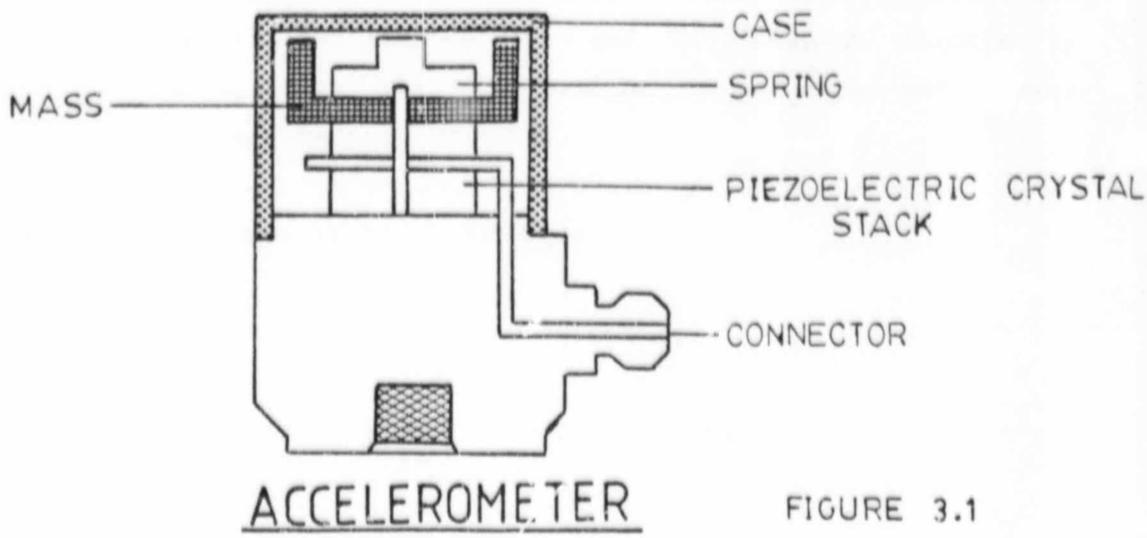


FIGURE 3.1

3.1.1 Advantages of Accelerometers

a) Accelerometers offer a broad frequency response, 1 Hz to well in excess of 50 kHz.

b) They offer good long-term reliability.

c) They are physically small in size compared to other transducers used. Note the mass of the accelerometer should be at most 10% that of the vibrating component, preferably lighter.

d) They are robustly constructed.

e) They have no screening problems.

f) They have a high resonant frequency well in excess of 35 kHz and hence have a high upper limit cut off frequency.

g) They have wide operating temperature range of $-250^{\circ}C$ to $+1\ 000^{\circ}C$.

h) From accelerometers one cannot only obtain acceleration spectra but also velocity and displacement spectra by integrating the acceleration trace once and twice respectively.

3.1.2 Disadvantages of Accelerometers

a) They require signal conditioning.

b) They are very sensitive to mounting.

c) They are sensitive to the length of the cable used.

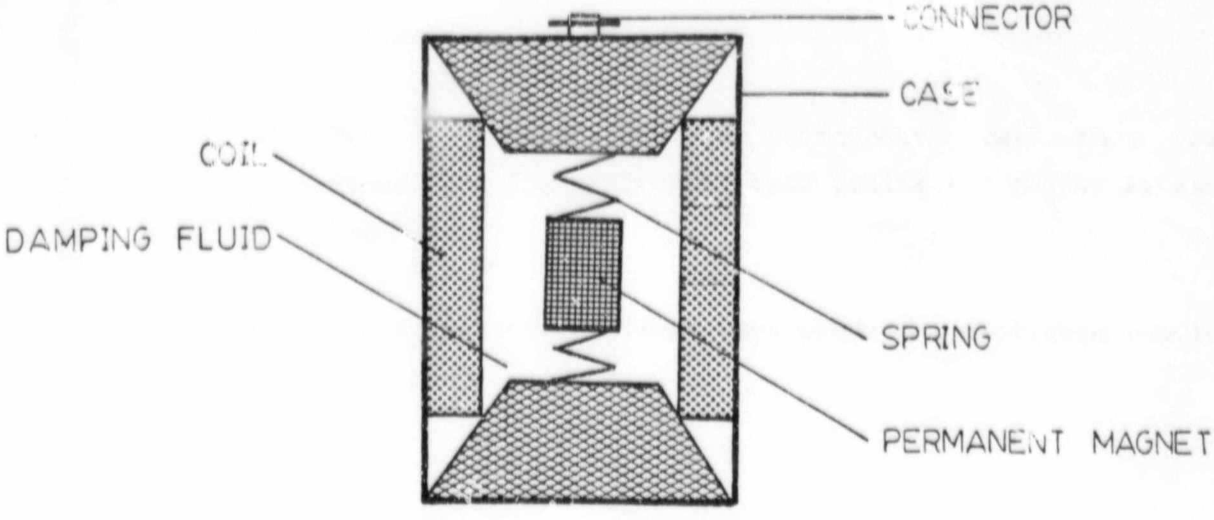
d) Falsified signals may arise due to dirty or badly fitted cable connections.

e) Spurious signals may arise due to ground loop effects.

f) They are unsuitable for extremely dirty or moist environments unless the connections are well sealed.

3.2 VELOCITY TRANSDUCERS

The main contacting type velocity transducer is termed seismic transducer. A permanent magnet moves within a fixed electric coil. As the magnet moves through the coil, an electric current is induced. This current is proportional to the displacement of the magnet within the probe. The entire spring/magnet assembly is immersed within oil or a similar fluid medium to provide damping. The spring-mass-damper system is designed to have natural frequency of between 8 to 10 Hz. This establishes a lower frequency limit of approximately 10 Hz. The upper frequency limit ranges from 1 000 Hz to 2 000 Hz. Figure 3.2 shows a cut away view of a velocity transducer.



VELOCITY TRANSDUCER FIGURE 3.2

3.2.1 Advantages of Velocity Transducers

a) No signal conditioning is required.

b) They are adaptable for portable hand-held instruments.

c) The cable length used is not critical.

d) Velocity is a good basis for condition monitoring as a parameter intermediate between displacement and acceleration. Velocity has a direct relationship to vibrational energy which is related to the deterioration.

3.2.2 Disadvantages of Velocity Transducers

a) They offer a comparatively low frequency range of 8/10 Hz to 1 000/2 000 Hz.

b) They are physically large and heavy.

c) Since they contain moving parts, they can wear out with time changing their sensitivity.

d) They are sensitive to stray magnetic fields and must be screened.

e) Due to their low upper frequency limit they are unsuitable for monitoring gear trains and roller element bearings.

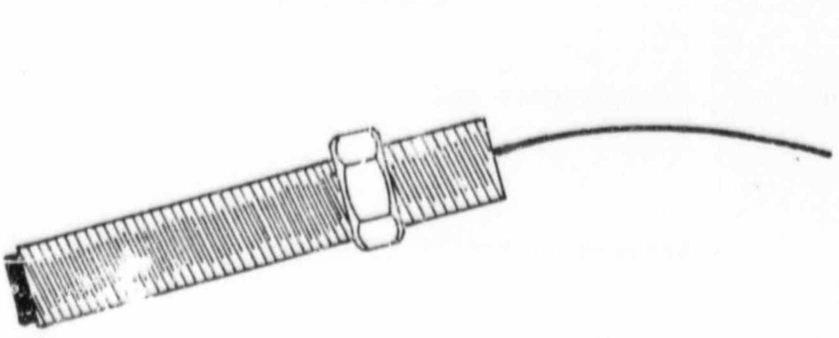
f) The comparatively large mass produces unreliable results on light structures.

g) The fluid medium inside the transducer sets an upper and lower operating temperature. These operating temperatures are directly related to the viscous properties of the fluid medium, i.e. the freezing and boiling temperatures of the fluid.

h) Some velocity transducers are only sensitive in one direction.

3.3 DISPLACEMENT TRANSDUCERS OR PROXIMITY PROBES

Proximity probes operate on an Eddy current principle. The Eddy current probe also contains an electrical coil, but no permanent magnet. Instead the coil is excited by a high frequency carrier, which induces an oscillating magnetic field around the coil. This magnetic field is disturbed by the proximity of a metal target. Small currents in the surface of the metal target are induced by the magnetic field. As the shaft moves relative to the sensor, the Eddy current energy changes, modulating the oscillator voltage. Figure 3.3 shows a typical proximity probe.



PROXIMITY PROBE FIGURE 3.3

3.3.1 Advantages of Proximity Probes

a) Proximity probes measure relative motion, i.e. the vibration of the shaft relative to the bearing casing when the bearing housing is too rigid for an accelerometer or velocity transducer.

b) They are very robust.

c) There are no moving parts.

d) They can be used in hostile environments.

e) Mainly used for shaft orbital analysis.

f) They can be used to obtain keyphasor or tachometer pulses so that the speed of the shaft can be ascertained. These N pulses per revolution are also used for phase analysis in the generation of Bode and Nyquist plots (where N = 1,2,.....X).

g) They are ideal for measuring sleeve-bearing, rotor and thrust pad movement.

3.3.2 Disadvantages of Proximity Probes

a) They offer a comparatively low frequency range of 0 to 1 500 Hz.

b) The installation of these probes is critical.

c) They are limited to shafts in plain bearings, and are not suitable for rolling element bearings.

d) Readings are sometimes susceptible to mechanical and electrical runout, i.e. shaft surface scratches, out-of-roundness and variation in electrical properties (permeability), all produce spurious signals. Surface treatment and run-out subtraction can be used to solve these problems.

e) They are very sensitive to variations of plating thickness on chromium plated shafts.

3.4 SHAFT RIDERS

The shaft rider consists of a rod assembly which extends through the bearing housing and literally rides on the shaft via a spring-loaded mechanism. The top of the rod, outside the bearing housing, is directly attached to a seismic transducer, usually a velocity pickup. The shaft rider measures shaft absolute vibration. Figure 3.4 shows a typical installation of a proximity probe.

3.4.1 Advantages of Shaft Riders

a) Measures shaft absolute vibration.

3.4.2 Disadvantages of Shaft Riders

a) Shaft riders directly contact the shaft causing wear.

b) They must be located in a lubricated area, which usually means going through a bearing casing.

c) They are very susceptible to "oil whip".

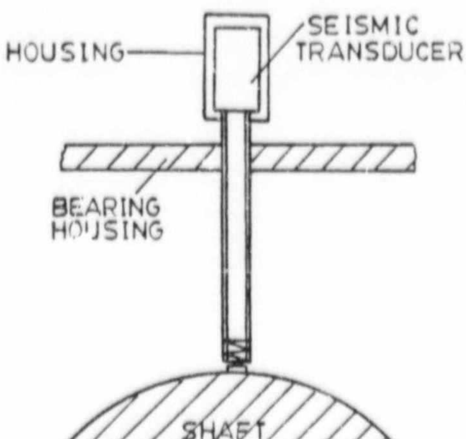
d) They offer a limited frequency response of 10 Hz to 120 Hz.

e) Due to moving parts and direct contact, sticking, slip, bounce, squeal and chatter can occur, providing some erroneous readings.

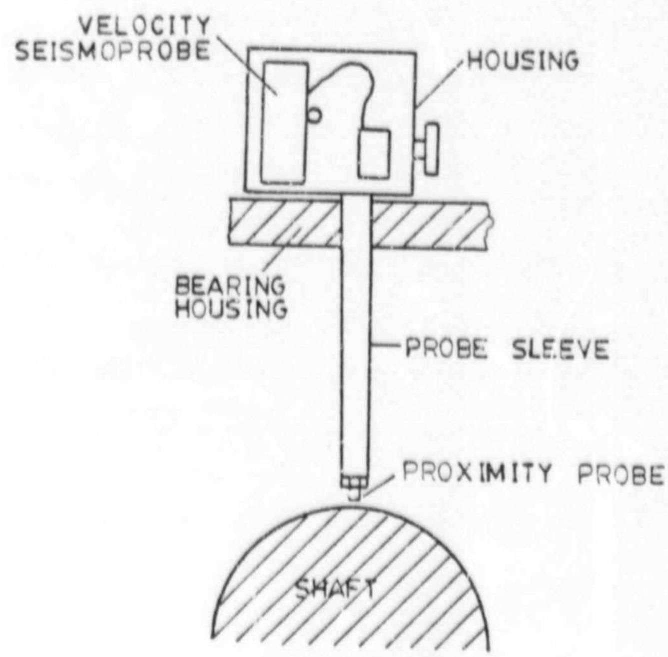
f) It is difficult to calibrate a shaft rider system.

3.5 DUAL PROBES

A dual probe is a combination of a proximity probe and a velocity transducer installed together at a common reference point. The proximity probe measures shaft relative motion while the velocity transducer measures bearing absolute motion. To obtain the shaft absolute motion, the velocity signal is first integrated once to displacement and then vectorially summed to the shaft relative displacement signal. An analog computer is employed to vectorially sum the two displacement signals. Figure 3.5 shows a typical installation of a dual probe.



SHAFT RIDER
FIGURE 3.4



DUAL PROBE
FIGURE 3.5

3.5.1 Advantages of Dual Probes

a) Same as for velocity and displacement transducers.

b) With the vectorial addition of the two displacement signals the shaft absolute motion can be obtained.

3.5.2 Disadvantages of Dual Probes

a) Same as for velocity and . transducers.

4. MODULE 1
DETECTION OF ROLLING ELEMENT BEARING DAMAGE BY STATISTICAL VIBRATION ANALYSIS

4.1 INTRODUCTION AND THEORY

Vibration signals may be considered to be statistical in nature. Therefore from the acceleration signals it is possible to derive a probability density of the instantaneous amplitudes. This distribution indicates the probability of occurrence of an acceleration of a particular amplitude. Rather than examining the actual probability density curve, it is more informative to examine the statistical movements of the data, from which the type of distribution may be assessed.

The probability density function p(x) may be expressed in mathematical terms:

$$P(x) = \lim_{\Delta x \; 0} \frac{P(x) - P(x + \Delta x)}{\Delta x} \qquad \text{Rogers (22)}$$

where: P(x) is the probability of occurrence of instantaneous amplitude values exceeding the level x. $P(x + \Delta x)$ is the probability of occurrence of instantaneous amplitude values exceeding the level $x + \Delta x$.

Kurtosis is a statistical term relating to the "peakiness" of the distribution of vibration amplitude. The idea behind the use of kurtosis is that some components degenerate as a result of the loss of gross areas of surface, so that progressively increasing periodic impulses are generated.

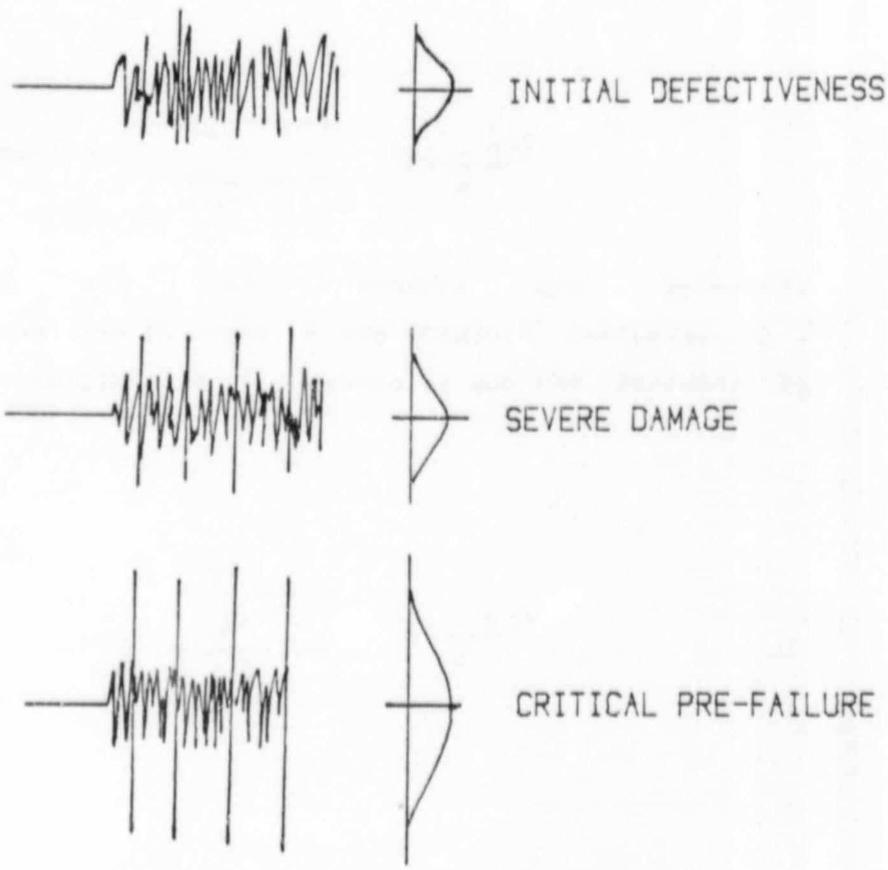
This is illustrated in Figure 4.1. It is primarily a measure of the wave shape. The kurtosis value for a Gaussian distribution of amplitudes, characteristic of the random amplitude noise generated by a new bearing, is 3. For a spiky signal, typical of a slightly damaged bearing, its value ranges between 3 and 20. Kurtosis is, therefore a method of allotting numbers to wave patterns, independent of the signal amplitude. A statistical distribution of such curves in the first order produces a normal form of distribution which tends to ignore the peaks of the shock. If the "statistical moments" are analysed in much the same manner as when obtaining first and second moments of area, the higher area moments become more sensitive to the "tails" of the distribution curve and accordingly register the size and frequency of the peaks - the very peaks that need to be monitored as a basis for condition appraisal.

The statistical moments of a set of data (x_i) which are of most use in defining the distribution are the first four namely:

i) First moment = $$\sum_{i=1}^{N} \frac{x_i}{N}$$

i.e. the mean value $\bar{x}$

ii) Second moment = $$\sum_{i=1}^{N} \frac{(x_i - \bar{x})^2}{N} = \sigma^2$$



PROGRESSIVELY INCREASING PERIODIC PULSES GENERATED FROM A DEGENERATING ROLLER OR BALL BEARING.

FIGURE 4.1

i.e. the variance and higher moments following the general form.

$$\text{jth moment} = \sum_{i=1}^{N} \frac{(x_i - \bar{x})^j}{N}$$

The third and fourth moments are generally non-dimensionalised by means of the standard deviation, σ, to give the coefficients of skewness B_1 and the kurtosis B_2 as follows:

$$\text{iii) } B_1 = \frac{1}{\sigma^2} \sum_{i=1}^{N} \frac{(x_i - \bar{x})^3}{N}$$

and

$$\text{iv) } B_2 = \frac{1}{\sigma^2} \sum_{i=1}^{N} \frac{(x_i - \bar{x})^4}{N}$$

The skewness indicates the degree of the probability density curve.

The kurtosis indicates the "peakiness" of the data. Kurtosis values for wave forms with zero skewness are:

Square wave	$B_2 = 1$
Sine wave	$B_2 = 1,5$
Gaussian	$B_2 = 3$
Random and impulsive	$B_2 > 3$

In general vibration signals from a bearing in a good condition have a Gaussian distribution and thus have a kurtosis value of three. Damaged bearings have a kurtosis value greater than three.

In the day to day operation of a power station the operating conditions of certain machines such as load or speed, may vary due to fluctuations in demand. To obtain the best results from a vibration condition monitoring programme, vibration data for a machine must be taken at the same mechanical conditions each time. It would be convenient to have a method of determining the condition of a machine with roller or ball bearings irrespective of the operating load or speed. Kurtosis offers a solution with some limitations. Kurtosis cannot be applied to bearings in close proximity to cavitation and to bearings where vibration is transmitted from other machines via the foundation or pipework.

4.1.1 Literature Review of Rolling Element Bearing Vibration Health Monitoring

Vibration analysis is based on the measurement of forces exerted by the bearings which excite their surroundings and cause structural vibration. The most common methods of analysing vibration data to diagnose bearing damage are by observing changes in the RMS level, the power spectrum and discrete frequency spectrum. Monk (23) observed that the RMS value increases with increasing bearing damage. He also suggests that a refinement to straight RMS level measurement is to observe changes in different octave, or one-third octave frequency bands. More detailed information can be obtained by narrow-band frequency analysis. Balderson (24) identified discrete bearing rolling frequencies and associated changes in their amplitude with particular bearing faults.

The Shock Pulse Method (SPM) system uses a piezo-electric accelerometer to measure the mechanical impact or shock pulse. The running surface of a bearing always has a certain roughness even when new, which causes low-level acoustic emission. As a rolling element bearing deteriorates, the rolling surfaces suffer damage. Cracks and pitting appear, small particles of metal come off and the wear debris is circulated within the bearing. As the fault areas and/or debris pass into the contact zones, they cause small, "knocks" which are transmitted into the bearing housing as a discontinuous shock pulses. The SPM meter employs an accelerometer tuned mechanically and electrically to a resonant frequency of 32 kHz. The compression wavefront or shock pulse caused by a mechanical impact sets up a dampened oscillation in the transducer at its resonant frequency. The peak amplitude of this oscillation is therefore directly proportional to the impact velocity and hence the condition of anti-friction bearings (25).

Another method which is used in bearing condition monitoring is the measurement of the crest factor. The crest factor is the ratio of peak level and RMS level, Collacott (26). The peak to RMS ratio can detect discrete bearing defects such as scratches, dents, nicks or inclusions. Bearings with defects exhibit peak/RMS values greater than unity, and the larger the defect the larger the ratio. Dyer and Stewart (27) have found the measurement of the crest factor to be partially insensitive to changes in bearing load and speed. They also acknowledge that incipient damage is most easily detected by changes in peak acceleration.

The Spike Energy Meter (SEM) measures three parameters of high frequency "pulses", namely, pulse amplitude, pulse rate and high frequency "random vibratory energy" associated with bearing defects. These three parameters are "electronically" combined into a single severity quantity called "g-SE".

The measured values of RMS, power spectral density, shock pulse and spike energy method are dependent upon bearing load, speed, housing tightness, quantity of lubricant and bearing clearance. It is therefore difficult to define the condition of a bearing from a single measurement. A statistical approach has proved the most successful method of quantifying these characteristic changes in peak acceleration levels and, unlike RMS and peak levels, it is independent of changes in load and speed. Dyer and stewart (27).

Rogers (22) and Dyer et al (27) both indicate that early damage gives rise to changes mainly in the lower frequency bands, whereas with more advanced damage the higher frequencies are affected most. However, only by observing the distribution of Kurtosis values in a selected band is it possible to predict the severity of damage. They both recommend the following frequency bands; 3 Hz - 5 kHz for incipient damage and 5 - 10 kHz for advanced damage.

4.2 OBJECT

The purpose of the study undertaken in this module was:

1) To determine the effectiveness of kurtosis as a means of monitoring the condition of roller and ball bearings.

2) To determine if load and speed has any effect on the kurtosis value obtained from a new and a damaged bearing.

3) To develop software for existing hardware so that the kurtosis value could be obtained.

4.3 APPARATUS

Recorders

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

Transducers

Two B&K accelerometers type 4371. One for the vertical and horizontal directions on the bearing housing. One Bently Nevada proximity probe type 7200 with a sensitivity of 8V/mm, used for speed indication.

Analysers

Wavetek 100A FFT analyser

Amplifiers and Power Supplies

Two B&K charge amplifiers type 2635
Bently Nevada -24V DC power supply

Filters

Wavetek 24D tracking filter
Kron-hite low/high pass filter

Demodulator

Bently Nevada oscillator/demodulator type 7200

Computer

Hewlett Packard desk top computer type HP9836

Test Rig

Bearing rotor test kit. See Figure 4.2

Details of test bearing

SKF type SY504M

Number of balls = 8

Pitch diameter = 34 mm

Ball diameter = 8 mm

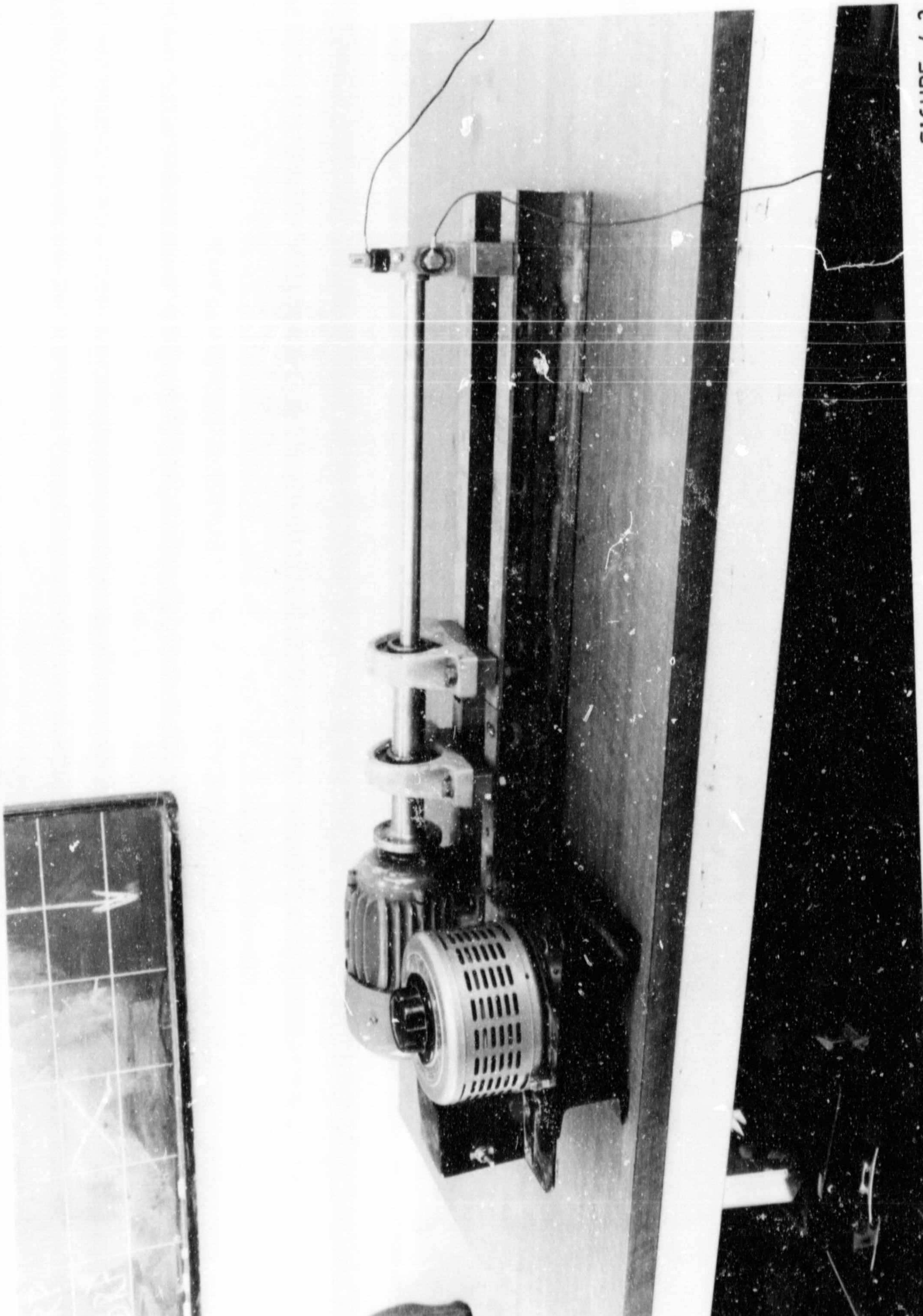


FIGURE 4.2

KURTOSIS BEARING ROTOR TEST KIT

4.4 EXPERIMENTAL PROCEDURE

4.4.1 Data Acquisition

The equipment used for this experiment was configured as shown in Figure 4.3. Accelerometers were mounted in the vertical and horizontal directions on the NDE bearing plummer block. The acceleration signals from the two accelerometers were recorded on a seven channel TEAC tape recorder via two B&K charge amplifiers.

A proximity probe was used as a tachometer and the output fed into the Wavetek 24D tracking filter where the speed was displayed digitally.

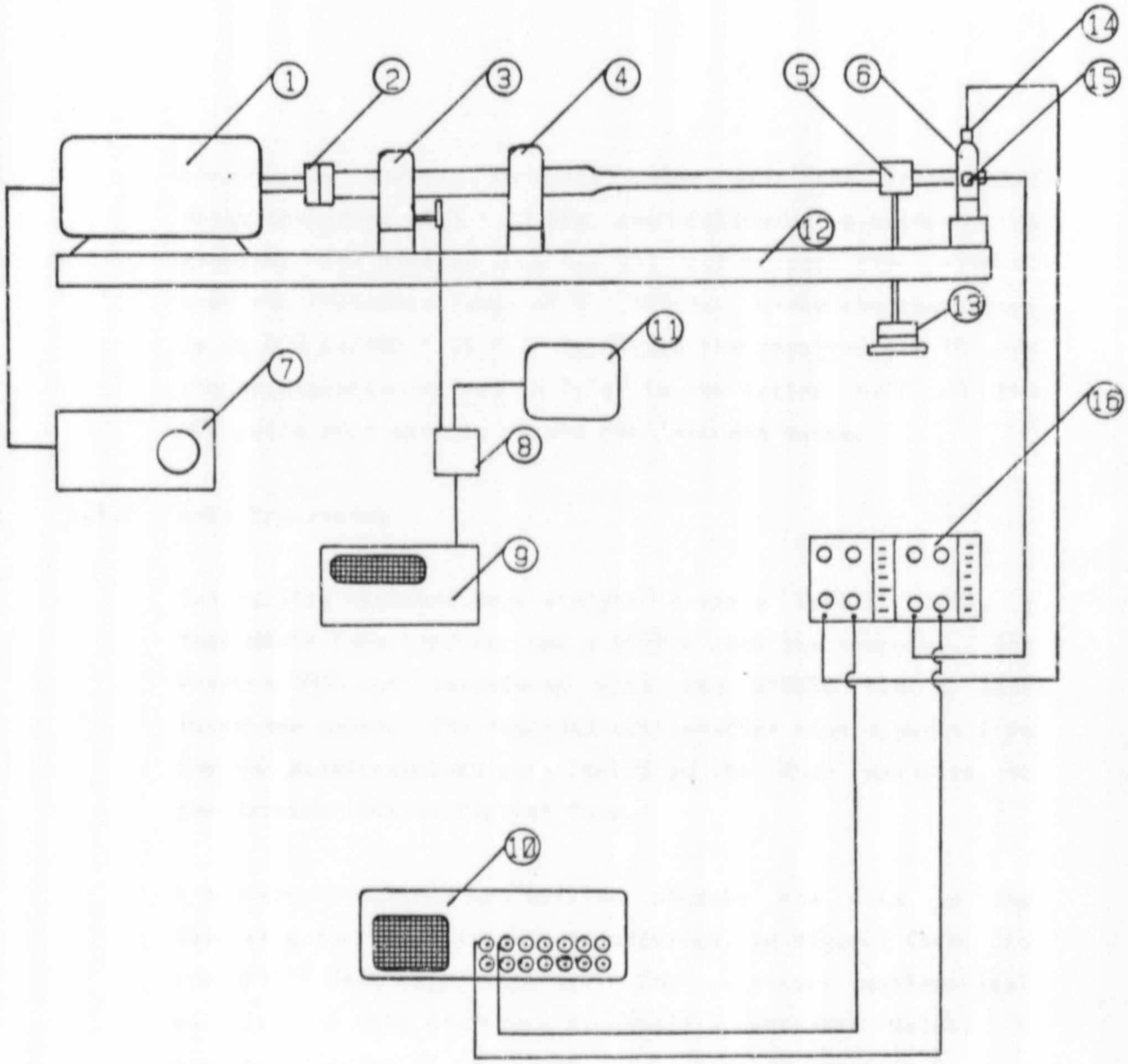
The motor speed was varied using a variable voltage transformer.

Different loads were applied to the bearing using the mechanism shown in Figure 4.3. The load was varied by changing the mass on the loading mechanism.

The above procedure was applied to a new bearing and to a damaged bearing. The original bearing was damaged by injecting fly-ash into it.

A 5 kHz frequency range was chosen on the Nicolet 100A for the new bearing. This frequency range was chosen so that incipient damage could be detected. The Kron-hite low/high pass filter was used as a high pass 3 Hz filter so that the recommended 3 Hz - 5 kHz frequency range could be used, Rogers (22). In the case of the damaged bearing a 10 kHz frequency range was chosen on the Nicolet 100A. In order to

DATA ACQUISITION



1. MOTOR
2. COUPLING
3. BEARING
4. BEARING
5. LOAD MECHANISM
6. TEST BEARING
7. VARIABLE VOLTAGE TRANSFORMER
8. OSCILATOR/DEMODULATOR
9. WAVETEK 24D TRACKING FILTER
10. TEAC 7 CHANNEL FM TAPE RECORDER
11. -24V DC POWER SUPPLY
12. BED PLATE
13. WEIGHTS
14. ACCELEROMETER (Vertical)
15. ACCELEROMETER (Horizontal)
16. B&K CHARGE AMPLIFIERS

FIGURE 4.3

obtain the correct Kurtosis value over the recommended frequency range of 5 - 10 kHz the following procedure was adopted. The Nicolet 100A has 400 cells per time window, over the frequency range of 0 - 10 kHz. Hence the resolution is 10 000 Hz/400 = 25 Hz. To obtain the required 5 - 10 kHz the acceleration values in "g's" in the latter half of the 400 cells were used to obtain the Kurtosis value.

4.4.2 Data Processing

The results obtained were analysed using a Wavetek 100A, a TEAC MR-30 tape recorder and a HP9836 desk top computer. The Wavetek 100A was interfaced with the HP9836 via a IEEE interface cable. The recorded acceleration time signals from the two accelerometers were stored in the four memories of the Wavetek 100A in digital form.

A computer programme was written whereby the data in the Wavetek's four memories was transferred, in digital form, to the HP9836 desk top computer. The necessary mathematical calculations were performed to obtain a Kurtosis value. A computer listing of programme can be seen in Appendix 1.

4.5 OBSERVATIONS AND RESULTS

Care was taken to align the motor, shaft and plummer blocks so as not to introduce false anomalies into the system.

The results obtained from the tests are displayed graphically in Figures 4.4 and 4.5.