FACTORS AFFECTING BRINELLING AND ITS DETECTION IN CYLINDRICAL ROLLER ELEMENT BEARINGS

BRIAN HOLMAN SMITH

N H DIP (MECH)

DISSERTATION SUBMITTED IN COMPLIANCE WITH THE REQUIREMENTS FOR THE MASTER'S DIPLOMA IN TECHNOLOGY IN THE DEPARTMENT OF MECHANICAL ENGINEERING AT TECHNIKON NATAL

SUPERVISOR : DR G J RAE - PhD., B.Sc., M.I.Mech.E., C.Eng., Eur.Ing.

CO-SUPERVISOR: DR D COERTZE - DSc (PUCHE) MBL (UNISA)

I HEREBY DECLARE THAT THIS DISSERTATION REPRESENTS MY OWN WORK BOTH IN CONCEPTION AND EXECUTION.

B Smith
....

SUBMITTED AT DURBAN

FEBRUARY 1992

ACKNOWLEDGEMENTS

I would like to acknowledge the following people and organisations, without whose assistance and support this research project would not have been possible.

Dr G J Rae, for his interest and guidance as a subject specialist. Especially for the opportunities he created for me within Eskom, encouraging me to always strive for the best, and engendering a professional approach towards assignments.

Dr D J Coertze, Technikon Natal. This document would not have attained the standard it has without his considerable input and guidance in terms of the research methodology.

Mr P Du Plessis, for his interest and comments and for patiently proof reading many draft copies of this document.

Mrs E. Powell, for many hours of typing and the promptness and efficiency with which it was done.

Fellow members of the Stress and Vibration sub-section for their comments and assistance.

For advice and support from various members of staff of:

Corrosion and Surface Engineering

Analytical Chemistry

Applied Electronics

Physical Metallurgy

Library Services

Photographic

Statistics

Workshops

Graphics

ABSTRACT

Brinelling exists in two forms, namely true brinelling and false brinelling. Bearing damage due to false brinelling is a common problem and is known to occur on power stations in Eskom. An example of false brinelling has recently been identified on the ejector pump motors at Lethabo Power Station. True brinelling does not pose a problem to Eskom, and has therefore not been included in this study

False brinelling is a special form of fretting and is found to occur in stationary roller element bearings subject to vibration. The vibration creates load fluctuations and varying degrees of relative motion within the bearings. As a result, small depressions or flutes are formed in the race (Plate 1.1) which eventually lead to increased operational vibration and premature bearing failures.

Previous studies do not make any reference to factors affecting false brinelling in horizontally mounted bearings. The need for a better understanding of false brinelling has increased with the cold storage and partial cold storage of under utilized power stations.

This investigation approaches the problem of false brinelling from a new angle. Horizontally mounted bearings have been used for testing as opposed to vertically mounted bearings used by other researchers. The effects of static bearing load, impedance, vibration amplitude and direction on false brinelling have been investigated.

The study has shown that bearing impedance and load do not have a significant influence on the rate of bearing damage, while vibration

amplitude and direction contribute directly to the rate of damage.

The effects of vibration amplitude, direction and cycles have been combined to create a model for assessing the extent of false brinelling in horizontally mounted bearings.

False brinelling in vertically mounted roller element bearings is dependent on the dynamic load or force experienced by the loaded elements. This investigation has shown that in horizontally mounted bearings, the amplitude and direction of vibration on the unloaded elements are the determining factors. A considerable difference therefore appears to exist between factors affecting false brinelling in vertically and horizontally mounted bearings.

The model developed in this study provides an indication of bearing damage which may be expected to occur in horizontally mounted bearings subject to single frequency vibrational excitation. This model, together with the information obtained from the literature study, provides a base from which Eskom personnel can determine preventative measures required to reduce the possibility of bearing damage occurring as a result of false brinelling. These models are not limited in application to Eskom, but may be applied to any stationary roller element bearing exposed to vibration.

A number of additional factors have arisen from the investigation which may influence the process of false brinelling. Recommendations have therefore been made to address these factors in order to establish their significance.

Plate 1.1 Inner race of roller element bearing showing typical damage caused by false brinelling. The "scars" or flutes (A, B, C, and D) are formed as a result of relative motion between roller element and race.

UITTREKSEL

Brinellering kom voor in twee vorme, naamlik ware brinellering en valse brinellering. Laerbeskadiging as gevolg van valse brinellering is 'n algemene probleem in verskeie industrieë en ook in kragstasies van Eskom. 'n Voorbeeld van valse brinellering is geïdentifiseer op die ejekteurpompmotore van Lethabo Kragstasie. Ware brinellering blyk geen probleem vir Eskom te wees nie, daarom is dit nie in hierdie studie ingesluit nie.

Valse brinellering is 'n spesiale vorm van uitvreting wat plaasvind in stilstaande rollaers wat onderhewig is aan vibrasie. Die vibrasie veroorsaak lasfluktuasies en wisselende grade van relatiewe beweging in die laers. Die gevolg is dat klein holtes of groewe gevorm word in die laerring (Plaat 1.1) wat uiteindelik lei tot verhoogde operasionele vibrasievlakke en voortydige laerdefekte.

Vorige studies maak geen melding van faktore wat valse brinellering in vertikaal gemonteerde laers beïnvloed nie. Die behoefte aan 'n beter begrip van valse brinellering het toegeneem deurdat sommige kragstasies onderhewig is aan gedeeltelike of totale koue berging.

Hierdie studie ondersoek die effek van statiese laerlas, impedansie, vibrasie-amplitude en rigting van valse brinellering in horisontaalgemonteerde laers. Die resultate toon dat laer impedansie en las nie 'n merkwaardige invloed het op die tempo van laerbeskadiging nie, maar dat vibrasie amplitude en rigting wel 'n invloed het.

Die effek van vibrasie amplitude, rigting en siklusse is gekombineer om 'n model te vorm vir die beraming van die mate van valse brinellering in horisontaal gemonteerde laers.

Valse brinellering in vertikaal gemonteerde rollaers is afhanklik van die dinamiese las of kragte wat die belaste elemente ondervind. Hierdie ondersoek het getoon dat in horisontaal gemonteerde laers, die amplitude en rigting van vibrasie op die onbelaste elemente, die bepalende faktore is. As gevolg hiervan, bestaan daar 'n aansienlike verskil tussen faktore wat valse brinellering in vertikaalgemonteerde en horisontaalgemonteerde laers beïnvloed.

Die model ontwikkel in hierdie studie gee 'n aanduiding van die laerbeskadiging wat kan plaasvind in horisontaalgemonteerde laers wat onderhewig is aan vibrasie veroorsaak deur 'n enkele frekwensie. Hierdie model, tesame met die inligting verkry uit die literatuurstudie, verskaf 'n basis waarvolgens Eskom-personeel voorkomende maatreëls kan bepaal vir die voorkoming van moontlike laerbeskadiging. Hierdie model is nie beperk tot aanwending binne Eskom nie, maar is toepaslik vir enige stilstaande rollaers blootgestel aan vibrasie.

'n Aantal addisionele faktore wat die proses van valse brinellering kan beïnvloed, het uit die ondersoek navore gekom. Aanbevelings word dus gemaak ten opsigte van verdere navorsing.

TABLE OF CONTENTS PAGE

PAGE

PAGE

LIST OF TABLES

LIST OF FIGURES

LIST OF FIGURES IN APPENDICES

LIST OF PLATES PAGE

LIST OF SYMBOLS

Force	Newton	N
Load	kilogram	kg
Displacement	metre	m
Velocity	millimetre per second	mm/s
Acceleration	metre per second squared	m/s^2
Impedance	force/velocity	N.s/mm
Frequency	Hertz	Hz
Time	Seconds	s
Coefficient of determination		R^2
Revolutions per minute		rpm

CHAPTER ONE

THE PROBLEM AND ITS SETTING

1.1 INTRODUCTION

False brinelling is a special form of fretting and is found to occur in stationary roller element bearings subject to vibration. The vibration gives rise to relative motion within the bearing, resulting in small depressions or flutes being formed on the bearing race. These flutes eventually lead to increased operational vibration and premature bearing failures.

The effects of false brinelling are of particular concern to Eskom at present. The cold storage and partial cold storage of certain power stations has introduced additional problems in this regard, over and above those encountered during normal power station operation.

An average conventional power station has in excess of 1000 electric motors larger than 7.5 kW. The cold storage of a single plant will therefore result in approximately 2000 bearings from this area alone standing for extensive periods of time without operating. These bearings will be exposed to possible damage through false brinelling.

Bearings damaged in this way are likely to go unnoticed until the equipment is recommissioned. Severely damaged bearings would fail very quickly, or at least produce excessive vibration levels, necessitating the expense of replacement and possibly causing delays in the recommissioning process. Those bearings less severely damaged are likely to cause additional problems and possible outages in the ensuing weeks and months. The forced outage of a single 600 MW unit can cost Eskom R 1 000 000 per day in lost revenue.

Studies have been conducted previously into false brinelling in vertically mounted bearings, and a model exists for determining bearing life under such conditions (Pittroff 1964). However, no reference has been found relating to the characteristics of false brinelling in horizontally mounted bearings. An understanding of the factors affecting false brinelling in horizontally mounted bearings is therefore required in order to develop a method for assessing its presence and severity.

The ability to assess the existence of false brinelling on Eskom plant, and a knowledge of the factors affecting this process will enable preventive measures to be taken. This will reduce maintenance costs and improve plant reliability.

Eskom's Engineering Investigations (E-I) department is a multi-disciplinary test and research facility and as such is ideally suited for undertaking such an investigation.

1.2 PROBLEM STATEMENT

This study evaluates the individual effects of static load, bearing impedance, vibration amplitude and direction on flute depth due to false brinelling in horizontally mounted roller element bearings, for the purpose of establishing a method for predicting the presence and severity of false brinelling on Eskom plant.

1.3 HYPOTHESIS

It is hypothesised that static load, vibration amplitude, direction and bearing impedance will have a direct influence on the severity of false brinelling, and as such the inter-relationship of these factors may be used as a means of predicting its occurrence and severity.

1.4 DELIMITATIONS

The study will concentrate on damage due to false brinelling under vibrational excitation. Although true brinelling damage can occur, it is not a common problem on Eskom plant. True brinelling is most likely to result from negligence during the maintenance or transportation of equipment.

The study will be restricted to cylindrical roller element bearings. Roller element bearings are more susceptible to false brinelling damage than ball bearings due to their geometrical shape and larger internal clearances, and will therefore be first to suffer damage.

Unlubricated bearings will be used in order to exclude the complex and unpredictable effects of lubrication. Both Pittfoff (1964:6) and Du Randt (1991:4) made use of unlubricated bearings in separate investigations into false brinelling in vertically mounted bearings.

The study will include single frequency sinusoidal excitation only. As most vibration transmitted to stand-by machines originates from other rotating machinery, operating at a constant speed, the use of a discrete forcing frequency will provide a valid model.

The research will be restricted to horizontally mounted bearings although literature on vertically mounted bearings has been included and reviewed.

1.5 ASSUMPTION

As bearings are manufactured to international standards, it is assumed that the effect of any variation in bearing material hardness or composition from one bearing to another will be negligible.

1.6 DEFINITIONS

1.6.1 Basic Load Rating

"The basic load rating is defined as that load which will produce a total permanent deformation of the rolling element and race way on the most heavily stressed rolling element/race way contact of 0,0001 of the rolling element diameter" (SKF General Catalogue 1981:62).

1.6.2 Bearing Clearance

For a radial bearing, the bearing clearance will be the total distance through which one bearing race can be moved relative to the other in a radial direction.

1.6.3 Dynamic Bearing Load

The dynamic bearing load is the fluctuating load or force experienced by the elements of a bearing as a result of vibratory motion.

1.6.4 False Brinelling

False brinelling is a special case of fretting found to occur in stationary roller element bearings. Relative motion is a prerequisite for the occurrence of false brinelling (Pittroff 1964), and may be the result of vibration, low angle radial oscillation, or a combination of these.

1.6.5 Fretting

Repeated small amplitude relative sliding between two surfaces in contact in the presence of oxygen results in a special form of erosion/corrosion known as fretting.

1.6.6 Impedance

Mechanical impedance is the ratio of force to velocity and is a measure of the resistance a system has to move at a particular velocity.

1.6.7 Nominal Bore

Nominal bore is the inside diameter of the inner bearing race.

1.6.8 Oscillation Angle

The angle rotated by a shaft while fluctuating between a clockwise and anti-clockwise direction, measured in degrees.

1.6.9 Root Mean Square (rms)

The rms value of a single frequency vibration amplitude is equal to 0.707 times the 0 - Peak value.

1.6.10 Static Bearing Load

The total mass carried by a bearing measured in kilograms.

1.6.11 True Brinelling

True brinelling occurs on the surface of a material when the elastic limit has been exceeded, usually as the result of an impact.

1.6.11 Vibration Amplitude

A measure of the velocity with which a system is vibrating in mm/s .

1.7 OUTLINE OF THE CHAPTERS

CHAPTER 2

The literature review covers aspects on both false brinelling and fretting related investigations. The topics include relative motion, lubrication, bearing pre-load, velocity of sliding and vibration amplitude. Two models relating to false brinelling in vertically mounted bearings have also been discussed.

CHAPTER 3

The general procedure sets out the type of data that is required, the methods of obtaining these data and processes used for analyses. Preferences for certain methods and approaches are discussed in order to assist and simplify further research. Particular attention is paid to the sample design which discusses in detail, methods of excitation, the assessment of flute depth, lubrication, selection of test parameters and development of the test rig .

CHAPTER 4

The results include the data from each of the tests as well as the calculations required to determine the directional contributions. The analytical and graphical models, for assessing the presence and severity of false brinelling in horizontally mounted bearings, have also been included in this chapter.

CHAPTER 5

A method for assessing false brinelling in horizontally mounted bearings has been developed and is discussed in this chapter. The effects of humidity, lubrication, bearing clearance and multi-frequency excitation are also discussed.

CHAPTER 6

Recommendations for future work include the effects of lubricants, multi-frequency and uni-directional excitation and an alternative method for assessing the existence of false brinelling. The requirements for obtaining comparable results are discussed.

CHAPTER TWO

REVIEW OF RELATED LITERATURE

2.1 INTRODUCTION

It is necessary at the outset to establish a clear understanding of the terms false brinelling and fretting, and how these two are related.

Fretting occurs when two metal surfaces in contact experience oscillating relative motion. In the primary process metallic wear particles are produced due to abrasion, fatigue fracture and material transfer. These particles quickly react to the environment, eg. air, lubrication and moisture, and undergo varying degrees of chemical change. The oscillating motion grinds these particles between the surfaces resulting in further abrasion and increased wear rates.
A special case of this fretting mechanism is found to occur in stationary roller element bearings subject to vibration and is referred to as false brinelling.

A close relationship therefore exists between these two terms. For this reason literature concerning both fretting and false brinelling has been included in the review.

2.2 FRETTING RELATED INVESTIGATIONS

The fretting process is a relatively well researched topic and literature covering a number of aspects of this phenomena is readily available. Those aspects most likely to contribute to an understanding of the false brinelling process have been included in this review.

2.2.1 Dependence of Fretting on Relative Motion

Relative motion is a prerequisite for the occurrence of fretting. Tomlinson (1927:472-483) undertook experiments to investigate the effects of relative motion in steel interfaces.

Displacements in the region of $10^{-4}\mu m$ were investigated and fretting damage was evident. Tomlinson suggests that the reason for fretting occurring under such minute displacements is that molecular cohesion is probably taking place. If this explanation is correct then it is likely that molecular cohesion will also be present in the interface between rolling element and race of a bearing, and may well play a part in the process of false brinelling. Relative motion or "amplitude of slip" is also seen as an important variable by Waterhouse (1982:6).

2.2.2 The Effect of Contact Load on Fretting

The contact load between two elements has been shown to affect the severity of fretting. Bill (1982:168) comments that generally fretting wear will increase nearly proportionally to contact load providing that the interface motion is held constant. A correlation between contact load and fretting wear suggests that bearing load, whether dynamic or static, may well influence the process of false brinelling.

As wear increases between curved and flat surfaces, as with cylindrical roller element bearings, the resultant pressure at the point of contact will be reduced (Waterhouse 1982:6). If the pressure is being reduced the effective load is decreasing and therefore presumably the rate of brinelling should decrease, assuming excitational levels remain constant.

2.2.3 The Effect of Frequency on Fretting

Frequency alone does not have an influence on the process of fretting. Fretting damage can occur at almost any frequency (Budinski 1982:50). Damage does not depend on the rate at which some displacement may oscillate, but rather on the fact that relative motion has occurred. Waterhouse (1982:6) supports these views by stating that frequency is of lesser importance as a variable in the fretting process.

2.3 ROLLER ELEMENT BEARINGS

It has been established that roller element bearings are more susceptible to false brinelling than ball bearings. This is due to the geometric shape of the roller elements, and the larger internal clearances in comparison to an equivalent size of ball bearing. The larger internal clearances allow for increased relative motion between inner and outer race, thereby inducing larger forces at the interface between roller element and race.

An investigation was carried out by Pittroff (1964:1-11) into false brinelling in vertically mounted cylindrical roller element bearings type NJ 204. These bearings have a nominal bore of 20mm and a static load carrying capacity of 7,35 kN.

2.3.1 Bearing Lubrication

In order to eliminate this complex factor and reduce the time required for testing, Pittroff removed all lubrication from the bearings before testing.

2.3.2 Relationship Between Flute Width and Stress Reversals

Pittroff (1964:6) found that the flute width increased parabolically with the number of stress reversals. This is due to the curvatures of the roller element and race, and is expected to hold true regardless of a bearing being vertically or horizontally mounted.

2.3.3 Relationship Between Load and Rate of Damage

Damage per unit time increases with increasing load. Pittroff's finding (1964:6) agrees with the fretting related investigations, and assists in clarifying the relationship between fretting and false brinelling.

Similar tests were carried out on vertically mounted roller bearings by Du Randt (1991). The bearings used were NU 313 roller element bearings having a nominal bore of 65 mm, considerably larger than the 20 mm nominal bore of the NJ 204 bearings used by Pittroff.

The main thrust of Du Randt's investigation was to establish the effect of velocity amplitude and time of exposure on false brinelling in the ejector pump motors at Lethabo Power Station. Recommendations were to be made towards the development of an improved technique for predicting the occurrence of false brinelling in vertically mounted bearings. Du Randt concluded that velocity amplitude, as a measure of the dynamic load, could be used as a criteria for the prediction of false brinelling on these particular units.

By comparison, Pittroff's results show a 3,3 newton change per μm of flute depth and Du Randt's a 7,1 newton change per μm of flute depth. The difference in force per μm of flute depth may possibly be attributed to the difference in bearing sizes used by the two researchers.

Du Randt's investigation is important in that bearings of larger nominal bore than 50 mm were tested. The load information from both these investigations should prove useful in assessing false brinelling in vertically mounted bearings in the Eskom environment.

2.3.4 False Brinelling and Elastic Deformation

The existence of a changing elastic deformation is a prerequisite for the occurrence of false brinelling. Pittroff (1964:5) explains the relative motion responsible for false brinelling as being that motion which takes place due to deformation of the rolling element subject to an alternating load. This effect can more easily be visualised by partially depressing a tennis ball against a flat surface and noting the change in shape and contact area. It is shown that significant relative motion can occur from loads as small as 0,03 % of static capacity.

It can therefore be concluded that false brinelling will only occur where loaded rolling elements are in contact with the race. Applying this reasoning to horizontally mounted bearings, damage will only be expected to occur in an arc of 120°. This is generally the angle of contact of the loaded elements.

2.3.5 The Effect of Frequency on False Brinelling

Frequency does not appear to have any direct influence on false brinelling (Pittfoff 1964:6). As was concluded in the fretting related investigations, it is not the frequency but the degree of relative motion which influences the extent of damage.

2.3.6 Bearing Pre-load as a Means of Preventing False Brinelling

Where the pre-loading forces are in the magnitude of the bearing static capacity, no false brinelling will occur. Pittroff (1964:7) reduced the bearing clearance to zero in order to apply pre-loads and found that loads in the region of bearing static capacity were required to eliminate damage. This method of reducing false brinelling will not be suitable for use in Eskom as the majority of bearings require specific radial clearances in order to operate. Bearing clearance is especially critical in cylindrical roller element bearings.

2.3.7 Relationship Between Flute Width and Bearing Clearance

An increase in bearing clearance gives rise to an increase in the resulting flute width in a linear relationship (Pittroff 1964:7). With vertically mounted bearings the shaft is usually free to swing as a pendulum, the degree of swing being controlled by the amount of radial clearance in the bearing. However, in horizontally mounted bearings the shaft rests on the elements and no free play exists, regardless of bearing clearance. It is therefore expected that bearing clearance will be of less significance in horizontally mounted bearings.

2.4 MODEL FOR ESTABLISHING BEARING LIFE IN VERTICALLY MOUNTED BEARINGS

A graphical method exists for establishing bearing life in vertically mounted bearings under false brinelling conditions. This was developed by Breward (1973:1-3) from the mathematical relationship given by Pittroff (1964:9). Pittroff's equation includes the dynamic load expressed in terms of the bearings static capacity, the curvature of the bearing elements and the number of stress reversals. Pittroff also determines experimentally the maximum fluting a bearing may sustain without affecting its operation.

Breward's graphical method is detailed in Appendix A. This method is however subject to the same limitations which were applied to Pittroff's investigation. The method should only be applied to vertically mounted bearings of nominal bore less than 50 mm.

2.5 ANTI-FRETTING AND ANTI-WEAR PROPERTIES OF LUBRICANTS

No relationship exists between the anti-wear properties of a grease and its anti-fretting corrosion characteristics. Christian and McConnell (1971:21) found that the degree of fretting increased with a collective increase in load, speed and angle of oscillation but other variations either singly or collectively had no significant effect. A typical test made use of an 89 kg load, a 6° oscillation and a frequency of 10.25 Hz. Load is shown to have the greatest effect.

Considerable advances have been made in the field of lubrication and synthetic oils, particularly in the form of extreme pressure (E.P.) additives. It is felt that this area is worth investigating at a later stage as a possible means of reducing the effects of false brinelling.

2.6 SUMMARY

The literature review has shown that a close relationship exists between false brinelling and fretting. The common factors being the dependence on relative motion, the influence of contact load on the rate of damage and that frequency has no direct influence.

Roller element bearings are more susceptible to false brinelling than ball bearings due to the geometric shape of the rollers, and the larger internal clearances that roller bearings have.

It was also established that no relationship exists between the anti-wear properties of a grease and it's anti-fretting properties.

Although investigations into false brinelling have been carried out on vertically mounted bearings for which a model for calculating bearing life was established, no reference is made to horizontally mounted bearings.

CHAPTER THREE

GENERAL PROCEDURE

3.1 INTRODUCTION

A number of factors required consideration during the development of an appropriate procedure for assessing false brinelling in horizontally mounted bearings. Literature relating to both false brinelling and fretting was used to establish an understanding of factors likely to influence the rate of bearing damage.

The present investigation makes use of horizontally mounted bearings as opposed to vertically mounted bearings used by both Pittroff (1964) and Du Randt (1991). Information obtained from these studies could therefore only be used as a guide in the construction of a test procedure. In order to ensure that the investigation was relevant to Eskom, bearing conditions at Lethabo Power Station were assessed and included in the construction of test parameters.

3.2 DATA USED

3.2.1 Primary Data

The primary data is the information collected directly from the experiments and includes the following:

flute width	... μm
load	... kg
vibration amplitude	... mm/s rms
direction of vibration	... vertical, horizontal and axial
bearing impedance	... N.s/mm
bearing clearance	... μm

Flute depth, calculated from the measured flute width, is the criterion used for determining the extent of bearing damage resulting from false brinelling.

3.2.2 Secondary Data

The secondary data includes the results of other research and information from the literature review. Information relating to factors affecting false brinelling were of particular importance in the development of the test procedure and testing equipment.

The following information was utilized:

Effect of: relative motion on fretting
contact load
frequency
number of cycles
changing elastic deformation
radial clearance
lubrication

3.3 CRITERIA FOR THE ACCEPTABILITY OF DATA

3.3.1 Flute Clarity

Flutes must be clearly distinguishable on the bearing race. This is important for the measuring and assessing of bearing damage. As flute width is a direct measure of bearing damage, any confusion or unclarity in this area will only create questionable data.

3.3.2 Number of Flutes

Only one clear flute may be present at each point of roller contact. The existence of two or more flutes at any one point indicates that the shaft or rolling elements have moved, and as such will not produce comparable or repeatable results.

3.3.3 Bearing Rotation

The bearing may not be rotated during or after the false brinelling process as this will damage the flute edges and render the measurement of flute width extremely difficult.

3.4 METHODOLOGY

An exploratory study has been undertaken in order to establish the effects and interdependence of bearing impedance, vibration amplitude, direction and load on flute depth due to false brinelling in horizontally mounted cylindrical roller element bearings.

Experimental and analytical research will establish and compare the size of the flutes resulting from a change in either bearing impedance, vibration amplitude, direction or load. The individual contribution that these factors make to the false brinelling process will be assessed. This will enable the construction of a model for predicting the presence and severity of false brinelling in horizontally mounted roller element bearings.

3.5 SAMPLE DESIGN

Bearings at various Eskom Power Stations were examined in order to establish realistic values and parameters for testing. An appropriate experimental procedure was then constructed, and an overview of the test schedule is included in Table 3.1.

3.5.1 Selection of Bearing Type

A cylindrical roller element bearing NU 309 was selected for testing as representative of a typical intermediate bearing size commonly used in Eskom. This bearing has a nominal bore of 45 mm, a static load rating of 41,5 kN, and is commonly used in the 15 kW electric motors on power stations. A roller element bearing was selected in preference to a ball bearing as the geometric shape of the rollers, and the larger internal clearances, make these bearings more susceptibility to damage through false brinelling.

3.5.2 Selection of Parameter Ranges for Testing

3.5.2.1 Load

Static loads between 11 kg and 170 kg were applied to the bearing shaft during testing. The rotor of a 15 kW motor was weighed and found to have a mass of 27 kg. So as to allow for additional loading which might arise from couplings, pulleys and overhung shafts, the bearing mass was set at 56 kg when this parameter was held constant.

Table 3.1 Experimental test conditions for assessing the effects of static load, impedance, vibration amplitude and direction on false brinelling.

TEST	CYCLES	LOAD	IMPEDANCE	VIBRATION
1	R	C	C	C V A
2	C	R	C	C V H
3	C	C	R	C H A
4	C	C	C	R V H
5	C	C	C	R V A
6	C	C	C	R H A

C = condition held constant

R = condition varied through a range

A = axial excitation

H = horizontal excitation

V = vertical excitation

The application of loads as large as 170 kg is representative of situations where bearings support larger items such as fan impellers.

3.5.2.2 Frequency

The frequency of excitation was held constant at 50 Hz throughout the tests. This is a common frequency of operation of electrical motors on power stations and as such, is a common frequency of transmitted vibration.

3.5.2.3 Bearing impedance

Impedance levels ranged between 20 N.s/mm and 150 N.s/mm. The impedances of a series of motors at Lethabo Power Station were measured in order to establish an appropriate range. These measurements have been tabulated together with the ranges used in the experiments in Table 3.2.

3.5.2.4 Vibration amplitude and direction

Typical transmitted vibration levels recorded at various power stations were in the region of 1,5 mm/s rms at 50 Hz. However, individual machines have been known to experience transmitted levels well in excess of 10 mm/s rms. For this reason the amplitude of vibration tested extended from 1 mm/s rms to a maximum of 22 mm/s rms. Vibration was examined in all three directions, vertical, horizontal and axial, as vibration can exist in any direction.

Table 3.2 Comparison between impedance levels used during tests and impedance levels measured at Lethabo Power Station.

DIRECTION	IMPEDANCE N.s/mm	
VERTICAL	LOWER RANGE	UPPER RANGE
M	33	666
T	104	122
HORIZONTAL		
M	100	250
T	19.3	133
AXIAL		
M	119	158
T	27.7	149

M = Impedance values measured at Lethabo Power Station

T = Impedance values used for testing in laboratory

3.5.3 The Use of Unlubricated Bearings

Both Pittroff (1964) and Du Randt (1991) made use of unlubricated bearings in their investigations. The removal of lubrication eliminates an otherwise complicated and unpredictable factor, and also reduces the time required for testing. Unlubricated bearings have therefore been used in the present investigation. Petroleum ether was used for cleaning as this solvent is compatible to the particular lubrication applied to the bearings and was recommended by the supplier. For the purpose of this investigation the factors of 15 and 30 for grease and oil lubrication respectively, will be applied as recommended by Pittroff (1964:8). It should be noted that these values were obtained in 1964 for vertically mounted bearings. Further investigation will be required in order to establish more appropriate factors for horizontally mounted bearings and modern lubricants.

3.5.4 Selection of Excitational Source

A rotary shaker with adjustable weights was used to induce the vibration levels on the test rig. Waterhouse (1982:7) lists three methods of actuating fretting conditions:

1. mechanically, eg. an eccentric cam.
2. an electromagnetic shaker.
3. rotating out of balance weights.

The out of balance weights are recommended by Waterhouse for frequencies between 25 Hz and 50 Hz.

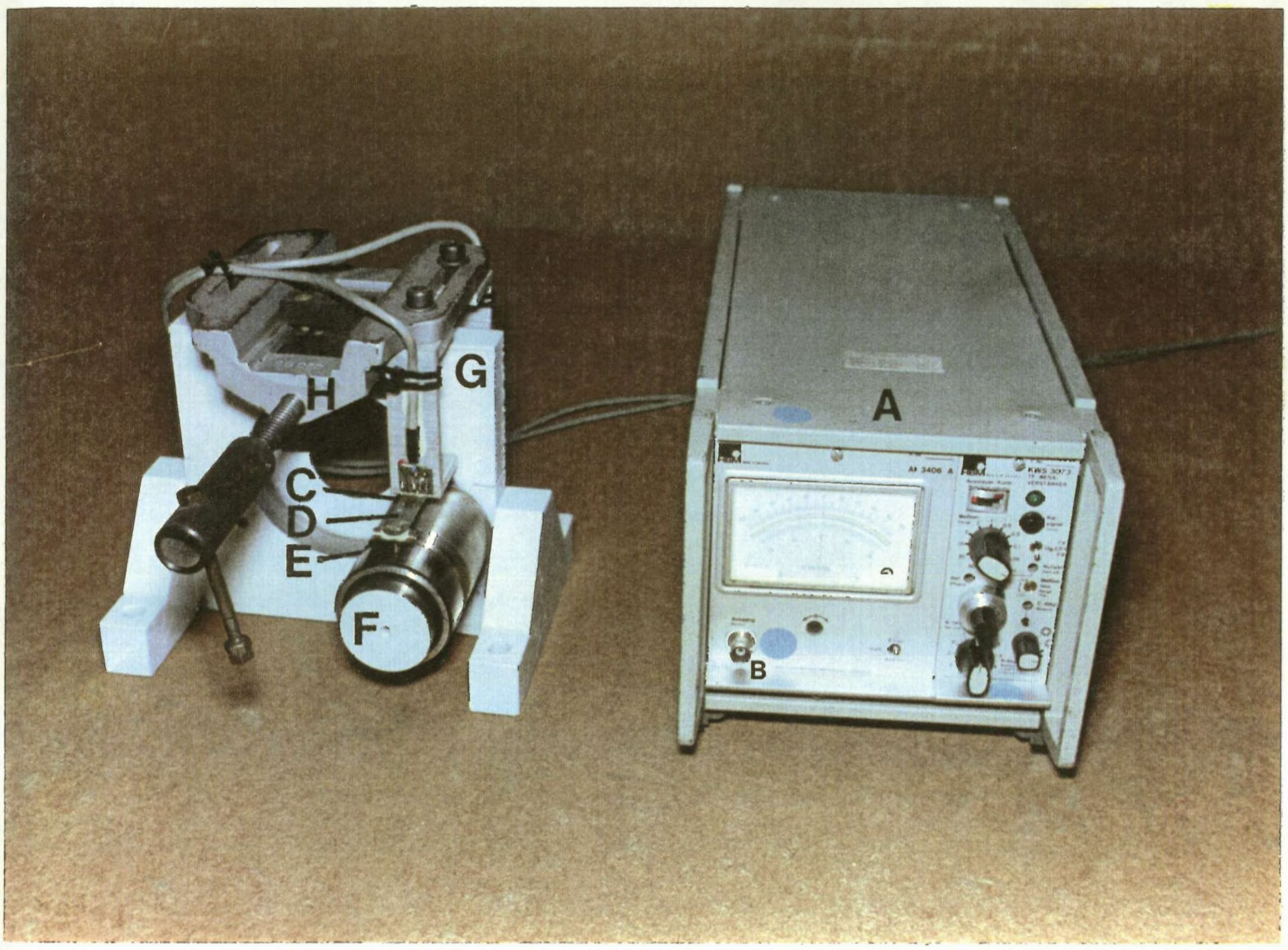
Both the eccentric cam and electromagnetic method of excitation provide uni-directional excitation unless multiple systems are implemented. This would be expensive and complicated. As the excitation of plant is seldom uni-directional, and as rotary shakers and controls are readily available, this method of excitation has been selected.

3.5.5 Measurement of Bearing Damage

Bearing damage was assessed in terms of flute depth which was calculated once the flute width had been measured using a calibrated profile projector. In order to validate these calculations a number of flute depths were measured using a strain gauged cantilever and stylus especially constructed for this application (Plates 3.1 to 3.3).

In order to provide a rigid base for the cantilever system, a dummy shaft was welded to the side of a spare housing onto which the bearing inner race was fitted. A probe was fixed below the relevant cantilever and the system calibrated using a micrometer. The calibration curve is presented in Figure 3.1.

The voltage output was connected to the frequency analyser and examined in the time series over a 30 second window. By slowly rotating the bearing race, the depth of flutes could be measured. These results were compared with those calculated from the flute widths.



A	Amplifier	E	Stylus
B	Voltage output	F	Dummy shaft
C	Strain gauge	G	Bearing housing
D	Cantilever	H	Clamp

PLATE 3.1 Strain gauged cantilever system and amplifier used for measuring radial bearing clearance and flute depth during false brinelling investigation.

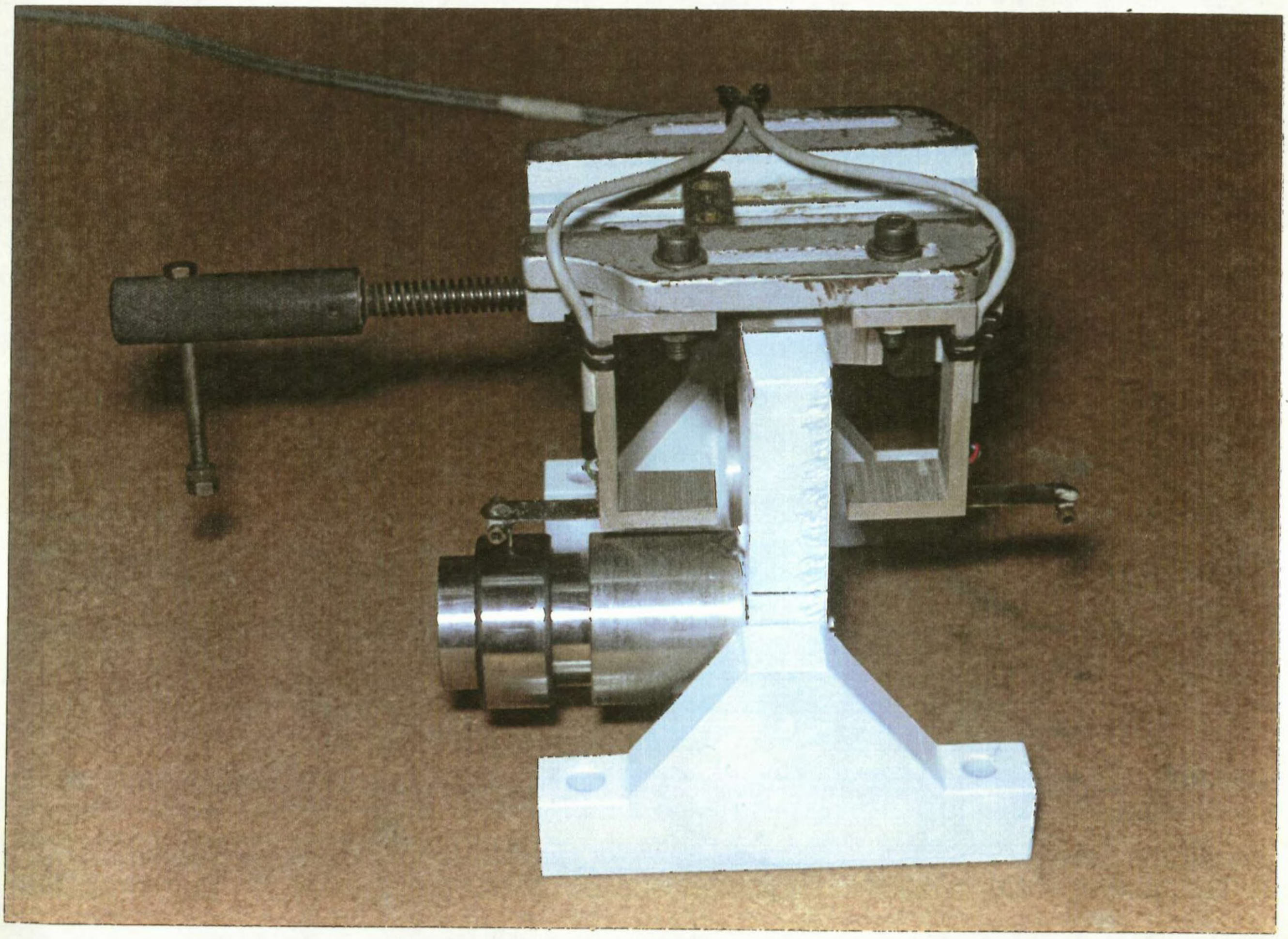
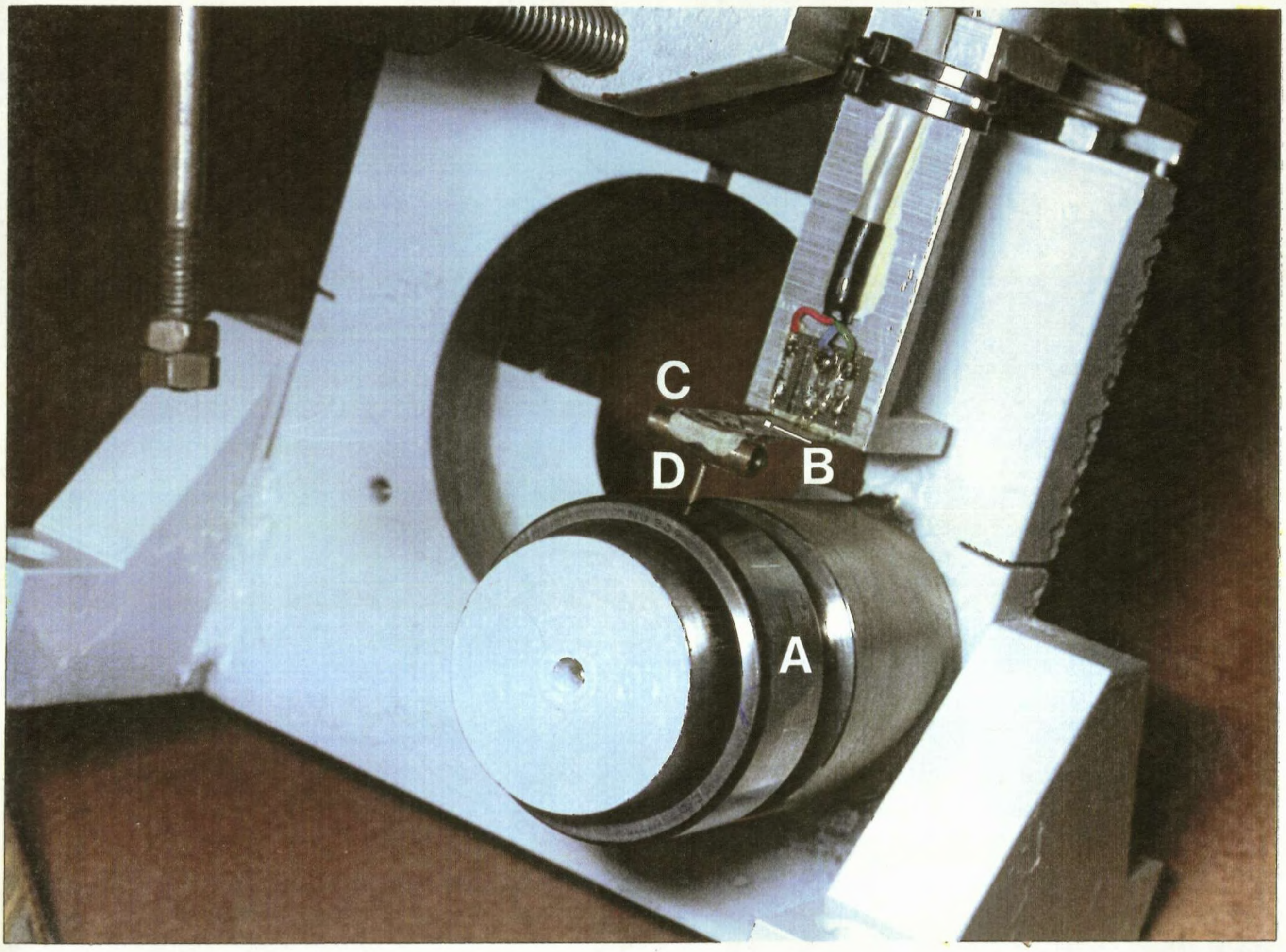


PLATE 3.2 Side view of strain gauged cantilevers fitted to bearing housing.



A	Bearing inner race	C	Cantilever
B	Strain gauge	D	Stylus

PLATE 3.3 Strain gauged cantilever and stylus positioned for flute depth measurement.

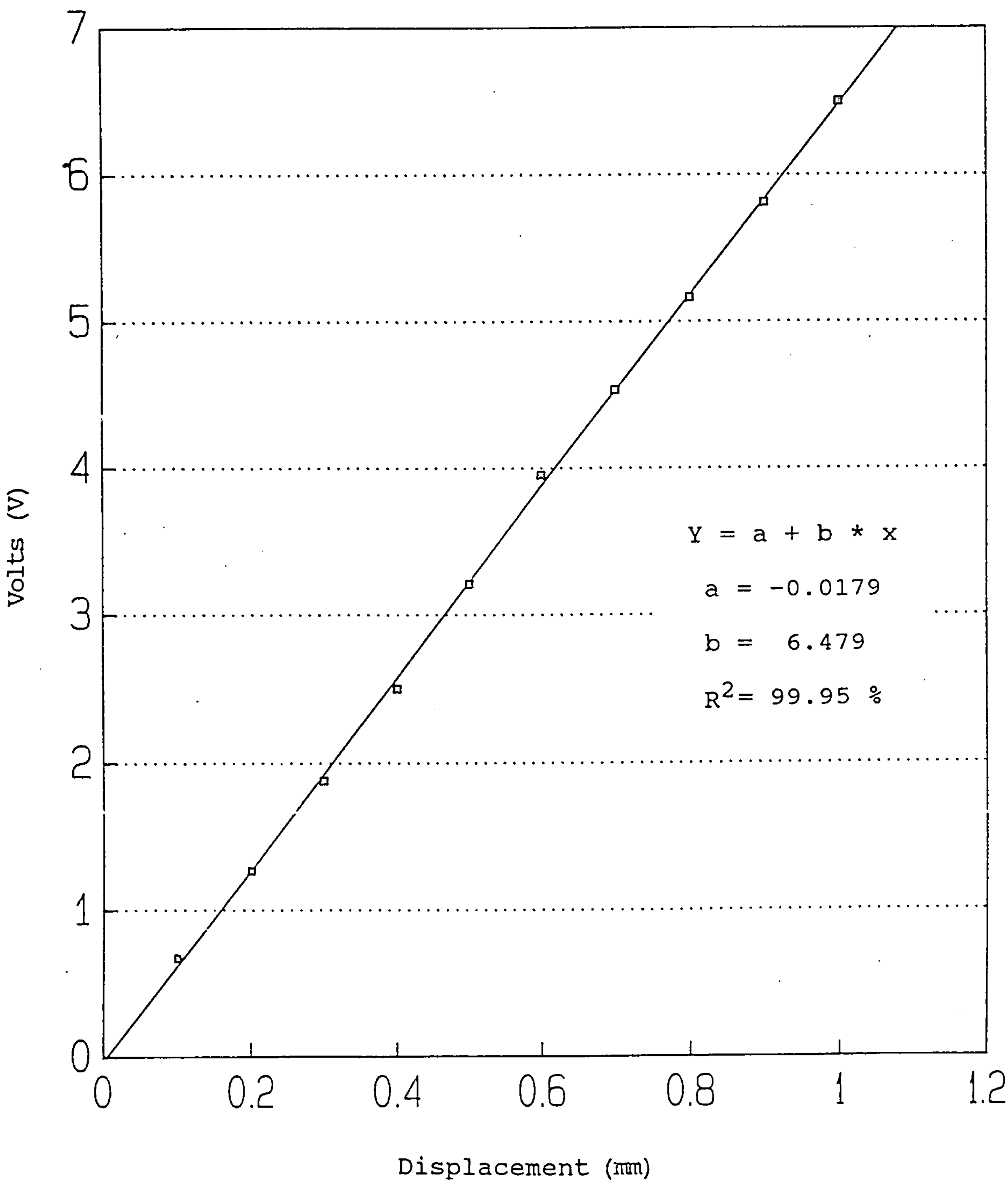


Figure 3.1 Calibration curve of cantilever for flute depth measurement.

Although the strain gauge method was partially successful, the method of calculating flute depth from the measured flute width proved to be more reliable.

The calculation was obtained from Pittroff's study (1964:4).

For flute depth D on inner ring:

$$D = \frac{b^2}{4} \times \frac{(A + B)}{(A \times B)}$$

where b = flute width (mm)

A = Inner race diameter (mm)

B = Roller element diameter (mm)

Du Randt (1991:12) later used the same formula to determine flute depth in his investigation. Both Pittroff and Du Randt verified their calculations using a Taylor-Hobson surface measuring instrument to obtain a physical measurement of flute depth.

3.5.6 Bearing Clearance and its Effect on False Brinelling

Bearing clearance has a direct influence on the severity of false brinelling (Pittroff 1964:7). An accurate method of ascertaining this parameter was therefore required. In view of the small tolerances allowed, these being 30 μm to 45 μm for normal clearance, it was decided that strain gauged cantilevers would provide the necessary accuracy. Two cantilevers were manufactured and fixed to a clamp which could then simply be attached to the bearing housing (Plate 3.2). Strain gauges were applied to both sides of the cantilever and connected in a half-bridge configuration. Amplifiers were used to supply a voltage reading. The system was calibrated using a micrometer to relate displacement to voltage output. The calibration curve is presented in Figure 3.2. With the cantilever clamped in position over the bearing housing, the shaft was lifted upwards using elastic ropes attached to each end. The resulting change in voltage was measured and the bearing clearance calculated.

3.5.7 Measurement of Vibration Amplitude and Bearing Impedance

Vibration levels were measured using B & K piezoelectric transducers type 4370, B & K amplifiers type 2635A and a Hewlett Packard dual channel dynamic signal analyser type 35660A. Calibration of this system was achieved using a Rion VE10 calibrator. The signals were recorded on magnetic tape using a TEAC 7 channel tape recorder. The same accelerometers, analyser and amplifiers were used in conjunction with a PCB 086B20 impact hammer and type 480 amplifier to obtain bearing impedance. Figure 3.3 shows a schematic layout of the instrumentation.

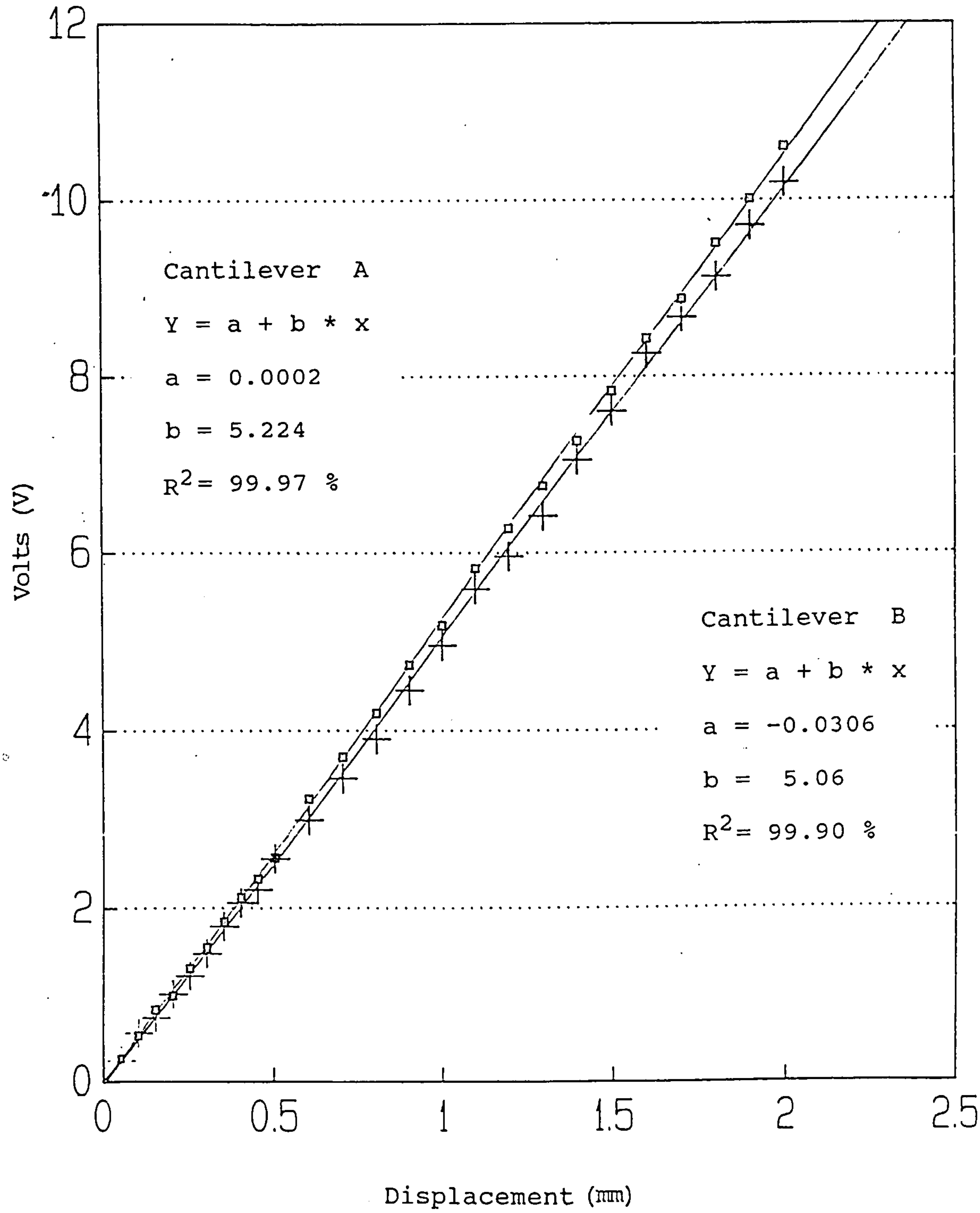


Figure 3.2 Calibration curves of cantilevers for radial bearing clearance measurement.

3.5.8 Test Rig Development

For the test rig to be acceptable, it was decided that the damage induced on the bearing race under identical test conditions should be repeatable to within 10 %. Due to the complex mechanisms involved in false brinelling, a number of changes to the rig configuration were required before this criterion was met. The final rig configuration is shown in Plate 3.4 to Plate 3.6 together with test equipment.

The test rig was initially designed to have a centrally mounted rotary shaker and six bearing housings located around the shaker as shown in Figure 3.4. Masses were fitted equally onto both sides of the shafts which were unrestrained. This configuration gave rise to a number of undesired forces and relative motion due to shaft rocking.

In order to eliminate the rocking mode, the masses were repositioned between two bearing housings as shown in Figure 3.5. This setup proved unsuccessful due to alignment complications. Problems were also experienced with the load application, as this required a split shaft which compounded the alignment problems.

Figure 3.6 shows a simply supported shaft with the mass positioned to one side. This configuration was intended to alleviate the alignment problems as well as the shaft rotation and rocking. The shaft rotation and rocking was significantly reduced, however alignment remained difficult and time consuming.

The bearing housing was finally moved to the centre of the plate, and the load applied through a swing arm mechanism as shown in Figure 3.7. This configuration was found to give repeatable results, required no time consuming alignment and effectively eliminated any shaft rotation or rocking.

Impedance could be adjusted in the axial and horizontal directions by raising or lowering the steel base on the threaded bars. No adjustment was available for impedance in the vertical direction.

As Pittroff (1964:5) indicated that changing elastic deformation is a prerequisite for the occurrence of false brinelling, it was expected that damage would occur at the points of contact of the loaded elements. This however was not the case. The loaded elements occupy an arc of approximately 120° positioned centrally in the lower half of the bearing. The unloaded elements are found in the remaining arc which extends across the upper half of the bearing. It was in this unloaded region in the upper bearing half that damage was consistently observed. This discrepancy was of some concern and it was felt that the matter should be further investigated before proceeding with the planned investigation.

In order to ensure that a positive contact between the elements and the race was beyond doubt, all the rolling elements were removed, except for one, which was positioned directly below the vertical centre of the shaft. A series of six tests were carried out with velocity amplitudes equating to between 1 % and 20 % of the static load rating of the bearing. No physical relative motion was observed during the tests. The only relative motion which should have been present would have arisen from the elastic deformation at the point of contact between the rolling element and race. Pittroff (1964:4) gives 0.03 % of the bearing static load as the lower limit required for initiating false brinelling. These conditions therefore fulfil his criteria for the onset of false brinelling, however still no damage was evident on the race or on the roller.

It may be concluded that either the element was not actually experiencing a cyclic loading, or that elastic deformation is not a prerequisite for the occurrence of false brinelling.

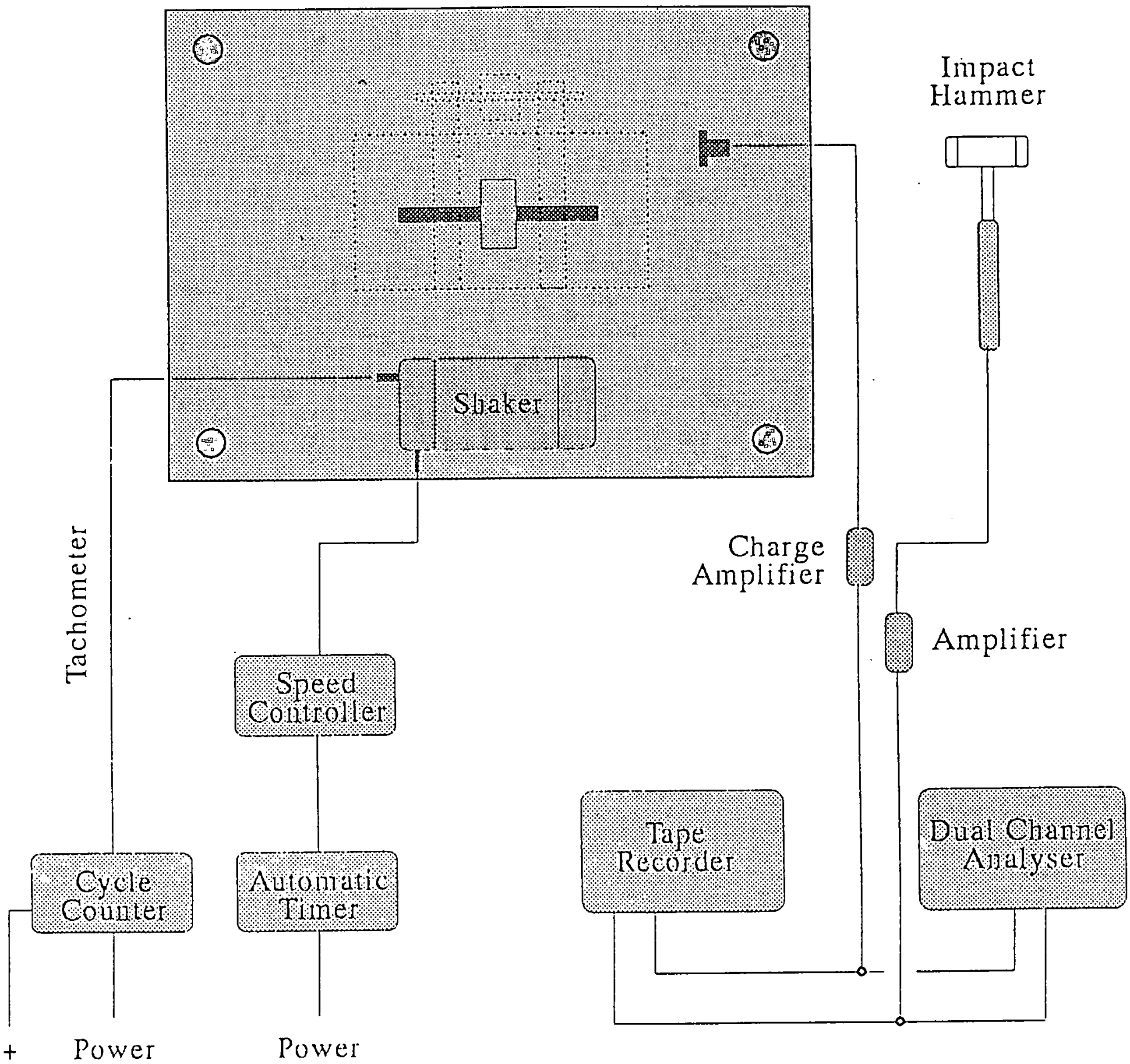
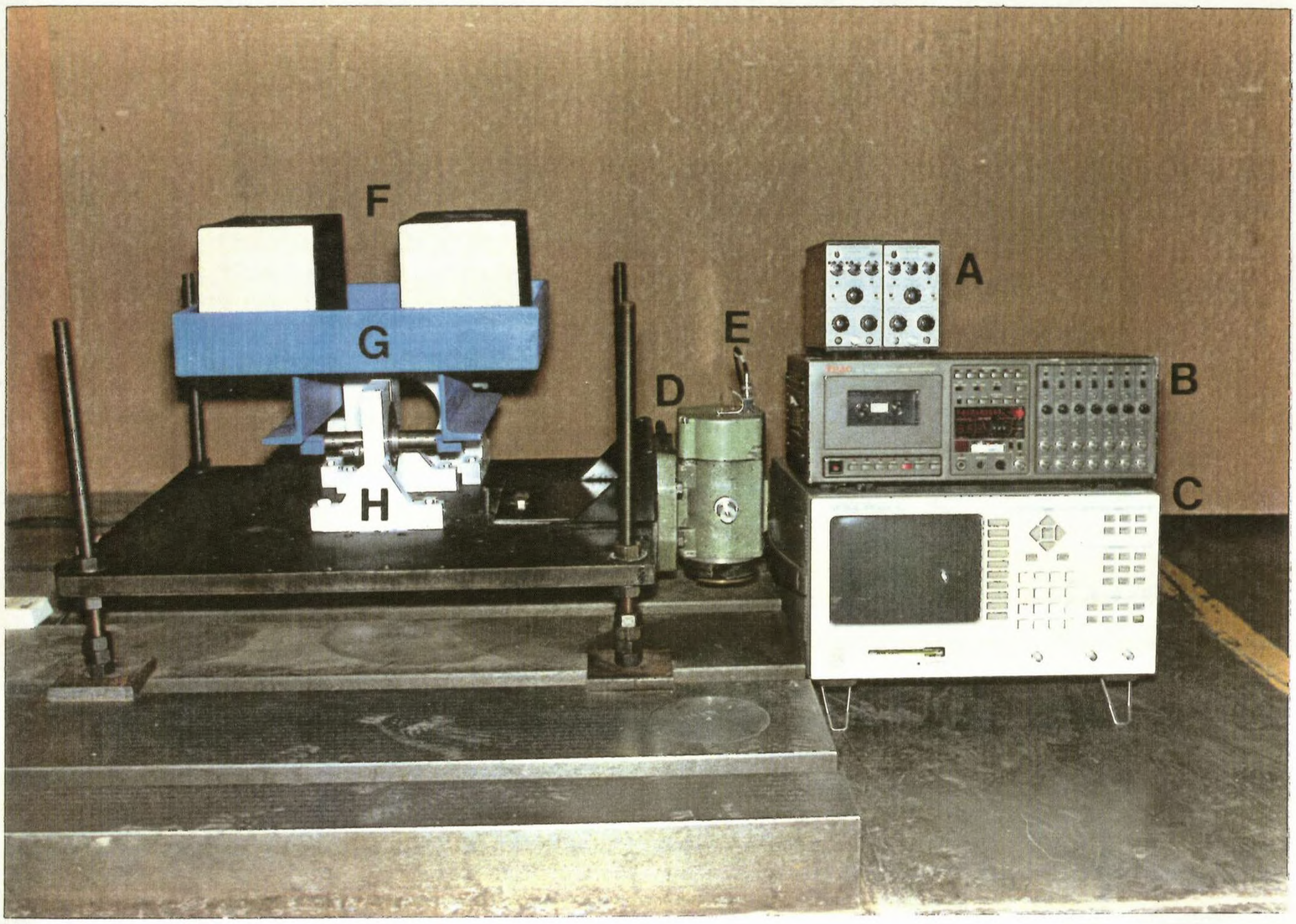


Figure 3.3 Schematic layout of test instrumentation for false brinelling investigation.



A	Charge amplifier	E	Tachometer
B	Tape recorder	F	Mass
C	Analyser	G	Swing arm
D	Rotary shaker	H	Bearing housing

Plate 3.4 Test rig and equipment utilized for measurement and data capture during false brinelling investigation.

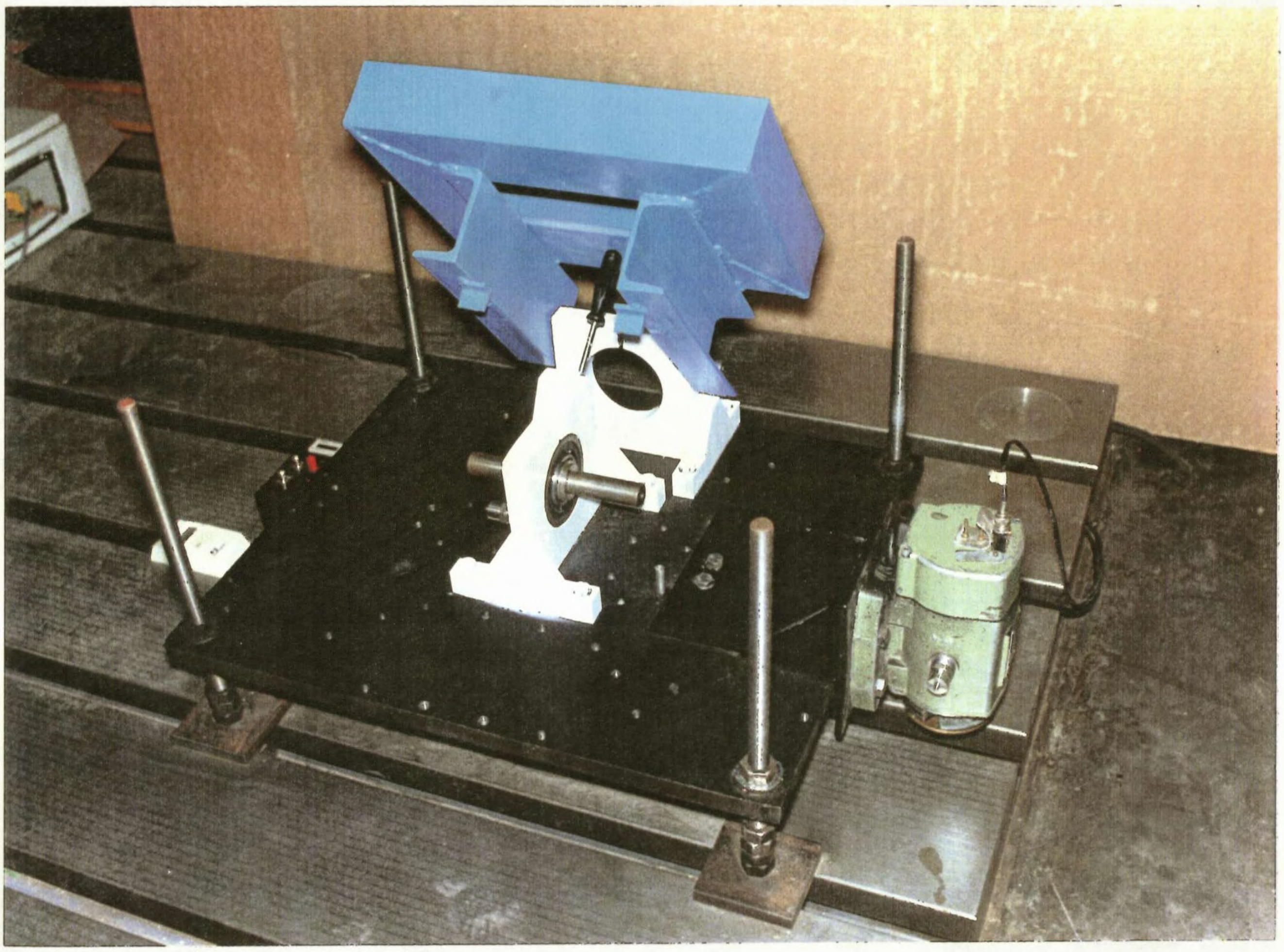


Plate 3.5 Elevated view of swing arm system used to apply static load to bearings under test conditions.

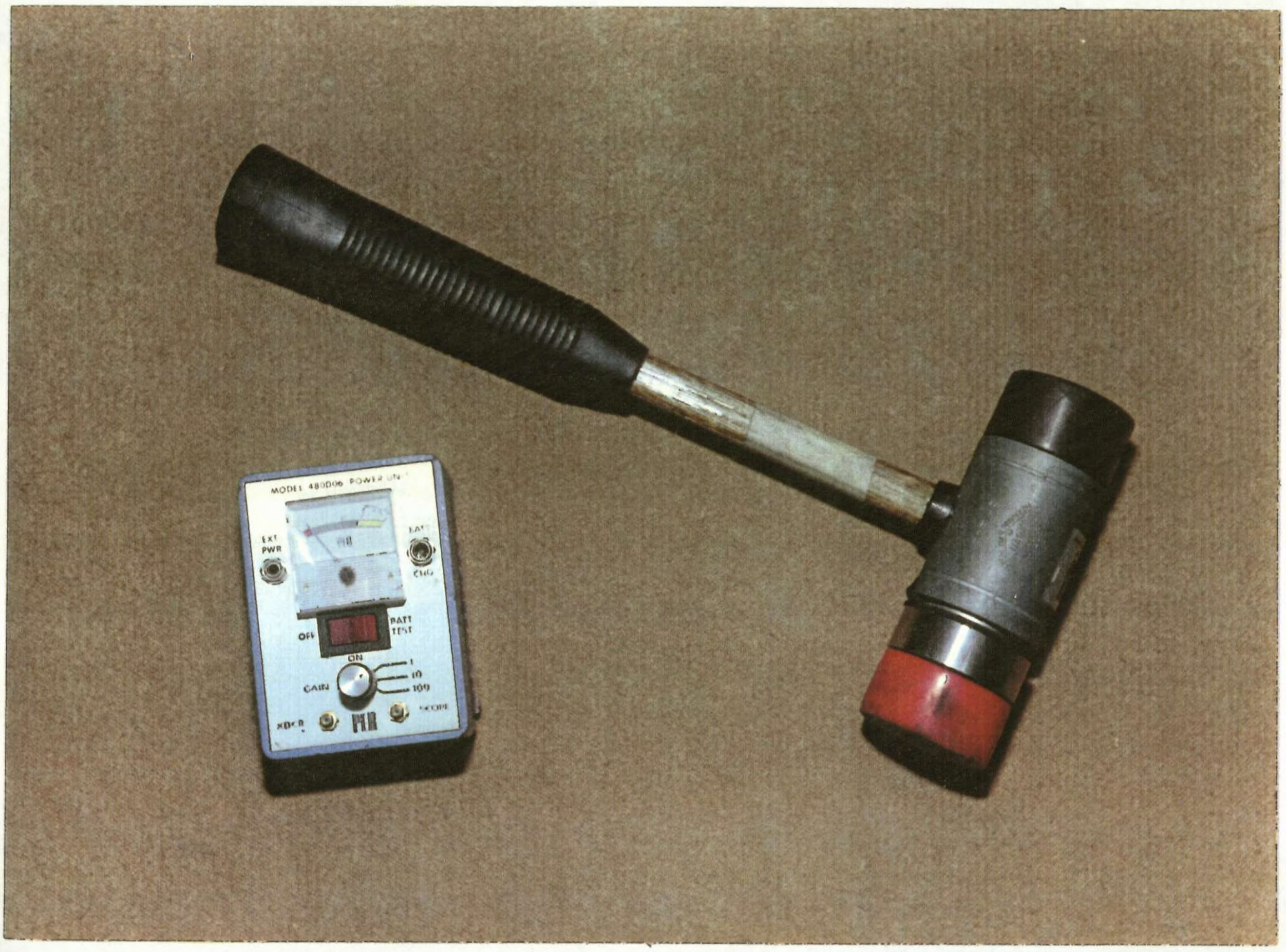


Plate 3.6 Force hammer and amplifier for assessing system impedance.

Figure 3.4

Initial test rig configuration.

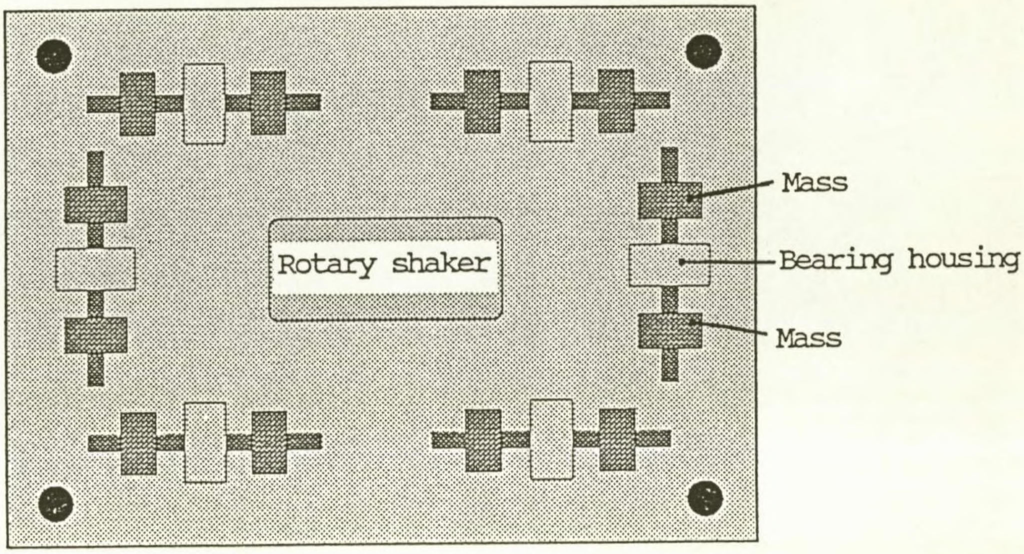


Figure 3.5

First modification to test rig configuration.

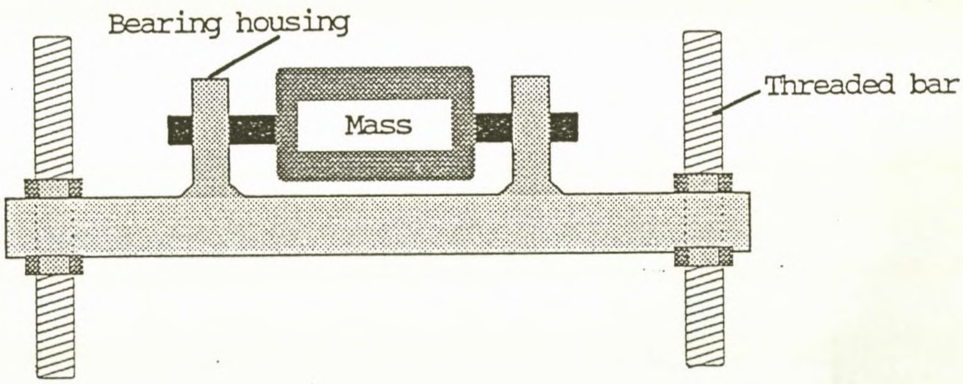
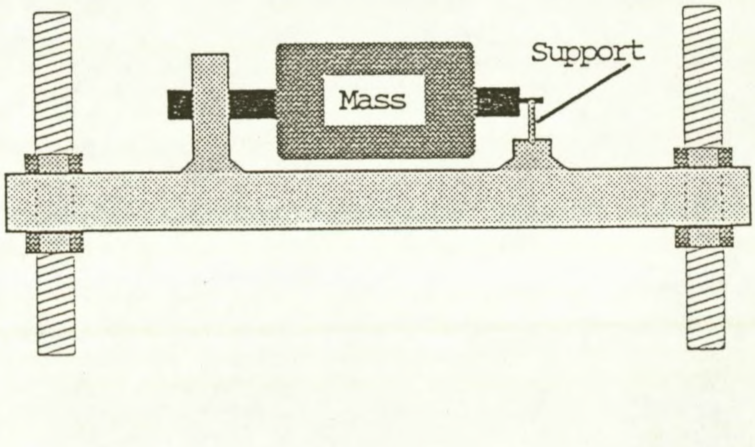
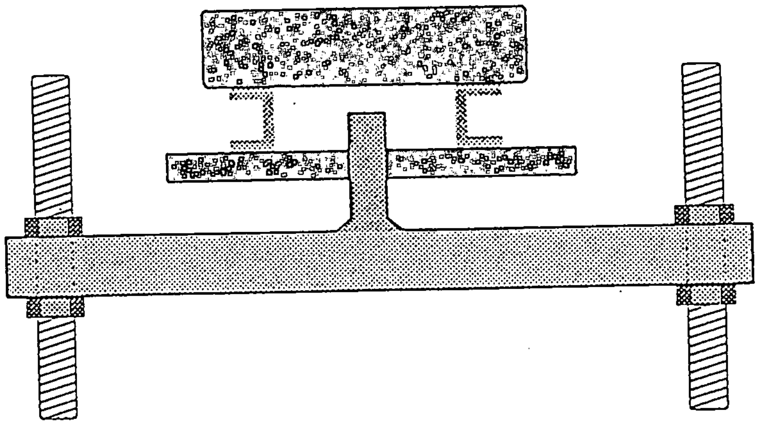


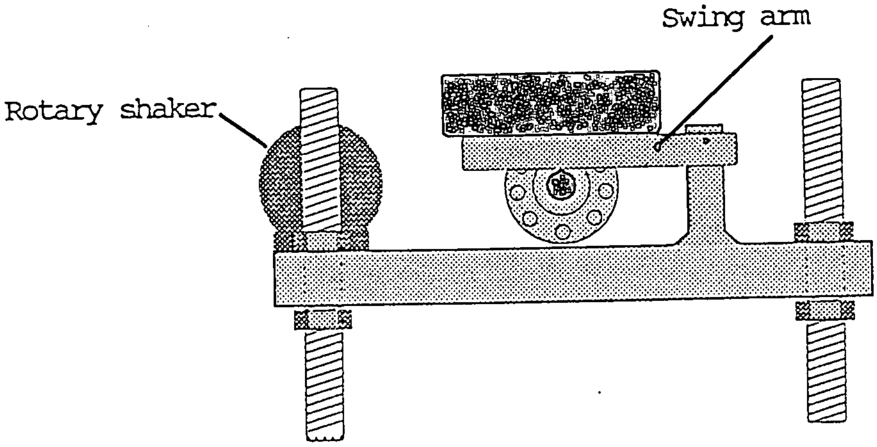
Figure 3.6

Second modification to test rig configuration.





Front view



Side view

Figure 3.7 Final test rig configuration

CHAPTER FOUR

EXPERIMENTAL RESULTS

4.1 INTRODUCTION

Tests were carried out in order to establish the effects of bearing impedance, static load, vibration amplitude and direction on false brinelling in horizontally mounted bearings. All relevant data from these tests are presented in Table 4.1 and Table 4.2. An explanation of the data appearing in these tables has been included below, using the first row of values in Table 4.1 as an example.

These data show that bearing number 67 experienced 0.36 million cycles of vibration from the rotary shaker which was mounted in the vertical/axial direction. The static bearing mass was 56.5 kg and the vibration amplitudes were 8.9 mm/s in the vertical direction, 0.97 mm/s in the horizontal direction and 13.5 mm/s in the axial direction. Vibration amplitudes were measured on the bearing housing. The depth of the largest flute that was formed during the test was 0.5 μm. Bearing impedance, measured in three directions on the bearing housing, was 114.943 N.s/mm in the vertical direction, 58.48 N.s/mm in the horizontal direction and 81.3 N.s/mm in the axial direction. The parameter indicated on the left of the table, in this case cycles, is the parameter which was varied during the set of eight tests. All other parameters were held as steady as possible.

Table 4.1 Results of impedance, load and cycle tests from false brinelling investigation.

	BEARING NO	Cycles (million)	EXCI-TATION	Bearing Mass (kg)	Amplitude (mm/s) RMS on housing			Depth of flute (micro meter)	Impedance (N.s/mm)		
					V	H	A		V	H	A
CYCLES	67	0.36	V.A.	56.50	8.90	0.97	13.50	0.50	114.94	58.48	81.30
	65	1.04	V.A.	56.50	9.25	0.63	13.50	1.10	114.94	58.48	81.30
	68	0.72	V.A.	56.50	9.20	1.43	13.30	1.20	114.94	70.00	81.30
	64	2.10	V.A.	56.50	9.30	0.82	13.70	1.60	114.94	58.48	81.30
	69	3.00	V.A.	56.50	9.52	0.92	13.80	1.83	114.94	58.48	81.30
	71	3.50	V.A.	56.50	9.20	1.03	13.50	3.10	114.94	58.48	81.30
	72	5.16	V.A.	56.50	9.30	0.94	13.40	2.80	114.94	58.48	81.30
	66	15.83	V.A.	56.50	9.03	1.10	13.60	4.20	114.94	58.48	81.30
LOAD	61	4.20	V.H.	11.00	8.80	10.30	0.86	1.50	114.90	58.80	81.30
	73	4.20	V.H.	56.60	8.80	11.60	0.91	1.50	117.60	63.77	78.74
	88	4.20	V.H.	78.00	8.80	11.10	0.86	1.30	113.60	58.80	76.90
	89	4.20	V.H.	88.00	8.80	11.80	0.39	1.88	113.60	58.80	76.00
	55	4.20	V.H.	101.00	8.80	13.60	0.50	1.41	149.25	61.70	77.50
	59	4.20	V.H.	124.00	8.70	13.00	0.39	1.52	116.27	56.18	73.50
	56	4.20	V.H.	147.50	8.80	14.30	1.00	2.50	112.35	58.48	83.00
	58	4.20	V.H.	170.00	8.70	13.00	0.40	2.20	114.94	58.13	81.30
IMPEDANCE	96	4.20	H.A.	56.50	0.76	10.60	7.60	2.10	104.00	19.30	27.70
	95	4.20	H.A.	56.50	0.85	9.30	8.50	2.50	101.00	33.60	38.21
	93	4.20	H.A.	56.50	0.68	8.80	8.30	2.40	101.00	62.11	69.44
	90	4.20	H.A.	56.50	0.70	8.40	8.10	2.10	114.90	101.00	102.00
	91	4.20	H.A.	56.50	0.70	9.10	8.60	1.00	114.90	101.00	102.00
	98	4.20	H.A.	56.50	0.37	9.20	8.20	2.50	121.95	133.30	149.25
	99	4.20	H.A.	56.50	0.30	9.70	8.40	1.50	121.95	133.30	149.25
	100	4.20	H.A.	56.50	0.40	8.70	8.50	1.00	121.90	86.00	97.00

V = Vertical
H = Horizontal
A = Axial

Table 4.2 Results of directional velocity amplitude tests from false brinelling investigation.

Group	BEARING NO	Cycles (million)	EXCI-TATION	Bearing Mass (kg)	Amplitude (mm/s) RMS on housing			Depth of flute (micro meter)	Impedance (N.s/mm)		
					V	H	A		V	H	A
VELOCITY V.H.	40	4.10	V.H.	56.50	3.65	6.78	0.13	0.81	114.94	58.48	81.30
VELOCITY V.H.	54	4.20	V.H.	56.50	4.31	7.42	0.30	0.83	114.94	58.48	81.30
VELOCITY V.H.	85	4.20	V.H.	56.50	5.90	8.30	0.35	1.50	114.50	70.00	81.30
VELOCITY V.H.	42	4.20	V.H.	56.50	6.00	10.40	0.21	2.06	114.94	58.48	81.30
VELOCITY V.H.	38	4.20	V.H.	56.50	8.80	13.10	0.52	2.32	114.94	58.48	81.30
VELOCITY V.H.	45	4.20	V.H.	56.50	11.70	18.30	1.05	3.85	114.94	58.48	81.30
VELOCITY V.H.	43	4.20	V.H.	56.50	12.50	20.30	0.86	4.80	114.94	58.48	81.30
VELOCITY V.H.	39	4.20	V.H.	56.50	14.50	22.40	1.90	4.62	114.94	58.48	81.30
VELOCITY V.A.	51	4.20	V.A.	56.50	2.84	0.28	4.27	0.41	114.94	58.48	81.30
VELOCITY V.A.	50	4.20	V.A.	56.50	3.43	0.35	5.17	0.41	114.94	58.48	81.30
VELOCITY V.A.	49	4.20	V.A.	56.50	4.68	1.22	7.24	1.09	114.94	58.48	81.30
VELOCITY V.A.	48	4.20	V.A.	56.50	5.50	0.89	8.10	1.64	114.94	58.48	81.30
VELOCITY V.A.	47	4.20	V.A.	56.50	6.38	1.00	9.47	3.67	114.94	58.48	81.30
VELOCITY V.A.	86	4.20	V.A.	56.50	7.70	0.37	12.40	4.00	114.94	58.48	81.30
VELOCITY V.A.	53	4.20	V.A.	56.50	9.95	1.43	14.80	4.70	114.94	58.48	81.30
VELOCITY V.A.	52	4.20	V.A.	56.50	11.20	1.26	16.40	5.20	114.94	58.48	81.30
VELOCITY H.A.	82	4.20	H.A.	56.50	0.28	1.50	1.00	0.24	114.94	58.48	81.30
VELOCITY H.A.	81	4.20	H.A.	56.50	0.52	2.90	2.00	0.38	114.94	58.48	81.30
VELOCITY H.A.	80	4.20	H.A.	56.50	0.48	5.00	3.20	0.35	114.94	58.48	81.30
VELOCITY H.A.	79	4.20	H.A.	56.50	0.86	8.10	5.50	1.62	114.94	58.48	81.30
VELOCITY H.A.	78	4.20	H.A.	56.50	0.90	9.80	7.00	3.20	114.94	58.48	81.30
VELOCITY H.A.	75	4.20	H.A.	56.50	1.20	11.30	8.70	4.50	114.94	58.48	81.30
VELOCITY H.A.	76	4.20	H.A.	56.50	1.50	14.20	11.30	6.00	114.94	58.48	81.30
VELOCITY H.A.	77	4.20	H.A.	56.50	0.67	14.80	10.80	6.53	114.94	58.48	81.30

V = Vertical
H = Horizontal
A = Axial

Results of the individual tests are presented graphically in Figures 4.1 to 4.5. Stepwise regression and simultaneous equations were used in order to obtain the directional contribution of vibration, these calculations have been included in appendix B.

4.2 THE EFFECT OF STATIC LOAD ON FALSE BRINELLING

Static loads ranging from 11 kg to 124 kg resulted in only a marginal increase in the rate of bearing damage (Figure 4.1). Application of the 147.5 kg load was accompanied by a change in the ratio of vertical to horizontal impedance. This gave rise to increased horizontal and axial vibration levels and is likely to be the reason for the larger flute formed during this particular test. It was decided to include the results of the 147.5 kg and 170 kg load tests in spite of the effect these loads had on the impedance ratio. Information at these higher loads is required as roller bearings are not restricted to low load applications as in electrical motors but are often used to support larger items such as fan impellers. The results of these tests indicate that static bearing load does not make a significant contribution to the process of false brinelling in horizontally mounted bearings.

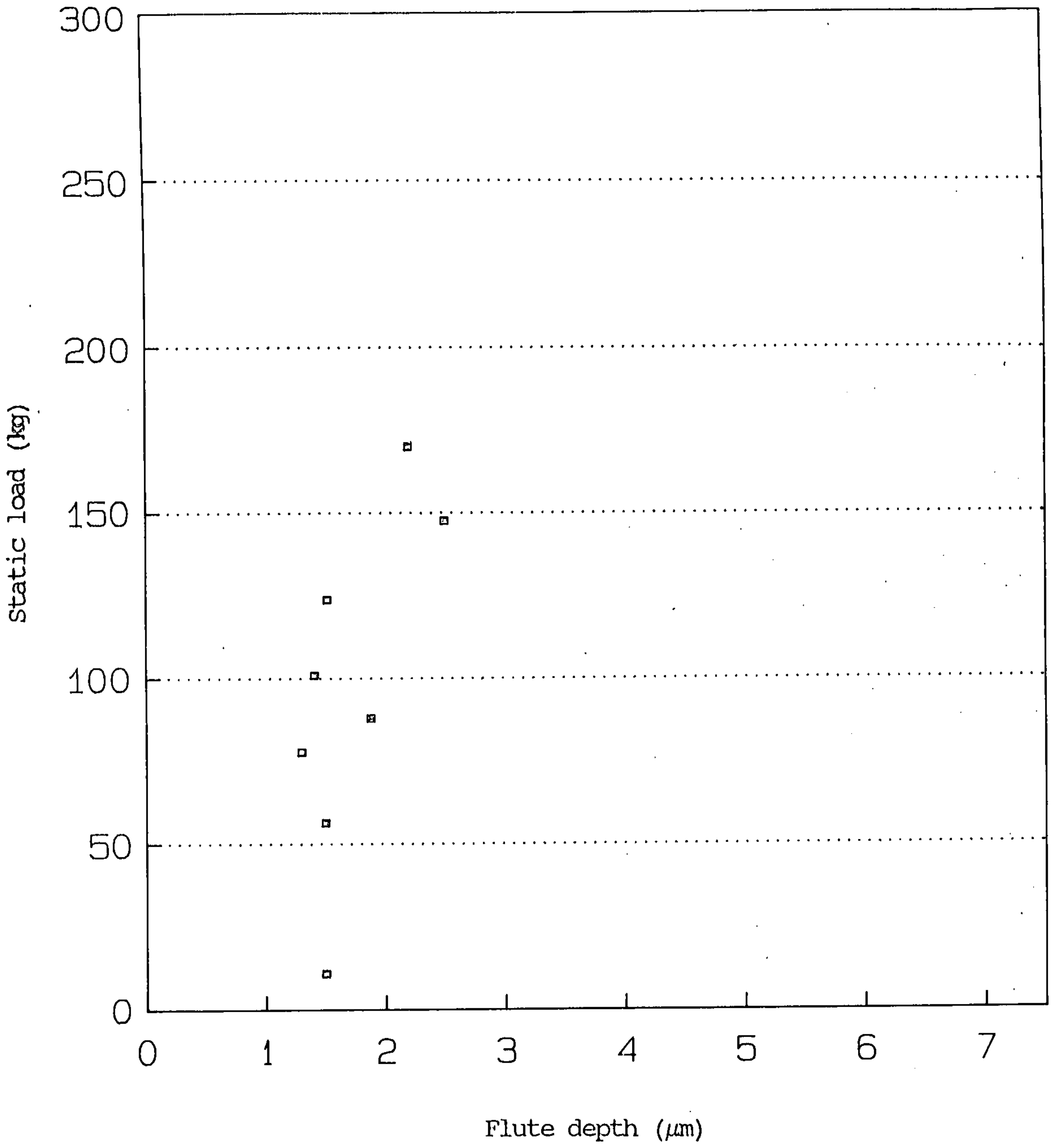


Figure 4.1 The effect of static load on false brinelling.

4.3 THE EFFECT OF IMPEDANCE ON FALSE BRINELLING

The results of the impedance tests are presented graphically in Figure 4.2. The scatter in data arose due to climatic changes which did not affect any of the other sets of tests. Previous researchers have made no reference to the use of controlled climates during testing, or to the effect of humidity levels on false brinelling. Humidity was therefore not considered as a factor that required to be controlled or monitored during the development of the test procedure.

Test results did not appear to be affected significantly by humidity levels less than approximately 50 %. As it was not possible to install a humidity chamber given the time schedule and budget, a 50 % humidity level was accepted as the permissible limit, above which test data were ignored. The scatter in results may be ascribed to the smaller influence of humidity below this 50 % cut off.

These results do not show any clear trend, indicating that bearing impedance does not have a significant influence on the rate of false brinelling in horizontally mounted bearings.

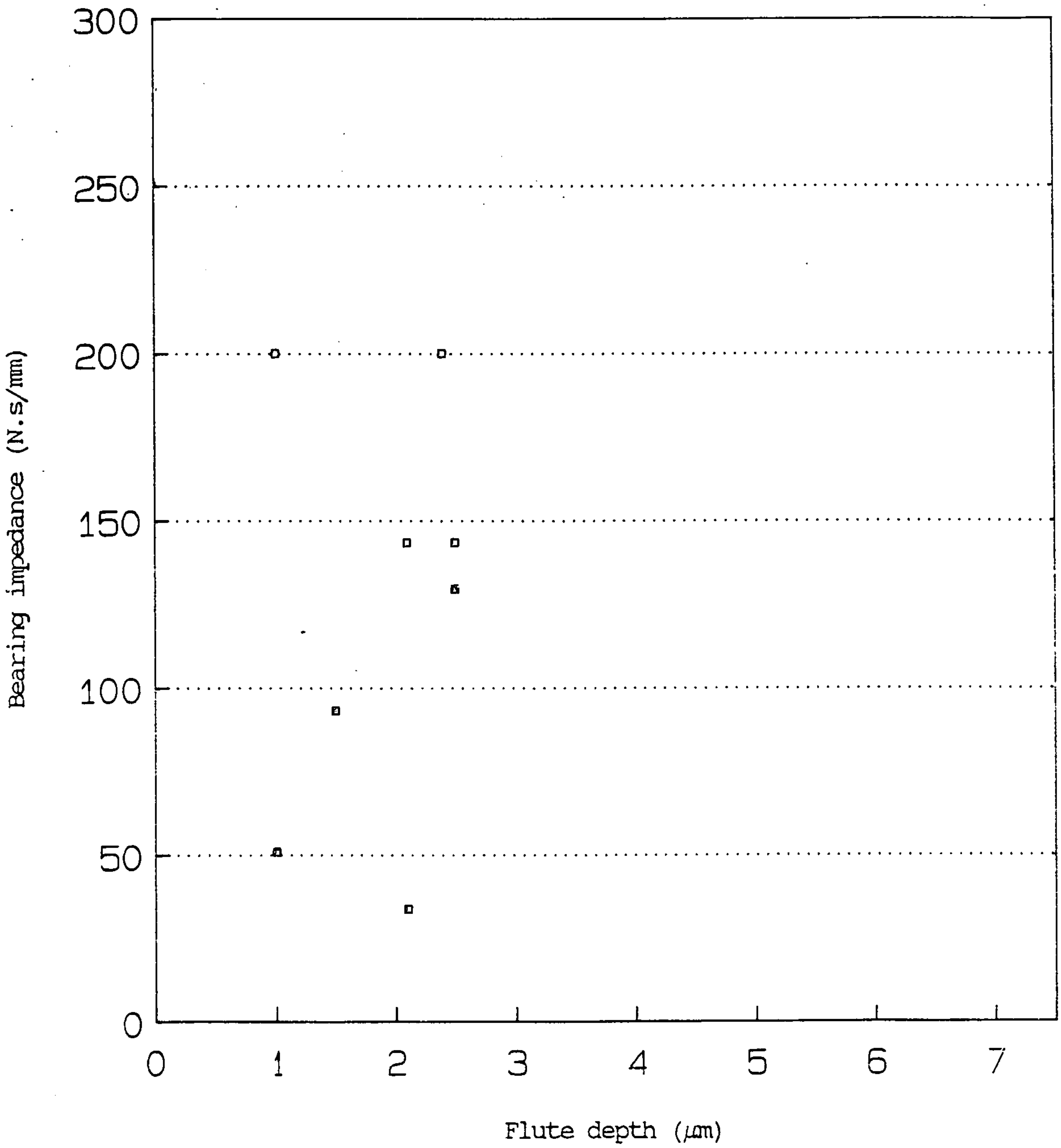


Figure 4.2 The effect of bearing impedance on false brinelling.

4.4 THE EFFECT OF VIBRATION AMPLITUDE AND DIRECTION ON FALSE BRINELLING

Vibration amplitude tests were carried out in three different directional combinations, these being vertical/horizontal, vertical/axial and horizontal/axial. This was done in order to determine any difference in directional contribution. The resultants of the two directional excitation were related to flute depth and the results are presented graphically in Figures 4.3 to 4.5.

An increase in the amplitude of vibration was accompanied by an increase in flute depth. The results also show the three excitational combinations to have different slopes and Y intercepts. This indicates that the direction of excitation does have an influence on the severity of false brinelling in horizontally mounted bearings.

4.5 THE EFFECT OF CYCLES ON BEARING DAMAGE

The rate of bearing damage decreases as the size of the respective flute increases (Figure 4.6). This is due to the curvatures of the roller and bearing race. As the depth of flute increases, so too does the contact area between roller and bearing race. The increasing contact area requires a greater volume of material to be removed for an equal increase in flute depth. The result will be a decrease in the rate of flute depth propagation, assuming all variables remain constant. This relationship between the number of cycles experienced by a bearing and flute depth agrees with the findings of Pittroff (1964:6).

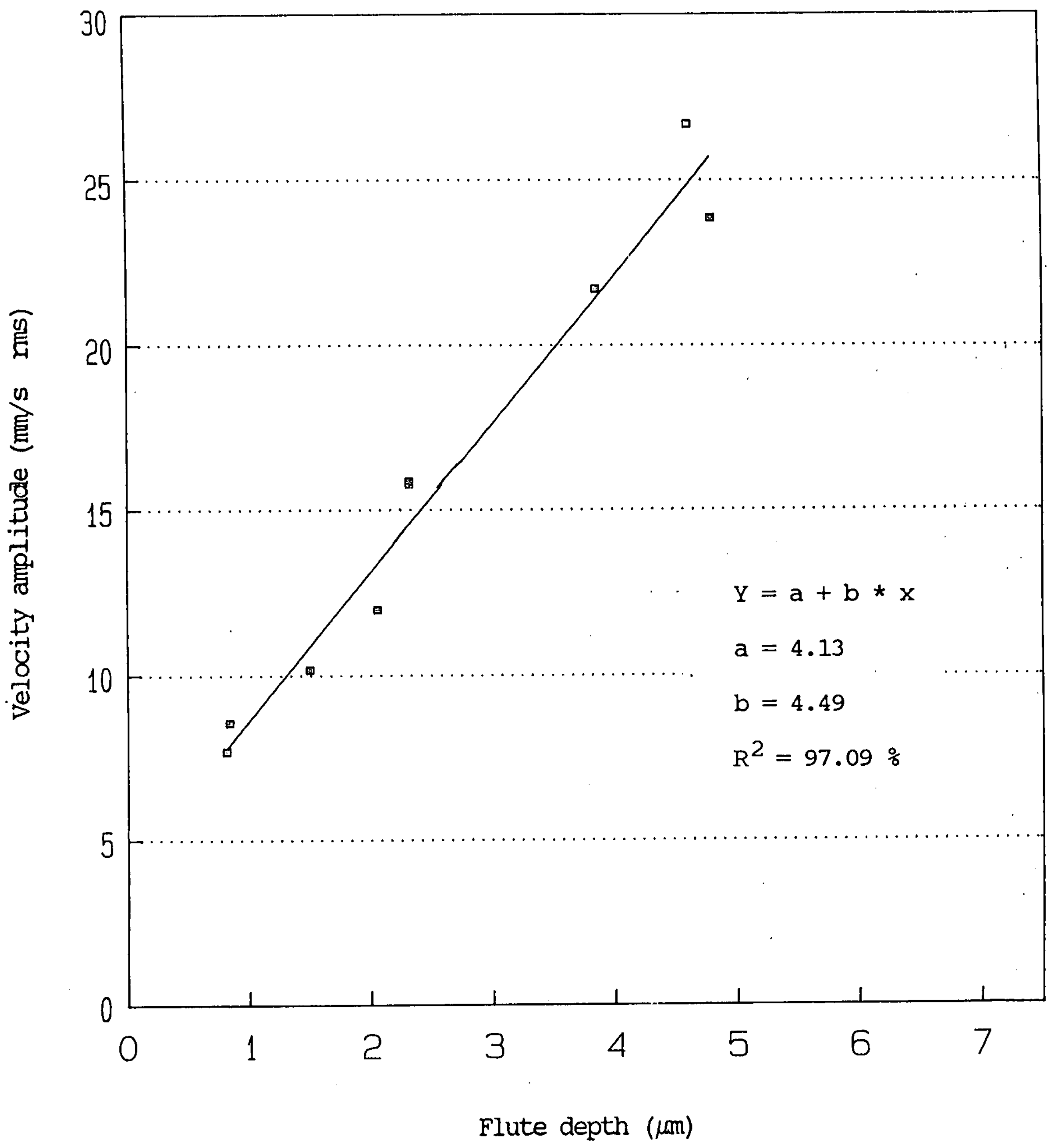


Figure 4.3 Vertical/horizontal vibration resultant versus bearing damage.

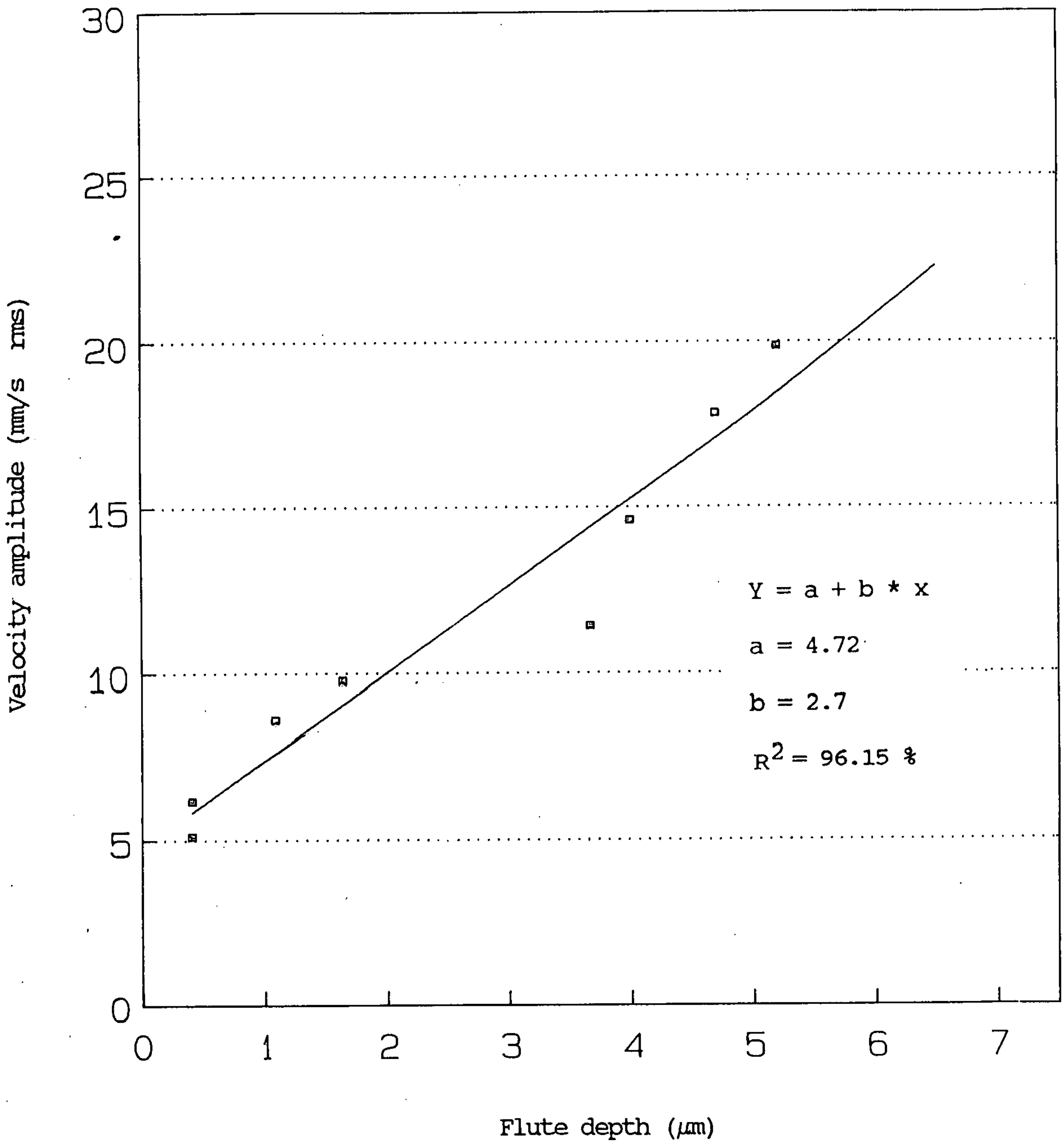


Figure 4.4 Vertical/axial vibration resultant versus bearing damage.

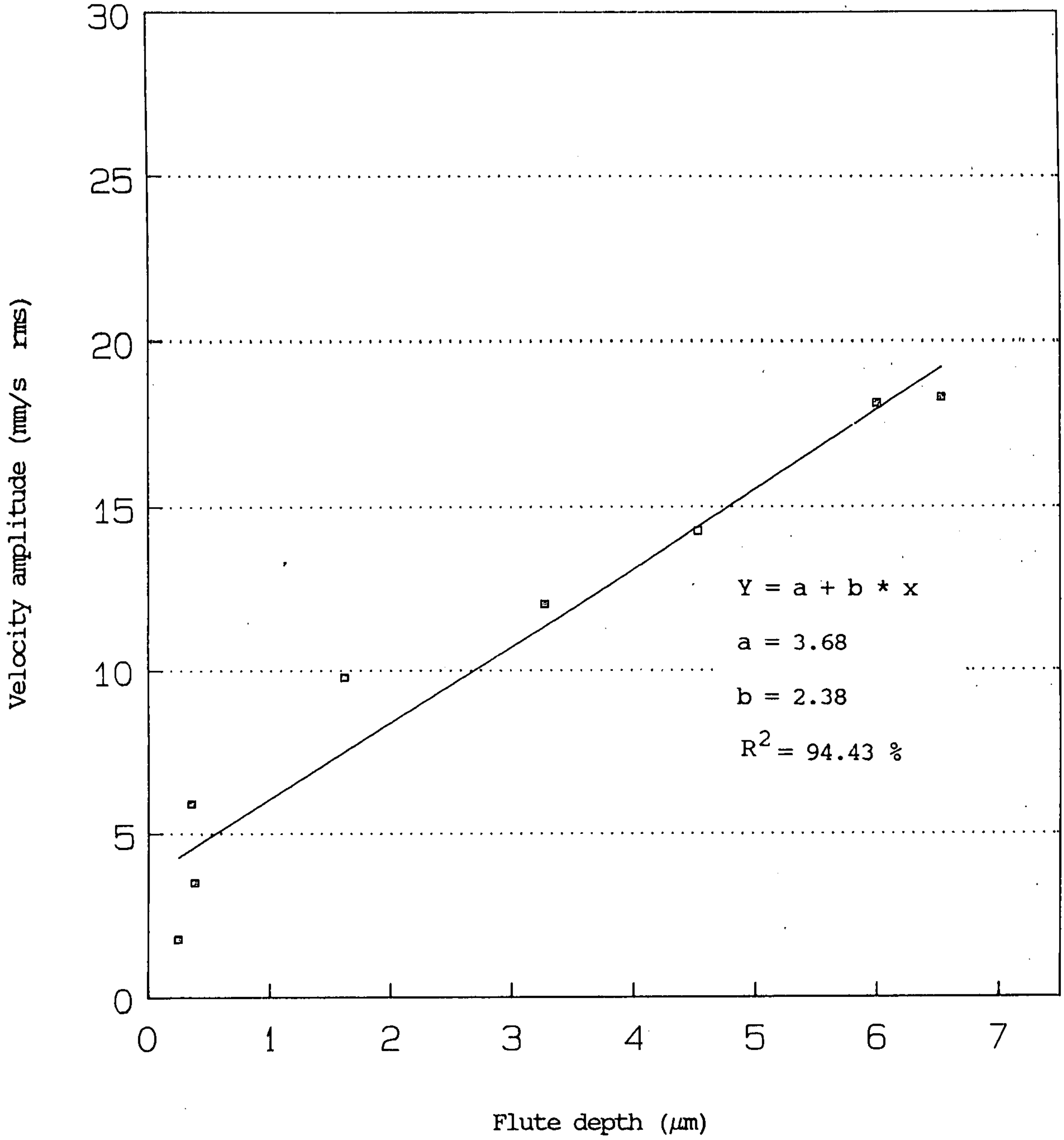


Figure 4.5 Vertical/axial vibration resultant versus bearing damage.

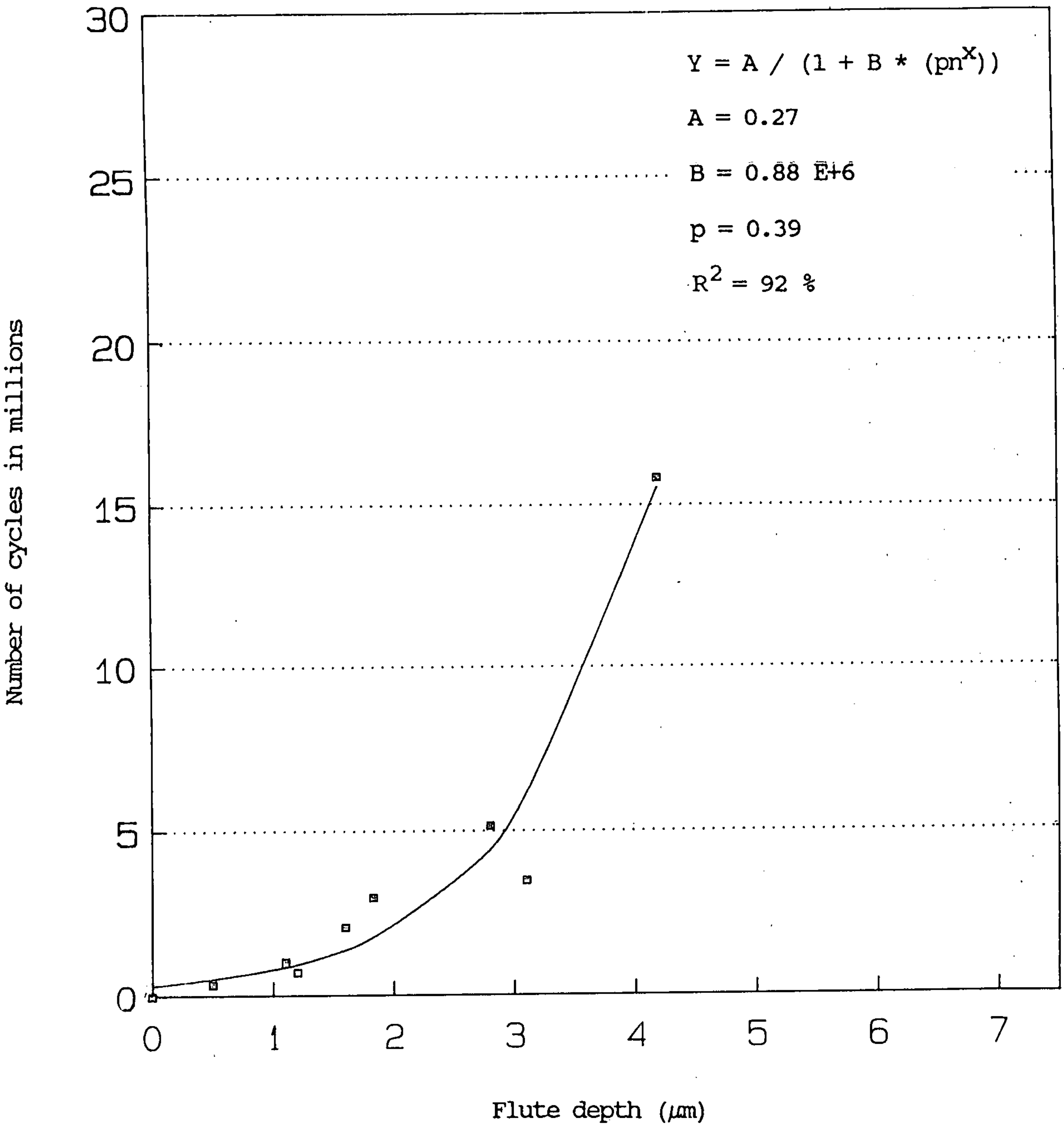


Figure 4.6 Relationship between flute depth and the number of cycles experienced by a bearing.

4.6 RELATIONSHIP BETWEEN VERTICAL, HORIZONTAL AND AXIAL VIBRATION

The relationship between vertical, horizontal and axial vibration was calculated by means of simultaneous equations and these results confirmed using stepwise regression (Appendix B). Stepwise regression calculated the coefficients to be as follows:

Vertical....0.08914

Horizontal..0.10744

Axial.......0.21817

R^2..........88.8 %

The corresponding ratio between vertical, horizontal and axial directions is 1 : 1.21 : 2.45. This compares favourably to the ratio of 1 : 1.65 : 2.4 obtained using simultaneous equations. The ratio obtained from the stepwise regression has been used in developing a method of establishing the presence and severity of false brinelling in horizontally mounted bearings. Figure 4.7 shows the weighted velocity line for an excitation of 50 Hz after 4.2 million cycles.

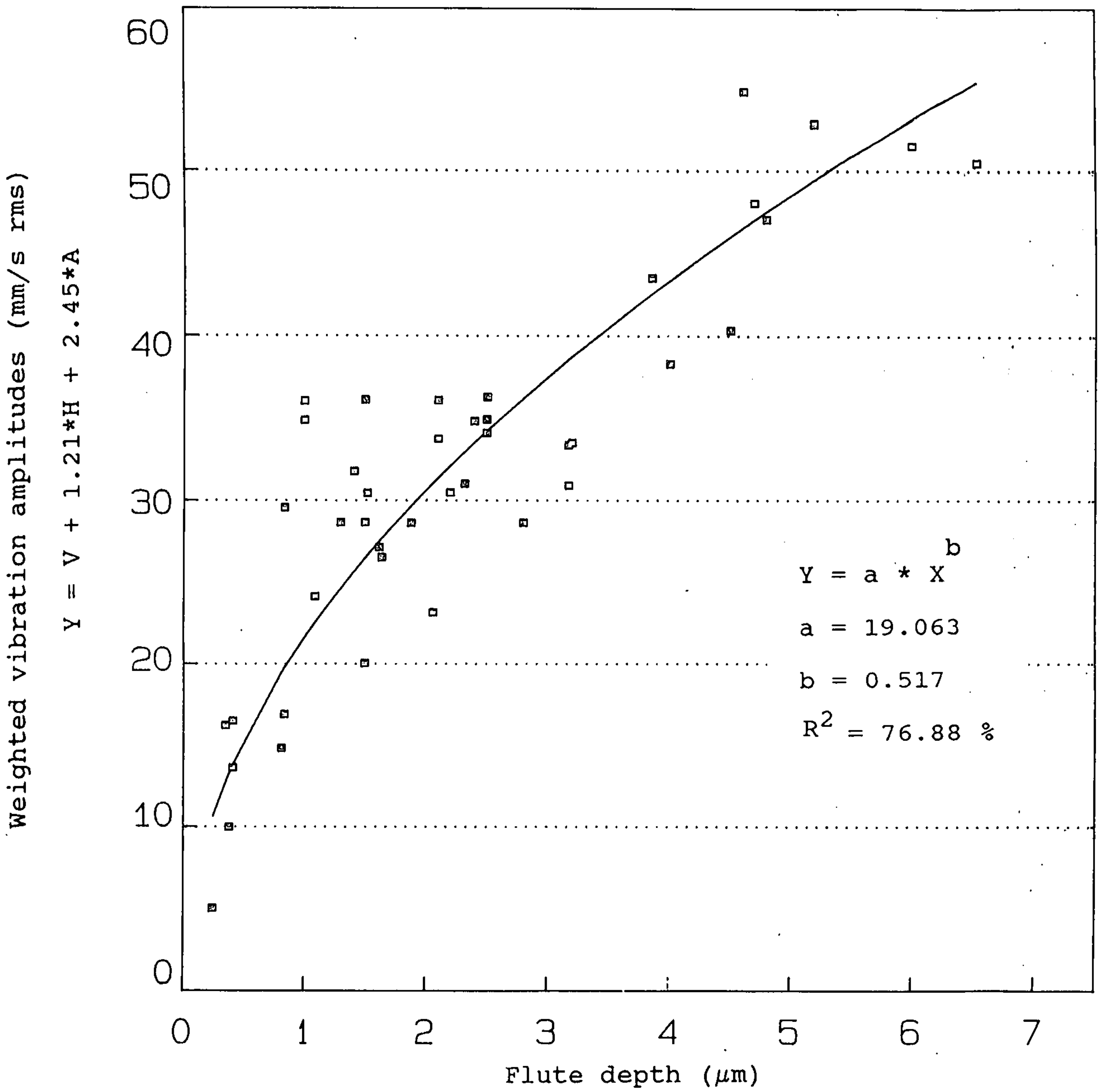


Figure 4.7 Directional vibration amplitude curve (D-VAC) relating weighted velocity amplitudes at 4.2 million cycles to bearing damage in unlubricated roller element bearings.

4.7 VALIDITY OF ASSUMPTION

It was assumed that as bearings are manufactured to international standards, the effect of any variation in bearing material hardness or composition from one bearing to another would be negligible. The high coefficients of determination obtained for the vibration amplitude, direction and cycle tests, indicate that this assumption was correct.

4.8 IDENTIFICATION OF FALSE BRINELLING DAMAGE

The scars or flutes formed as a result of false brinelling occur on the bearing race at the point of contact between roller element and race. Flutes may either be polished (Plate 4.1), or have a red/brown colouring from the oxidised wear particles (Plate 4.2).

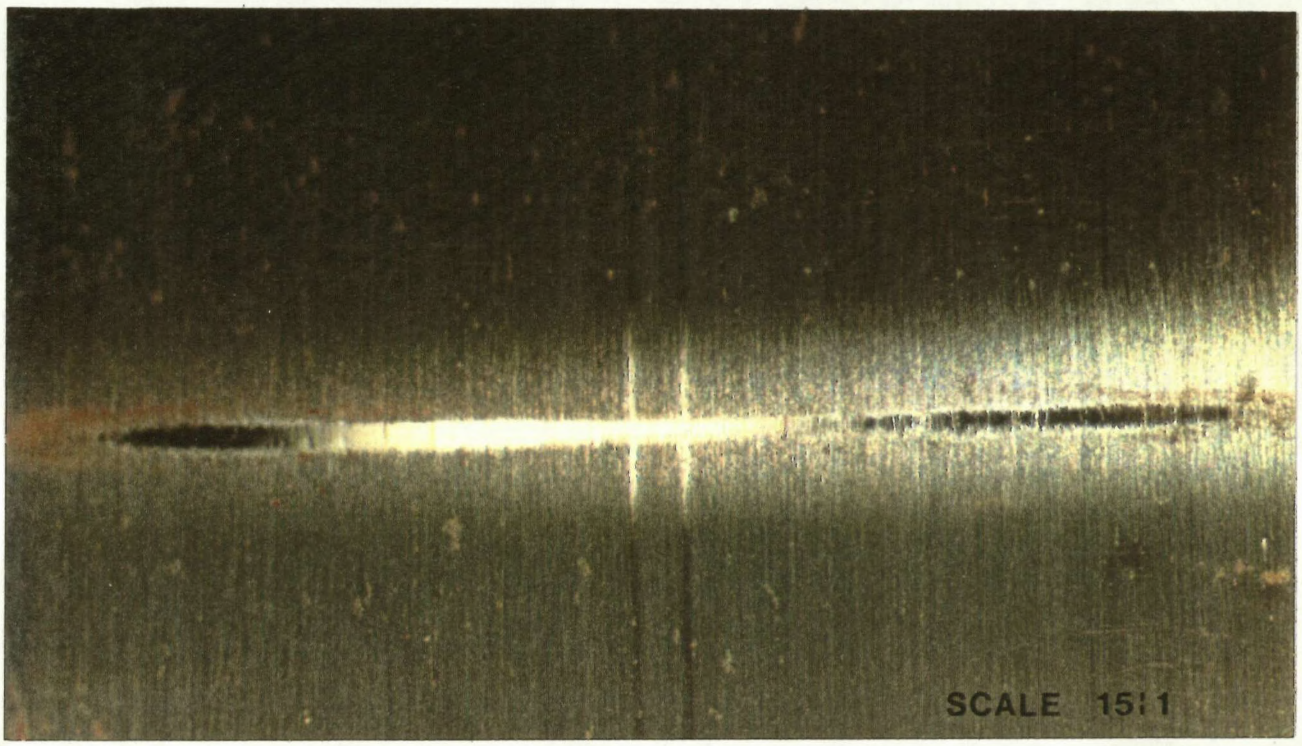


Plate 4.1 Flute on inner race of roller bearing showing no evidence of oxidised debris. (Mag. x 15).

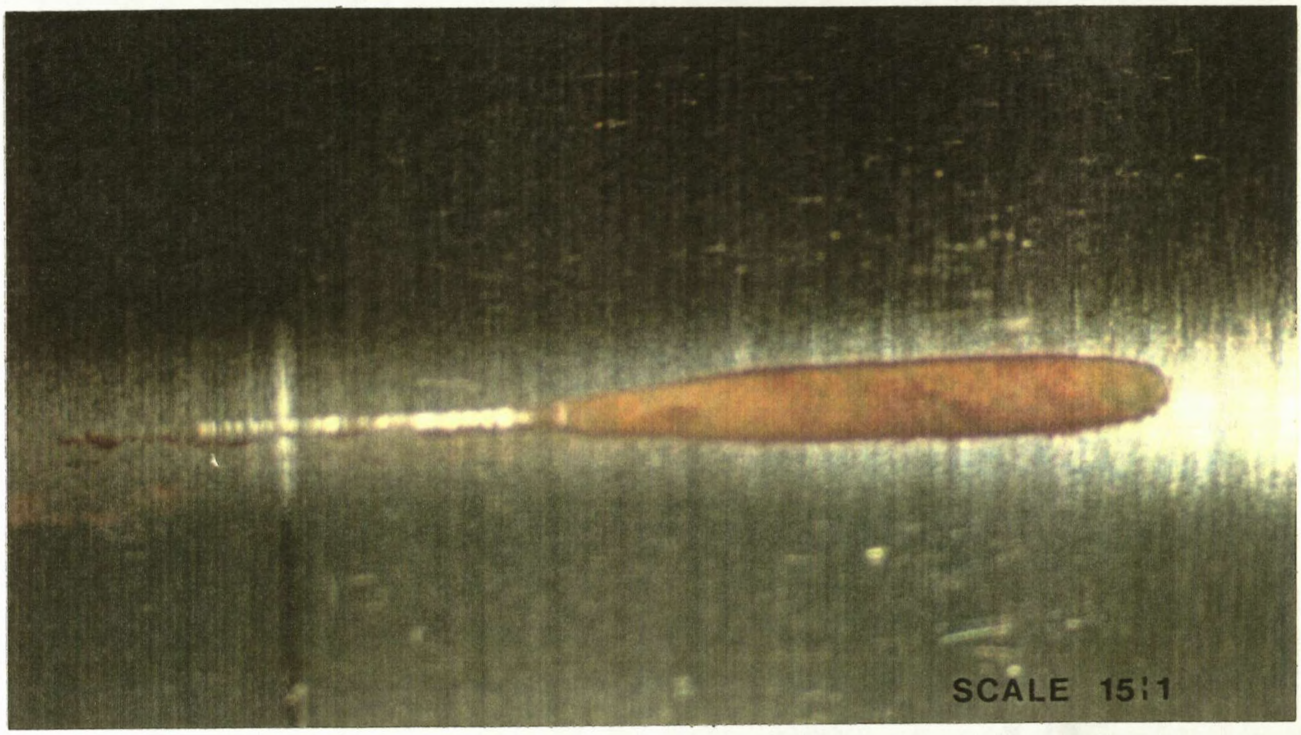


Plate 4.2 Oxidised debris deposits in flute of roller bearing.

(Mag. x 15)

CHAPTER FIVE

GENERAL DISCUSSION

5.1 INTRODUCTION

Of particular interest to Eskom, is the ability to establish the likelihood of false brinelling occurring on power station plant, and the relevant knowledge to implement appropriate preventative measures. Only limited information is available relating to factors affecting false brinelling, and no reference has been located relating specifically to horizontally mounted bearings. The test results presented in Chapter 4 have been discussed in the light of existing literature, with a view to establishing an appropriate procedure for assessing false brinelling on Eskom plant.

5.2 THE EFFECT OF HUMIDITY ON FALSE BRINELLING

A discrepancy was noted in the results of the impedance tests. Large variations in the extent of bearing damage did not relate in any way to the changing impedance level. As no change had been made to the other parameters under investigation, it followed that an additional factor was influencing the rate of bearing damage.

Most of the parameter testing was carried out in the winter months on the highveld, during which time humidity levels did not vary considerably, and tended to be in the region of 20 %. However, impedance testing only began in summer and was accompanied by almost daily rainstorms. Humidity levels could fluctuate between 15 % and 70 % in a single day. It was found that humidity levels in excess of approximately 50 % had a significant influence on flute depth, and that flutes could be as much as four times deeper than at humidity levels below 50 %. This value compares well with the 60 % critical humidity level for corrosion of ferrous alloys given by both Shreir (1965:18.4) and VGB Kraftwerke (1981:14). Previous researchers have not indicated that controlled climates were used during testing. There has also been no reference to the effect of humidity levels on false brinelling.

Eskom power stations and pump storage schemes operate in a wide variety of geographical locations including coastal areas. For this reason it is important that Eskom personnel be aware of the influence of humidity, and include this factor when considering preventative measures relating to false brinelling. Most importantly it should be ensured that bearing lubrication is free from any abnormal ingress of moisture.

5.3 DIRECTIONAL VIBRATION AMPLITUDE CURVE (D-VAC) FOR ASSESSING FALSE BRINELLING IN HORIZONTALLY MOUNTED BEARINGS

Static bearing load and impedance have been shown to have little or no effect on false brinelling in horizontally mounted bearings (Figures 4.1 to 4.2). Velocity amplitude, direction and cycles influence the rate of false brinelling directly, and have therefore been used to construct a model for establishing the extent of false brinelling in such bearings (Figure 5.1).

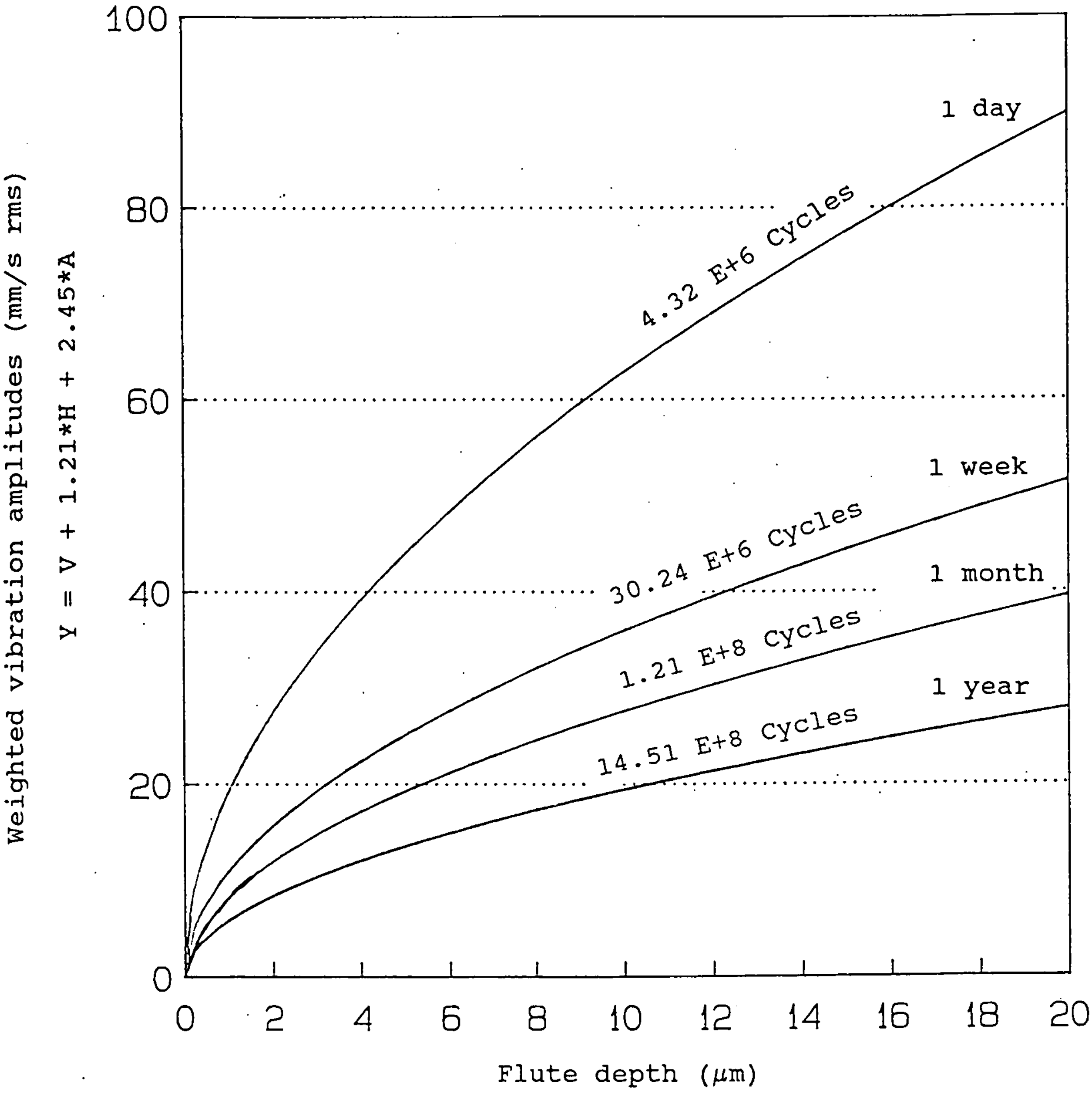


Figure 5.1 Directional vibration amplitude curves (D-VAC) calculated for predicting bearing damage in unlibricated horizontally mounted roller element bearings. The curves relate to the number of cycles after 1 day, 1 week, 1 month and 1 year at an excitation of 50 Hz.

The relative directional contributions enabled the effect of the sum of the directionally weighted velocities at 4,2 million cycles to be established (Figure 4.7). Relating this information to the effect of cycles on flute depth, it was possible to calculate directionally weighted velocity lines at any particular number of cycles (Appendix B). In order to provide a simple graphical aid, the lines relating to a 50 Hz excitation for a period of 1 day, 1 week, 1 month and 1 year have been calculated and are presented in Figure 5.1.

5.4 THE EFFECT OF LUBRICATION ON FALSE BRINELLING

Pittroff (1964:8-9) has shown that grease retards the rate of false brinelling by a factor of 15, and oil by a factor of 30. The model constructed for predicting bearing damage in horizontally mounted bearings (Figure 5.1), does not include the effect of lubrication, as all testing was carried out on unlubricated bearings. As it was decided to use the lubrication factors derived by Pittroff, values for flute depth given in Figure 5.1 should be divided by 15 for grease lubricated bearings, and by 30 for oil lubricated bearings. It should be stressed that these values were established for vertically mounted bearings and should be seen merely as a guide. The effects of lubricants most commonly used by Eskom power stations have yet to be assessed, and will be investigated as a separate project on completion of the present investigation.

5.5 BEARING CLEARANCE

False brinelling in horizontally mounted bearings arises due to the vibration of unloaded roller elements. If this vibration is eliminated for one reason or another, it is fair to conclude that no damage will occur. As bearing clearance is reduced, the elements will have increasingly smaller space in which to vibrate, until eventually they are held in place by the inner and outer race. Such a situation is unlikely to occur within Eskom as roller element bearings are generally fitted with a minimum internal clearance. It would however suggest that bearing clearance may possibly have a small effect on false brinelling in horizontally mounted bearings.

In vertically mounted bearings, radial clearance will directly influence the amount of free play experienced by the suspended rotor. An increase in radial clearance will increase the force with which the rotor and inner race vibrate against the roller elements and outer race. The increased force significantly increases the grinding effect of the eroded particles between roller element and race, thereby affecting the rate of bearing damage. For this reason, bearing clearance will be of greater significance in vertically mounted bearings than in horizontally mounted bearings.

5.6 MULTI-FREQUENCY EXCITATION

Although frequency itself does not have an effect on false brinelling, the rate of damage will be influenced by the degree of relative motion between roller and race. It is expected that a certain vibration amplitude at 100 Hz will have twice the effect of the same vibration amplitude at 50 Hz. However, due to the exponential relationship between cycles and flute depth (Figure 4.6) a certain vibration amplitude at 50 Hz will not result in the same damage as from half the vibration amplitude at 100 Hz. Further tests will be required before an accurate method of addressing multi-frequency excitation can be established.

5.7 CONCLUSION

It was hypothesised that static load, vibration amplitude, direction and bearing impedance would have a direct influence on the severity of false brinelling, and as such the inter-relationship of these factors could be used to predict its presence and severity. Static load and impedance did not significantly influence the rate of bearing damage. However, the effects of vibration amplitude and direction contributed directly to the rate of false brinelling. Vibration amplitude and direction were used to create a method of assessing the extent of bearing damage in horizontally mounted bearings subject to false brinelling (Figure 4.7). The hypothesis has therefore been accepted.

The purpose of this investigation was to establish a method for predicting the presence and severity of false brinelling on Eskom plant. It has been shown that a significant difference exists between false brinelling in vertically and horizontally mounted bearings. False brinelling in vertically mounted roller element bearings is dependent on the dynamic load or force experienced by the loaded elements. In horizontally mounted bearings, the amplitude and direction of vibration on the unloaded elements are the determining factors.

The method developed in this study for assessing false brinelling provides an indication of the extent of damage that may be expected to occur in horizontally mounted roller element bearings.

Vertically mounted bearings appear to be more vulnerable to damage through false brinelling than horizontally mounted bearings. This is due to the pendulum action and momentum of the vertical shaft, which is capable of inducing larger interface forces than a single unloaded roller element, as found in a horizontally mounted bearing. The results of this study and the literature review have shown that for assessing false brinelling in vertically mounted bearings, the method developed by Breward in 1978 is currently the most applicable. However, due to the advancements in lubrication technology, a reassessment of the factors for grease and oil lubrication is recommended.

CHAPTER SIX

RECOMMENDATIONS FOR FUTURE WORK

6.1 INTRODUCTION

Although a number of factors affecting false brinelling in horizontally mounted bearings have been established and quantified during this investigation, a number of new aspects have been identified and require further investigation.

6.2 THE INFLUENCE OF LUBRICATION

It is recommended that lubricants most commonly used within Eskom be tested in order to establish their effects on false brinelling. It is also recommended that lubricants containing extreme pressure (E.P.) additives be investigated as a possible method of reducing the likelihood of false brinelling occurring. If a lubricant can be identified that will maintain long term separation between surfaces of the rolling elements and race, the possibility of false brinelling occurring will be reduced. Tests should also be carried out under high humidity conditions in order to establish whether or not lubrication alone is sufficient to overcome the effects of moisture.

6.3 MULTI-FREQUENCY EXCITATION

Collins (1981:491-494) explains one method of assessing total fretting damage in the presence of multiple frequencies. This method may be applicable to false brinelling, however tests should be carried out in order to establish an appropriate method of assessing false brinelling in the presence of multi-frequency excitation.

6.4 THE EFFECT OF BEARING SIZE

In vertically mounted bearings, the rate of false brinelling depends largely on the force with which the roller elements vibrate against the bearing race. It is most likely that in horizontally mounted bearings, the force in the contact region between roller and race will also influence the rate of false brinelling. As the shaft is cradled by the bearings and is unable to swing, as does a vertical bearing shaft, the only factor influencing the interface force will be roller mass. It is therefore recommended that various sizes of bearings be tested in order to establish the significance of this factor.

6.5 ALTERNATIVE METHODS OF ASSESSING FALSE BRINELLING

This research has been successful in establishing one method of assessing false brinelling in horizontally mounted bearings. It is however an indirect method, and is dependent on an assessment of the factors affecting false brinelling and not on a measurement of the process itself. If a method could be established of monitoring the process directly, the influence of complex factors such as humidity, lubrication and multi-frequency excitation could be eliminated.

As it is the vibration of unloaded roller elements between inner and outer bearing race that causes damage, a measure of energy loss between housing and shaft is unlikely to provide an indication false brinelling in horizontally mounted bearings. No significant relative motion between shaft and housing was detected during testing, or any changing trend in vibration amplitude between the two.

It is by now well known that false brinelling is a special form of fretting, requiring relative motion in the form of sliding or scratching. As such the process will emit high frequencies from approximately the 100 kHz range and well into the MHz range. It is suggested that the method of acoustic emission monitoring be investigated as a possible means of establishing the presence of false brinelling. Such a method should be applicable to both vertically and horizontally mounted bearings.

Acoustic emission monitoring is still very much in a development stage and requires relatively sophisticated equipment. The ability to measure vibration already exists at most of Eskom's power stations and is a relatively simple procedure. The method of detecting false brinelling by monitoring vibration amplitudes, as proposed in this research, is therefore a more practical and useable system at this stage.

6.6 AXIAL SLIDING

During the preliminary investigations which involved vibration measurements on various types of plant at Lethabo Power Station, it was noted that pump impellers and motors connected to common systems experience a special form of excitation. The hydraulic pulses created by the pumps in operation were transmitted through the fluid, causing the stationary pump impellers to pulsate, and phase changes of up to 180° between shaft and housing. Such conditions give rise to significant relative motion and it is recommended that the effects of this be investigated separately.

6.7 REQUIREMENTS FOR OBTAINING COMPARABLE RESULTS

In order to be able to compare the results of future research, a number of aspects should be adhered to.

6.7.1 Humidity

The control of humidity is essential. Humidity should remain between 15 % and 40 % and certainly not be allowed to exceed 50 % at any stage. It would be preferable to have an isolated test chamber for the bearing in which the humidity could be closely controlled to within a few percent.

6.7.2 Lubrication

Where tests are being carried out on unlubricated bearings, the cleaning process is critical. Satisfactory results cannot be obtained unless the bearings are dismantled and thoroughly wiped with a solvent compatible to the particular lubricant used on the bearing. Ultrasonic cleaning with a variety of solvents did not prove to be successful.

6.7.3 Stability

The bearing shaft cannot be allowed to rock in an axial direction, or experience low angle radial oscillation. Both these movements induce additional relative motion which did not form part of this investigation. These two motions are not typical of that experienced by Eskom plant.

7. LITERATURE REFERENCES

Bill, R.C. 1982. Review of factors that influence fretting wear, in Materials evaluation under fretting conditions, STP 780, edited by S.R. Brown, Philadelphia: ASTM, 1982, p. 165 - 181.

Breward, M.J. 1973. Bearing life under false brinelling conditions. Ball bearing journal, no. 174 , p. 1 - 3.

Budinski, K.G. 1982. Evaluation of fretting corrosion by means of a new device for the control of oscillation amplitude, in Materials evaluation under fretting conditions, STP 780, edited by S.R. Brown, Philadelphia: ASTM, 1982, p. 49 - 65.

Christian, J.B. and McConnell, B.D. 1971. The analysis of mechanical variables influencing fretting corrosion in grease lubricated bearings, in Proceedings of the 26th annual meeting of American Society of Lubrication Engineering. preprint no. 71 AM 1B-1, Illinois: ASLE, 1971, p. 21 - 27.

Collins, J.A. 1981. Failure of materials in mechanical design, New York: John Wiley and Son, Inc.

Corrosion, edited by L.L. Shreir. 1965. London: George Newnes Ltd. 2 vol.

Du Randt, W.C. 1991.Brinelling in roller and ball bearings with reference to the development of techniques to identify the presence and severity of brinelling in bearings. Unpublished dissertation.

Pittroff, H. 1964. Fretting corrosion caused by vibration with rolling bearings stationary, ASME paper no. 64-lub-21.

SKF General Catalogue, 1981. Sweden: Elanders.

Tomlinson, G.A. 1927. Rusting of steel surfaces in contact, in Proceedings of the Royal Society. vol. 115A, 472 - 483.

VGB Guidelines: Preservation of power plant systems, 1981. VGB-R 116 He. Essen: VGB Technische Vereinigung der Grosskraftwerksbetreiber E.V. 43 p.

Waterhouse, R.B. 1982. Occurrence of fretting in practice and its simulation in the laboratory, in Materials evaluation under fretting conditions, STP 780, edited by S.R. Brown, Philadelphia: ASTM, 1982, p. 3 - 16.

APPENDIX A

BREWARDS MODEL FOR DETERMINING BEARING LIFE

A method of assessing bearing life in vertically mounted bearings under false brinelling conditions was developed by Breward in 1973 from the mathematical relationship derived by Pittroff in 1964.

The limit for false brinelling for satisfactory operation at speeds up to the bearing limits specified in the SKF General Catalogue, are given as 1/10 000 of the roller element diameter. Twice this depth is allowed where subsequent operation is at half the limiting speed or lower.

The critical flute width is calculated from eq 1. Using eq 2, the equivalent static bearing load is calculated. The number of load reversals N_V required to reach the limiting flute width b is obtained from Figure A1.

Equation 3 allows the bearing life to be calculated under either grease or oil lubrication.

Permissible bearing flute width $b = 0.018 * k * D$ in mm..... eq 1

where n = speed of rotation in rpm

$k = 1$ when $\frac{1}{2}n_{max} < n \leq n_{max}$ (n_{max} = limiting speed)

$= 2$ when $n \leq \frac{1}{2}n_{max}$

D = rolling element diameter

Nv = number of load reversals required to reach the limiting flute width

Equivalent static load $P_v = X_0F_{rv} + Y_0F_{av}$..........eq 2

where F_{rv} = amplitude of vibration in radial direction (kg)

F_{av} = amplitude of vibration in axial direction (kg)

X_0 = radial factor from SKF General Catalogue

Y_0 = axial factor from SKF General Catalogue

If the equivalent static bearing load P_v is less than F_{rv} use $P_v = F_{rv}$.

The allowable bearing life under either grease or oil lubrication may be calculated as follows:

$$Lv = \frac{Nv * A}{3600 * f} \text{ hours} \quad \text{.................................eq 3}$$

where Lv = bearing life in hours

A = 1 for unlubricated bearings

= 15 for grease lubrication

= 30 for oil lubrication

f = frequency of vibration (Hz)

It has been shown that dynamic bearing loads below 0.16 % of the static load rating will not cause damage to the raceways.

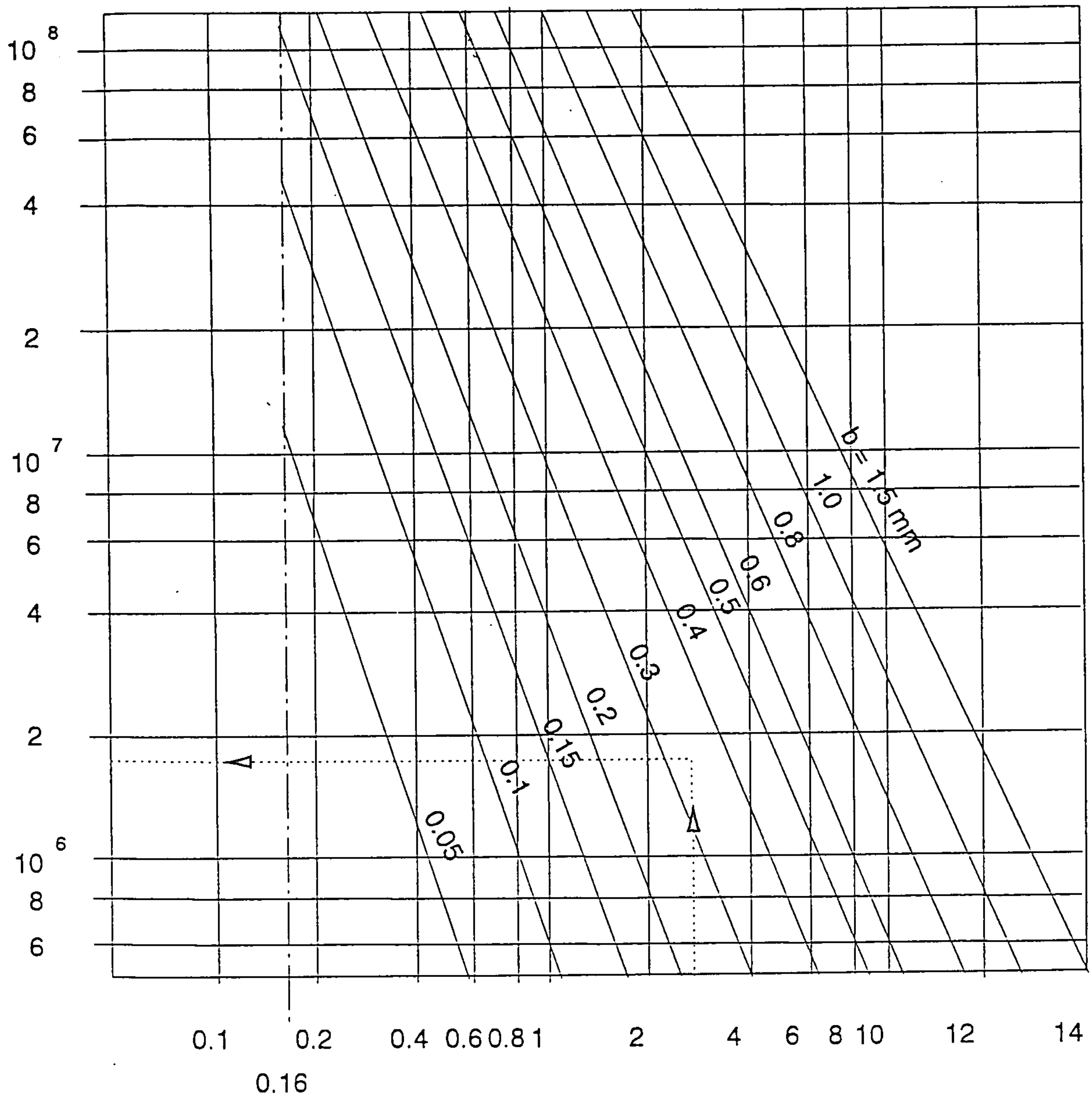


Figure A1 Breward's diagram for calculating bearing life in vertically mounted bearings under false brinelling conditions.

APPENDIX B

CALCULATION OF DIRECTIONAL VIBRATION CONTRIBUTION

The individual contribution of each direction has been calculated using simultaneous equations. From the directional tests the following formulas were obtained:

Vertical/horizontal	$y = 4.49\ X + 4.13$	...........eq.1
Vertical/axial	$y = 2.7\ X + 4.72$	...........eq.2
Horizontal/axial	$y = 2.38\ X + 3.68$	...........eq.3

For a vertical/horizontal resultant of 12 mm/s the vertical and horizontal amplitudes were 6 mm/s and 10.4 mm/s respectively. The arithmetical sum of these two amplitudes is 16.4 mm/s.

In order to obtain the percentage contribution of the vertical and horizontal amplitudes to the resultant, the directional amplitude is divided by the sum of the two

Vertical contribution $6/16.4 \times 100 = 36.5\ \%$

Horizontal contribution $10.4/16.4 \times 100 = 63.5\ \%$

The vertical/horizontal resultant of vibration gave rise to a rate of damage of 4.49 mm/s/μm with a y intercept of 4.13.

Therefore : $0.365\ V + 0.635\ H = 4.49 + 4.13$

$0.365\ V + 0.635\ H = 8.62$ eq.4

Similarly for the vertical/axial direction :

$6.38 + 9.47 = 15.85$

Vertical contribution $6.38/15.85 = 40.2\ \%$

Axial contribution $9.47/15.85 = 59.8\ \%$

Therefore $0.402\ V + 0.598\ A = 2.7 + 4.7$

$0.402\ V + 0.598\ A = 7.4$ eq.5

And for the horizontal/axial direction

$9.8 + 7 = 16.8$

Horizontal contribution $9.8/16.8 = 58.3\ \%$

Axial contribution $7/16.8 = 41.7\ \%$

Therefore $0.583\ H + 0.417\ A = 2.38 + 3.68$

$0.583\ H + 0.417\ A = 6.06$ eq.6

Rearranging equations 4,5 and 6 in terms of V, H and A respectively

$$V = 23.62 - 1.74\ H \;\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{eq.7}$$

$$H = 12.37 - 0.67\ V \;\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{eq.8}$$

$$A = 10.39 - 0.71\ A \;\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{eq.9}$$

Substituting eq.9 in eq.7

$$V = 23.62 - 1.74\ (10.39 - 0.71\ A)$$

$$V = 5.54 + 1.235\ A \;\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{eq.10}$$

Substituting eq.8 in eq.10

$$V = 5.54 + 1.235\ (12.37 - 0.67\ V)$$

$$V = 20.81 - 0.827\ V$$

$$V = 11.39$$

Substituting the value of V in eq.8

$$A = 12.37 - 0.67\ (11.39)$$

$$A = 4.738$$

Substituting the value of A in eq.9

$$H = 10.39 - 0.71\ (4.738)$$

$$H = 7.025$$

These values represent the weighting required for each value to obtain the same result, therefore the contribution each one makes will be the inverse of these values.

$$1/V = 0.087$$

$$1/A = 0.241$$

$$1/H = 0.14$$

Using the vertical direction as a reference these three directions relate to each other in the following ratio

$$V : H : A = 1 : 1.6 : 2.42$$

The effect of the axial vibration is therefore 2.42 times and horizontal vibration 1.6 times more severe than the vertical vibration.