

CRANFIELD UNIVERSITY

SEYED ALI MIRHADIZADEH

Monitoring hydrodynamic bearings with Acoustic Emission and
vibration analysis

School of Engineering
Department of Power & Propulsion

PhD
Academic Year: 2007 - 2012

Supervisor: Professor David Mba
June 2012

CRANFIELD UNIVERSITY

School of Engineering
Department of Power & Propulsion

PhD

Academic Year 2007 - 2012

SEYED ALI MIRHADIZADEH

Monitoring hydrodynamic bearings with Acoustic Emission and
vibration analysis

Supervisor: Professor David Mba
June 2012

This thesis is submitted in partial fulfilment of the requirements for
the degree of PhD

© Cranfield University 2012. All rights reserved. No part of this
publication may be reproduced without the written permission of the
copyright owner.

ABSTRACT

Acoustic emission (AE) is one of many available technologies for condition health monitoring and diagnosis of rotating machines such as bearings. In recent years there have been many developments in the use of Acoustic Emission technology (AET) and its analysis for monitoring the condition of rotating machinery whilst in operation, particularly on high speed machinery. Unlike conventional technologies such as oil analysis, motor current signature analysis (MCSA) and vibration analysis, AET has been introduced due to its increased sensitivity in detecting the earliest stages of loss of mechanical integrity.

This research presents an experimental investigation that is aimed at developing a mathematical model and experimentally validating the influence of operational variables such as film thickness, rotational speed, load, power loss, and shear stress for variations of load and speed conditions, on generation of acoustic emission in a hydrodynamic bearing. It is concluded that the power losses of the bearing are directly correlated with acoustic emission levels. With exponential law, an equation is proposed to predict power losses with reasonable accuracy from an AE signal.

This experimental investigation conducted a comparative study between AE and Vibration to diagnose the rubbing at high rotational speeds in the hydrodynamic bearing. As it is the first known attempt in rotating machines. It has been concluded, that AE parameters such as amplitude, can perform as a reliable and sensitive tool for the early detection of rubbing between surfaces of a hydrodynamic bearing and high speed shaft.

The application of vibration (PeakVue) analysis was introduced and compared with demodulation. The results observed from the demodulation and PeakVue techniques were similar in the rubbing simulation test. In fact, some defects on hydrodynamic bearings would not have been seen in a timely manner without the PeakVue analysis.

In addition, the application of advanced signal processing and statistical methods was established to extract useful diagnostic features from the acquired AE signals in both time and frequency domain. It was also concluded that the use of different signal processing methods is often necessary to achieve meaningful diagnostic information from the signals. The outcome would largely contribute to the development of effective intelligent condition monitoring systems which can significantly reduce the cost of plant maintenance.

To implement these main objectives, the Sutton test rig was modified to assess the capability of AET and vibration analysis as an effective tool for the detection of incipient defects within high speed machine components (e.g. shafts and hydrodynamic bearings).

The first chapter of this thesis is an introduction to this research and briefly explains motivation and the theoretical background supporting this research. The second and third chapters, summarise the relevant literature to establish the current level of knowledge of hydrodynamic bearings and acoustic emission, respectively. Chapter 4 describes methodologies and the experimental arrangements utilized for this investigation. Chapter 5 discusses different NDT diagnosis. Chapter 6 reports on an experimental investigation applied to validate the relationship between AET on operational rotating machines, such as film thickness, speed, load, power loss, and shear stress. Chapter 7 details an investigation which compares the applicability of AE and vibration technologies in monitoring a rubbing simulation on a hydrodynamic bearing.

Keywords:

Acoustic Emission, Hydrodynamic Bearing, Condition Monitoring, Vibration Analysis, PeakVue

ACKNOWLEDGEMENTS

In the name of God, the most Gracious, the most Compassionate

I would deeply like to express my appreciation and gratitude to my supervisor **Professor David Mba** for his guidelines, time and effort throughout my research. He was a great source of information throughout my time at Cranfield University, and specially for helping me to find the right direction when I thought all avenues were exhausted.

My deepest sincere gratitude to **Professor Stefan Kaczmarczyk**, **Dr Jonathan Adams** (Head of Department) and **Mr Richard Albany-Ward** for their valuable support and advice throughout my time at Northampton University specially for helping me to manage my life and my PhD, without their support, this study would not have been possible to complete.

I am deeply indebted to my parents and family for all their support, prayer, love and encouragement. Without their knowledge, wisdom, and guidance I would not have the goals I have to strive and be the best to reach my dreams. Without their unreserved support, completion of this study would not have been possible.

I would also like to express my gratitude to the many people for their help. Particular mention goes to those individuals and bodies listed below:

Professor Pericles Pilidis (Cranfield University, UK)

Dr. Bannister (Cranfield University, UK)

Dr. Victor A.KARASEV (CIAM)

Dr. Mike Ross (Altom Power)

Mr. Josh Redmond (Cranfield University, UK)

Mr. Scott Booden (Cranfield University, UK)

Dr. Seyed Mohseni (Samad Power Ltd)

Mr. Cranford Johnstone (Emerson Process Management)

Many individuals and institutions have contributed information that was used to do this work. Their contributions are gratefully acknowledged. Last, but not least, I would like to express my thanks to the staff members of the Department of Power and Propulsion and my colleagues for their help.

TABLE OF CONTENTS

ABSTRACT	i
ACKNOWLEDGEMENTS.....	iii
LIST OF FIGURES.....	ix
LIST OF TABLES	xv
LIST OF EQUATIONS.....	xvii
LIST OF ABBREVIATIONS	xix
1 Introduction.....	1
1.1 Project Objectives.....	2
1.2 Scientific Contribution	4
1.3 Publications	5
1.3.1 Journal Publications	5
1.3.2 Conference Publications	5
1.3.3 Books	6
2 Journal Bearing (Hydrodynamic Bearing).....	7
2.1 Typical Failures in Hydrodynamic Journal Bearings	9
2.2 Wear	9
2.3 Scoring.....	9
2.3.1 Scoring of White Metal Journal Bearing	10
2.3.2 Scoring of Lead-Bronze Journal Bearing.....	11
2.4 Erosion.....	11
2.4.1 Erosion damage of White Metal Journal Bearing	12
2.4.2 Erosion damage to the Axial Oil Grooves in a Journal Bearing.....	12
2.5 Corrosion	13
2.6 Lubricant Degradation Deposits.....	13
2.7 Overheating	14
2.8 Electrical Pitting	15
2.9 Fatigue.....	15
2.9.1 Fatigue of White Metal Journal Bearing	16
2.9.2 Premature Fatigue of White Metal Journal Bearing Caused by Vibration.....	17
2.10 Fretting.....	18
2.11 Wiping.....	18
2.11.1 Start-up Wiping Damage	19
2.12 Shaft	19
2.13 Theory and Design of the Journal Bearing.....	20
2.13.1 Minimum Film Thickness.....	22
2.13.2 Eccentricity Ratio	22
2.13.3 Power Loss	23
2.13.4 Pressure Distribution	23
2.13.5 Sommerfeld Number	25

2.13.6 Clearance.....	27
2.13.7 Specific Pressure	27
2.13.8 Journal Bearing Data.....	28
2.14 Shear Stresses in the Fluid.....	28
2.15 Calculation methods for hydrodynamic journal bearings.....	35
3 Acoustic Emission	39
3.1 Background of AE in Bearings	40
3.2 Acoustic Emission and Friction in Journal Bearings	44
3.3 Literature Review.....	46
4 Test Rig Design.....	57
4.1 Test Rig	57
4.1.1 Hydrodynamic Bearing	58
4.1.2 Journal Test Shaft.....	59
4.1.3 Bearing Oil Supply System.....	60
4.1.4 Elimination of the Noise of the Rotating Shaft.....	60
4.1.5 The Hsu-Nielsen source test.....	61
4.1.6 Hydraulic pump	64
4.1.7 Pressure gauge	64
4.1.8 Load pistons.....	65
4.2 Application	66
4.2.1 Sensors Location	66
4.2.2 Data Acquisition Process and Instrumentation.....	67
4.2.3 AE data acquisition system	68
4.2.4 Temperature Measurement.....	70
4.2.5 Vibration acquisition system.....	71
4.2.6 Displacement Measurement.....	71
4.2.7 CSI 4500 Machinery Health™	73
4.2.8 NI DAQ.....	74
5 NDT Technologies.....	77
5.1 Vibration Analysis	77
5.1.1 Vibration characteristics and monitoring methods.....	79
5.2 PeakVue	81
5.2.1 The generation of stress waves	83
5.2.2 The detection of stress waves.....	84
5.2.3 The PeakVue Methodology	84
5.2.4 Application: bearing fault detection	88
5.3 Eddy currents measurements	89
6 Operational Results and Discussions.....	93
6.1 Operational Experimental Results.....	96
6.1.1 Power Losses Results.....	96
6.1.2 Film Thickness Results	98
6.1.3 Shear Stresses Results.....	100

6.1.4 Acoustic Emission (AE) Results	102
6.1.5 Power Losses, AE and Shear Stresses in the Fluid	104
6.1.6 Modelling AE with bearing parameters.....	125
7 Incipient Hydrodynamic Bearing Failure Detection.....	129
7.1.1 Observation of the signals in time and frequency domains	134
7.2 Vibration Results.....	150
7.2.1 Case Study: Demodulation and PeakVue	150
7.2.2 PeakVue Results for Rub Simulations.....	154
7.3 Displacement Results	164
8 Conclusions.....	169
8.1 Recommendations for Future Work	171
REFERENCES.....	175
APPENDICES	193
Appendix A	193
Appendix B	196
Appendix C	201
Appendix D	203
Appendix E	213
Appendix F.....	215

LIST OF FIGURES

Figure 2-1	Pressure distribution in a journal bearing.....	8
Figure 2-2	Two diametrically opposed axial grooves at 90° to the load line...	8
Figure 2-3	Wire wool Damaged Collar (Neale Consulting Engineers).....	10
Figure 2-4	Scoring of White Metal Journal Bearing [17].....	11
Figure 2-5	Scoring Lead-Bronze Journal Bearing [17]	11
Figure 2-6	Erosion Damage of White Metal Journal Bearing (Neale Consulting Engineers).....	12
Figure 2-7	Axial Oil Groove Damage (Neale Consulting Engineers).....	13
Figure 2-8	Bearing Surface with Ammonium Succinate [17]	14
Figure 2-9	Damage due to Overheating [16].....	14
Figure 2-10	Electric Pitting (Kingsbury)	15
Figure 2-11	Initial Deterioration (Neale Consulting Engineers).....	16
Figure 2-12	Severe Damage to the Bearing (Neale Consulting Engineers)	16
Figure 2-13	Poor bonding of white metal and the bearing shell (Neale Consulting Engineers).....	17
Figure 2-14	Bonding defect between the edges of the dovetail and white metal (Neale Consulting Engineers).....	18
Figure 2-15	Fretting (Neale Consulting Engineers)	18
Figure 2-16	Start-up Wiping (Neale Consulting)	19
Figure 2-17	Kinematic viscosity of lubricant.....	21
Figure 2-18	Illustration of rotation [25]	22
Figure 2-19	Pressure distribution [25].....	24
Figure 2-20	Journal bearing with two axial oil grooves	27
Figure 2-21	Shear Stress distribution	29
Figure 2-22	Normal and shear stresses in cylindrical coordinates.....	30
Figure 2-23	Comparison graph for power losses Vs rotational speed at 2kN.	37
Figure 2-24	Comparison graph for Minimum film thickness Vs rotational at 2kN	38

Figure 2-25	Comparison graph for Shear stresses Vs rotational speed at 2kN	38
Figure 3-1	Schematic diagram of an acquisition system used to monitor AE activity in an operational steam turbine unit [68]	51
Figure 4-1	Test rig layout	57
Figure 4-2	Flexible Coupling	58
Figure 4-3	Hydrodynamic Bearing.....	59
Figure 4-4	Test rig sketch.....	59
Figure 4-5	Noise Eliminator-Nylin 6/6.....	61
Figure 4-6	The Hsu-Nielsen source	62
Figure 4-7	Attenuation tests on the test rig.....	63
Figure 4-8	Dimensions and features of the pushing cylinder	64
Figure 4-9	Hydraulic pump and pressure gauge installed on the test rig	65
Figure 4-10	sketch of applied load positions.....	65
Figure 4-11	location of AE sensor and thermocouple sensors on bearing..	66
Figure 4-12	Location of AE and thermocouple sensors on test bearing (right side)	67
Figure 4-13	Data acquisition process diagram	68
Figure 4-14	Basic setup of an AE sensor	69
Figure 4-15	Continuous AE signal acquisition system.....	69
Figure 4-16	Location of AE and thermocouple sensors type on test bearing..	70
Figure 4-17	VK-202A1 Driver and Sensor	72
Figure 4-18	External Mounting VL Sensor.....	73
Figure 4-19	CSI 4500 Machinery Health TM Monitor [122]	74
Figure 4-20	4.2.8 NI PCI-MIO-16XE-50 DAQ	74
Figure 4-21	Labview diagram for measurement	75
Figure 5-1	Generation of stress waves by impact and their propagation along a metal plate Source [122]	84
Figure 5-2	Demodulation process [164]	86
Figure 5-3	Peak Value detection using a high sampling frequency [114]	86

Figure 5-4	Outline of the PeakVue methodology for bearing fault detection	89
Figure 5-5	Eddy current sensors position.....	91
Figure 6-1	The layout of the sensors in the bearing	95
Figure 6-2	Temperature for each thermocouples and effect of rises due to speed at 10kN load condition	95
Figure 6-3	Power Losses vs rotational speed at SP 68 and SP220	96
Figure 6-4	Power Losses vs rotational speed at SP 460.....	97
Figure 6-5	Power Losses vs Viscosity for Different type of Oils	98
Figure 6-6	Sommerfeld Number vs Film Thickness at all viscosities.....	99
Figure 6-7	Film thickness vs rotational speed at SP68 & SP220 oil.....	100
Figure 6-8	Film thickness vs rotational speed at SP460 oil	100
Figure 6-9	Shear stress vs rotational speed at SP68 & SP220 oil	101
Figure 6-10	Shear stress vs rotational speed at SP460 oil	101
Figure 6-11	Typical test sequence for a load of 10kN	102
Figure 6-12	Acoustic emission vs rotational speed at SP460 oil	103
Figure 6-13	Acoustic emission vs rotational speed at SP68 & SP220 oil .	103
Figure 6-14	Power Losses vs Acoustic Emission at SP68 & SP220 oil....	104
Figure 6-15	Power Losses vs Acoustic Emission at SP460 oil	105
Figure 6-16	AE r.m.s and total power loss for varying speed 2kN	108
Figure 6-17	Typical averaged AE r.m.s and total power loss for varying speed and constant load conditions	108
Figure 6-18	AE r.m.s and power loss for varying speed and load conditions (2kN, 6kN, 10kN) at varying dynamic viscosities.....	109
Figure 6-19	Power Losses vs Shear Stress at SP68 & SP220 oil	110
Figure 6-20	Power losses vs Shear Stress at SP460 oil.....	110
Figure 6-21	Velocity distribution in the oil film [159].....	110
Figure 6-22	Pressure and viscous forces acting on an element of lubricant [159]	111
Figure 6-23	Velocity gradient in the oil film [159]	113
Figure 6-24	Laminar flow of fluid in clearance space [159].....	113
Figure 6-25	Power losses vs film thicknesses at SP68 & SP220 oil.....	115

Figure 6-26	Power losses vs film thicknesses at SP460 oil	115
Figure 6-27	Influence of Minimum film thicknesses on Power Loss for 10kN load condition	116
Figure 6-28	Acoustic emission vs film thicknesses at SP460 oil by increasing the speed [750rpm to 4500rpm]	117
Figure 6-29	Acoustic emission vs film thicknesses at SP68 and SP220 oil by increasing the speed [750rpm to 4500rpm]	117
Figure 6-30	AE r.m.s levels at an applied load 10kN for varying viscosities	118
Figure 6-31	AE r.m.s and shear stress for varying speed and constant load conditions of 6kN	120
Figure 6-32	AE r.m.s and shear stresses for varying speed and constant load conditions of 10kN and different dynamic viscosities	121
Figure 6-33	Shear stresses, rotational speed, dynamic viscosity (SP220), power loss (bottom of bearing) and minimum film thickness for load condition 10kN	123
Figure 6-34	AE r.m.s, rotational speed, dynamic viscosity (SP220), bottom power losses and total loss for load condition 10kN	124
Figure 6-35	AE r.m.s and Power loss bottom (P_{Btm}) for varying dynamic viscosity	126
Figure 7-1	Load position on ball bearing at the drive end	129
Figure 7-2	Typical AE test sequence at 1500 rpm	130
Figure 7-3	AE sensors position	132
Figure: 7-4	Influence of rubbing on hydrodynamic bearing and shaft	132
Figure 7-5:	Typical AE time traces background noise; 1500rpm and no applied load	133
Figure 7-6	AE RMS levels during rub tests at load cycle 6 kN	135
Figure: 7-7	AE RMS levels during rub tests at load cycle 4 kN	135
Figure: 7-8	Typical AE waveform for 4kN and 1500rpm balanced loads .	136
Figure: 7-9	Typical AE waveform for 6kN and 1500rpm balanced loads .	136
Figure: 7-10	Modulated AE waveforms due to rubbing for 4kN load	137
Figure: 7-11	Modulated AE waveforms due to rubbing for 6kN load	137
Figure: 7-12	Influence of rubbing on hydrodynamic bearing close to channel 2	139

Figure: 7-13	Influence of rubbing on hydrodynamic bearing close to channel 1.....	140
Figure: 7-14	Close-up view of a selected region in figures 7-11 and 7-12..	141
Figure 7-15:	different time compression graph	142
Figure 7-16:	Two FFT at different time at 6kN	143
Figure 7-17:	Two FFT at different time at 4kN	143
Figure 7-18:	AE signature with corresponding frequency spectrum at 4kN	144
Figure 7-19:	AE signature with corresponding frequency spectrum at 6kN....	145
Figure 7-20	Spectrogram plot associated with 4kN and 6kN load condition... ..	147
Figure 7-21	Spectrogram analyses for stimulated rub at 4kN load condition.. ..	148
Figure 7-22	Spectrogram analyses for stimulated rub at 6kN load condition.. ..	149
Figure 7-23	Vibration and AE sensors position.....	151
Figure 7-24	Vibration waveform at 1000 rpm with FFT frequency 16.67Hz.... ..	152
Figure 7-25	Demodulation spectrum at 1000 rpm and 1kN	153
Figure 7-26	PeakVue Spectrum at 1000 rpm (16.67 Hz) and 1kN	153
Figure 7-27	Comparison between the Demodulation and PeakVue	154
Figure 7-28	Vibration levels during rub tests at 4kN and 6kN condition....	155
Figure 7-29	Acceleration compression for load conditions	156
Figure 7-30	FFT spectrum with balanced, 4kN and 6kN load conditions..	157
Figure 7-31	Comparison frequency spectrum at balanced condition with different techniques.....	158
Figure 7-32	Comparison frequency spectrum at 4kN load condition with different techniques.....	159
Figure 7-33	Comparison frequency spectrum at 6kN load condition with different techniques.....	160
Figure 7-34	Comparison frequency spectrum at 6kN load condition with different techniques up to 300Hz.....	161
Figure 7-35	Comparison frequency spectrum at 4kN load condition with different techniques up to 300Hz.....	162

Figure 7-36	Test rig layout.....	164
Figure 7-37:	Typical eddy current displacement sensor installation	164
Figure 7-38	Typical displacement waveform at 2kN	165
Figure 7-39	Typical Orbit graphs at 750 rpm	166
Figure 7-40	One revolution at 4500 rpm and 2kN.....	166
Figure 7-41	Displacement waveforms and their frequency domains at 2kN...	167
Figure 7-42	Displacement waveform and frequency domain at 4500rpm for 2kN load	167
Figure 7-43	P-P Displacement for channel 3 and Channel 4 and 2 σ m change in shaft displacement at 4500 rpm and 10kN	168

LIST OF TABLES

Table 2-1	Oil types density at 20°C [Appendix C)	21
Table-2-2	Summerfield number characteristic.....	25
Table 2-3	Relationship between Sommerfeld number and other parameters	26
Table 2-4	Journal Bearing Data	28
Table 2-5	RAIMONDI Boyd graphs based method at 2kN.....	36
Table 2-6	ESDU METHOD for varying speed conditions at 2kN.....	36
Table 2-7	BANNISTER METHOD for varying speed conditions at 2kN	36
Table 2-8	JB DATABOOK METHOD for varying speed conditions at 2kN..	36
Table 2-9	The Relative percentage changes in Film Thickness (MFT) for varying Speed conditions	37
Table 4-1	Nylin6/6 specification	61
Table 4-2	Attenuation results in Journal Bearing Test Rig	63
Table 4-3	A physical acoustics Pico sensor Characteristics	68
Table 4-4	Demonstrating characteristics of Type J and K thermocouples ..	71
Table 5-1	Typical faults and defects in shafts and journal bearings can be detected by vibration monitoring [97]	79
Table 5-2	The basic components of a vibration monitoring system	79
Table 5-3	Standard identifies evaluation zones of shaft vibrations	80
Table 5-4	Key different between PeakVue and Demodulation	87
Table 6-1	Temperature, AE r.m.s and Film Thickness for different location of Journal Bearing	94
Table 6-2	Minimum film thickness and power loss at different load condition and speed on bearing for SP 68.....	105
Table 6-3	Minimum film thickness and power loss at different load condition and speed on bearing for SP 220.....	106
Table 6-4	Minimum film thickness and power loss at different load condition and speed on bearing for SP 460.....	107
Table 6-5	The Castrol Alpha SP oil range of high quality lubricants at different grade (Δ) with viscosity factor.....	128
Table 7-1	AE activities at symmetric load condition for SP 460	130

Table 7-2	The Relative percentage changes in AE RMS for varying Load between Rubbing and Symmetric load conditions.....	131
-----------	--	-----

LIST OF EQUATIONS

(2-1).....	21
(2-2).....	23
(2-3).....	24
(2-4).....	25
(2-5).....	27
(2-6).....	29
(2-7).....	31
(2-8).....	31
(2-9).....	31
(2-10).....	31
(2-11).....	31
(2-12).....	31
(2-13).....	31
(2-14).....	32
(2-15).....	32
(2-16).....	32
(2-17).....	32
(2-18).....	32
(2-19).....	32
(2-20).....	32
(2-21).....	33
(2-22).....	33
(2-23).....	33
(2-24).....	33
(2-25).....	33
(2-26).....	33
(2-27).....	33
(2-28).....	34

(2-29).....	34
(2-30).....	34
(2-31).....	34
(2-32).....	34
(2-33).....	34
(2-34).....	34
(6-1).....	108
(6-2).....	110
(6-3).....	111
(6-4).....	111
(6-5).....	112
(6-6).....	112
(6-7).....	112
(6-8).....	112
(6-9).....	112
(6-10).....	112
(6-11).....	113
(6-12).....	114
(6-13).....	114
(6-14).....	114
(6-15).....	114
(6-16).....	114
(6-17).....	126
(6-18).....	127
(6-19).....	127
(6-20).....	127
(6-21).....	127

LIST OF ABBREVIATIONS

D	Diameter of journal	m
e/ε	Eccentricity for an aligned bearing	m
g	Acceleration due to gravity	m/s^2
H	Power loss	W
N	Frequency of shaft rotation	rev/s
W	Radial load	N
h_{min}	Minimum film thickness at closest approach for aligned bearing	m
Ra	Surface roughness value	m
ψ	Attitude angle (angle between load line and line of centres)	deg
P	The specific pressure	Pa
R	The radius of the bearing	m
ω	The rotating speed of the shaft	rad/s
u	The coordinates of the velocity vector	-
v	The coordinates of the velocity vector	-
w	The coordinates of the velocity vector	-
h	The lubricant film thickness in the z-direction	-
c	Radial clearance	m
L	Journal axial length	m
Ta	Ambient temperature	$^{\circ}C$
$R1$	Sensor 1 right side of bearing	$^{\circ}C$

$R2$	Sensor 2 left side of bearing	$^{\circ}\text{C}$
$L1$	Sensor 1 right side of bearing	$^{\circ}\text{C}$
$L2$	Sensor 2 left side of bearing	$^{\circ}\text{C}$
β_1	Top brg. Arc	Deg
β	Bottom brg. Arc	Deg
μ	The dynamic viscosity of the fluid	Pa.s
u	The velocity distribution of the fluid along the boundary	m/s
y	The height of the boundary	m
ρ	Density	cSt
$\tau_{r\theta}$	Shear stress distribution in the bearing	MPa
R_1	Shaft radius	m
R_2	Bearing radius	m
f	Friction factor	-
P_{Btm}	Power loss in btm. Pad	kW
AE_{rms}	Acoustic emission r.m.s	V
Λ	Viscosity factor	-

1 Introduction

The need for a reliable condition monitoring technique (CMT) is essential for predicting failure prior to any catastrophic disaster and preventing sudden shutdowns of industrial machinery. The former is particularly important in the oil and gas sector where unexpected plant shutdown can result in major economic loss, while an unexpected failure can result in fatalities in civil aviation. Hydrodynamic (journal) bearings are one of the most critical components of high speed rotating machinery in the oil and gas industries. Condition monitoring can be referred to as an incremental measure of a machine's ability to safely and efficiently perform its predefined duty [1]. There are different techniques available for rotating machinery condition monitoring such as; vibration monitoring, oil debris analysis, thermography, motor current analysis and acoustic emissions (AE).

Condition monitoring, through the use of vibration analysis is widely established for a range and classification of rotating machinery. RMS (root-mean-square) acceleration, maximum acceleration, skewness and kurtosis are the most common parameters which are usually considered in vibration analysis.

As the time between detection and failure can be very short, there is a need for a new diagnostic technique that can offer earlier damage detection, thereby allowing the operators sufficient lead time to take corrective actions.

Failure symptoms are often detected when the noise level increases or settles at a specific frequency. Increased viscosity of the lubricants also contributes to a noise level increases [2, p.10]. Increased noise could also be attributed to an increased load on the elastic body. The noise could be a manifestation of the appearance and propagation of cracks, fracture processes, cavitation phenomena and phase changes, or other plastic deformations. [2, p.11]. Compared to monitoring machinery's temperature, Acoustic Emission technology (AET) allows for early detection and proactive implementation of remedial measures to prevent a complete breakdown. Detecting the early stages of failure is helpful to the operator or owner of the engine or machinery

[2, p.13]. Noise levels can range from audible to ultrasonic. When there is too much background noise to detect any abnormality, changes in the ultrasonic sound levels, using an AE sensor to detect them, would be advantageous. Albers et al. suggest that both frequency and amplitude can be used to detect failing bearings earlier [2, p.10].

Aside from the basic visual examination, the opportunities for prognostics must be coupled to new monitoring techniques such as AE, which could indicate problems early within the assembly. AET is gaining ground as a power tool in the condition monitoring of rotating equipment. Based on the British standard; Acoustic Emission is defined as "physical phenomenon occurring within material a spontaneous elastic energy released within a material or by a process in the form of elastic waves" [3]. The application of AET in research and industry is well-documented [4-14].

According to Prieto [15], AET has advantages when compared with other evaluation methods. First, defects are easier to detect because of improved signal-to-noise ratio. Second, AET can monitor efficiently machinery that is rotating at low speeds. Third, acoustic emission can transcend background noise (<20 kHz) [15, p.36]. Fourth, since it can detect abnormalities in operations at high frequencies, it is far more sensitive than vibration techniques. Fifth, detection can continue while the machinery is running. Finally, the AE technique can be used to make an assessment at almost any time regardless of rotational speed [15, p.37].

There are drawbacks though when using AET to detect defects in bearings.

To establish a margin or standard, an experiment must first be conducted. AE can be attributed to the transmission length.

To conduct AE tests, one needs sophisticated and sensitive instruments. Finally, compared to vibration Analysis, AE emits a weaker signal [15, p.38].

1.1 Project Objectives

The project objectives are listed as follows:

- This thesis presents an experimental investigation that is aimed at developing a mathematical model and experimentally validating the influence of operational variables such as film thickness, speed, load, power loss, and shear stress for variations of load and speed conditions, on generation of acoustic emission in a hydrodynamic bearing. It is concluded that the power losses of the bearing are directly correlated with acoustic emission levels. With exponential law, an equation is proposed to predict power losses with reasonable accuracy from an AE signal.
- Detecting mechanical faults in rotating machines, particularly hydrodynamic bearings, has been recognised as important for preventing sudden shut downs. This experimental research presents a comparative study between AE and Vibration to diagnose the rubbing on the hydrodynamic bearing at high rotational speeds. As it is the first known attempt in rotating machines. It has been concluded, that AE parameters such as amplitude, can perform as a reliable and sensitive tool for the early detection of rubbing between surfaces of a hydrodynamic bearing and high speed shaft. With reasonable approaches, this model will allow us to interpret the result of rubbing units, as well as predict the behaviour of rotating machines and the features of tribology pairs in industrial experiments.
- The application of vibration (PeakVue) analysis was introduced and compared with demodulation. The results observed from the demodulation and PeakVue techniques were similar in the rubbing simulation test. In fact, some defects on hydrodynamic bearings would not have been seen in a timely manner without the PeakVue analysis.
- When impacting occurs, many other sources of signal are present. Some examples are rubbing of the ball bearing in supporters, contaminants in the oil or grease, friction between surfaces, surface roughness, skidding, etc. These are all sources of AE as background noise. Sometimes the background noise signal can be as significant as the signal produced by the phenomenon measured. Although these levels can be equal, it is still

possible to distinguish both signals and therefore reduce the background noise. This was one of the fundamental aims of this thesis.

- Finally, to study the application of advanced signal processing techniques to enhance the diagnostic effectiveness of AE and vibration signals. It was also concluded that the use of different signal processing methods is often necessary to achieve meaningful diagnostic information from the signals. The outcome would largely contribute to the development of effective intelligent condition monitoring systems which can significantly reduce the cost of plant maintenance.

1.2 Scientific Contribution

This Research has made significant contributions in the area of condition monitoring of high speed rotating machinery.

Firstly, this research programme can be considered as the only comprehensive publication that have involved running bearing tests within the parameters of a hydrodynamic bearing (power loss, friction factor, shear stress and film thickness) rotating shafts at high speed and load conditions with AE activity and other monitoring tools. Secondly, the applicability of the AE and vibration technologies were studied for rubbing circumstances on the hydrodynamic bearing. Thirdly, it has been demonstrated that the capabilities of vibration integrated with the PeakVue analysis techniques can detect impact in the hydrodynamic bearing in the presence of rubbing. It has been shown that PeakVue techniques are more useful than the standard spectral analysis in early fault detection for high rotational speed of hydrodynamic bearings. Fourthly, the results from this investigation have shown that bearing impacts are more likely to be detected at an early stage and the development of the fault can be monitored using AE signal analysis. Fifthly, the results also revealed that the use of PeakVue with AE analysis provided a successful combination of a time-frequency spectrum to describe whether the AE signals have periodicity components or not. Finally, with exponential law, an equation is proposed to

predict power losses with reasonable accuracy from an AE signal systems without prior knowledge of the relative rotational speeds or load conditions.

1.3 Publications

The findings of this study were published in several international journals, Conferences and contributed in two books. Furthermore, these publications have been cited in journal publications.

1.3.1 Journal Publications

- 1) Mirhadizadeh,S.A.; Moncholi,E.P.; Mba,D, Influence of operational variables in a hydrodynamic bearing on the generation of acoustic emission; Tribol.Int., 2010, 43, 9, 1760-1767
- 2) Mirhadizadeh,S.A, D. Mba, Observations of acoustic emission in a hydrodynamic bearing, Journal of Quality in Maintenance Engineering (JQME), Volume 15 Issue 2, pp. 193-201, 2009

1.3.2 Conference Publications

- 1) Mirhadizadeh,S.A, D. Mba ;Acoustic Emission in lightly loaded hydrodynamic bearings 22nd International Congress and Exhibition on Condition Monitoring and Diagnostic Engineering Management (COMADEM 2008), ISBN 978-80-254-2276-2, 355-360
- 2) Mirhadizadeh S.A, Mba, D, Observations of Acoustic Emission in a hydrodynamic bearing, 2nd International conference on Technical Inspection and NDT, TINDT 2008, Tehran, Iran, 21-22 Oct 2008
- 3) Mirhadizadeh S.A, Mba, D, Development of Acoustic Emission Technology For Condition Monitoring and Diagnosis Of Journal Bearing; Journal: CM 2009 and MFPT ;CM-MFPT-0188-2009.R1;17-Apr-2009
- 4) Mirhadizadeh,S.A, Esmaili M, Refahi Oskuei A. "Acoustic Emission as a powerful Non- Destructive Technique (NDT) for applying diagnosis

evaluation of rotating machines" 3rd Rotating Equipment in Oil & Power Industry , Razi Conference Centre ,Tehran ,Iran ,Nov 2010

- 5) Mirhadizadeh,S.A, Esmaili, M. , and Refahi Oskuei, A, " Development of Acoustic Emission Technology For Condition Monitoring& Diagnosis Of Journal Bearing ", Regional Confererence of Mechanic Engineering ,Azad University,March 2011.- ISSN 1735-5540
- 6) Mirhadizadeh,S.A, Esmaili M., Refahi Oskuei A, "Acoustic emission monitoring of journal bearings: The effect of transverse loading, viscosity and spindle (rotary) speed" Regional conference of Mechanical Engineering , Azad University, March 2011.- ISSN 1735-5540
- 7) Mirhadizadeh,S.A, Esmaili M., "Development of the use of Acoustic Emission Technology in hydrodynamic bearings" 2nd Rotating Equipment in Oil & Power Industry , Razi Conference Centre ,Tehran ,Iran ,May 2010- ISSN 1735-5540

1.3.3 Books

- 1) Mirhadizadeh. S.A (contributor), Mba, D (contributor); Acoustic Emission Method for Monitoring Rotating Machinery; Acoustic Emission", ISBN 979-953-307-372-8 (Under Revision).
- 2) Mirhadizadeh. S.A (contributor); Defect Diagnostics of Rotatory Machinery Using Acoustic Emission Methods. Nashre Daneshgahi with Cooperating Pers Sanco Co, 66418586.

2 Journal Bearing (Hydrodynamic Bearing)

The aim of the journal bearing is to allow the shaft to rotate easily and transmit its loads to the machine's frame. The journal bearing has a 'sacrifice mission' in the machine; it must be the bearing and not the shaft or the machine's frame, which is wearing for maintenance and economic reasons. Journal bearings are designed to work a predetermined number of cycles. The bearing's life can be determined by load, rotational speed, temperature, maintenance, vibration, lubrication and other factors.

A failure of the journal bearing causes the collapse of the machine via the wearing of the shaft or the frame. The cost of replacing a shaft or a pedestal is much larger than that of replacing the journal bearing.

Bearings that have to support rotating shafts are categorised into four types: rolling contact, hydrostatic, hydrodynamic and magnetic [16, p.3]. Of particular interest in this report are the failures associated with hydrodynamic journal bearings. By definition, hydrodynamic journal bearings transmit load onto a 'self-renewing film or lubricant.' [16, p.7], and usually support radial loads. Another type of bearing that supports the axial load is referred to as a thrust bearing [16, p.7].

Hydrodynamic bearings function because the oil forms a wedge-shaped film between the journal and bearing. The oil's resistance to the flow and its adhesion to the journal as it is rotated will form a wedge-shaped film. The principle is illustrated in Figure 2-1. The integrity of the bearing is dictated by the formation of the wedge-shaped film [16, p.7].

For trouble-shooting and evaluation purposes it is essential that mechanics or engineers can spot the tell-tale signs of failure. A visual examination is often the first step in determining if failure is imminent; however, an oversight may cause severe damage to other parts of the assembly. Most industrial machines use hydrodynamic lubricants to operate. The materials of choice for bearings are often metals or Babbitt [17, 19-23], because of their low hardness and melting

points ,making sure that if the bearing fails the damage is just isolated to the bearing [17, p.8].

As stated previously, the type of bearing used in this research is a hydrodynamic bearing which self-generates a film thickness and pressure as it rotates. Figure 2-1 shows the effect of a pressure gradient in a hydrodynamic bearing with a stationary bearing and a moving shaft. This gives rise to the film pressure which is present in an oil film bearing.

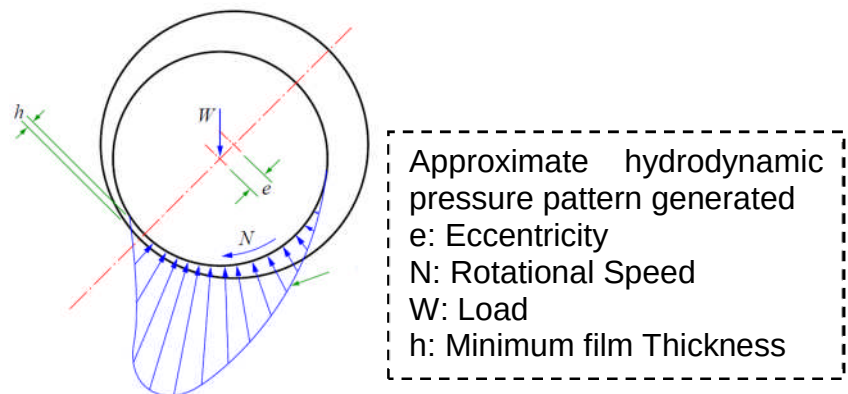


Figure 2-1 Pressure distribution in a journal bearing

As the velocity within the hydrodynamic bearing increases, so does the pressure. This is noted in the standard pressure distribution profile, see figure 2-2.

As the journal bearing starts to rotate, and with the lubrication being supplied into the bearing via the twin axial grooves. Grooves at both the joints (see Sketch 2-3), make it equally suitable for shaft rotation in either direction. The rotation of a shaft starts to create a wedge of film, which separates the shaft from the bearing housing.

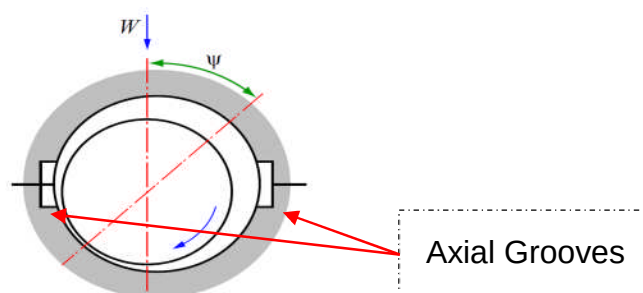


Figure 2-2 Two diametrically opposed axial grooves at 90° to the load line

In order to understand hydrodynamic bearings on high speed shafts, it was decided to study typical failures in Hydrodynamic Journal Bearings. Understanding the causes of each defect or damage would make one more cautious about abusing the material or putting substandard materials into the assembly.

2.1 Typical Failures in Hydrodynamic Journal Bearings

Any of the following failures which occur in bearings when left undetected could be costly to the company owning the machine. It is important for technicians or mechanics to know about basic failure symptoms in order to initiate proactive solutions. Overlooking these symptoms could affect the rest of the parts within the assembly.

2.2 Wear

Journal bearings do not experience wear, especially when they are lubricated properly. Failures are more likely caused by high flow rates that would inadvertently erode the flow restrictors. If wear does occur, it is usually during start-up and shutdown when the speed is too low to produce sufficient fluid pressure to support the bearing surfaces on a lubricant film. [18]. If debris becomes embedded in the Babbitt, wear would occur during start-ups [18].

2.3 Scoring

A severe form of wear is wire wooling. This happens when debris is packed into the Babbitt's surface and creates grooves on the journal surface. When this happens, the journal bearing is unable to create the wedge-film that would ensure smooth operation [18, 19-23]. Extensive damage is evident in both the shaft and the bearing, and black scabs often appear. As such, an alternative name - "black scab failure" - is also used to describe the problem. The product from the defect is similar to wire wool [18, 19-23].

The damage occurs when a particle is embedded in the bearing material and continues to rub against the bearing surface. The Temperature rises rapidly and

the dirt that comes into contact with the bearing material creates hard chromium carbides. The process continues until it eats through the journal collar. Wire wool failures are often encountered in bearings made with white metal or copper [17]. Figure 2-3 shows a wire wool damaged journal collar.

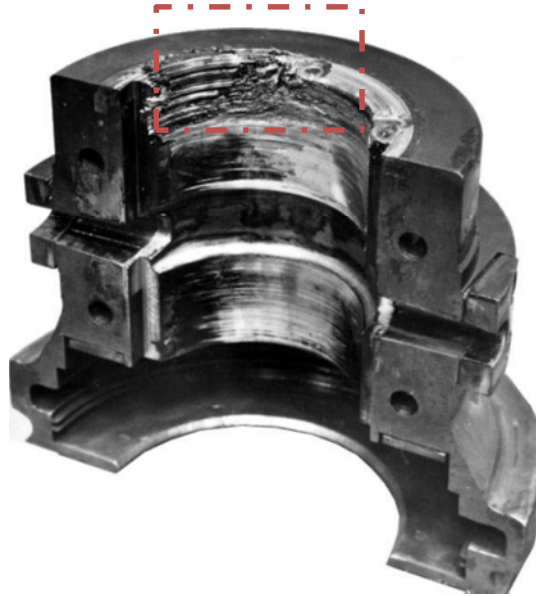


Figure 2-3 Wire wool Damaged Collar (Neale Consulting Engineers)

There are several ways in which scoring can occur in hydrodynamic bearings.

2.3.1 Scoring of White Metal Journal Bearing

Debris or particles usually end up embedded on the journal surface. This usually happens when lubricants are contaminated with minute, hard particles. They are small enough to enter the oil film, but slightly oversized to transcend without bridging the film. The small particles that cause the damage might not be visible, but the grooves created are evident around the bearing circumference. Scoring will occur if there is inadequate filtration in the lubricant or the lubricant has not been changed. Figure 2-4 shows an example of how a scored journal bearing face would appear.

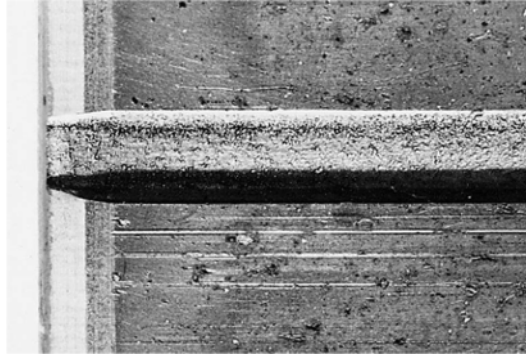


Figure 2-4 Scoring of White Metal Journal Bearing [17]

2.3.2 Scoring of Lead-Bronze Journal Bearing

The scoring on the lead-bronze is severe, and the scoring follows the path of the rotation. The cause of this type of scoring is contaminated lubricants. Failure to clean out the crankcase before starting the machine could also cause the scoring. Figure 2-5 shows the scoring pattern on a lead-bronze journal bearing.

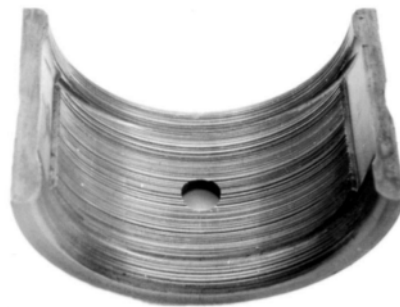


Figure 2-5 Scoring Lead-Bronze Journal Bearing [17]

2.4 Erosion

Erosion damage often looks like Babbitt voids, with smooth edges usually following the direction of the rotation. The problem would most likely occur in stationary bearings. When Babbitts are affected, they are often thought to be damage caused by cavitation. However, since the erosion is caused by an obstruction in lubricant flow, the damage could be widespread. The solution to the problem is to eliminate the cause of obstruction and ensure that the flow of lubricants is streamlined through the bearing [16, p.23].

2.4.1 Erosion damage of White Metal Journal Bearing

Erosion damage in this case is caused by small particles managing to enter through the oil feed holes. When erosion damage happens, the colour of the white metal is dulled and some spots appear abrasive. Unlike in scoring, the dirt particles that cause erosion are smaller than the minimum oil film thickness. The problem could also be confused with cavitation damage. As such, it is recommended that further microscopic examination be considered [17] Figure 2-6 illustrates erosion damage seen in the white metal journal bearing.

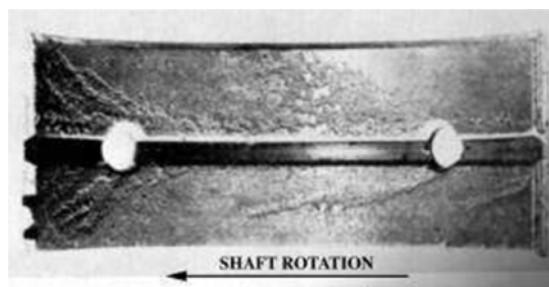


Figure 2-6 Erosion Damage of White Metal Journal Bearing (Neale Consulting Engineers)

2.4.2 Erosion damage to the Axial Oil Grooves in a Journal Bearing

As in the previous discussion, contaminated lubricant is the common cause of erosion. In this example, the particles are far too large to enter the oil film; particles trapped in the groove and eat away at the edges. Usually, bearings have grooves chiselled at 45 degree angles to allow extraneous particles to escape. Aside from contaminated lubricants, poor bearing design can also cause the problem. Figure 2-7 shows a sample where the axial grooves are damaged.

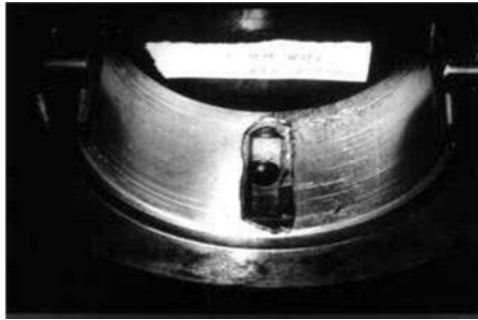


Figure 2-7 Axial Oil Groove Damage (Neale Consulting Engineers)

2.5 Corrosion

Corrosion attacks the bearing lining and creates latticework patterns. The affected area appears faded or discoloured, and rust can also occur at the collars. This defect can be differentiated from electrical pitting because the patterns are random and uneven. Corrosive materials can contaminate the lubricant through: decomposition of oil additives; acidic oxidation products formed in service; water or coolant in lube oil; and direct corrosive contamination. To avoid corrosion, housing seals, oil additive packages, and oil reservoir operating temperatures must be reviewed [16, p.24].

2.6 Lubricant Degradation Deposits

The damage to the bearing is caused by the chemical composition of the corrosion inhibitors added to lubricants. In Figure 2-8, the substance that damaged the bearing is a deposit of Ammonium Succinate - this is formed because there is a leak in the oil barrier shaft seals. This chemical is commonly used in turbine lubricants as an anti-corrosive additive. It is quite difficult to avoid corrosion as corrosive materials also form part of the composition of lubricants. The best way is for lubricant companies to reformulate their lubricants, minimizing the usage of corrosive additives [17]. Finally, to ascertain what type of chemicals attack the bearing surface, the diagnosis should include chemical analysis.

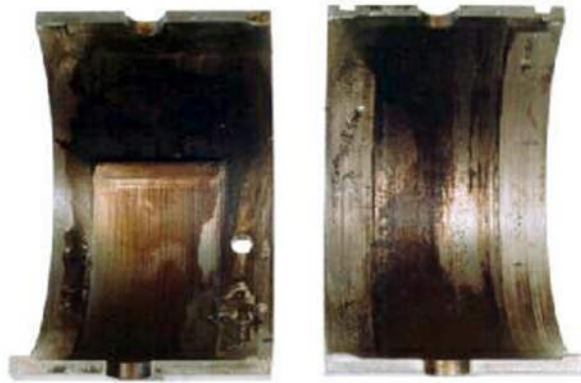


Figure 2-8 Bearing Surface with Ammonium Succinate [17]

2.7 Overheating

Overheating is manifested as discoloration, wiping or deformation [16, p.18]. Overheating is the result of improper lubricant selection; inadequate lubricant supply; interrupted fluid film; and boundary lubrication (p.18). Other causes of overheating include improper bearing selection; HP lift system failure; [poor collar, runner or journal surface finish; insufficient bearing clearance; excessive load; over-speed; and harsh operating environment.] [16, p.18]

To remedy the problem, one must ensure that there is enough lubricant for the bearing. This can be verified at the bearing manufacturer. Cracks must be eliminated and the shoe surface restored to its original condition. If within tolerance values, the bearings could be reused [16, p.18-19]. Figure 2-9 shows how overheating can damage metal surfaces.

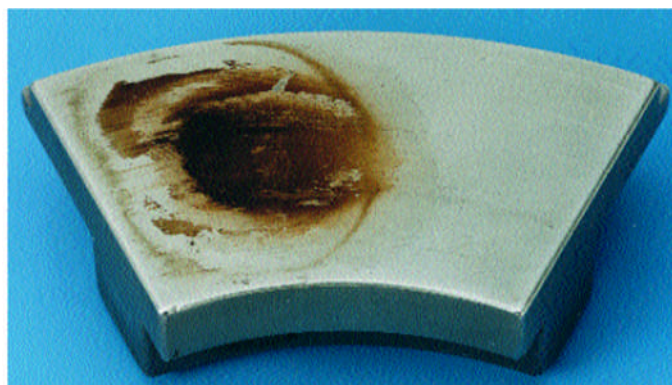


Figure 2-9 Damage due to Overheating [16]

2.8 Electrical Pitting

Electrical pitting usually appears as rounded pits on the bearing lining, and the colour of the pitting is different from the original surface. Pitting will occur in areas where lubricant is sparse and the damage progresses into abrasions until the bearing surface is unable to support oil films. The bearing may recover but the deterioration of the part continues. Electrical pitting is usually the consequence of intermittent electrical arcing between stationary and moving parts. The most severe damage usually occurs at the Babbitt. Another possible source of electrical pitting is electrostatic or electromagnetic. Electromagnetic shaft current causes more damage than electrostatic shaft current - deterioration is slower in electrostatic shaft current. Electrostatic shaft current is caused by charged lubricant, charged drive belts, or impinging particles. [16, pp.19-20]

Grounding brushes or straps can remove the electrostatic shaft current. Bearing isolation is also recommended. [16, p.20] To eliminate electromagnetic shaft current, the components should be demagnetised, lubricants changed, and all oil reservoirs should be thoroughly cleaned and flushed. Bearing finish must be refurbished and the journal shoes replaced [16, p.21]. Figure 2-10 illustrates electric pitting.

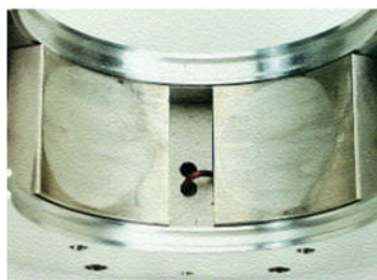


Figure 2-10 Electric Pitting (Kingsbury)

2.9 Fatigue

Fatigue may manifest itself in the Babbitt as hairline cracks. The cracks would appear to be in the same direction as the rotation. Fatigue usually occurs when there is a concentrated cyclical load putting much stress on the part. Conditions

that promote concentrated cyclical load include misalignment; journal eccentricity; imbalance; bent shaft; thermal cycling; and vibration. [16, p.21] Another cause of fatigue is extreme temperature - when the bearing is subject to unusually high temperature, the material composition is weakened [16, p.22].

To remedy the problem, the lubricant must be replaced, and all the parts of the assembly should be thoroughly cleaned and flushed. The Babbitt and the shoes should be replaced if severely damaged. If not, puddle-repair and refinishing should be applied [16, p.22, 19-23].

2.9.1 Fatigue of White Metal Journal Bearing

Bruising and cracking may show up on the surface of the bearing subjected to extreme loading. If the fatigue is confined to one area of the bearing, this implies that there is a misalignment in the installation of the bearing. The only solution to this problem is to reduce the load. Figures 2-11 and 2-12 show different stages of fatigue in white metal journal bearing.



Figure 2-11 Initial Deterioration (Neale Consulting Engineers)

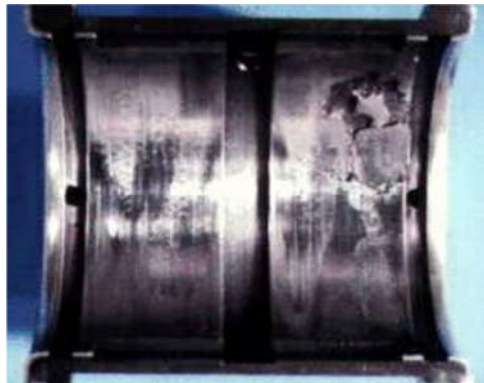


Figure 2-12 Severe Damage to the Bearing (Neale Consulting Engineers)

2.9.2 Premature Fatigue of White Metal Journal Bearing Caused by Vibration

In premature fatigue, aside from hairline cracks, the material also has bubble-like formations at the central oil groove. This is a manifestation of poor bonding and the vaporization of the oil trapped within the cast-iron bearing [17]. The problem is caused by inadequate bonding of the white metal of the bearing to the bearing shell, thus reducing the fatigue strength [17]. To prevent the problem, bearing shells must not be made from cast iron. Cast iron has properties incompatible with white metal: the porosity of the metal absorbs oil and makes it difficult to remove when the bearing is retooled. An ultrasonic test should be conducted before the two parts are rebounded [17]. Figures 2-13 and 2-14 show the condition of a bearing with premature fatigue.

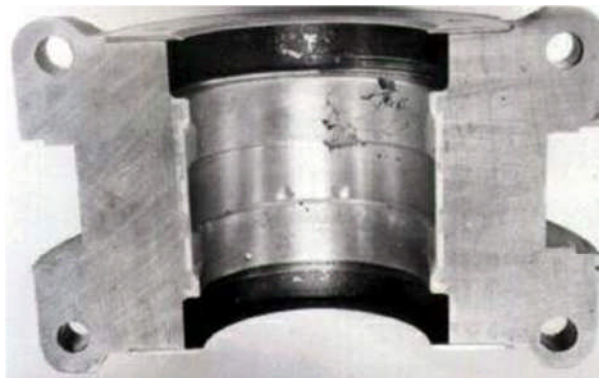


Figure 2-13 Poor bonding of white metal and the bearing shell (Neale Consulting Engineers)

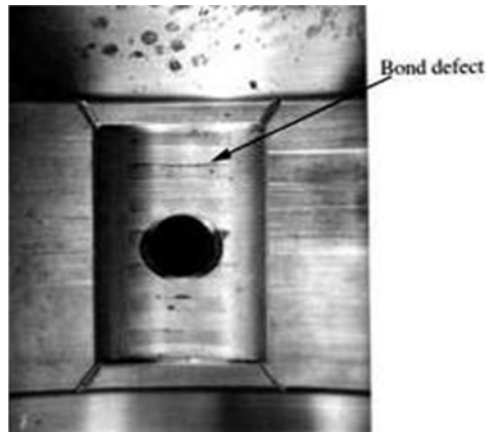


**Figure 2-14 Bonding defect between the edges of the dovetail and white metal
(Neale Consulting Engineers)**

2.10 Fretting

Fretting occurs when the bearing is subjected to external vibration. Usually, the damage occurs in an area where there is least lubrication. The damage is also caused by the oscillatory move at the points of contact between the journal and the bearing [17]. Figure 2-15 shows a severely damaged bearing with fretting at the centre.

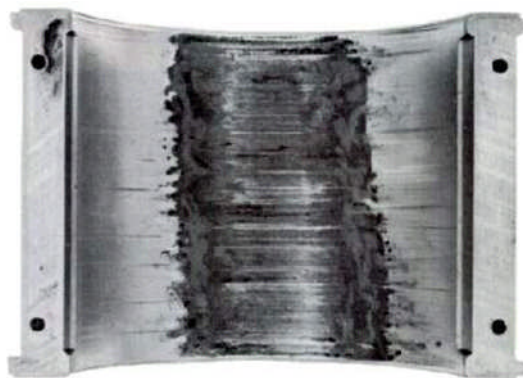


Figure 2-15 Fretting (Neale Consulting Engineers)

2.11 Wiping

Wiping usually happens when the bearing loses hydrodynamic fluid while operational. Different metals are fused together because the molten part fuses with the cooler one. Wiping can be triggered by electrical erosion or fatigue

damage [17]. Changing the bearing into one that could accommodate a heavier load without verifying the cause of wiping should not be considered.

2.11.1 Start-up Wiping Damage

This type of damage is usually caused by rubbing of the journal before the hydrodynamic lubricating film has developed. [17, 19-23] Smearing occurs along the direction of the rotation. However, this type of damage usually occurs in heavy loads. Minor wiping can sometimes be helpful in assisting the run-in as marks found at the edges signal misalignment. Figure 2-16 illustrated a bearing subjected to vertical loads.

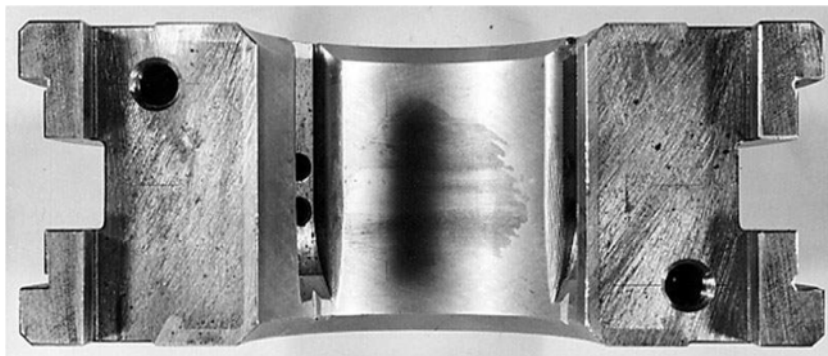


Figure 2-16 Start-up Wiping (Neale Consulting)

Knowing the basics about failure characteristics of parts could save the company millions and provide an immediate solution to the problem. Understanding the causes of each defect or damage can make one more cautious about abusing the material or putting substandard materials into the assembly. Adequate and clean lubrication are the keys to preventing the majority of failures.

2.12 Shaft

Shafts with a circular cross section are the simplest rotating elements inside the transmission designed to incorporate the bearings. Fatigue is the most common type of failure associated with transmission shafts. The fatigue crack normally initiates from the critical points where stress concentration is higher, i.e. the stress raiser points [22]. The keyways roots and the edges resulting from a

sharp change in the cross section area are among the most typical stress raiser points in circular cross section shafts [23], Berndt and Bennekom [24] categorised shaft fatigue into three groups, among which rotational bending is the most observed type of fatigue in the circular shaft; rotational bending fatigue normally occurs when the shaft is subjected to the oscillating tensile and compressive stress within its circumference.

2.13 Theory and Design of the Journal Bearing

In Journal bearings, there are five main input variables in the system: geometry, load, rotational speed, temperature and oil viscosity. Due to the dimensions that are used for journal length, inner diameter and radial clearance the geometry is fixed. Figure 2-17, represents the relationship between these factors and how the pressure from the load can distribute.

Between the rotating shaft and the bearing, there is an oil film used as a lubricant. This oil film, $h_{\min}$, is found at the point which is in line with the centre of the shaft and journal bearing, and is known as the minimum film thickness. Oil types are Castrol Alpha SP 460, Alpha SP 220 and Alpha SP 68. These oil types are a range of high quality solvent refined mineral oils; fortified with sulphur-phosphorous compounds to provide anti-wear and extreme pressure properties.

There are two main functions of a lubricant [25]:

1. **Lubrication.** The lubricant must be able to allow the motion of the moving part. Its presence avoids the metal-to-metal contact. The surface of metals is not plain, but is irregular with peaks and valleys. When two metals in relative movement are in contact, there is a severe friction effect between both surfaces, resulting in noise, losses and damage
2. **Shock Absorber.** The oil film also has stiffness and damping properties, which will avoid punctual hits being transmitted to the shaft. This task is especially important at the minimum oil film thickness point, where the oil layer must be capable of keeping both elements separated. The lubricant must be capable of absorbing impacts.

, there is a relationship between temperature and oil viscosity; the relationship depends on the oil type. Once temperature is measured, the viscosity can be read in the viscosity-temperature graphs provided by the oil manufacture. Their temperature-viscosity relationship can be assumed as linear in a logarithmic graph, see appendix C. As soon as the kinematic viscosity (ν) is calculated, the dynamic viscosity is obtained through the relationship:

$$\mu = \rho \times \nu \quad (2-1)$$

The density of the oil types at 20°C is shown here:

Table 2-1 Oil types density at 20°C [Appendix C]

Alpha SP 68	885 kg/m ³
Alpha SP 220	895 kg/m ³
Alpha SP 460	900 kg/m ³

Three different types of oil kinematic viscosity have been used in this research: Alpha SP68, with a base oil kinematic viscosity at 40°C and 100°C of 68cSt and 8.53cSt respectively; Alpha SP220, with a base oil kinematic viscosity at 40°C and 100°C of 220cSt and 18.7cSt respectively; and Alpha SP460, with a base oil kinematic viscosity at 40°C and 100°C of 460cSt and 30.5cSt respectively. This is demonstrated in Figure 2-18. This plot gives viscosities at a range of temperatures during tests.

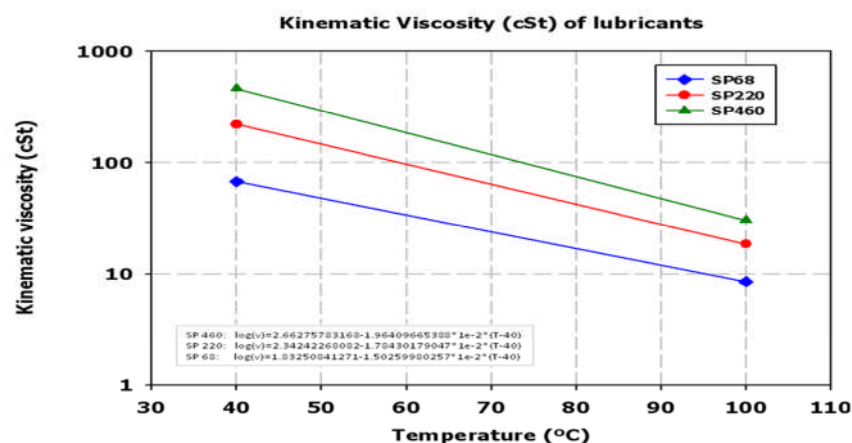


Figure 2-17 Kinematic viscosity of lubricant

Nevertheless, oil viscosity has been assumed to be constant with the temperature. The dynamic viscosity, μ , is the tangential force per area needed

to move a body in a tangential direction with respect to a fixed one and at a given speed. Thus, this is the viscosity value employed to calculate dynamic-associated factors, such as shear stresses and Sommerfeld Number.

2.13.1 Minimum Film Thickness

The distance, h_{min} , is found at the point which is in line with the centre of the shaft and journal bearing, as illustrated in the following diagram, and is known as the minimum film thickness. It is also noted that this point on the bearing is always found at a point away from the bottom of the arc in the direction of rotation.

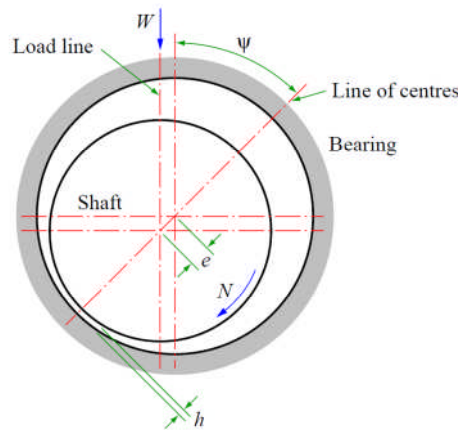


Figure 2-18 Illustration of rotation [25]

Minimum film thickness will be derived from the non-dimensional constants $\frac{h_{min}}{c}$ (depends on the Sommerfeld Number) and can be read on the Raimondi-Boyd graphs; see Raimondi-Boyd graphs in the appendix A. In order for the shaft to be allowed to build up minimum film thickness, there has to be a difference in the diameter of the shaft and the bearing, i.e. the bearing has to be slightly larger than the shaft.

2.13.2 Eccentricity Ratio

The eccentricity ratio, ϵ , is a fundamental parameter in the analysis of a hydrodynamic journal bearing. Its value, for an aligned bearing.

2.13.3 Power Loss

Power losses will be diverted from the non-dimensional constants (dependents on the Sommerfeld Number) read on the Raimondi-Boyd graphs; see Raimondi-Boyd graphs in the appendix A. These non-dimensional factors are obtained from the graphs. They will be $R \frac{f}{c}$, $\frac{h_{min}}{c}$ and angle ϕ is the location of the minimum film thickness. Once the constant $R \frac{f}{c}$ is known, the value of the friction factor f can be found. Then f factor is employed to calculate the Power Losses. In the case of a Newtonian fluid, there are friction losses as the journal bearing is rotating. Due to these losses, the temperature in the system increases. The friction factor f is obtained by means of graphs and depends on the Sommerfeld Number, this in turn depends on the operational conditions of the journal bearing such as; rotational speed, load, geometry and viscosity. Once the factor of friction is known, the power losses are calculated through the equation 2-2:

$$H = f \times P \times R \times \omega \quad (2-2)$$

where:

P is the specific pressure in Pa, is the relationship between the load and the area of the bearing ($L \times D$).

R is the radius of the bearing.

' ω ' is the rotating speed of the shaft in rad/s.

H is the power loss in watts

2.13.4 Pressure Distribution

The equation that defines that pressure distribution in a gap between two plates is known as the Reynolds generalised equation, see figure 2-20. To calculate this equation, it is assumed that [26]:

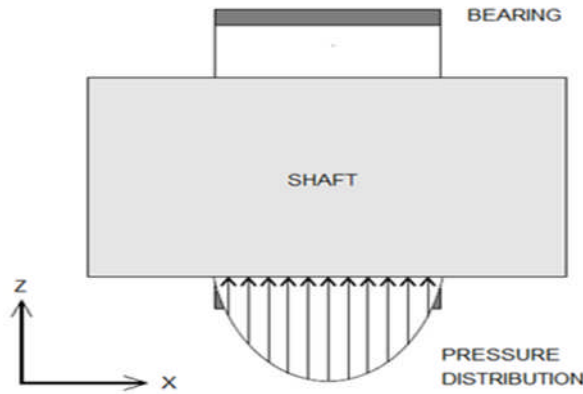


Figure 2-19 Pressure distribution [25]

- The effect of the slope on the film thickness is not considered, and it is assumed that the pressure distribution is the same as in the case of a developed journal bearing
- The fluid is considered as Newtonian, with constant density and constant viscosity in $\hat{x}$ direction
- Perfect adherence is assumed in the contact points of the surfaces – hence, the speed at the contact point between lubricant and inner surfaces is the same
- Gravity effect or inertial forces are not considered

Reynolds generalized equation for the distribution of pressures of a fluid between plates has the following expression [110]:

$$\begin{aligned}
 \frac{\partial}{\partial x} \left(\frac{\rho h^3}{12\mu} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left(\frac{\rho h^3}{12\mu} \frac{\partial p}{\partial y} \right) = & \quad (2-3) \\
 = \frac{\partial}{\partial x} \left(\frac{\rho h(u_a + u_b)}{2} \right) + \frac{\partial}{\partial y} \left(\frac{\rho h(v_a + v_b)}{2} \right) + \\
 + \rho(w_a + w_b) - \rho u_a \frac{\partial h}{\partial x} - \rho v_a \frac{\partial h}{\partial y} + h \frac{\partial p}{\partial t}
 \end{aligned}$$

where

$\hat{u}$, $\hat{v}$ and $\hat{w}$ are the coordinates of the velocity vector, ρ

' μ ' is the dynamic viscosity of the fluid,

ϕ is the pressure and

h is the lubricant film thickness in the z-direction

Since the analytical solution for the Reynolds equation exists for particular solutions, it is necessary that it is solved by numerical techniques. This equation was solved through finite differences by Raimondi and Boyd (1958), showing the final results as a function of the Sommerfeld number [25, 26].

2.13.5 Sommerfeld Number

The Sommerfeld number is a non-dimensional number which is a characteristic of a journal bearing in given working conditions and is calculated by the equation 2-4:

$$S = \mu \times \left[\frac{N}{P} \times \left(\frac{R}{c} \right)^2 \right] 10^{-9} \quad (2-4)$$

Journal Speed	N (rev/s)
Radial Clearance	c (m)
Journal Diameter	D (m)
Journal Axial length	L (m)
Specific Pressure on Bearing	P (lb/in²)
Oil Viscosity @ 20°C	μ (kg/m³)

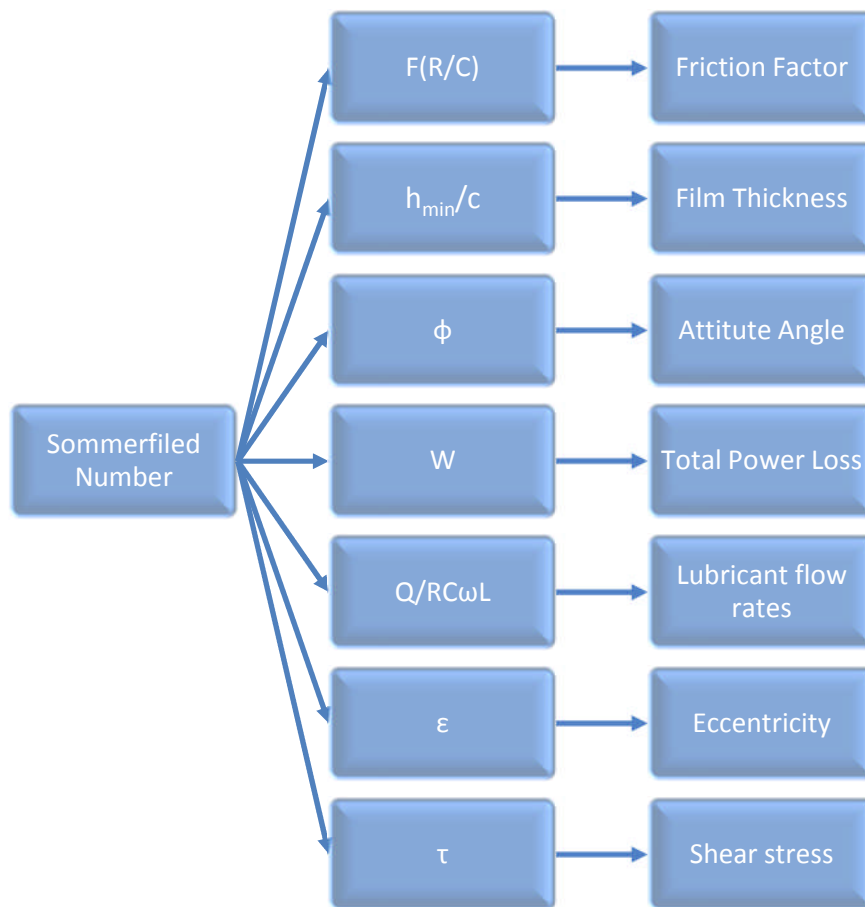
Table-2-2 Sommerfeld number characteristic

The film thickness, like the power losses, can be calculated with the Raimondi-Boyd graph, which is dependent on the Sommerfeld Number and is therefore dependent on the known input variables. This number is characteristic of every single bearing in every working condition. As soon as the Sommerfeld number is calculated, the rest of the parameters are found in different graphs. Another parameter in this relationship is eccentricity; although it is commonly calculated by other methods used to calculate film thickness, it is not necessary to calculate it. The film thickness can be directly calculated from the Sommerfeld Number in the Raimondi-Boyd graphs. Therefore, its appearance is merely for purposes of orientation [26].

It appears that, the behaviour of a radial bearing depends on the pressure distribution of the film thickness.

Note: graphs exist where a relationship is plotted between the Sommerfeld number and constants resulting from the Reynolds equation solution by Raimondi and Boyd. Those constants are shown in sketch 2-3

Table 2-3 Relationship between Sommerfeld number and other parameters



Power loss and shear stress will be calculated from the non-dimensional constants which depend on the Sommerfeld Number,. The Raimondi-Boyd graphs can be seen in appendix A.

The non-dimensional factors to be obtained from the graphs will be $F(R/C)$, h_{min} and angle ϕ of location of the minimum film thickness. The geometrical significance of the constants is explained in the sketch, shown in figure 2-21 [27]:

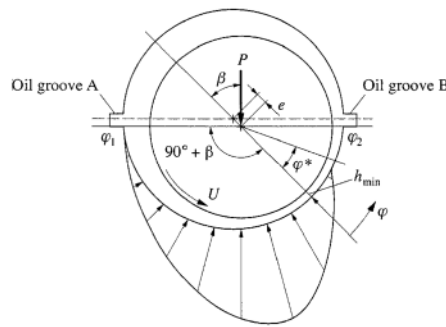


Figure 2-20 Journal bearing with two axial oil grooves

Once the constant Rf/c is known, the value of the friction factor, f_0 is calculated. The friction factor is a dimensionless parameter which is determined from the eccentricity ratio and Sommerfeld number of the bearing, and is employed to calculate the power losses of the bearing [27].

2.13.6 Clearance

The test bearing had a radius of 40mm, length of 80mm and a measured radial clearance of 0.17mm. The value for radial clearance, $\tilde{C}$ must be in the region of 0.4% to 0.5% of the shaft diameter. For this reason, a midpoint value was decided: i.e. $0.17\text{mm} = 0.42\% D/2$, where D =shaft diameter (mm).

2.13.7 Specific Pressure

The specific pressure (P) is one of the most important parameters in developing a hydrodynamic bearing, and based on the mechanical engineering design, it can be calculated as the imperial equation 2-5:

$$P = \frac{W}{2RL} \quad (2-5)$$

where:

W : applied load R : Radius of bearing L : Length of bearing

L/D ratio has been used to decrease the chance of a low specific pressure to adequate minimum film thickness being developed.

2.13.8 Journal Bearing Data

The hydrodynamic bearing test rig employed for this research has an operational speed range of 500rpm to 4500rpm, with a maximum load capability of 20kN.

Table 2-4 Journal Bearing Data

Data		Unit	
Journal Speed	w [rpm]		500
			1500
			2500
			3500
			4500
Radial Clearance	c[m]		0.00017
Journal Diameter	D [m]		0.08
Journal Axial length	L [m]		0.08
Bearing Load	P (kN)		2
			4
			10
Ambient Temperature	[°C]		20
Oil Viscosity @ 20°C	kg/m ³	Alpha SP 68	885
		Alpha SP 220	895
		Alpha SP 460	900
Top Brg. Arc	Deg		120
Bottom Brg. Arc	Deg		120

The material of the hydrodynamic bearing was bronze. The test bearing had a radius of 40mm, length of 80mm and a measured radial clearance of 0.17mm. See Table 2-4 for journal bearing test rig data.

2.14 Shear Stresses in the Fluid

A shear stress is defined as a stress that attempts to cause an internal friction to parallel or tangential faces of materials, and slide them relative to each other. In hydrodynamic bearings, when Newtonian fluids are moving along the solid border (boundary), a shear stress will occur [138]. The no-slip condition dictates that the speed of the fluid at the boundary is 0, but at some height from the boundary, the flow speed must equal that of the fluid. The region between these two points is known as the boundary layer. For all Newtonian fluids, the

shear stress is proportional to the strain rate in the fluid where the viscosity is the constant of proportionality. However, for non-Newtonian fluids, this is no longer the case as, for these fluids, the viscosity is not constant. The shear stress is imparted onto the boundary pursuant to this loss of velocity [139 to 146].

$$\tau_w = -\mu \frac{\partial u}{\partial y} \Big|_{y=0} \quad (2-6)$$

where

μ is the dynamic viscosity of the fluid,

u is the velocity distribution of the fluid along the boundary

y is the height of the boundary.

To calculate the shear stresses in the fluid, the formula postulated by Newton for straight and parallel flow is used. It shows a linear relationship between the shear stresses and the gradient of velocities, see figure 2-22.

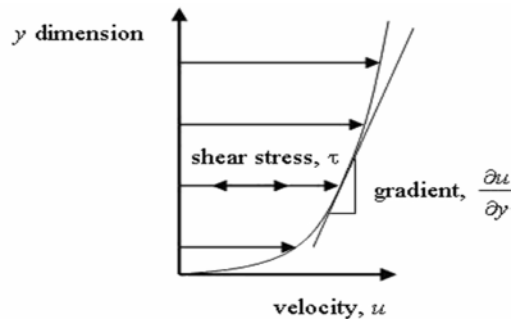


Figure 2-21 Shear Stress distribution

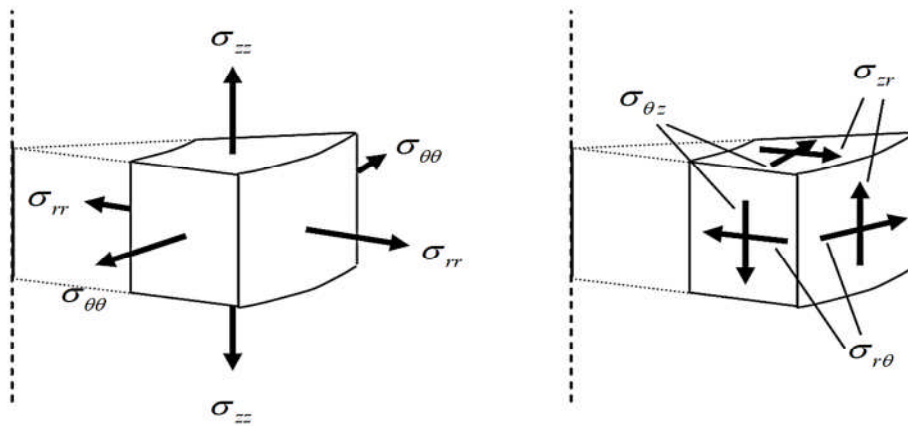


Figure 2-22 Normal and shear stresses in cylindrical coordinates

In the following calculation for shear stress and velocity profile, the inner cylinder (shaft) rotates at a constant speed, and the outer cylinder (hydrodynamic bearing) is stationary. The flow (oil) is assumed to be laminar. To obtain an expression for the velocity profile and shear stress distribution, the continuity, Navier-Stokes, and tangential shear stress equations have been simplified. These equations are written for incompressible flow with constant viscosity, which necessitates some terms being used to simplify the basic Navier-Stokes. These terms are listed below for further calculations [147, 148]:

- a) Steady flow: this assumption is constant for the angle of the outer cylinder (hydrodynamic bearing), which eliminates time variations in fluid properties and equations.
- b) Incompressible flow: by making this assumption, the governing equations of fluid flow can be simplified significantly, which causes the elimination of space variations in density ($\rho = \text{constant}$).
- c) Periphery symmetric flow: this assumption causes no changes in the properties in angular directions (θ and $\partial(.)/\partial\theta = 0$).
- d) No flow or variation of properties in the z direction: this term can cause obliteration in the z component of the continuity, Navier-Stokes, and tangential shear stress equations, except for hydrostatic pressure distribution ($\vartheta_z = 0$ and $\partial(.)/\partial z = 0$).

All of these assumptions will be applied to find an expression for velocity profile and shear stress distribution in a hydrodynamic bearing. The continuity equation in cylindrical coordinates (see figure 2-23) for constant density is [147]:

$$\frac{1}{r} \frac{\partial}{\partial r} (r v_r) + \frac{1}{r} \frac{\partial}{\partial \theta} v_\theta + \frac{\partial}{\partial z} v_z = 0 \quad (2-7)$$

Normal and shear stresses in cylindrical coordinates for constant density and viscosity:

$$\sigma_{rr} = -p + 2\mu \frac{\partial v_r}{\partial r} \quad \tau_{r\theta} = \mu \left[r \frac{\partial}{\partial r} \left(\frac{v_\theta}{r} \right) + \frac{1}{r} \frac{\partial v_r}{\partial \theta} \right] \quad (2-8)$$

$$\sigma_{\theta\theta} = -p + 2\mu \left[\frac{1}{r} \frac{\partial v_\theta}{\partial \theta} + \frac{v_r}{r} \right] \quad \tau_{\theta z} = \mu \left[\frac{\partial v_\theta}{\partial z} + \frac{1}{r} \frac{\partial v_z}{\partial \theta} \right] \quad (2-9)$$

$$\sigma_{zz} = -p + 2\mu \frac{\partial v_z}{\partial z} \quad \tau_{zr} = \mu \left[\frac{\partial v_r}{\partial z} + \frac{\partial v_z}{\partial r} \right] \quad (2-10)$$

r momentum:

$$\begin{aligned} & \rho \left[\frac{\partial v_r}{\partial t} + v_r \frac{\partial v_r}{\partial r} + \frac{v_\theta}{r} \frac{\partial v_r}{\partial \theta} - \frac{v_\theta^2}{r} + v_z \frac{\partial v_r}{\partial z} \right] \\ &= \rho g_r - \frac{\partial p}{\partial r} + \mu \left[\frac{\partial}{\partial r} \left[\frac{1}{r} \frac{\partial}{\partial r} (r v_r) \right] + \frac{1}{r^2} \frac{\partial^2 v_r}{\partial \theta^2} - \frac{2}{r^2} \frac{\partial v_\theta}{\partial \theta} + \frac{\partial^2 v_r}{\partial z^2} \right] \end{aligned} \quad (2-11)$$

θ momentum:

$$\begin{aligned} & \rho \left[\frac{\partial v_\theta}{\partial t} + v_r \frac{\partial v_\theta}{\partial r} + \frac{v_\theta}{r} \frac{\partial v_\theta}{\partial \theta} - \frac{v_\theta v_r}{r} + v_z \frac{\partial v_\theta}{\partial z} \right] \\ &= \rho g_\theta - \frac{\partial p}{\partial \theta} + \mu \left[\frac{\partial}{\partial r} \left[\frac{1}{r} \frac{\partial}{\partial r} (r v_\theta) \right] + \frac{1}{r^2} \frac{\partial^2 v_\theta}{\partial \theta^2} + \frac{2}{r^2} \frac{\partial v_r}{\partial \theta} + \frac{\partial^2 v_\theta}{\partial z^2} \right] \end{aligned} \quad (2-12)$$

z momentum:

$$\begin{aligned} & \rho \left[\frac{\partial v_z}{\partial t} + v_r \frac{\partial v_z}{\partial r} + \frac{v_\theta}{r} \frac{\partial v_z}{\partial \theta} + v_z \frac{\partial v_z}{\partial z} \right] \\ &= \rho g_z - \frac{\partial p}{\partial z} + \mu \left[\frac{\partial}{\partial r} \left[r \frac{\partial v_z}{\partial r} \right] + \frac{1}{r^2} \frac{\partial^2 v_z}{\partial \theta^2} + \frac{\partial^2 v_z}{\partial z^2} \right] \end{aligned} \quad (2-13)$$

$$g_\theta = g_r = 0 \text{ and } g_z = -g \quad (2-14)$$

After assumptions (c) and (d) are applied, the continuity equation reduces to

$$\begin{aligned} \frac{1}{r} \frac{\partial}{\partial r} (r v_r) + \frac{1}{r} \frac{\partial}{\partial \theta} v_\theta + \frac{\partial}{\partial z} v_z \\ \frac{1}{r} \frac{\partial}{\partial r} (r v_r) = 0 \end{aligned} \quad (2-15)$$

Because $\partial(\cdot)/\partial\theta = 0$ and $\partial(\cdot)/\partial z = 0$ by assumptions (c) and (d), then the equation will change to

$$r v_r = \text{constant} \quad (2-16)$$

By knowing v_r is zero at the solid surface of cylinder (Bearing), and then v_r must be zero everywhere. e) The fact that $v_r = 0$ reduces the Navier-Stokes equations further, as pointed out by cancellations.

So, the shear stress distribution by the last assumption will reduce to:

$$\tau_{r\theta} = \mu \left[r \frac{\partial}{\partial r} \left(\frac{v_\theta}{r} \right) + \frac{1}{r} \frac{\partial v_r}{\partial \theta} \right] \quad (2-17)$$

And the final momentum equations reduced to:

$$-\rho \frac{v_\theta^2}{r} = -\frac{\partial p}{\partial r} \quad (2-18)$$

$$0 = \mu \left[\frac{\partial}{\partial r} \left[\frac{1}{r} \frac{\partial}{\partial r} [r v_\theta] \right] \right] \quad (2-19)$$

But, since $\partial(\cdot)/\partial\theta = 0$ and $\partial(\cdot)/\partial z = 0$ by assumption (c) and (d), then v_θ is a function of radius only, and

$$\frac{d}{dr} \left[\frac{1}{r} \frac{d}{dr} [r v_\theta] \right] = 0 \quad (2-20)$$

Integrating once:

$$\frac{1}{r} \frac{d}{dr} [r\vartheta_\theta] = c_1 \quad (2-21)$$

Then

$$\frac{d}{dr} [r\vartheta_\theta] = c_1 r \quad (2-22)$$

By integrating again, a liquid velocity profile for Incompressible flow is given, as described in the following equation:

$$r\vartheta_\theta = c_1 \frac{r^2}{2} + c_2 \quad \text{or} \quad \vartheta_\theta = c_1 \frac{r}{2} + c_2 \frac{1}{r} \quad (2-23)$$

For Hydrodynamic Bearing test rig, the boundary conditions which are needed to evaluate c_1 and c_2 :

$$\vartheta_\theta = \omega R_2 \quad \text{at} \quad r = R_2 \quad (2-24)$$

$$\vartheta_\theta = 0 \quad \text{at} \quad r = R_1 \quad (2-25)$$

where:

- The velocity in the inner face of the journal bearing (static part) is obviously nil
- The velocity of the oil when in contact with the rotating shaft has the same velocity the shaft has,
- The slope of the velocity field has a minimum friction in the journal bearing contact zone, which means the velocity field line is tangential to the inner surface. It also indicates that there will be no shear stresses in the static part

By substituting these boundary conditions in the velocity profile equation

$$\omega R_2 = c_1 \frac{R_2}{2} + c_2 \frac{1}{R_2} \quad (2-26)$$

$$0 = c_1 \frac{R_1}{2} + c_2 \frac{1}{R_1} \quad (2-27)$$

After manipulation

$$c_1 = \frac{2\omega}{1 - (\frac{R_1}{R_2})^2} \text{ and } c_1 = \frac{-\omega R_1^2}{1 - (\frac{R_1}{R_2})^2} \quad (2-28)$$

By substituting in to expression, a liquid velocity profile for Incompressible flow is given, as described in the following equation:

$$v_\theta = \frac{\omega r}{1 - (\frac{R_1}{R_2})^2} - \frac{\frac{\omega R_1^2}{r}}{1 - (\frac{R_1}{R_2})^2} = \frac{\omega R_1}{1 - (\frac{R_1}{R_2})^2} \left[\frac{r}{R_1} - \frac{R_1}{r} \right] \quad (2-29)$$

$$v_\theta = \frac{\omega R_1}{1 - (\frac{R_1}{R_2})^2} \left[\frac{h_0}{R_1} - \frac{R_1}{h_0} \right] \quad (2-30)$$

After swapping velocity profile equation in equation (1), and with the boundary conditions, the shear stress distribution for a Newtonian fluid can be expressed as the following equation:

$$\tau_{r\theta} = \mu r \frac{d}{dr} \left(\frac{v_\theta}{r} \right) = \mu r \frac{d}{dr} \left\{ \frac{\omega R_1}{1 - (\frac{R_1}{R_2})^2} \left[\frac{r}{R_1} - \frac{R_1}{r} \right] \right\} \quad (2-31)$$

$$\tau_{r\theta} = \mu r \frac{\omega R_1}{1 - (\frac{R_1}{R_2})^2} (-2) \left[-\frac{R_1}{r^3} \right] \quad (2-32)$$

$$\tau_{r\theta} = \mu \frac{2\omega R_1^2}{1 - (\frac{R_1}{R_2})^2} * \frac{1}{r^2} \quad (2-33)$$

$$\tau_{r\theta} = \mu \frac{2\omega R_1^2}{1 - (\frac{R_1}{R_2})^2} * \frac{1}{h_{\min}^2} \quad (2-34)$$

where:

- $\tau_{r\theta}$ is shear stress distribution in the bearing
- μ is dynamic viscosity
- R_1 is shaft radius
- R_2 is bearing radius
- $h_{\min}$ is film thickness

The mathematical calculation of the shear stresses in the fluid, for straight and parallel flow is formulated in equation 2-34. The shear stress value is inversely proportional to the square of minimum film thickness base on the parabolic equation.

2.15 Calculation methods for hydrodynamic journal bearings

This part provides charts and calculation methods for the design of a hydrodynamic journal and for achieving greater accuracy from the comparison of these different types of method. The comparison is undertaken with an input load condition of 2kN with oil density of 900 kg/m³ for the following methods:

- ESDU Method
- Bannister Method
- JB Data Book Method (JB)
- RAIMONDI BOYD Method (RB)

The Raimondi Boyd (RB) [26] graphs, Bannister [28] and JB Databook (JB) [27] methods are set in the calculation of the Sommerfeld Number, and from that value, the other working conditions, such as friction factor, flow rate, eccentricity and minimum film thickness, are obtained by reading the graphs. The main difference is that, in the Bannister and JB method, it is necessary to calculate the eccentricity (ϵ) to calculate the minimum oil film thickness ($h_{\min}$) whereas, in the RB graphs based method, it is not necessary $\ddot{h}_{\min}$ is calculated from the non-dimensional $h_{\min}/c$ taken from the graphs. The ESDU [25] method is also different from the other established methods. This method starts with the calculation of the non-dimensional load factor ($W\phi$), and then again, the rest of the values are calculated through non-dimensional factors taken from graphs. Tables 2-5 to 2-8 demonstrate the comparisons for different types of minimum film thicknesses and on shear stress and power loss, using JB, the base method for varying speed conditions at the load of 2kN.

Table 2-5 RAIMONDI Boyd graphs based method at 2kN

N (rpm)	S	ε	Power losses (kW)	Shear stress (MPa)	$h_{\min}$ (μm)
750	0.94	N/A	0.34	0.017	144.5
1500	1.23	N/A	0.90	0.021	149.6
2500	1.27	N/A	1.56	0.022	151.3
3500	1.05	N/A	1.89	0.019	146.2
4500	0.91	N/A	1.98	0.017	142.8

Table 2-6 ESDU METHOD for varying speed conditions at 2kN

N (rpm)	S	ε	Power losses (kW)	Shear stress (MPa)	$h_{\min}$ (μm)
750	1.05	0.19	99.22	0.0182	137.7
1500	0.81	0.125	267.11	0.0219	148.75
2500	0.78	0.11	462.58	0.022	151.3
3500	0.95	0.13	526.78	0.0188	147.9
4500	1.08	0.145	585.94	0.0167	145.35

Table 2-7 BANNISTER METHOD for varying speed conditions at 2kN

N (rpm)	S	ε	Power losses (kW)	Shear stress (MPa)	$h_{\min}$ (μm)
750	0.94	0.22	0.29	0.0189	132.6
1500	1.23	0.165	0.78	0.023	141.95
2500	1.27	0.16	1.35	0.023	142.8
3500	1.05	0.19	1.56	0.020	137.7
4500	0.91	0.23	1.71	0.018	130.9

Table 2-8 JB DATABOOK METHOD for varying speed conditions at 2kN

N (rpm)	S	ε	Power losses (kW)	Shear stress (MPa)	$h_{\min}$ (μm)
750	0.94	0.14	0.13	0.017	146.2
1500	1.23	0.12	0.35	0.021	149.6
2500	1.27	0.08	0.61	0.021	156.4
3500	1.05	0.11	0.70	0.018	151.3
4500	0.91	0.15	0.77	0.016	144.5

Note that in ESDU method, the non-dimensional power losses factor is calculated instead of the Sommerfeld Number. In the following table 2-9, relative changes are demonstrated in minimum film thicknesses for different calculation methods with an average change of 4% at different speed conditions.

Table 2-9 The Relative percentage changes in Film Thickness (MFT) for varying Speed conditions

N (rpm)	$\Delta X\% = \frac{MFT_{JB} - MFT_{RB}}{MFT_{JB}}$	$\Delta X\% = \frac{MFT_{JB} - MFT_{Banister}}{MFT_{JB}}$	$\Delta X\% = \frac{MFT_{JB} - MFT_{ESDU}}{MFT_{JB}}$
750	1.16	9.3	5.81
1500	0	5.11	0.57
2500	3.26	8.7	3.26
3500	3.37	8.99	2.25
4500	1.18	9.41	-0.59

The results of this comparison are shown in the following comparison figures 2-24 to 2-26. As can be seen in these figures, the RB method results follow a similar tendency to the other methods. Therefore, it is accurate enough to consider the results obtained by means of this method as being results that have physical sense. Conclusions will be drawn by considering the results from the RB graphs based method, which is more accurate and reliable.

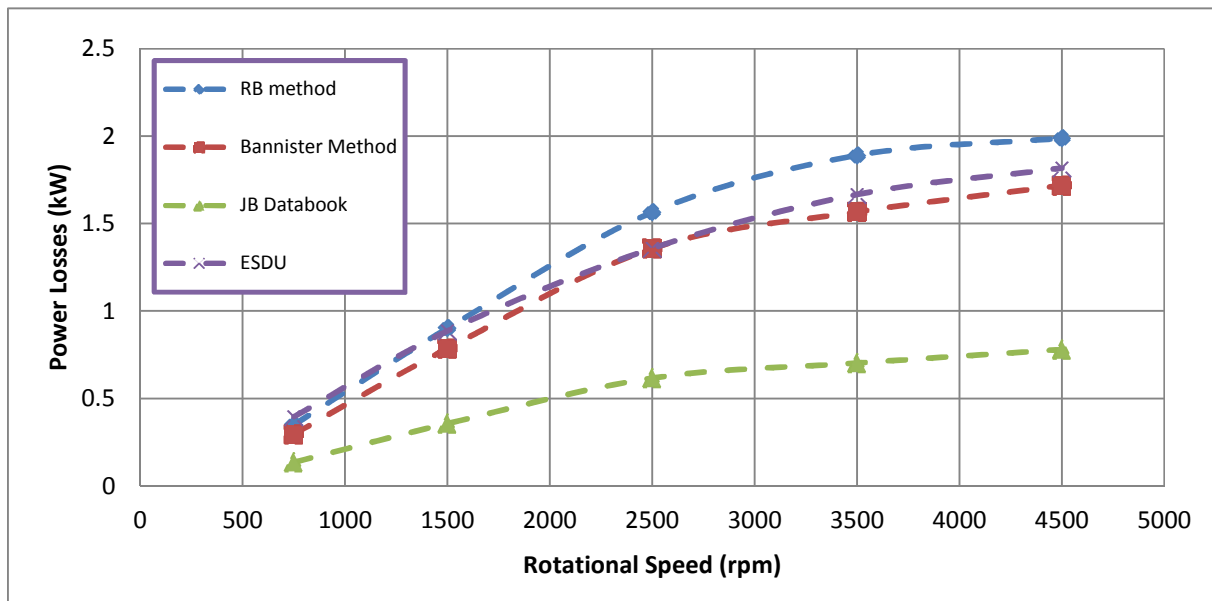


Figure 2-23 Comparison graph for power losses Vs rotational speed at 2kN

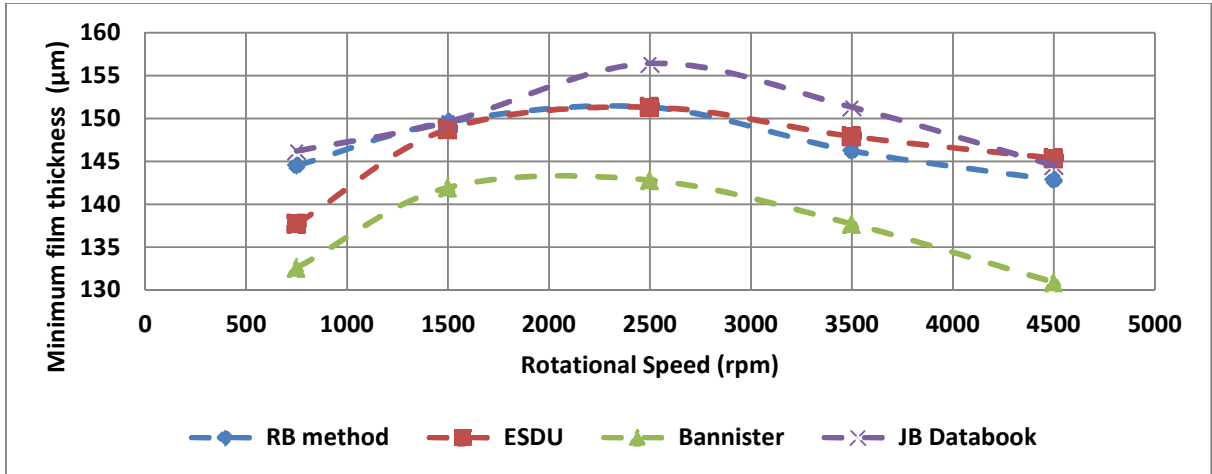


Figure 2-24 Comparison graph for Minimum film thickness Vs rotational at 2kN

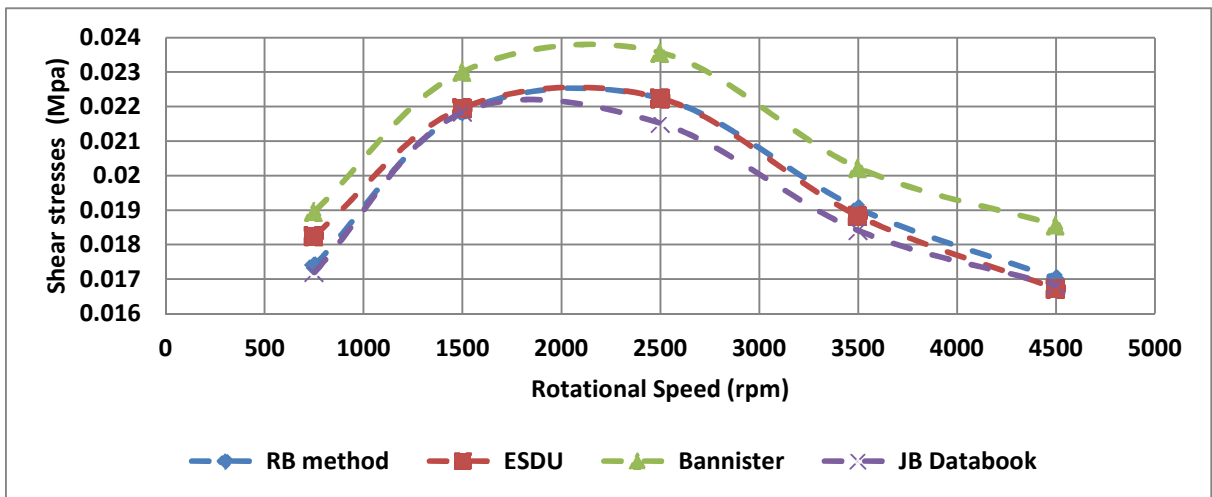


Figure 2-25 Comparison graph for Shear stresses Vs rotational speed at 2kN

3 Acoustic Emission

Acoustic Emission (AE) is an elastic wave which is produced as a result of a swift discharge of energy from a source within a material that is compelled by an externally applied stimulus [30]. It is a developing technological field and several research projects are in progress, to explore more options about, and possibilities of, AE [29]. Acoustic Emission technology (AET) has several applications: it can be used for tank bottom testing, pipeline testing, railroad tank car testing, reactor & high energy piping testing, aging aircraft evaluation, advanced materials testing, vessel testing and so on [31].

One further application of AE is in the field of journal bearings. Journal bearings, which are used in rotating machines, have a perilous weakness in that whenever they face an insufficient supply of lubricant, there is always the possibility of sudden shutdown or other faults developing [34].

In order to cope with this problem, several non-destructive testing methods, have been applied to reduce the maintenance costs of rotating machinery, and the possibility of accidents resulting from bearing damage could also be reduced. Many objectives have been achieved using other testing methods, such as the prototype testing method in which parts are often tested under high-stress conditions to induce early failures, etc. [29].

Over the last few years, since the bearings are the main cause for creating certain problems in rotating machines, the monitoring of bearing conditions in rotating machines has been gaining momentum, [40]. In order to diagnose bearing faults with vibration analysis, several techniques were developed, such as spectrum analysis, synchronised averaging, spectrum analysis, sum and difference frequencies analysis and high frequency resonance. [40]

AE techniques perform various functions in the analysis of bearing performance. They can be used to detect the anticipated failure that is caused by the intrusion of foreign particles in bearings [29]. Acoustic emission RMS, voltage, waveforms, and distributions of acoustic emission parameters are used

In order to diagnose the bearing faults with AE techniques for the analysis and characterization of AE sources. [30]

Traditionally Bearings faults were detected through signal processing of the measured vibration signals. These methods were known as RMS value, crest factor analysis and kurtosis analysis [40]. AE techniques are proving themselves to be a significant system to monitor problems in journal bearings. Also AE has now become a useful tool for the diagnosis of faults in the bearings of rotating machinery [32]. Along with fault detection, AE techniques can also be used to detect unwanted blinking or rubbing between the central shaft and the surrounding components [32]. Certain sources are highlighted as examples of AE in rotating machinery [51]. Among these techniques, some important ones are impacting, cyclic fatigue, friction, turbulence, material loss, cavitation, leakage, etc. [38].

In the field of AE, an important point to consider is AE signals [32]. Many techniques and signal processes have been applied to AE during the last few years, to facilitate detection of faults in rotating machinery [42, 43].

3.1 Background of AE in Bearings

AE was introduced by Kaiser in 1950. Initially, it was used for monitoring the defects in static or loaded structures and materials [38, 39], but soon it paved the way for the non-destructive monitoring of rotating machines. No matter what the size, loads or operational speed of the machinery, AE techniques have implications for detecting faults, but for large scale rotating machinery, it is preferred to establish AE in low speed plants incorporating rolling element bearings [38], because there is always the possibility that accurate analysis of the fault will not be attained due to low rotational speed and low level of background noise. The application of AE techniques in high power and large plants has not been sufficiently studied yet. The first document that accepts the significance of AE techniques in identifying bearing defects was published in 1969 by Balerston [50], who also observed that the AE signature is of two types: burst type emission and continuous type emission [50]; the first is related to the

seeded defects on the inner and outer race elements and the second happens whenever a bearing runs dry, i.e. without any lubrication.

The process of identifying the AE technique, as a significant method of monitoring bearing defects, was started in 1969 by Balerston [50], but in the following years several research studies focussing on AE techniques further explored this field when many possible uses of this technology were also discovered [50]. Ten years after Balerston's work, AETs were utilised, in order to monitor slow rotating antifriction bearings, by Rogers in 1979. Rogers's study provides several important results, and it was accepted that the AE system of monitoring bearing defects is much better than other testing systems. The new sources of AE signature were pointed out by researchers as being rubbing of the crack faces, grinding of the metal fragments in the bearing, impacts between the rolling elements and the damaged parts in the loaded zone. [48].

Later, in the early 80s, Catlin described the activity of acoustic emission for bearing failure diagnosis. He reported that the acoustic activity in a rolling element was associated with the existence of failure mechanisms or abnormal behaviour. The report also commented that different defects were represented in different ranges of frequencies. Moreover, Catlin's work also reported that the readings of the AE signal acquisition were mainly associated with journal bearing failure mechanisms, rather than other types of failure in the machine, such as misalignment, imbalance, external friction, shaft bending [49]. Catlin reported on the importance of the correct attachment of the sensors; he noted that the high frequency waves were rapidly attenuated. The correct placement of the sensor allowed the detection of the high frequency waves coming from the bearing, whereas the high frequency signals from other components of the machine were attenuated before being read by the sensor.

In 1983, Mathews reported that AE signals can be translated through four main parameters: RMS, Energy, Counts and Events. Counts are the number of times the value of an AE signal exceeds a threshold value.

In 1984, there was another innovative development in the field of AE when Yoshioka and Fujiwara revealed that the defects in bearings were being

identified by the AE parameters even before their appearance in the vibration acceleration range. During the studies of this period, certain other developments occurred in the field of AE techniques. A test was conducted on a thrust loaded ball bearing that resulted in the identification of AE generations [45]. In the coming year, the work performed by Yoshioka and Fujiwara was further investigated and reinforced by other researchers. It was discovered that AE techniques identify the fault in a bearing before their appearance in the vibration acceleration range, further studies were held based on this phenomenon and in 1988, Hawman and Galinaitis determined that AE- had advanced capabilities to perform early diagnosis of bearing defects due to the modulation of high-frequency AE bursts at the outer race defect frequency [45, 46].

Tandon and Nakra worked with maximum AE peak amplitudes and counts [44]. Their work stated that there was a relationship between AE, radial load, rotational speed and defect size. Their conclusion was that the higher the load of the rotational speed, the more AE counts and peak amplitudes to diagnose defects would appear; and, in the case of no failure, the signals were lower. They reported the counts were suitable for early defect size smaller than 250 μm . On the other hand, the AE peak technique was not concerned with the size of the defect – this was supported by Choudhury when he employed AE peak value signals to detect 500 μm to 1500 μm size defects [44].

Several studies were conducted in order to prove these findings, and the AE receiving sensor was placed at the outer race of the bearing directly. The results of these findings were further reinforced by other researchers. For example in 1993, Randall [50] conducted a study, and a similar effort was made by Holroyd in 2001 [49]. In the year 1988, Bagnoli et al. [51] designed a research project in which they investigated the demodulation of AE signatures at the defect rotational frequency (outer race) of a bearing [40]. They concluded that when there is no defect present in the bearing, then the periodicity of the passage of the balls beneath the load could be readily identified by observing the frequency spectrum of demodulated AE signatures.

[46]. The research study also revealed that when the defect is not present in the bearing, the intensity of the AE is also very low, and goes on increasing as soon as there is some defect in the bearings [51].

AE counts and their amplitudes for the outer race were also investigated in 1990 [52]. A resonant type transducer was used to accomplish the research, and the study pointed out that when there is an increase in load and rotational speed of the bearings, then the AE counts also increase. Another point that was discovered from this study was that when the defect in bearings is less than 250 μ m, only then could the AE counts be used for the detection of the defect. The AE counts were also analysed and tested to detect the defects in the bearings of different sizes and at different speeds, ranging from 500 to 1500 rpm. In 2000 [44, 52], a study was conducted, the results of which stated that: when the damage is low, the AE counts are also low, and as far as the speed of the damaged and undamaged bearings increases, the AE counts also increase. There were also outcomes from the research that prove that the load increase in a damaged or undamaged bearing has no noticeable effect on the AE [44, 52].

In 1990, variations in the AE count were studied for ball bearings of different sizes. The study was quite difficult to conduct and the researcher faced many hurdles in collecting the data [52]. As a result, some new methods and new AE counts were devised so that future studies could focus on them. Another study [53] focussed on the use of AE for the purpose of detecting damage to specified materials. In order to conduct this study, a trigger level of 10% of the maximum amplitude was employed.

Along with these studies, some research work was also conducted to learn the effects of vibration on AE on deep groove ball bearings with a diameter of 20 mm. When the groove was removed on the inner race, there was an increase in the stresses of the contact area and the situation resulted in fatigue failure [54]. During the fatigue test, the levels of vibration (RMS) were also monitored on a regular basis. An increase in the AE rate was also observed, resulting from the increase in vibration. There was still the need for a study that could identify the

most suitable threshold level for AE count diagnosis in rolling elements bearings [38]. One such study was performed in 2003 in which it ascertained that increase in speed affects the AE score, but that load and defect size have no influence over the value of AE [58]. The study also proposed that the AE count could be effectively used to monitor defects which have a length of up to 15 mm and width of 1 mm [39].

Another comprehensive study was conducted by Kakishima in 2000 which focussed on accessing AE and vibration on bearings [54]. Later, the techniques of AE were applied to a rotor dynamic system so that the multiple could be seeded [56]. It was observed that AE techniques were very sensitive to the defect conditions and other factors involved in the process.

In the case of the journal bearing, AE is a powerful technique in health diagnosis, by the use of parameters such as RMS and counts, as described by Mba [52, 58]. It was also demonstrated that, when working with counts, the quality of the monitoring diagnosis is independent of the threshold value chosen, and AE, measured in counts, has a direct relationship to the bearing state [58].

3.2 Acoustic Emission and Friction in Journal Bearings

Several research projects have been conducted in order to find out whether there exists any type of relationship between AE techniques and friction. For this purpose AE was measured in the specimens of severe metallic sliding. This research [43] revealed that there is a link between AE techniques and friction because in the dry sliding the presence of AE was proven [43]. Also discovered were some particular external variables upon which the rate of AE depends [43]. AE and friction relationship proves that AE can be used in modern condition monitoring. There are three characteristics that are monitored in order to diagnose the change in the friction regime: Total Count, Count Rate, and Amplitude or Energy Distribution of Acoustic Emission Pulses. [42] These parameters depend upon the surface and skin layer properties, and difference in these variable results in the loss of lubrication and change in friction pattern.

AE and friction play a vital role in journal bearings because plain bearings support the rotors of huge turbines. These bearings are also used in high powered internal combustion engines and electric motors. It is necessary that these bearings should be operated at high speed so that they can effectively support the high loadings [41]. The reliability of the plant or machinery largely depends upon the efficient working of plain bearings which is why great emphasis is placed on the proper monitoring of these bearings. It has been discovered that changes in the condition of bearings occurs due to several reasons including: irregularities in the temperature, volume, flow and contamination of lubrication; thermal effects and conditions of wear and fatigue etc. [41]. However, the change in bearing condition due to the above mentioned factors has still not been completely diagnosed. AE techniques can be used to properly diagnose changes in friction of bearings resulting from a change in lubrication or other factors.

Damage to plain bearings often results in the failure of high powered turbines and plants. Most of the failures occur in the turbine housings, rotors, guide vanes, rotor vanes. A sudden shutdown of machinery is caused by damage to plain bearings resulting from a friction loss [59]. Along with the AE testing methods, the damage to these bearings is monitored from various perspectives. Frictional changes are noted as one important reason for the damage of plain bearings [41, 42] because when there is a change in friction, a change occurs in the temperature of the metal bearings which finally results in damage [41]. The lubrication supply is another important factor that influences the working of plain bearings [41]. Plain bearings are affected whenever there is a change in the oil temperature that has been supplied to the bearings because the difference in temperature affects the working capability of the bearing. Other factors that have an effect upon the working of plain bearings include the pressure of lubricant at specific bearing points, volume flow of the lubricant and size, quantity and composition of wear debris particles [41].

There is another perspective for measuring and monitoring the damage of plain bearings, and that is Shaft Vibration Monitoring [41]. This system is used to

monitor a shaft in a bearing when the bearing is in a phase of non-steady running or experiencing variations in loads. Through this process, it is possible to identify the contact between the shaft and bearing metal when there is a crack in the oil film due to the roughness of the peaks [41]. This monitoring system has proven to be an effective and easily adopted monitoring system worldwide.

Temperature variations occur in the bearings metal; however, when they are supported by a high-power turbine, vibration resulting from temperature variation stops, and it becomes difficult to monitor the effect of temperature fluctuation on the condition of the bearings. The AE testing system is used so that the bearing vibrations can be diagnosed [41], and concludes that when there is a low-frequency vibration occurring in a turbine then the temperature of the bearing metal also increases, which can be measured between 1 ÅK and 10 ÅK [41]. It was revealed from the AE testing method that for the diagnosis of vibration in bearings, the peak value and density of vibration acceleration are not appropriate constraints [41, 64]. It has also been proven that the low-frequency vibration is present in the signals of vibration frequency and in normal circumstances this can affect the value by up to 25% [41, 64]. The AE testing method verifies the presence of friction in the water circulation pumps where the plain bearings are placed with water lubrication. When the water lubrication was reduced under normal conditions, there was also a change in the impulse density and spectrum [64].

The discussion about AE and friction in journal bearings can be summed up by concluding that the AE testing method is an effective and suitable method to determine changes in the friction as a result of changes that occur in the condition of journal bearings [41, 64].

3.3 Literature Review

Numerous research studies have been conducted in the field of AE. These studies highlight the role of AE in the diagnosis of friction in journal bearings or focus upon other problem areas. Some research has also focussed on the

relations between AE techniques and different surfaces; however, most research focussed on the detection of slip or tool engagement. None of the studies underlined the effects of surface geometry on AE generation in sliding motion. This situation became the background reason for conducting another research study to cover those points omitted by previous studies.

Keeping in mind the increasing interest in automation, research was conducted to underline the interaction between a probe and a surface using an AE testing system "Study of Acoustic Emission Generation in Sliding Motion" This research was conducted with the objective of investigating the AE signal emission during the sliding motion when a probe moves over the surface [65]. This study gives a review of the AE used in contact studies and describes the assumed mechanism of AE generation in sliding motion. The paper also discussed an experimental approach that can be used to test these ideas. There is an analytical discussion about the obtained data, and Conclusions are drawn. This research paper highlights the reasons why the geometrical properties of a surface have an effect on the characteristics of AE signals. The study focuses on the point that there is a great need for research work to be done in the field of AE, so that it will be able to accurately measure average surface roughness characteristics [65].

The comprehensive introduction provides sample information about AE signals and their possible use in monitoring surface asperities. It investigates the direct effect of the surface in the interaction process of sliding motion, it is crucial to measure the surface characteristics and for this purpose, AE techniques could be best employed to obtain the desired objectives. The study also points out that if AE is used for studying the sliding motion, then a probe that moves over the surface could be used. There are two basis sources, which are assumed to be primarily associated with the AE generation mechanism. These sources include the "impact of the moving probe with the surface asperities and deformation of asperities due to sliding motion." [65]. An experimental setup was used in the study that consisted of five major parts: a "device holder, a sensor level, an adapter, Acoustic Emission sensor and wave guide probe

piece. [65]. The sensor was attached to a Cincinnati milling machine on which the lever arms are adjustable so that they could be used to give an effective load at the probe end. A lead counterweight was also used at the lever arms so that the effective load could be changed. The sensor adapter was used to hold the AE sensors and probe piece. The wave guide probe was made up of two pieces screwed together [65].

Using this experimental set up, several combinations of filters and AE transducers were used and many tests were also performed to identify the best transducer/filter combination [65]. After conducting the tests, data was obtained that showed the activities of AE reaching their peak within the frequency range of 100 kHz to 400 kHz. This data was acquired from three different probe geometries at three different speeds [65] i.e. 8.33 in/min, 16 in/min and 30 in/min. The constant value for different probe geometries and speed was set to 0.5v. After data gathering, the results from and discussion of the study underline that there are three main parameters of AE techniques: the events, the mean peak and the RMS [65]. The study also revealed some important trends. One concerned needle and bar probe sensitivity towards the variation in grid size. Another showed that when there is an increase in the diameter of grid size, the event went on decreasing. And a third indicated that the event is also decreased when there is an increase in travel speed [65]. The same study also explained the relation of RMS to speed. It was discovered that when there is a rise in the speeds of the three probes, the RMS also went on increasing. This proves that the intensity of impact increases with increasing speed [65].

The results and discussion of the study conclude that rather than the deformation of asperities, the impact between a probe and the asperities of a hard surface are a more dominant source of AE. It was also proved that a flexible probe which possesses a small contact area is more sensitive to surface interactions. The AE parameters are particularly sensitive to the variations which occurred in the grid size [65]. Whenever there is some increase in speed, this has inverse effects on the event which went on decreasing as a result of less contact with the grids due to the dynamics of the probes [65].

When there is an increase in speed, the RMS continued to increase [65]. The variation in the grid size also affects the events; for example, a rise in the speed causes a decrease in the event because there is less contact with the asperities [65].

Leahy et al. [66] conducted an empirical investigation into the suitability of the AE monitoring technique for detecting seal-to-rotor rubbing in steam turbines. A 2 meter long and 4.5 tonne test rotor was used and rubbing was induced at various axial locations along it. The Rotor was supported at both ends by 70 (178 mm) hydrodynamic journal bearings. These were placed in a vacuum chamber and were connected by an external electric motor. The rotation speed of the motor was set at 3000 rpm for the experiments because this is a typical speed for steam turbines used in power generation in Europe. The fin geometry of the seals was also considered to be similar to those used in steam turbines.

The simulation of the seal-to-rotor rubs was done using four replica stainless steel seals placed at different locations. The seals could press against the rotor due to the linear motion converted from the angular motion of stepper motors. This enabled the seals to be moved linearly at in 0.1 mm steps. Slack in the thread was prevented by incorporating a spring into the assembly of the seals. A Parker XLi-25 unit was used to drive the stepper motors, which were controlled by a computer using Easi-Tools software via its serial port. Each stepper motor was controlled in turn but due to cable limitations only one seal was able to rub at a time. In addition, thrust bearings were incorporated into the seals using flexible couplings to protect against axial loads; guides were mounted at both vertical ends to prevent their displacement, and thermo-couples attached to the seals helped to prevent over-heating. Furthermore, real-time measurements were taken of the shaft vibrations in both horizontal and vertical directions as well as the temperatures of the bearing oil, white metal, artificial seal, and oil pressure [66].

AE transducers, from the Physical Acoustics Corporation (PAC), were used to make detection of the rubbing easier. They were attached directly to the seals using cyano-acrylate, and were mounted to the bearing casings using magnetic

clamps and marine grease. The transducers were connected to the computer via variable gain pre-amplifiers. The computer had three PCI-2 acquisition cards from PAC, was fitted with a hardware accelerator and used anti-aliasing filters. ADC controlling software was then used to calculate the Average Signal Level (ASL) in decibels at 10 ms intervals in real time.

The step size had to be reduced to 0.1 mm for the seals to establish contact with the rotor. Contact was characterized by a dramatic increase in acoustic emission activity at the seal transducer accompanied by lesser but still significant rises in the activity at the bearing transducers [66]. Initially, the AE activity was large, but then reduced slowly due to the gradual reduction in contact pressure from wearing. The transducers were therefore, able to detect the onset of induced rubbing. Moreover, the pre-amplifier setting of 20dB produced the best performance, and both the inboard and outboard seals generated equally pronounced responses. In short, it was shown that the AE technique can be used for bearing mounted transducers for detecting seal-to-rotor rubs operating at 1500 rpm on a 4.5 tonne test rotor, and at 3000 rpm on hydrodynamic journal bearings. The results suggest a potential application of the technology for the monitoring of power generation turbines.

A case study by Mba et al. [68] on an actual steam turbine confirmed that high-frequency AE can detect and verify shaft-seal rubbing. The steam turbine was operational at 217 MVA [68]. Moreover, the AE activity was found to be attributed to the contact, deformation, adhesion and ploughing of surface asperities on the rubbing surfaces of the rotor and stator." [68]. They used a commercially available broadband piezoelectric transducer, see figure 3-1.

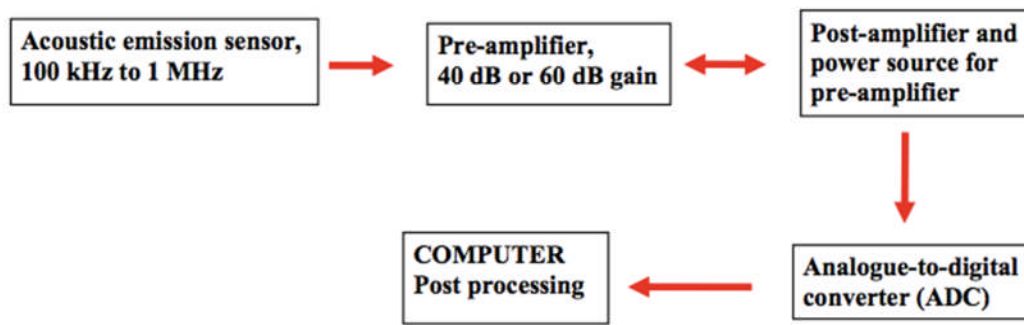


Figure 3-1 Schematic diagram of an acquisition system used to monitor AE activity in an operational steam turbine unit [68]

The acquisition board used was the PAC& AE-DSP-32/16 and the sampling rate was set at 4 MHz with 16-bit precision. An 8th order anti-aliasing filter was also used with a 3dB roll-off at 1.2 MHz. This arrangement provided 32,000 data points, and two sets of AE signatures were recorded at 4 MHz and 1 MHz, the first for corresponding frequency spectra and the second for AE data exceeding 0.032 sec, were sampled at 1MHz. The sensor was placed on top of the casing for the recordings taken at the four seal/gland positions to ensure identical transmission paths. A magnetic clamp was used to secure the transducer to the casing, and spring rings helped to maintain clearance between the seal and the rotor. In addition, a 1.75 V trigger level helped to minimise the AE noise data. [68]

High levels of shaft displacement occurred during operation on the generator and low-pressure cylinders of the turbine, which was observed using eddy current probes. During this operation, the steam exhaust temperature was at 18°C, the unit had a maximum load of 80 MW and rotational speed was 3000 rpm at 50 Hz with one shaft rotation period corresponding to 0.02s. [68]

The ability of AE transducers to successfully detect the rubbing induced on a rotor was also demonstrated in another study by Leahy et al. [68]. They investigated how AE technology could be used to detect seal-to-rotor rubbing in power generation turbines using a similar set-up of equipment as in an earlier study by Leahy et al. [67, 68] in the same year. Although the results confirmed the potential for AE with bearing mounted transducers to detect seal-to-rotor rubs, it was also observed that the placement of the transducer could also affect

the performance of AE technology. The placement of transducers is therefore an important consideration in terms of the possible sources of noise such as at the thrust faces. Nonetheless, the study showed that AE technology has the potential to be used as a complimentary tool for monitoring the integrity of power generation turbines [68].

In another study, the use of AE for shaft-seal rub detection, in large scale, 50 Hz turbine generators has been investigated by Sato [5] and Board [69], published in 1990 and 2000 respectively. In these papers, Board describes the use of Stress Wave Analysis (SWAN) testing as a diagnostic tool on a range of aircraft and industrial gas turbine engines for demonstrating the ability of the technique to detect various discrepant conditions accurately and to characterise damage severity using a graphical user interface. [69]

The SWAN technique is able to provide real-time measurements of friction and mechanical shock in operating machinery in general and dynamic load transfers between its moving parts. It is a high frequency acoustic sensing technology that can filter out background vibration and audible noise. The sounds are detected as they travel through the structure of the machinery at ultrasonic frequencies and then analysed. A piezoelectric crystal is used to convert the amplitudes of the stress signal into electrical signals prior to amplification and filtering, and the unwanted low frequency sound and vibration energies are then removed. The resultant Stress Wave Pulse Train (SWPT) is then analysed by a digital processor for calculating the peak level and the total energy content of any frictions or shocks. [69]

In this way, SWAN can then detect wear, defects, and early signs of damage. It will track their progression during failure of the machinery as the energy content of friction and shocks increases. SWAN can detect and measure this energy even for damages well below the level of degradation necessary for detection by vibration sensors and are therefore superior to vibration analysis. The discrepant conditions are identified by comparing the values of the Stress Wave Peak Amplitude (SWPA) and Stress Wave Energy (SWE) generated with the

machinery's normal operating conditions. SWAN can detect even slight friction or shock events. [69]

The three main tools used by SWAN are Stress Wave Energy (SWE), Stress Wave Amplitude Histograms (SWAH) and Stress Wave Spectral Analysis (SWS). The data is examined in the following three steps from each sensor:

- i. (SWE) A Stress Wave Energy operating history chart is created during routine operation and the readings are plotted against health indicating colour zones.
- ii. (SWAH) A Stress Wave Amplitude Histogram is drawn to check for a normal distribution at the lower end of the amplitude scale (which would suggest healthy machinery).
- iii. (SWS) Optionally, analysis of the Stress Wave Pulse Train can determine the spectral content whereby all vibrations are filtered out leaving only friction/shock modulating events. A horizontal line with no significant spectral lines indicates healthy machinery.

A wide range of damage levels can be quantified by SWE and a fault can be localised to identify the problematic bearing or gear or even an element within these. It can also detect abnormal pre-loads and lubricant contamination. SWAN is especially useful for detecting aperiodic events early, connected with lubrication problems, and SWS is especially useful for detecting abnormal dynamic loading and early minute levels of localised damage from fatigue. The article also gives some case histories to demonstrate SWAN's ability to detect various discrepant conditions such as bearing wear and surface damage, misalignment/imbalance abnormal dynamic loading, oil starvation, turbine rub, foreign object damage, etc. In addition, it is shown how SWAN can be used to isolate the root cause of these faults and to quantify the extent of the damage so that it can be trended and projected to show its remaining useful life. [69]

SWAN also has potential to be automated using artificial intelligence and this has been demonstrated for many discrepant conditions in several types of bearings and gears in complex kinematic assemblies. The software was also able to extract certain features from the large quantity of data and process it to

facilitate accurate decision making. This includes its use in aircraft applications. [69]

Sato recommended analysing with acoustic emission (AE) techniques to provide a direct means to detect early signs of problems in rotary machines. It is an alternative to the traditional indirect methods such as analysing pressure, temperature and vibrational signals, which are unable to make detections until the defect develops to an advanced stage. They work by detecting elastic waves that are generated when a solid is deformed and self-destructs under the pressure of a force that is externally applied. A single sensor is capable of monitoring a range of frequencies based on arrival time differences of the signals. Using multiple sensors however, can enable the source of the signals to be located. In short, AE techniques enable early detection of abnormalities, an incipient malfunction to be detected, monitoring of operational malfunctions, coupled with the ability to determine the source of a malfunction. [5]

The main concern AE techniques address is in ensuring the reliability of machinery through accurate diagnosis. They can help to improve reliability and operability in installations, especially to help prevent industrial accidents. Many large-scale plants in particular contain rotary machines where reliability can be critical and there is a need to ensure all the structures, equipment and materials are able to withstand the continual stresses that they have to endure while in operation. AE techniques can diagnose such mechanical malfunctions as abnormalities in the rotors from rotor cracks and slackening, bearing damage and rubbing of contacts between the rotors and stators. The latter is usually caused by abnormal vibrations and it can often interrupt the operation of the machine. Bearing damage can be caused by an overload or by an increase in the lubricating oil temperature, and it necessitates unscheduled shutdowns to avoid accidents. As for rotor cracks, these were typically detected using periodic non-destructive tests such as ultrasound measurements and magnetic field tests but as for the other two types of problems, such methods are unable to detect the problems early enough. [5]

In Sato's Experiment, he induced a rub condition on a 350 MW steam turbine at a normal operational speed. AE sensors were mounted on adjacent journal bearings whilst continuous rubbing was introduced at various rotor locations between the bearings, whilst the steam turbine rotated at 3000 rpm.

Sato studied the AE method of measurement and presented the results of a study in which this method was used in combination with unusual high-frequency sounds to make an early detection and diagnosis of mechanical malfunctions in rotary machines. The experiment involved simulations of rubbing in steam turbines and installing test samples at various locations, diagnosing damaged bearings and utilising a wireless based AE monitoring method for abnormal conditions in the rotor. It was found that the AE technique was effective in detecting mechanical malfunctions, especially for rubbing, bearing damage and rotor abnormalities. For rubbing, it was found that extracting the frequency components of rotations from processed signals following detection was successful and the location of the problem could be identified using two separate sensors. The major causes of journal bearing damage, i.e. bearing tilts and wipes, could also be detected by observing the wave shapes from the AE signals. A wireless AE sensor was then developed for detecting cracks on rotary parts whilst in operation, for early detection of abnormalities and malfunctions. [5]

The observation of amplitude modulation highlights that continuous rubbing is fundamentally linked to rotor dynamics and the AE response measured at the bearings will change consequently., Sato Therefore determined that detection of this AE modulation (Envelope) is an effective signal processing technique for detecting rubbing in rotating machines. [5]

4 Test Rig Design

In this chapter, the design and operation of the test rig is explained. In order to investigate the feasibility of the use of the AE technique for journal bearings, an experimental test rig was required – this was accomplished by modifying an existing test rig. The rig was designed by Sutton [165] to investigate the oil film force coefficient of a journal bearing in aligned and misaligned conditions. It involved three bearing types, with a test bearing (Hydrodynamic Bearing) in the centre. Morton [166] used this type of illustration to predict power loss, oil flow, load capacity, and temperature of the bearing material and oil. Mukhtar [71] modified the test rig to investigate the feasibility of its use to detect the early stages of journal bearing deterioration, by using a purpose-built miniature ultrasound transducer.

4.1 Test Rig

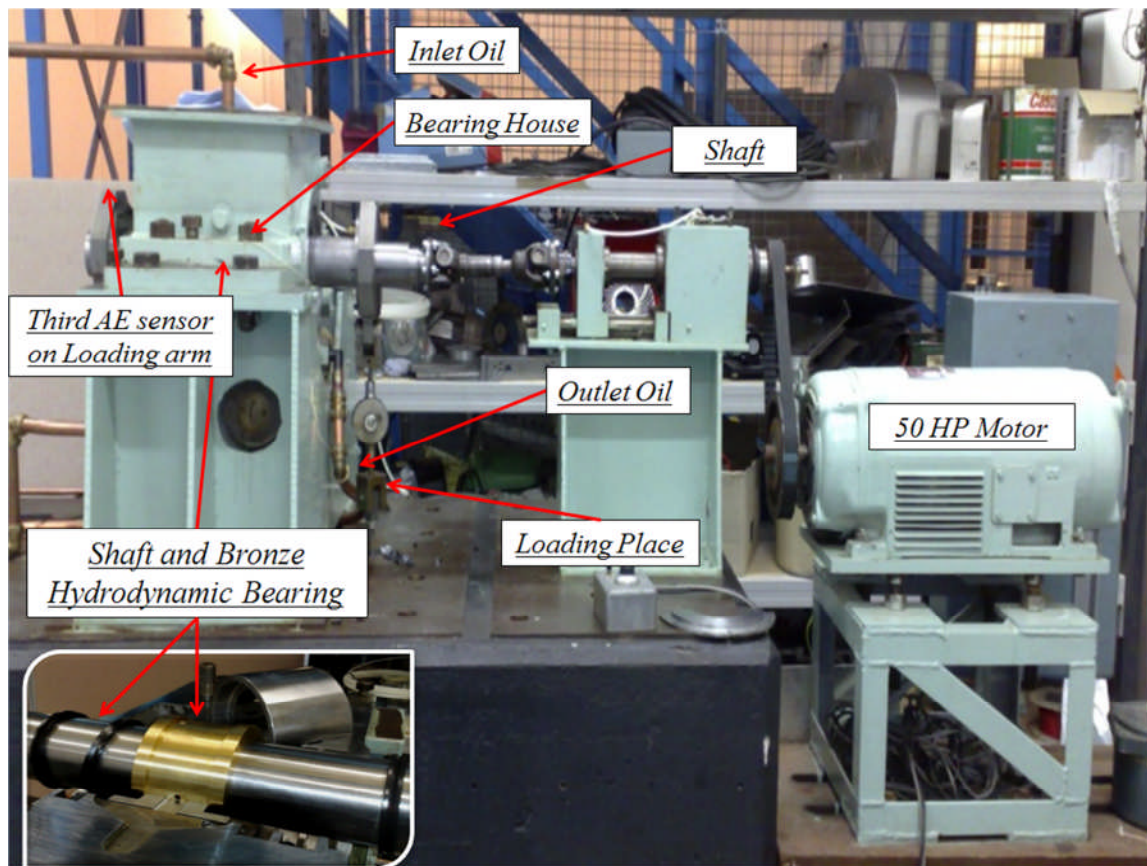


Figure 4-1 Test rig layout

Figure 4-1 depicts the shaft side view of the test rig employed to investigate shaft failure and associated AE characteristics. The distribution of the system elements is shown in the picture figure 4-1; the hydrodynamic bearing test rig employed for this study has an operational speed range between 10 rpm and 5000 rpm, with a maximum load capability of 20 kN.

A radial load was applied to the shaft through the overhang bearing by a hydraulic system (Hi-Force HYDRAULICS-HS 102HAND pump-single speed-working pressure: 700 BAR).

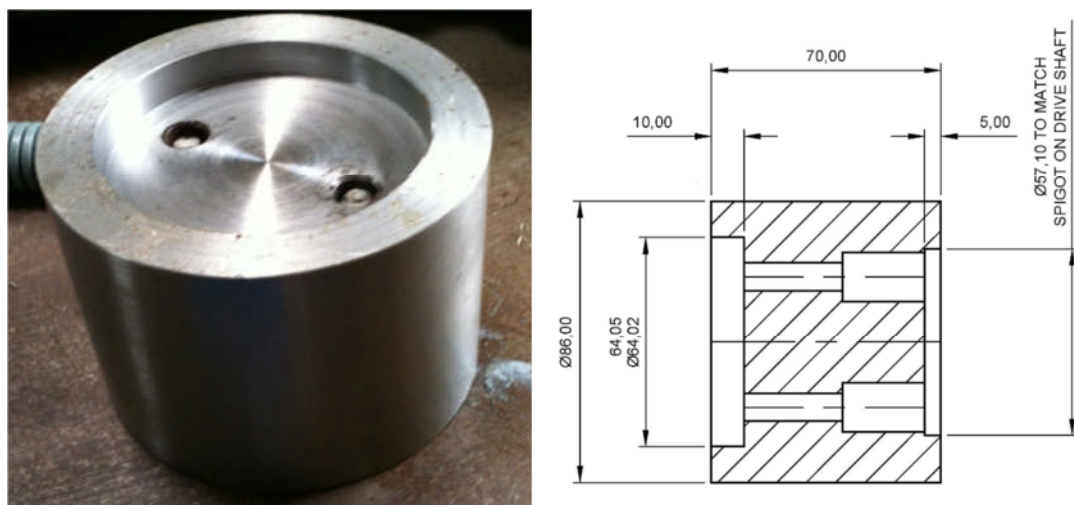


Figure 4-2 Flexible Coupling

A flexible coupling was employed to absorb any vibration as a result of attaching the shaft to the geared motor, see figure 4-2.

4.1.1 Hydrodynamic Bearing

Bronze was used for the manufacture of the journal bearing, see figure 4-3, so as to have lower surface hardness compared to the journal shaft material's hardness. The length and diameter of the bearing is 80 mm [$L/D=1$], with a surface roughness of approximately $3\text{ }\mu\text{m}$ and a measured radial clearance of 0.17 mm.

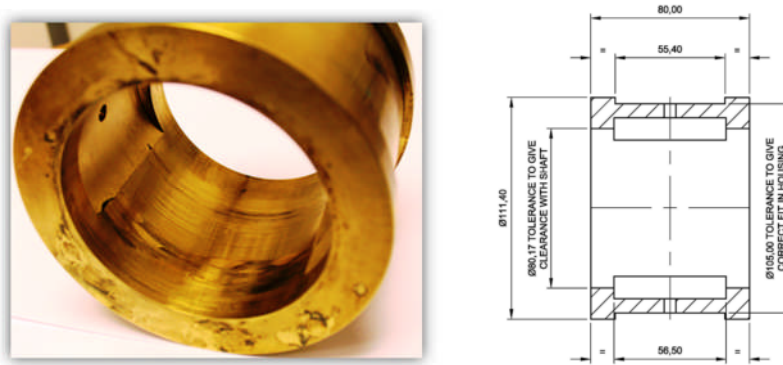


Figure 4-3 Hydrodynamic Bearing

4.1.2 Journal Test Shaft

A schematic of the journal test shaft is shown in figure 4-4. The total length of the shaft was chosen taking into consideration the position of two load bearings on a pedestal foundation, i.e. 120 mm, and the diameter of the shaft between load bearings is 80 mm and has a ground surface finish see appendix B. The two ends of the shaft have a 70 mm diameter, ground and honed until a sliding fit is obtained with SKF N214 rolling element bearings. The material used for the shaft manufacturing is mild steel so as to have a greater surface hardness than the hydrodynamic bearing.

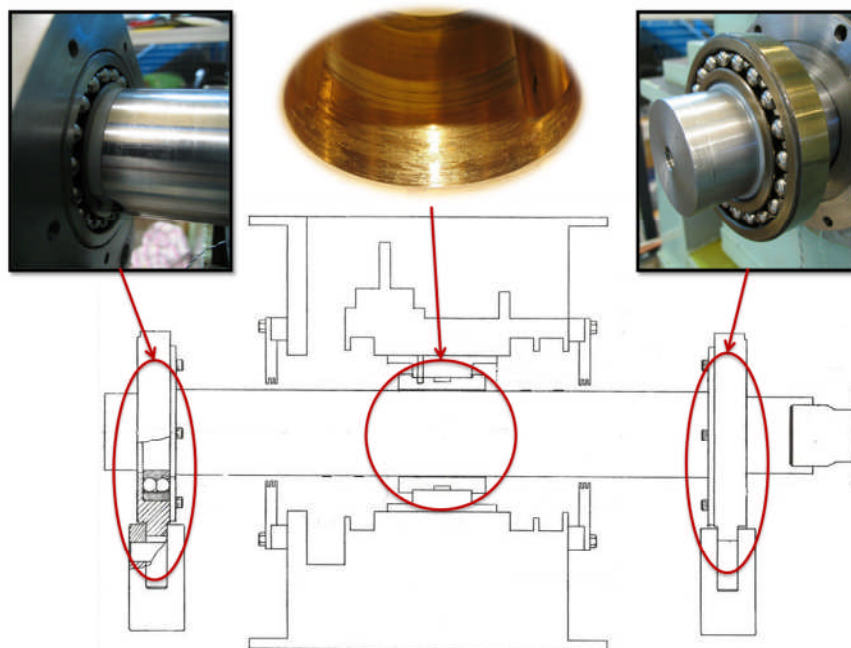


Figure 4-4 Test rig sketch

4.1.3 Bearing Oil Supply System

To provide lubrication oil to the journal bearing, a framework schematic was built to support the oil supply system. The framework was bolted to a concrete block and oil bath, with the gear pump on the floor. The oil supply is shown in figure 4-1. Lubricating oil is pumped from the oil bath through the framework to the top of the bearing house via a fine oil filter to trap any impurities from the lubricant.

To understand the influence of viscosity, three different oil types were employed:

- Alpha SP68, kinematic viscosity 68cSt and 8.53cSt at 40°C and 100°C respectively
- Alpha SP220, kinematic viscosity 220cSt and 18.7cSt at 40°C and 100°C respectively
- Alpha SP460, kinematic viscosity 460cSt and 30.5cSt at 40°C and 100°C respectively

4.1.4 Elimination of the Noise of the Rotating Shaft

According to the preliminary results, it was noted that the signals coming through the shaft from ball bearings in loaded places affect the elastic waveforms which were recorded in the journal bearing side sensors. For controlling, reducing and eliminating these noises, and registering mere elastic waveforms specifically from the journal bearing during experimental tests, Nylon 6/6 was used between the ball bearings in loading places and the shaft. The specification for Nylon 6/6 is detailed in table 4-1.

In a preliminary study case, the noise eliminator showed, on average, decreases of 500% in the noise amplitude forms which were recorded in the journal bearing sides from ball bearings in the loading places

Table 4-1 Nylin6/6 specification

Typical Properties	ASTM Test Method	Units	Nylon 6/6
Density	D792	lbs/cu in	0.0412
Specific Gravity	D792	----	1.14
Water absorption, 24 hours, 73°F (23°C)	D570	%	8.5
Tensile Strength 73°F	D638	psi	12,400
Elongation 73°F	D638	%	90
Flexural Strength, 73°F	D790	psi	17,000
Flexural Modulus, 73°F	D790	psi	4.1 X 10 ⁵
Izod Impact Strength, Notched, 73°F	D256	----	R120 - M79
Heat Deflection 66psi	D648	degrees F	455
264psi	D648	degrees F	194
Max. Temperature Long Term	----	degrees F	170
Short Term	----	degrees F	355

Figure 4-5 indicates different composing segments of the shaft, ball bearing, loading places and assembly of the Nylon 6/6.

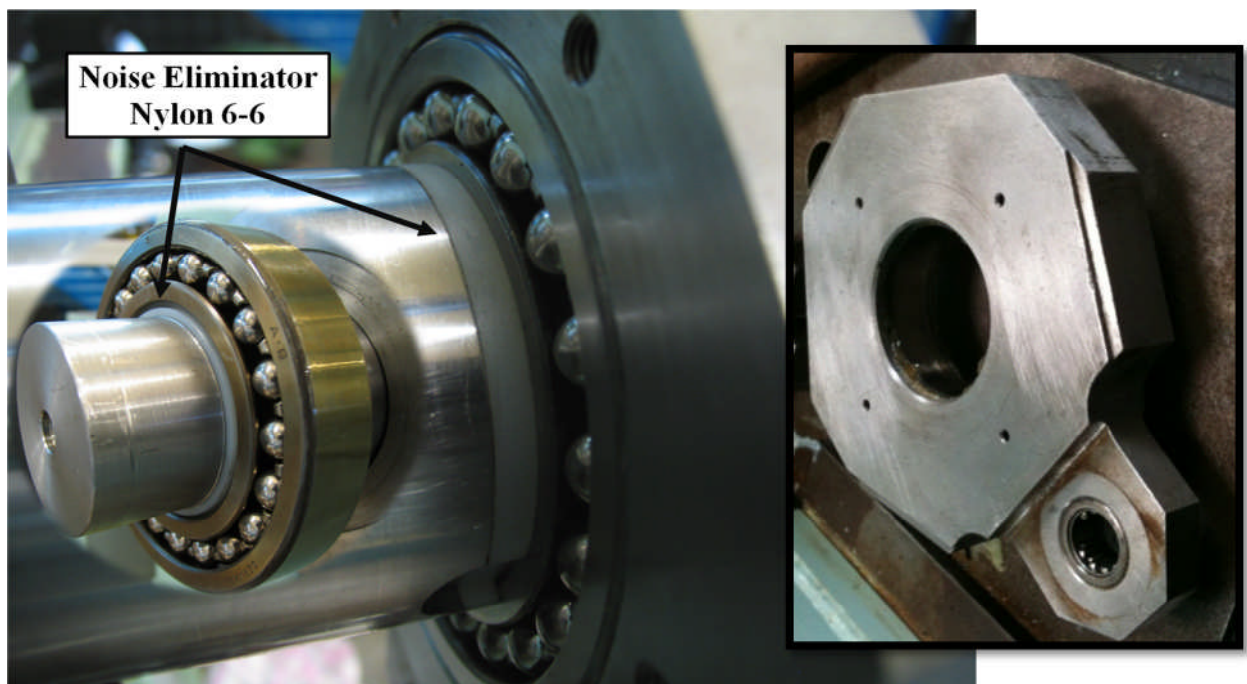


Figure 4-5 Noise Eliminator-Nylin 6/6

4.1.5 The Hsu-Nielsen source test

During AE employment, either under laboratory conditions or in industry, an attenuation of the measurement path used must be carried out. One of the

basic calibration methods was developed by Hsu-Nielsen and consists of a pencil lead break device to simulate an acoustic emission event using the fracture of a brittle graphite lead in a suitable fitting, see figure 4-6 [72].

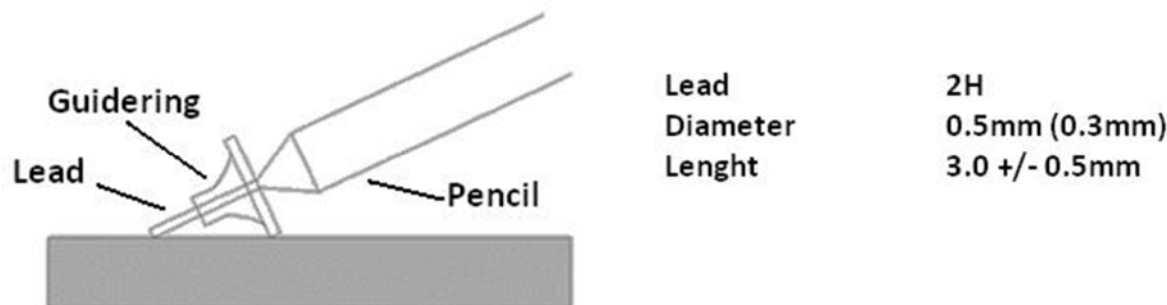


Figure 4-6 The Hsu-Nielsen source

Attenuation tests were undertaken (Hsu-Nielsen) on the test rig at various defined locations, see figure 4-7 and table 4-2. The Hsu-Nielsen test involves breaking lead (changes in signal can be due to variations in the lead) [73]. Typically, hard lead (2H) with a diameter of 0.5 mm (or 0.3 mm) and length (3.0 $\pm$ 0.5) mm is used) at a defined point i this is an established method of simulating AE. From the results presented in table 4-2, it was clear that AE could be transmitted across the plastic strip (for example; point G). However, attenuation across the strip to test the bearing shaft was in the order of 35dB, which would imply a 55-fold reduction in AE amplitude levels across the strip. This is of particular significance as all unwanted AE contributors must be eliminated (such as the rolling element bearings onto which loads are applied to the shaft). Attenuation tests were undertaken with the oil supply to the bearing switched on, i.e. oil was flowing into and out of the bearing.

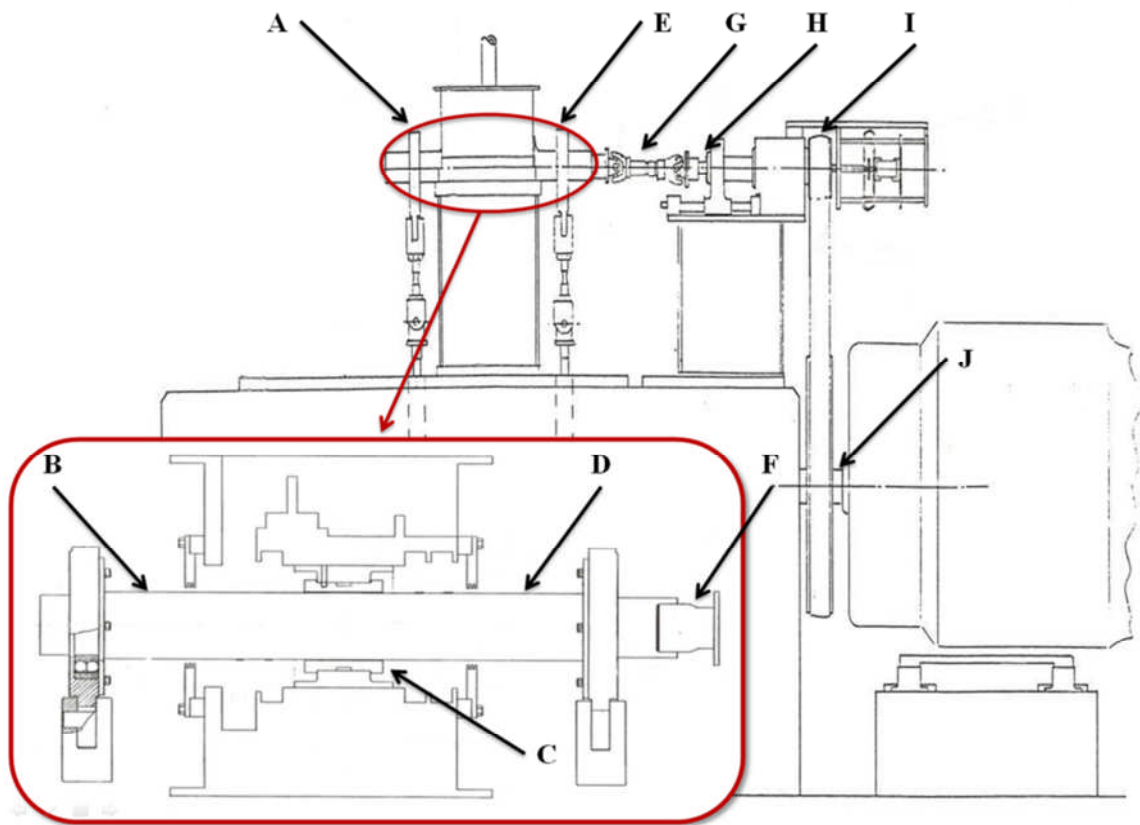


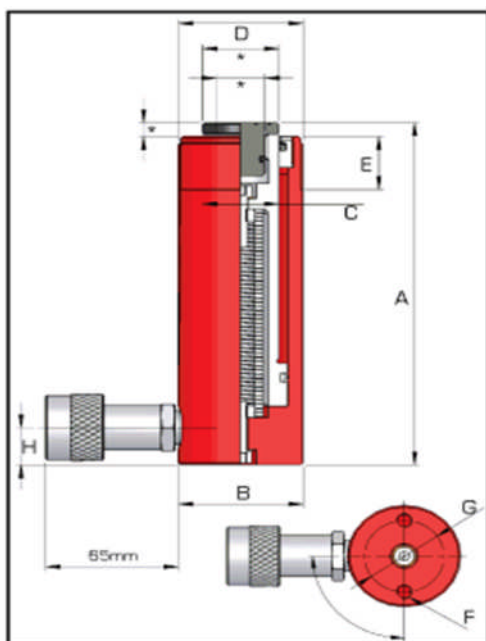
Figure 4-7 Attenuation tests on the test rig

Table 4-2 Attenuation results in Journal Bearing Test Rig

Attenuation Nelson Breaks	A	B	C	D	E	F	G	H	I	J
AE Amplitude Ch1(dB)	35	78	-	78	78	40	73	-	-	-
AE Amplitude Ch2(dB)	43	75	-	-	75	37	71	-	-	-
AE Amplitude Ch3(dB)	78	41	-	-	-	-	-	-	-	-

óóNo detectable emission

In order to apply a constant and reliable load, a pushing cylinder was installed. The picture below shows a loading force of HHS102 with a maximum working pressure of 700 bar, and the physical dimensions of these hydraulic jacks was 131mm × 57mm and oil capacity 81 cm³, see figure 4-8, onto the load plate linked to the rotating shaft on either side of the test bearing, with features and size of the cylinder taken from the Hi-Force catalogue.



Model Number	Hi-Force HS 102
Capacity (tonnes)	10
Stroke (mm)	56
Oil cap. (cm ³)	81
Cyl. Eff. Area (cm ²)	14.4
Mass (kg)	2.4
DIMENSIONS (mm, unless otherwise stated)	
A	131
B	57
C	35
D	21.0
E	27
F	M8
G	40
H	19.0

Figure 4-8 Dimensions and features of the pushing cylinder

4.1.6 Hydraulic pump

The cylinder is pushed by a hydraulic pump equipped with a pressure gauge to control the load applied. Also, the pressure gauge is checked during the test to ensure that there is not any leakage or pressure loss.

4.1.7 Pressure gauge

In order to read the pressure applied to the piston, a pressure gauge was installed in the pump, see Figure 4-9. Since a constant load test is to be carried out, it is very important to ensure that the reading in the gauge remains constant during the entire test. Also, in the event that there was any leakage the load can be readjusted as soon as the leak is noted.



Figure 4-9 Hydraulic pump and pressure gauge installed on the test rig

4.1.8 Load pistons

Load is applied at both sides of the shaft in order to avoid negative misalignment effects. As is shown in the sketch below (figure 4-10), the journal bearing is placed in the centre of the shaft. The points where loads are applied are located on both sides and at the same distance.

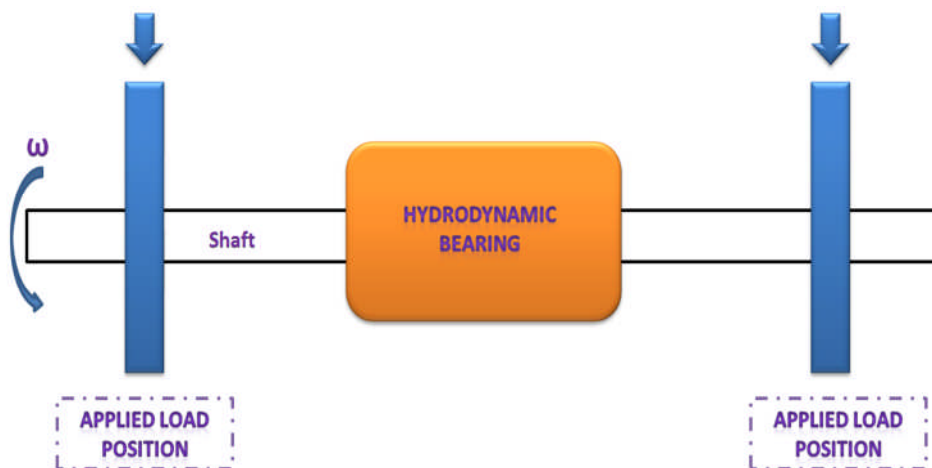


Figure 4-10 sketch of applied load positions

The load application system consists of two pneumatic pumps (manual). These pumps push a piston placed in the contact points. In order not to damage the shaft, each piston actuates over a ball bearing. Therefore, load is transmitted,

but pistons do not enter in contact with the shaft, avoiding any possible damage due to the contact.

4.2 Application

4.2.1 Sensors Location

There are 3 types of sensors attached to the machine, each one responsible for the registering of different signals:

- 4 J type thermocouples that record the temperature of the journal bearing
- 1 K type thermocouple that records the oil bath temperature
- 3 Acoustic Emission sensors to detect Acoustic Emission events
- 2 Eddy current sensors that record the shaft displacement.
- 1 Vibration accelerometer in the high frequency range

The layout of the sensors in the bearing is as shown in figures 4-11 and 4-12:

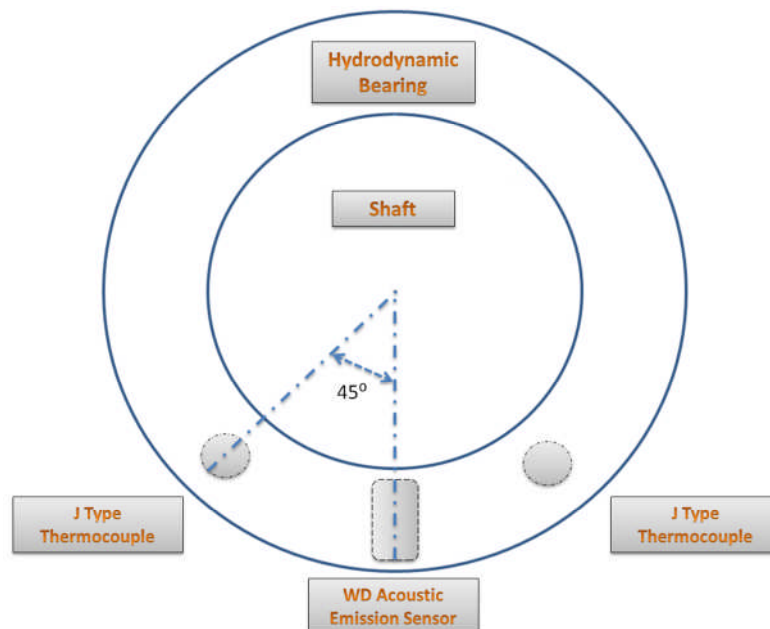


Figure 4-11 location of AE sensor and thermocouple sensors on bearing

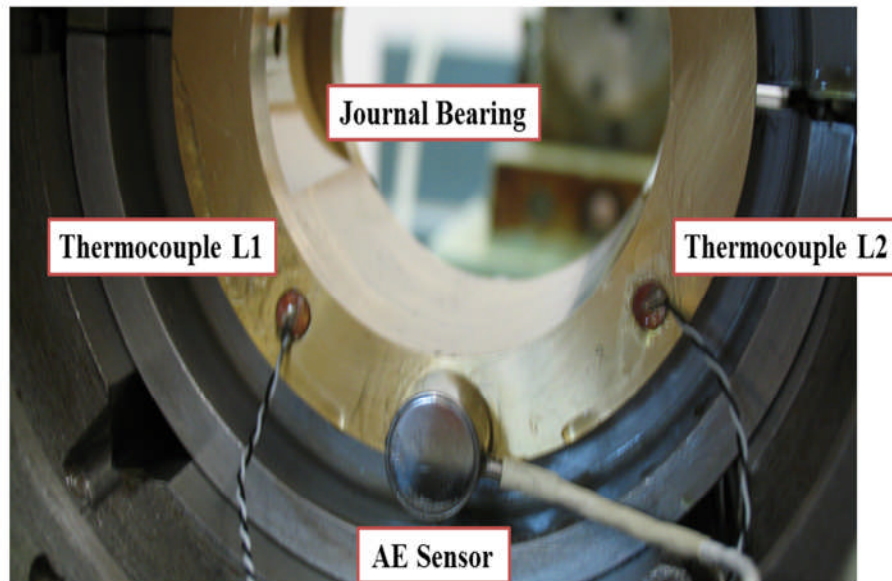


Figure 4-12 Location of AE and thermocouple sensors on test bearing (right side)

The layout is the same in each side of the bearing. The sensors are located in the load application points, and the oil bath thermocouple is located close to the oil bath, but not represented in these figures.

4.2.2 Data Acquisition Process and Instrumentation

Figure 4-13 shows the diagram of the data acquisition process.

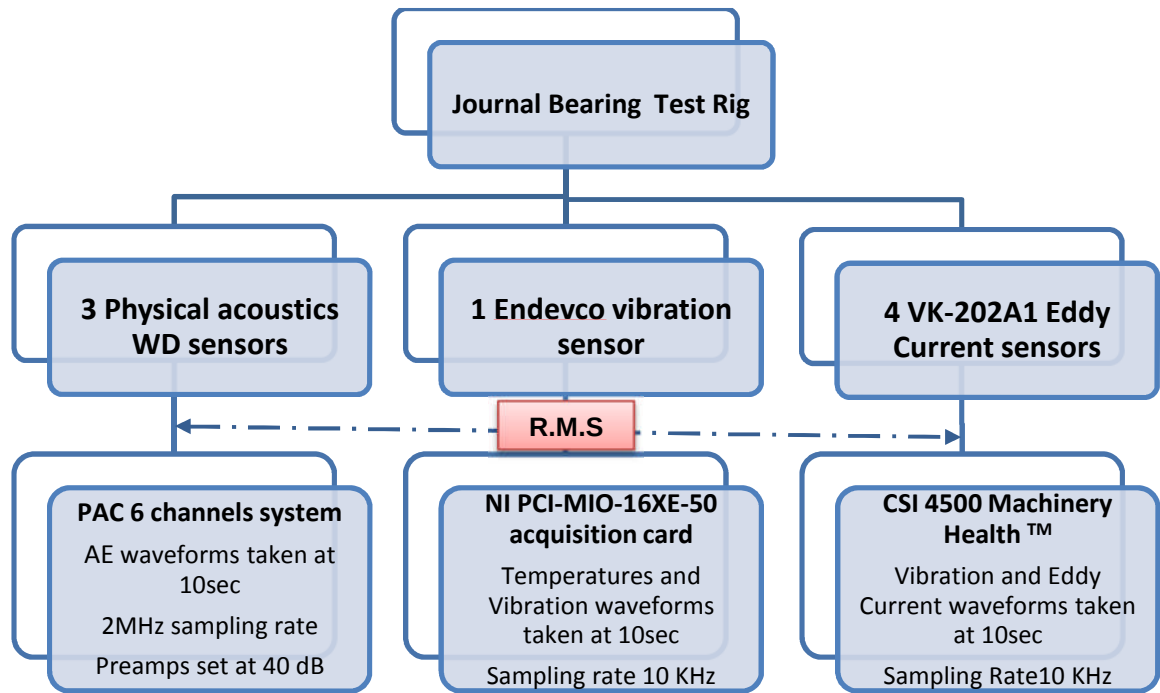


Figure 4-13 Data acquisition process diagram

4.2.3 AE data acquisition system

The AE system employed commercially available piezoelectric sensors (PHYSICAL ACOUSTICS, type rWD) with an operating range of 100-750 KHz, see table 4-3 and figure 4-15. These sensors (WD) are identical in physical, electrical and response characteristics; except that two signal leads are brought out to attach to a differential pre-amplifier [74]. By using a differential preamplifier, common mode noise is eliminated, resulting in a lower noise output from the preamplifier, and a higher electrical noise rejection in difficult and noisy environments. Noise improvements in the range of 2dB or higher can be expected using a differential sensor and preamplifier.

Table 4-3 A physical acoustics Pico sensor Characteristics

Model	Dimensions (dia x ht) mm/ inches	Operating Temperature(°C)	Peak Sensitivity dB ref. 1V/(m/s) [1V/mbar]	Operating Frequency Range (kHz)
WD	18 x 17 / .7 x .65	-65 to 177	55 ⁺ [-62.5]	100-1000

For calibration certificates for WD type sensors see appendix E. Two of these sensors were placed directly onto the test bearing at each axial end, see figure 4-12, and the third WD sensor was placed on the loading arm. Since the signals from piezo-electric materials are not strong enough to travel long distances through the cable, it is necessary to amplify the transducer signal to increase the signal amplitude and match the high impedance of the piezo element to the low impedance of the signal cable [74, 75], see figure 4-16. The signal is amplified by four pre-amplifiers 40dB gain.

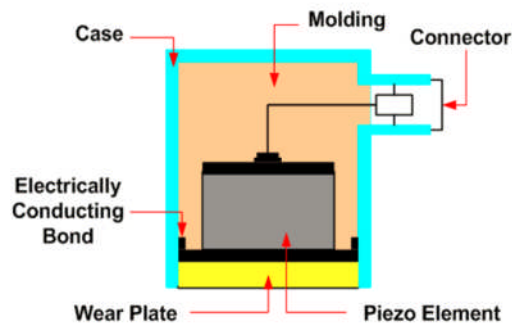


Figure 4-14 Basic setup of an AE sensor

The amplifiers' output is connected to a 6 channel system (AE win) which is installed in a computer next to the test rig. The acoustic sensors were connected to a data acquisition system through a preamplifier; set at 40dB gain. The system was continuously set to acquire AE RMS values over a time constant of 10 ms (millisecond) at a sampling rate of 2 MHz.

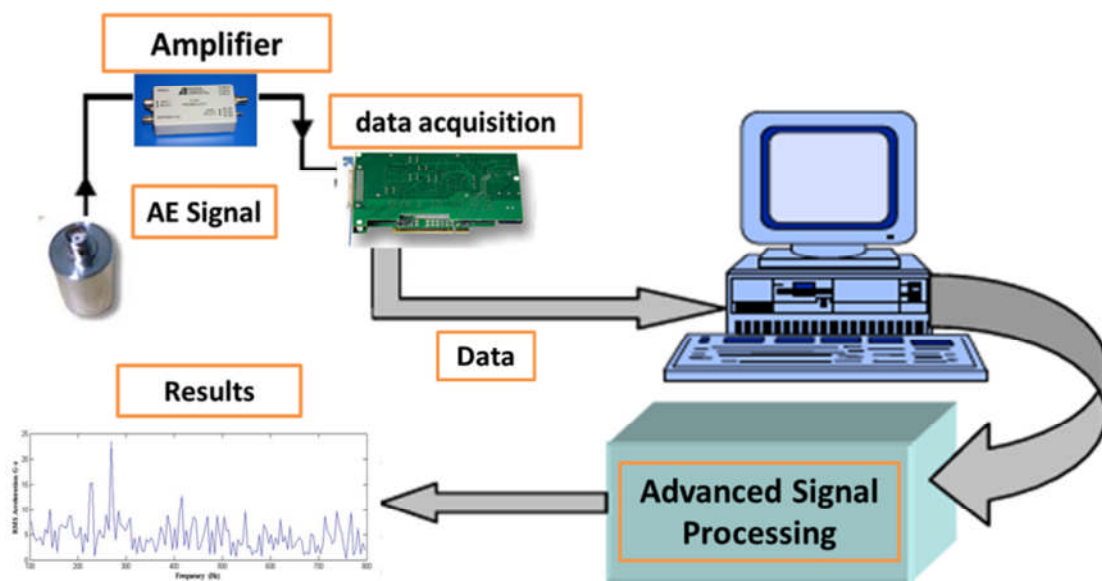


Figure 4-15 Continuous AE signal acquisition system

Figure 4-15 presents the AE measurement system developed for this research. It consists of AE sensor, preamplifier and a physical acoustic computer. The signal from AE sensor is amplified by the preamplifier and then the input signal transferring to the computer which has a built-in data-streaming capability, allowing AE waveforms to be continuously transferred to the computer hard disk at a sampling rate up to 2 MHz. The stored AE signal can be replayed and processed using advanced digital computing methods.

4.2.4 Temperature Measurement

Five thermocouples (four J-type by the side of the hydrodynamic bearing and one K-type on the oil bath) with Solid Wire Section and Diameter of 0.03 mm / 0.2 mm , Conductor Insulation of PTFE Twisted , Length of 1000 mm and operating range of 0 to 200°C (See Table 4-4), have been attached to the bearing race adjacent to the AE sensors (see figure 4-17) to enable the measurement of the temperature throughout the tests. Changes in temperature have a direct influence on film thicknesses and AE activities,

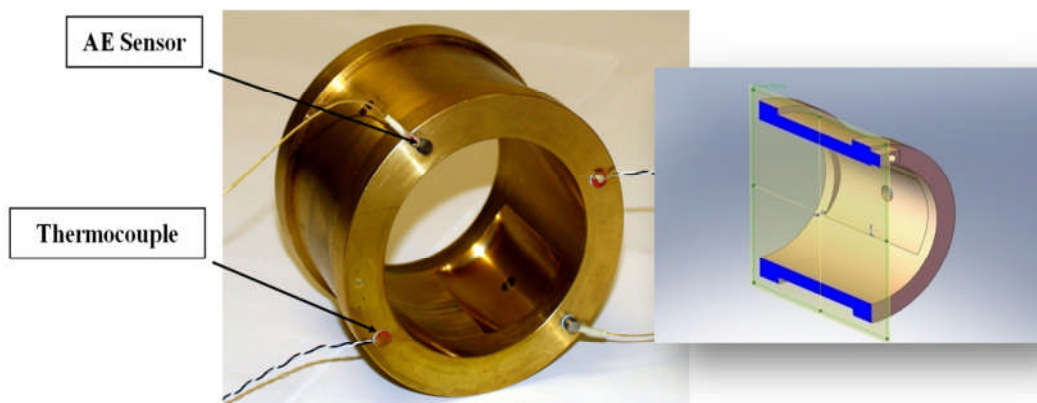


Figure 4-16 Location of AE and thermocouple sensors type on test bearing

J-Type Sensor: These are the ones attached to the bearing. They are manufactured with the combination iron-constantan (nickel-copper alloy). Its range is 0°C to +200°C, is good enough to monitor the temperatures in the bearing and its sensitivity is around 55 (mV/°C).

K-Type Sensor: This is the one responsible for the oil bath reading. It is manufactured with the combination of chromel-alumel (nickel-chromium and

nickel-manganese-aluminium-silicon). The range of temperatures is 0°C to +200°C. Sensitivity is about 41 (mV/°C).

Table 4-4 Demonstrating characteristics of Type J and K thermocouples

Attribute Type	Attribute Value	Attribute Value
Type	Thermocouple, Type J	Thermocouple, Type K
Operating Temperature Range	0 to 200°C	0 to 200°C
Standard Type	Type J	Type K
Termination Type	Lead	Lead
Accuracy Class	±1.5°C or ±0.004xT	±1.5°C or ±0.004xT
T Range	50°C to 750°C	50°C to 1100°C

4.2.5 Vibration acquisition system

To record the vibration data, an accelerometer (Model 236 Isobase accelerometer, Endevco Dynamic Instrument Division) with an operating frequency range between 10 Hz to 8 kHz was attached to the top bearing house in order to obtain a clearer signal from the test rig. The sampling rate for one complete revolution at every minute was applied at 10 kHz. The signal was sent to a scope box which calculates the RMS value of the original signal. This RMS value was sent directly to the 6 channel physical acoustic system where the vibration RMS, monitors synchronically with AE RMS, ASL and absolute energy for the same time driven rate, with the purpose of keeping under control its level and trend in real time for further analysis.

4.2.6 Displacement Measurement

In eddy current testing, a circular coil carrying an AC current is placed in close proximity to an electrically conductive specimen. The alternating current in the coil generates a changing magnetic field, which interacts with the test object and induces eddy currents. The penetration depth of eddy currents is indicated by a parameter known as the 'skin depth'. This is dependent upon operating frequency and specimen conductivity and permeability, and is typically between 5mm and 1mm. In the journal bearing test rig, the skin depth has been defined as 2mm. For most inspection applications, such as journal bearing, eddy current probe frequencies in the range 1 kHz to 3MHz are used. Usually, the

displacement sensors are mounted on the rotational shaft of the bearing test rig to determine the eccentricity, relative shaft displacement and vibration direction and vertical orientation. VK-202A1 displacement transducer and sensors are shown in figure 4-18.

VK-202A Transducer is the non-contact transducer system which consists of VL sensor, VW extension cable and VK driver. This transducer system utilizes eddy currents and measures the gap between VL sensor and measured object (shaft), without physical contact, its output voltage signal is proportional to distance. This transducer system offers 2,000 μ m (approx. 80mils) linear range for JIS SCM440 (AISI 4140 Steel) as a standard calibration material, and is the most suitable system designed to measure high speed shaft vibration, eccentricity and thrust position. Field wiring from driver to connecting instrument, such as a monitor, generally uses 3-wire shielded cable and it can be extended up to 500m (max)



Figure 4-17 VK-202A1 Driver and Sensor

The high frequency signal is supplied to the sensor from an oscillator in the driver. These high frequency signals generated by the rotational shaft will be captured by VL sensors. These signals are driven with sensor cable to VK-202A1 displacement transducer, sensor and then to the Digital Multimeter. The CSI 4500 Machinery Health TM Monitor will record these signals for further analysis (figure 4-19 demonstrate a block diagram of this process).

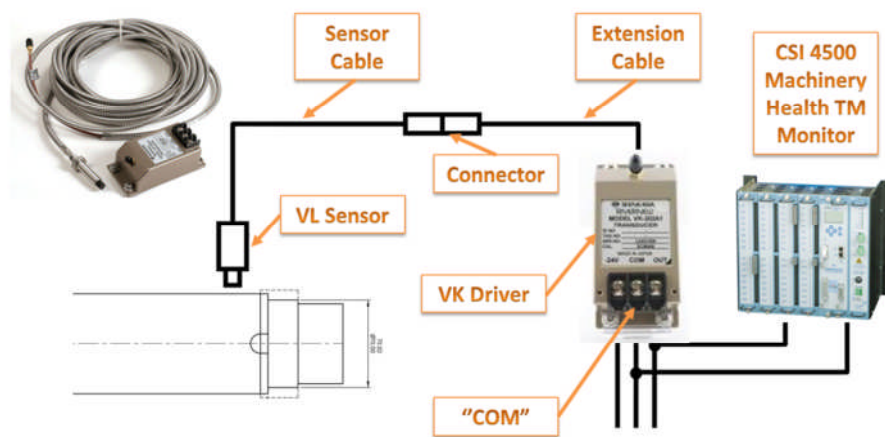


Figure 4-18 External Mounting VL Sensor

4.2.7 CSI 4500 Machinery Health™

The CSI 4500 machinery health monitoring system provides machinery performance prediction that complements an existing protection system, providing real-time feedback to both maintenance and operations. As part of Emerson's PlantWeb digital plant architecture, the CSI 4500 delivers real-time machinery health information when and where it is needed. Integration with the plant's process automation system and AMS[®] Suite: Machinery Health Manager Software empowers decision-making and reduces machinery health faults. When both prediction and protection are required, Emerson delivers with a complete asset management strategy for the most critical rotating machines. The CSI 4500M machinery health monitor is the most common model for applications in power, refining, oil & gas, petrochemical, chemical, pulp and paper, and offshore industries. The CSI 4500M Monitor (with up to 32 sensor input channels, 16 tachometer input channels and up to 16 relays) offers the best choice in flexibility and cost per channel [122,135].

The CSI 4500M Monitor includes the Trip Advisory Monitor for tier 2 critical machines, The Production State Monitor for continually changing machine operating states, and is upgradeable to the CSI 4500T Monitor, which includes transient analysis for turbines, see figure 4-20.



Figure 4-19 CSI 4500 Machinery Health™ Monitor [122]

4.2.8 NI DAQ

While the CSI 4500 Machinery Health™ Monitor, is for monitoring vibration, for journal bearing and displacement (Eddy Current) for shaft, the NI acquisition card can also be used for measuring of vibration and temperatures for journal bearing as a comparable tool, see Figure 4-21.

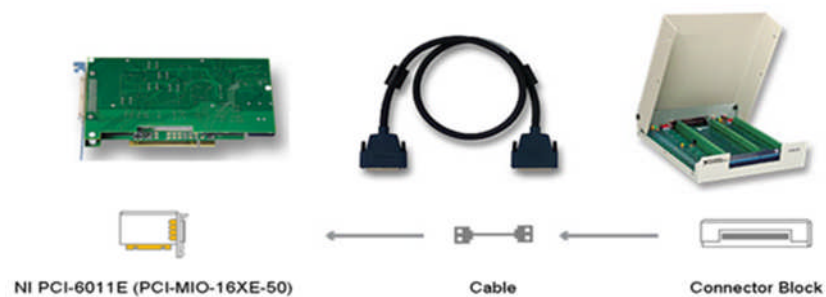


Figure 4-20 4.2.8 NI PCI-MIO-16XE-50 DAQ

Since the data acquisition device does not have direct signal connectivity, it is necessary that a shielded I/O connector block acts as an interface between sensors/signals and NI PCI-MIO-16XE-50 data acquisition device. The shielded I/O connector block provides access to the inputs and outputs of the measurement hardware. The shielded I/O connector block has 68 pins that can connect to 68-pin accessories with the SH6868 shielded cable or the R6868 ribbon cable. The cable can transmit signals between the data acquisition device and the signal connections on the shielded I/O connector block. Software is required to interface with the hardware and to collect, present and store measurements for further analysis. The DAQ board is compatible with a

variety of programming languages, including Labview, C/C++, Visual Basic and .NET. The following code has been modified and developed as labview provides the easiest integration with NI hardware, see figure 4-22.

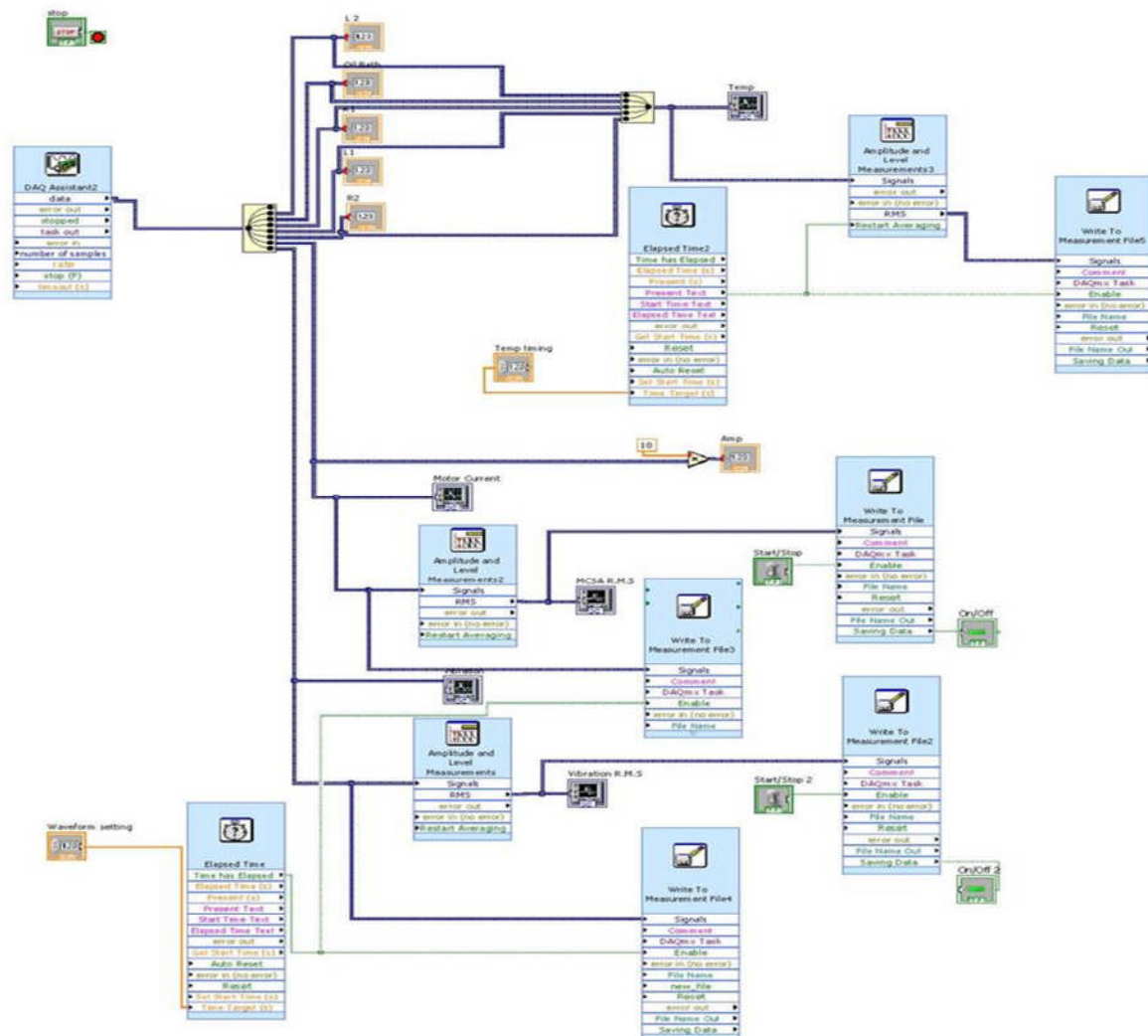


Figure 4-21 Labview diagram for measurement

To determine that vibration waveforms are more comparable to CSI 4500 Machinery Health™ Monitor, the same arrangement, sampling rate (10 KHz) and time driven rate (60 Sec) has been applied in labview.

5 NDT Technologies

The technology employed to improve the understanding of information contained in signals from sensors (in analogue format) will be described in this section.

5.1 Vibration Analysis

Vibration monitoring is a procedure for detecting the presence of vibrations and identifying their characteristics in mechanical machinery. This thesis is concerned with the vibration monitoring of high speed rotating machines such as shafts and (hydrodynamic) journal bearings. A shaft is a thick rod with a high rigidity modulus that is able to rotate about its own axis on bearings, and which is arranged to carry a load at one end with a couple at the other [105]. Bearings are essential components in machines that operate at high speed. They have high load carrying capacity and are low in cost, and they are designed to allow relative positioning and rotational freedom while transferring the speed [111]. On one hand, this makes them suitable for supporting rotors in industrial machinery with high loads, but on the other, failures in the journal bearings can lead to large losses and major safety problems, hence the need to monitor them and diagnose any problems as early as possible [103].

Examples of rotating machinery are motors, pumps, turbines, compressors and gearboxes. All such machines exhibit some constant degree of vibration as a feature of their dynamics due to being in rotatory motion. These vibrations occur either axially or radially and their magnitude can be affected by alignment and balancing elements in the rotating parts as well as due to other faults and deficiencies. The concern is usually with excessive vibrations that indicate problems in the machinery while in other cases it may be necessary to eliminate vibration altogether in order to prevent what is called 'high cycle fatigue'.

Vibration monitoring is an important practice in predictive maintenance as it can determine the condition of mechanical equipment and identify problems early at the developing stage before they could otherwise become serious and cause

unscheduled downtime [95]. Vibration monitoring can detect, for example, mechanical looseness, eccentricity, worn or damaged gears and deteriorating or defective bearings. Regular vibration and other types of monitoring should therefore be undertaken as a routine part of equipment maintenance. By trending the vibration levels, it is also possible to detect poor maintenance such as an improper installation or replacement of a bearing or an inaccurate alignment of the shaft. Vibration monitoring can therefore reveal, for example, how well a shaft has been aligned and balanced, or the condition of a bearing. In fact, misalignment and imbalance along with thermal growth are the most common problems in mechanical systems [100].

Some typical faults detected using vibration monitoring for shafts and journal bearings are listed in the table 5-1. Problems with mechanical machinery containing high-speed rotating parts are often due to bearing failure. In a study on machines greater than 75kW, over 40% of motor failures were attributable to problems related to bearings [108]. Journal bearings can suffer, for example, from 'oil whirl' (or 'oil whip'). This is an instability that occurs when the bearings are loaded lightly and the shaft whirls at a frequency that is close to half of the rotor angular speed [94]. Vibration monitoring is usually an effective procedure and it has the advantage of being non-intrusive. It is often in combination with other non-intrusive condition monitoring techniques such as acoustic emission and thermographic imaging [110].

Vibration monitoring is not considered to be the suitable tool, for low-speed rotating machinery because of the typically low energy loss rates and difficulty in measuring their fundamental frequency of operation [92]. However, low-speed journal bearings are an exception as they have been shown to be capable of being subjected to vibration monitoring using band pass filtering to eliminate noise [99].

Table 5-1 Typical faults and defects in shafts and journal bearings can be detected by vibration monitoring [97]

Typical fault	
Shafts and rotors	Unbalance Bent shaft Misalignment Eccentric journals Loose components Rubs Cracked shaft Blade loss Blade resonance
Journal bearings	Oil whirl Oval or barrelled journal Journal/bearing rub

A bearing is considered as damaged if its peak level differential and number of occurrences exceed a certain limit when compared to similar bearings.

5.1.1 Vibration characteristics and monitoring methods

The basic components of a vibration monitoring system are given in table 5-2. The steps involved are the same for all condition monitoring techniques, which are data, acquisition, data processing and data assessment [91]. Vibration monitoring can be undertaken either during normal operation or during the start-up and shut-down phases.

Table 5-2 The basic components of a vibration monitoring system

Signal pickup (transducer)	Signal analyser	Vibration analysis software	Computer for conducting the Analysis and storing data
----------------------------	-----------------	-----------------------------	---

During analysis, two components of the vibration signals are of importance for monitoring, namely frequency and amplitude. Standard methods for measuring vibration include the overall vibration, phase, acceleration enveloping, high frequency detection and other sensor resonant technologies [109].

Excessive horizontal vibration is usually a good indicator that there is an imbalance while vertical vibration suggests stiffness from mounting and gravity. Axial vibration indicates the possibility of misalignment and bent shaft problems. Ideally therefore, there should only be very little axial vibration. The readings taken for each of the three planes would determine the actual cause of the excessive vibration. For this, it is recommended to take these readings as close as possible to the bearings and to avoid taking readings on the case, which could indicate that the vibration is occurring due to resonance or looseness instead. Trending the readings helps to evaluate the severity of the vibration and a method of enveloping can be used to filter out low frequency vibration signals due to rotation. The forces that act on the journal bearings can also be modelled as a spring and damper system. Devising a model of their normal vibration behaviour can be useful for comparing experimental data with the simulation results to test for instability in the motion of the bearings. Changes in the position or distance relative to the normal value are its displacement.

The general requirement for measuring and evaluating vibration in machines consisting of rotating shafts is governed by ISO 7919. This standard identifies four evaluation zones for a qualitative assessment of shaft vibrations, as shown in the table 5-3.

Table 5-3 Standard identifies evaluation zones of shaft vibrations

Zone	Description
A	Typical of vibrations in newly commissioned machines
B	Acceptable (for unrestricted long-term operations)
C	Unsatisfactory (for continuous long-term operations)
D	Severe enough to cause damage

Often, special displacement probes are used to monitor vibrations in shafts relative to the machine's casing, especially for machines containing fluid film bearings. Displacement is detected if the machine and the shaft do not move

together. Sensors also exist to detect the rate of change in displacement, i.e. they function as simple velocity sensors, but they prove to be ineffective at very low frequencies below 10 Hz [109].

Although velocity meters can be used for measuring vibration, it is normally necessary to apply a filter to such a device for any practical use [96]. The filter provided adequate resolution but tends to be tedious to operate. The main instrument now used for measuring and analysing of vibration is the Fast Fourier Transform (FFT) spectrum analyser, which has powerful diagnostic abilities. FFT based spectra also have the capability to provide the maximum possible resolution (Randall, 2011: 156). FFT spectrum analysers have now become more competitive in cost to the traditional analysers though they remain complicated to use [113, p1].

5.2 PeakVue

PeakVue is short for 'peak values', which refers to the peaks of the stress waves generated by mechanical objects. PeakVue (or stress wave) data analysis is a method that is used exclusively to measure stress wave activity in a metallic component [128]. Stress waves are defined as the transient elastic waves generated on the surface of a material as a consequence of friction between two mating parts. The breaking of surface irregularities or asperities and relative motion of entrapped broken debris will result in generation of stress waves. PeakVue provides a measure of true peak accelerations at high frequencies. The purpose of the analysis is to detect and quantify the stress wave activity relative to the energy and rate of repetition so that any faults or impending faults can be recognised and assessed as early as possible [128].

PeakVue analysis has proven to be especially effective for identifying bearing defects [122] and defects with gears. Often, the severity of the defect can also be detected, as was shown by Emerson (2011) for race, rolling element and cage related defects. It can also identify demodulation in transitional frequency bands. That is, it can work at operational speeds of between 50 and 6000 rpm [116]. It is also proven to be a far superior method for detecting faults than

demodulation methodology [129] because it is able to maintain true signal levels that occur at the point of impact. For example, Ying-bo et al. [125] compared its performance to the traditional method based on three-band enveloping demodulation for detecting a bearing fault. The Matlab simulation results showed PeakVue to have improved the speed and accuracy of detection over the traditional method. However, PeakVue is still best used as a complementary tool to detect a range of conditions and faults rather than as the panacea of condition monitoring tools [128]. The following is a summary of key advantages of the PeakVue methodology:

- ❖ Faults can be detected on either fast or slow moving machinery.
- ❖ It can identify demodulation in transitional frequency bands.
- ❖ It is able to maintain true signal levels at the point of impact.
- ❖ PeakVue is trendable; as it captures the true amplitude [116] so it is suitable for routine condition monitoring
- ❖ It is best suited to detect high amplitude stress waves with short duration [116]
- ❖ The results are not so dependent on the filter settings.

The PeakVue methodology has already proven to be superior to traditional methods for detecting faults and to detect faults quickly so that losses can be avoided. As an example, at Siemens Ltd, the PeakVue methodology enabled them to completely eliminate demodulation methodology and it proved to be especially reliable for detecting anti-friction bearings and gear related problems, even in machinery operating at 20 rpm [116]. Hauck & Truiempi [117] mention another incident in which PeakVue was able to detect a bad bearing in the drive end of a motor by showing an abnormal reading. As a precaution, the maintenance engineers increased its lubrication but when the vibration level exceeded the normal level by 30 times and the PeakVue readings reached 117G-s, the indication was taken very seriously. The problem was that

lubrication was failing to reach the bearing; this could have seized the machinery and caused severe damage to the motor, costing more than \$40,000 to repair. The bearing was promptly replaced.

5.2.1 The generation of stress waves

In many cases involving rotating machinery, technical faults manifest themselves as vibration with mode excitation and as stress wave initiation [122]. Stress waves are generated by abrasive wear and tear, looseness, friction, loss of lubrication, contamination, cavitation, metal to metal impacting, fatigue cracking, faults in bearings or gears, etc. For example, a series of stress waves could be generated if there is a defect such as fatigue cracking on the bearing raceway as a result of an impact by the rolling element. If there is a cracked gear tooth, or pitting in an anti-friction bearing race this causes the roller to make an impact. In detecting bearing faults, care must be taken to identify the cause of the stress wave that may or may not necessarily be due to a fault with the bearing.

These waves usually propagate away from the location of the defect in numerous directions. The wave propagation then causes a ripple effect on the surface of the machine, as illustrated in figure 5-1 for one large and one small object, which results in a lower and higher stress wave frequency respectively. This in turn develops a response output in a sensor to enable the total movement to be detected. The emissions from the stress waves typically last only microseconds, or at most a few milliseconds. It is therefore a short-term transient phenomena but its duration is affected by the nature and severity of the fault and the relative location of the sensor. Usually, the dominant frequency band of the resulting stress wave packets lie in the band between 1 and 50 kHz [122].

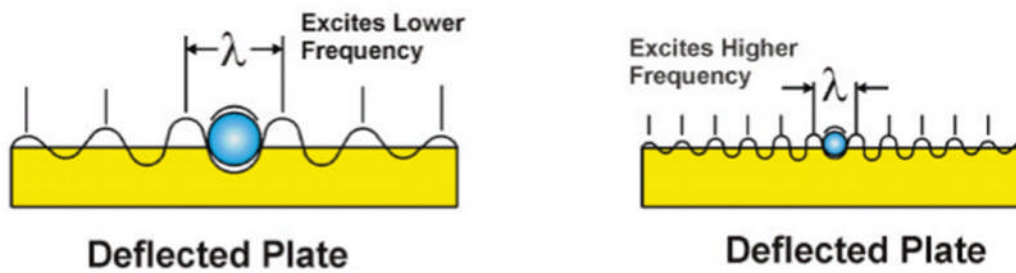


Figure 5-1 Generation of stress waves by impact and their propagation along a metal plate Source [122]

5.2.2 The detection of stress waves

The detection of stress waves is made possible using motion sensors. These can detect either absolute motion or relative motion. Accelerometers, AE sensors or strain gauges can be used but accelerometers are the most common devices used and they are typically having 100 mV/g of sensitive of accelerometers. Their output is in either g or velocity units. Such sensors are able to detect faults more quickly and accurately than a manual detection. However, a correct interpretation of the signals is essential. The maintenance engineer must be skilled in reading the sensor output information. With the advent of sophisticated electronic based sensors, the detection and interpretation process has become complex.

5.2.3 The PeakVue Methodology

As described above, the PeakVue methodology is characterised by the capturing of peak values of the stress waves that are generated. These stress waves are present due to the metal to metal contact that occurs during the early phase of a failure. Following a PeakVue analysis, further diagnosis of any fault is then made using spectral measurements [122]. The spectral analysis is undertaken, in order to obtain the repetition frequency. The typical method used to process the stress wave signals using PeakVue and spectral analysis in sequence is detailed in section 5-2.3.2. However, the four process steps listed below are the defining components of the methodology [95]. In addition, it

should be noted that only high frequency data is used to extract the signal and no enveloping or rectifying takes place using a low-pass filter.

1. A high-pass filter is used to remove low-frequency content in the acceleration signal.
2. The signal is made to pass through an analogue to digital converter.
3. Each sample of the signal is carefully scrutinised so that only the amplitude level exceeding a certain trigger level is assigned a digital value.
4. The digital information is then processed by an FFT algorithm so that the spectrum only displays fundamental peaks and harmonics with a frequency equal to the pulse.

The most important parameters that should be captured from stress wave activity for PeakVue analysis are the following:

- ❖ Amplitude of event
- ❖ Approximate time taken during the occurrence of the detected event
- ❖ Periodic/Non-periodic rate at which the events occur

For the third parameter, it is important to capture the rate of the event relative to specific fault frequencies that are dependent on the speed of the rotating machine as well as the specific component in focus. For trending the measurements, the PK-PK (peak-to-peak) waveform is mainly used, as in PdM (Predictive Maintenance) software. This is the distance between the positive and negative peaks, which is independent of the analysis bandwidth and it has been shown to be reliable for detecting impact related faults [122].

5.2.3.1 Demodulation vs. PeakVue

The PeakVue methodology was developed by the CSI (Computational Systems Incorporated) division of Emerson Process Management. The PeakVue methodology is similar to the Envelope demodulation analysis, also known as

High Frequency Resonance Technique (HFRT) or demodulated resonance analysis, in that the resonance zones are isolated using a digital high pass or band pass filter. An envelope detector is not used in the final stage [114, 164]. The basic structure for enveloping analysis of vibration signals has been illustrated in figure 5-2.

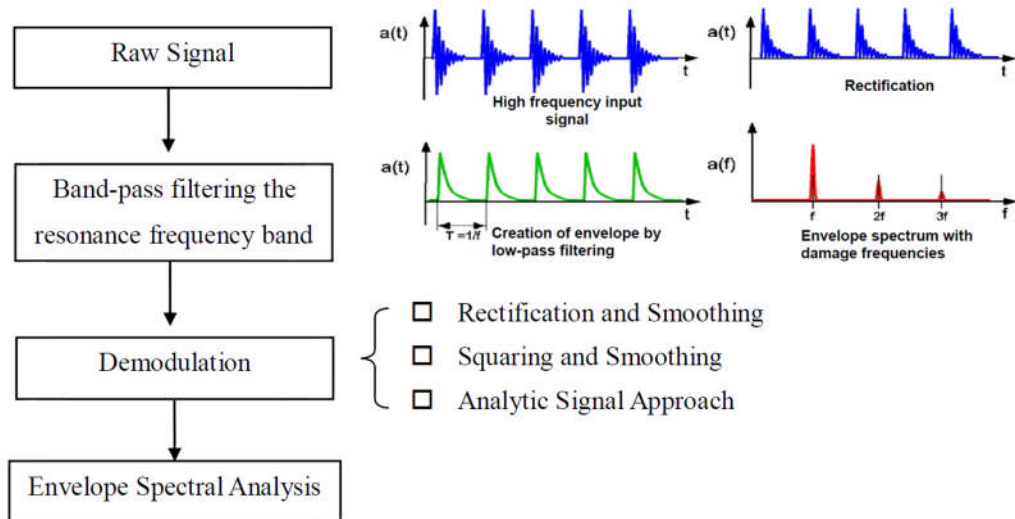


Figure 5-2 Demodulation process [164]

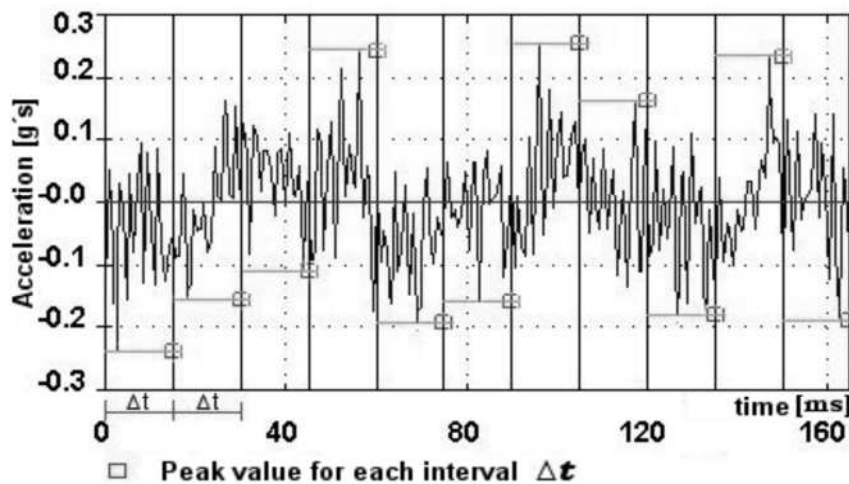


Figure 5-3 Peak Value detection using a high sampling frequency [114]

High frequency sampling of the order of 100 kHz is used to capture the peak value at the normal sampling time for each time interval, as in the example illustration in figure 5-3. In comparison, the amplitudes obtained using PeakVue techniques tend to be higher than when using demodulation, which makes it

easier to detect faults. The table 5-4 summarises some of the key differences between PeakVue and standard demodulation. It serves not only to contrast the two methodologies but also to highlight further important characteristics of the PeakVue methodology.

Table 5-4 Key different between PeakVue and Demodulation

Feature	Demodulation	PeakVue
Use of demodulation	Yes	No
Type of filter used	Low pass and high pass	Only high pass
Values measured	Average values	Peak values
What is detected?	Frequency	Amplitude and frequency
Is it trendable?	Only in isolated cases	Yes
Indication of severity?	None	Amplitude correlates with severity of fault
Diagnostic information	Contained in the spectrum	Contained in both spectrum and waveform

5.2.3.2 Signal processing

The vibration sensor produces an analogue signal; this is normally directed through an analogue signal processing unit first. The signal is then converted into a digital form and processed further, digitally. Usually, a high order low pass filter is used prior to the analogue to digital conversion in order to remove any components of the signal that exist at higher frequencies than the Nyquist frequency, which is half of the sampling rate. This also ensures that the digital converted form of the original analogue signal is correct in that a reconstruction of the band limited analogue signal before the conversion could still be made if necessary using the digital signal. A low pass filter either at or just below the Nyquist frequency, is not used. If an accelerometer is used as the sensor, its output is measured in g or m/s^2 units, and processing the signal may also involve converting the signal from having acceleration to velocity units using an analogue integrator.

As this is a an analysis of peak values, only the absolute maximum values of the digital data block are used, as experienced by the time waveform for each time increment according to the sampling rate. After acquiring a block of digital data at a constant sampling rate of a certain length, further processing of the signal takes place. For rotating machinery, this is usually done by applying a Fast Fourier Transform (FFT) algorithm in order to construct the spectrum in

acceleration or velocity based units. Spectral analysis is undertaken because it can separate band-limited signals into periodic components with respect to the machine's turning speed. Auto-correlation analysis can also be applied to the digital data block that represents the time based waveform. Its usefulness is in analysing the waveform while implementing PeakVue analysis to identify the main source of the periodicity.

During signal processing, a fault could be indicated by excessive activity when compared to the previous condition monitoring of the part or to other similar parts at known discrete fault frequencies. If a bearing fault is identified in the PeakVue and normal velocity spectral data, then it is expected that both methods be used for trending the signal values to identify the severity of the fault.

5.2.4 Application: bearing fault detection

An example of a PeakVue data analysis tool is the Model 682A05 Bearing Fault Detector. It is specially designed for detecting faults in ball bearings using high frequency PeakVue data [129]. This model provides a 4mA to 20mA output signal, which can then be monitored using a conventional process monitoring tool and it allows for at least 6 revolutions of the machine to be monitored for stress wave activity. A similar secondary signal is also produced for low frequency vibration. The claimed advantage of this model is that it can be operated continuously, it is described as compatible with existing process monitoring equipment and is easier to use so requires less training to operate.

The methodology employed by this model is typical of the general methodology of PeakVue analysis for bearing fault detection, see figure 5-4. The outline below was prepared from the description of the methodology. The requirement of obtaining the absolute g-level over at least 15 revolutions is specific to the 682A05 model. Testing of the device revealed that the output provided is representative of the peak g-levels experienced by an accelerometer for high frequency components of the signal. It was concluded that the peak g-level is a reliable detector for detecting faults and their severity.

For detecting bearing faults, it is recommended the block time should be sufficient to provide an adequate resolution of the lowest fault frequency, which is called the cage fault [133, 134]. This justifies the minimum of 15 revolutions. During the bandwidth analysis, it is recommended to set the high pass filter to twice the highest gear mesh in order to help capture possible back lashing. Trending of the data helps to identify consistency in the pattern of peak values and facilitates the severity assessment for the bearings. The FFT algorithm helps to analyse the repeatable impacting presence related to the machine's turning speed, and the auto-correlation methodology allows for extracting periodic signals from those comprising of significant non-periodic noise.

Figure 5-4 Outline of the PeakVue methodology for bearing fault detection

1. SELECTION OF ANALYSIS BANDWIDTH

Should be sufficient to contain the expected maximum fault frequency with few harmonics



2. SELECTION OF HIGH PASS FILTER

Equal or greater than the maximum frequency selected



3. SELECTION OF NUMBER OF LINES FOR THE SPECTRAL ANALYSIS

Ensures data capture during at least 15 revolutions



4. PEAKVUE TIME BLOCK OF DATA

Consists of absolute peak values from the observed time waveform for each time increment



5. PERFORMANCE OF SPECTRAL ANALYSIS

Peak g-level is noted for trending and assessing fault severity

5.3 Eddy currents measurements

The eddy current non-destructive technique is a well understood, widely used tool in predictive maintenance. As with some other methods, like Remote Field Testing, Flux Leakage or Barkhausen noise, eddy currents are based on electromagnetism [149]. Eddy currents are flows created by electromagnetic

induction where induction is the physical relationship between the electric flow and magnetic field. When an alternating current passes through a coil, there is flux variation as there is a current variation, which leads to the generation of electromotive forces (emfs). Then, an alternative magnetic field develops in the proximities of the coil. This magnetic field raises its peak value when the current reaches its peak value, and collapses when the alternating value of the current crosses the zero-value. Since eddy currents are based on the generation of magnetic fields, their origins are found within the first electromagnetic laws, proposed by Michael Faraday in 1831 and Heinrich Lenz in 1834. Before the development of electromagnetic theories, in 1824, Oesterd discovered that when an electric current is flowing in a conductive coil, it creates a magnetic field able to move a needle placed in the centre of the coil – it was the discovery of the relationship between electricity and magnetism [154, 155]. Lenz's law, based on the balance of energy of the non-conservative forces in electromagnetic induction, proposes that the direction in which an induced current moves is always opposite to the direction in which the source that generated it is. It was proposed that "The induced EMF acts in such a way as to oppose the change in flux" i.e. it acts to generate currents in the circuit whose associated magnetic effect would counteract the external change [152, 153].

On the other hand, Michael Faraday proposed the relationship between the emf and the magnetic flux [154] - his law reports that any change in the magnetic flux passing through a coil will result in the induction of voltage in the proximities of the coil. This induced voltage is named emf. Based on the works of the pioneers in magnetic induction works, it was during the World War II period when this technique was mainly developed, especially in the aircraft industry.

The technique consists of the generation of eddy currents in a coil placed close to the target element, but not in contact. When an AC current passes through a conductive coil, it induces a circulating current, known as an eddy current, on the surface of the coil. These eddy currents will create impedance dependent on the machines. This generated impedance is dependent on the magnitude and phase of these currents, and impedance generation is the basis of the eddy

current testing tool, so an increment of the impedance will cause a decrement of the eddy currents. The record of the voltage signals from eddy current sensors, and their display, is the last step in the eddy currents-based predictive maintenance.

Eddy currents are normally used in defect detection. When there is a defect in the surface close to the sensor, the distance between the coil and the element is modified, so that it results in a variation of the impedance, which is the same as a modification in the eddy current flow [151].

Eddy currents are also employed to calculate displacements of elements. The gap between the sensor and the target element is measured according to the value of the impedance. The employment of this technique is widely used in rotating machinery to calculate shaft displacements when they are spinning. In figure 5-5, the location of the eddy current displacement sensors in a shaft displacement test are shown.

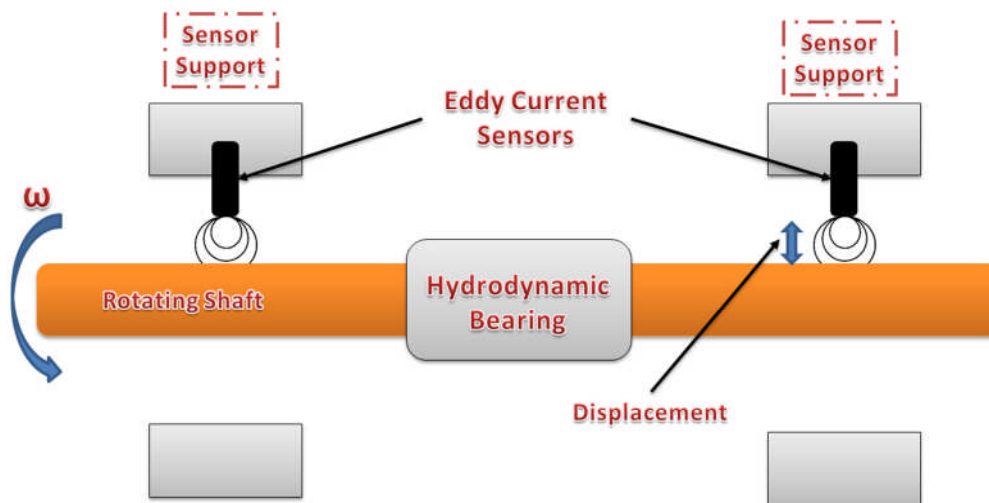


Figure 5-5 Eddy current sensors position

To put it briefly, the eddy current test is a quite widely used tool in condition monitoring for surface defects and elements displacement, based on electromagnetic impedance. The employment of this technique has several advantages, especially important being the fact that it is very sensitive in the detection of small, superficial or near surface defects, it provides on line results when performing, and the equipment is portable. However, there are some

limitations in its use, which are that only conductive materials can be tested with this method, and their surface must be accessible. This method requires a good surface treatment and finish of the target object, otherwise, the surface roughness can interfere with the accuracy of results. Probably, the greatest disadvantages lie in the fact that it is a superficial method, so the depth at which failure mechanisms can be detected is limited and failures cannot be detected when the failure growth or development is parallel to the coil sensor direction [157].

6 Operational Results and Discussions

In developing an understanding of the relationship and influence of operational variables on AE over a much broader operating range, such as oil viscosities, speeds and load conditions further research was required. Parameters needed to be added in more complete forms to fully study the sensitivity of AE in comparison with other parameters such as power loss, friction factor, shear stress, and film thickness. The results are described in the following topics.

This investigation involved varying the bearing rotational speed for fixed loads and also investigating the influence of different viscosities whilst recording the AE RMS levels. Prior to performing the tests, the temperature of the test bearing was raised to approximately 40°C; this was achieved by running the test-rig in excess of 1000 rpm at 1kN. As soon as the desired temperature was reached, the test sequence began. All efforts were undertaken to ensure no air bubbles were captured within the oil flow circuit.

The tests conditions investigated included five rotational speeds (750 rpm, 1500 rpm, 2500 rpm, 3500 rpm and 4500 rpm) three oil types (SP68, SP220 and SP460) and three load conditions, (2kN, 6kN, 10kN). Load on the test hydrodynamic bearing was achieved with two hydraulic mechanisms either side of the bearing. Two hydraulic cylinders were connected to two loading bearings (ball bearings) which were fixed on the shaft either side of the hydrodynamic test bearing. These were placed either side of the bearing to ensure uniform loading along its length. Given that these bearings are known to emit significant AE noise, a circumferential groove was cut into the shaft and a plastic strip was placed between the inner race of the bearing and the shaft. This ensured no AE transmission from the rolling element bearings could influence measured AE from the journal bearing. In addition, the drive motor was connected to the drive shaft via a belt drive thus eliminating AE noise generated from the electric motor though the belt did introduce its own vibration.

Table 6-1 Temperature, AE r.m.s and Film Thickness for different location of Journal Bearing

Load (kN)	Oil Type	Speed (rpm)	Time Duration	Tempreture R1(°C)*	Tempreture R2(°C)*	Tempreture L1(°C)*	Tempreture L2(°C)*	Film Thickness (μm)	Oil Bath Tempreture(°C)	r.m.s Ch1 (Volt)	r.m.s Ch2 (Volt)	r.m.s Ch3 (Volt)
2	SP 220	750	10	35.25	40.17	38.98	36.22	0.0339	25.51	0.007	0.006	0.009
2	SP 220	1500	10	46.8	52.88	50.69	46.64	0.0476	27.1	0.062	0.052	0.06
2	SP 220	2500	20	57.38	65.29	62.37	55.36	0.0559	29.59	0.185	0.16	0.165
2	SP 220	3500	20	65.18	76.15	73.51	65.23	0.0567	39.1	0.42	0.36	0.32
2	SP 220	4500	20	75.5	88.2	81.91	72.14	0.0574	42.38	0.64	0.54	0.46

Load (kN)	Oil Type	Speed (rpm)	Time Duration	Tempreture R1(°C)*	Tempreture R2(°C)*	Tempreture L1(°C)*	Tempreture L2(°C)*	Film Thickness (μm)	Oil Bath Tempreture(°C)	r.m.s Ch1 (Volt)	r.m.s Ch2 (Volt)	r.m.s Ch3 (Volt)
6	SP 220	750	10	43.62	47.78	46.58	42.81	0.0097	27.33	0.0102	0.0098	0.014
6	SP 220	1500	10	57.66	52.44	55.15	48.61	0.0164	29.83	0.065	0.06	0.067
6	SP 220	2500	15	62.43	68.66	65.52	59.15	0.01858	33.27	0.182	0.173	0.177
6	SP 220	3500	15	73.91	79.37	76.24	68.77	0.01848	40.29	0.36	0.345	0.32
6	SP 220	4500	12	89.49	83.18	86.57	78.9	0.01997	43.97	0.55	0.51	0.48

Load (kN)	Oil Type	Speed (rpm)	Time Duration	Tempreture R1(°C)*	Tempreture R2(°C)*	Tempreture L1(°C)*	Tempreture L2(°C)*	Film Thickness (μm)	Oil Bath Tempreture(°C)	r.m.s Ch1 (Volt)	r.m.s Ch2 (Volt)	r.m.s Ch3 (Volt)
10	SP 220	750	10	46.35	51.44	49.4	45.15	0.0078	29.06	0.0121	0.0116	0.0155
10	SP 220	1500	15	54.21	59.33	57.61	50.86	0.0122	30.49	0.061	0.058	0.07
10	SP 220	2500	15	65.4	71.17	68.21	63.17	0.0146	37.76	0.171	0.167	0.175
10	SP 220	3500	20	78.12	83.1	80.19	73.34	0.0135	41.86	0.34	0.331	0.33
10	SP 220	4500	10	86.27	94.28	93.7	88.41	0.0094	44.35	0.46	0.48	0.5

*

R1: Sensor 1 Right side of bearing

R2: Sensor 2 left side of bearing

L1: Sensor 1 Right side of bearing

L2: Sensor 2 Left side of bearing

Acquisition of AE and temperatures values were acquired continuously over duration of 10min for every speed, viscosity and load condition tested, see table 6-1. Values presented, hence forth are averaged over the 10min duration, in addition, standard deviations are also detailed (see table 6-1 and appendix D). Over 10,000 data files, averaged for calculating minimum film thicknesses, power loss, shear stress and friction factor. The temperature rises for a typical thermocouple at each side of the hydrodynamic bearing are presented in figure 6-1. These significant rises in temperature are due to running the journal bearing at varying rotational speeds at the constant radial load of 2kN, 6kN and 10kN. From observations, see figure 6-2, it is evident that by increasing speeds the oil bath temperature also showed a significant rise.

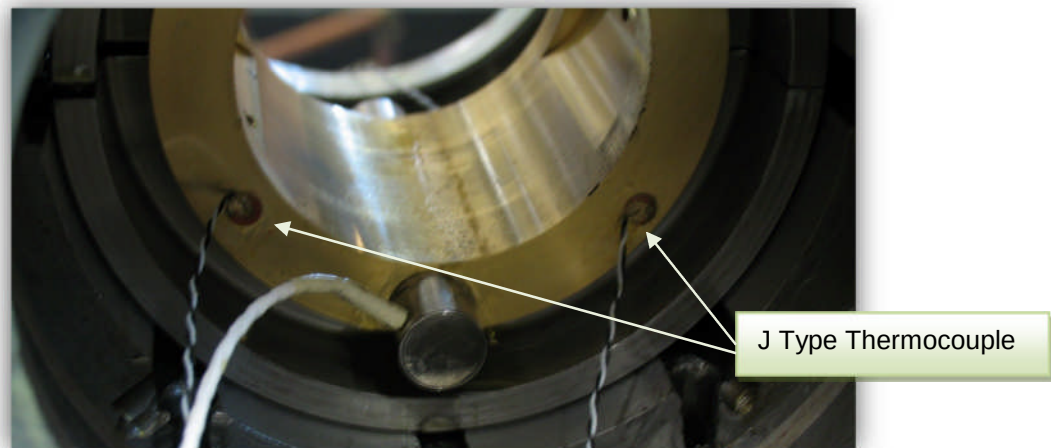


Figure 6-1 The layout of the sensors in the bearing

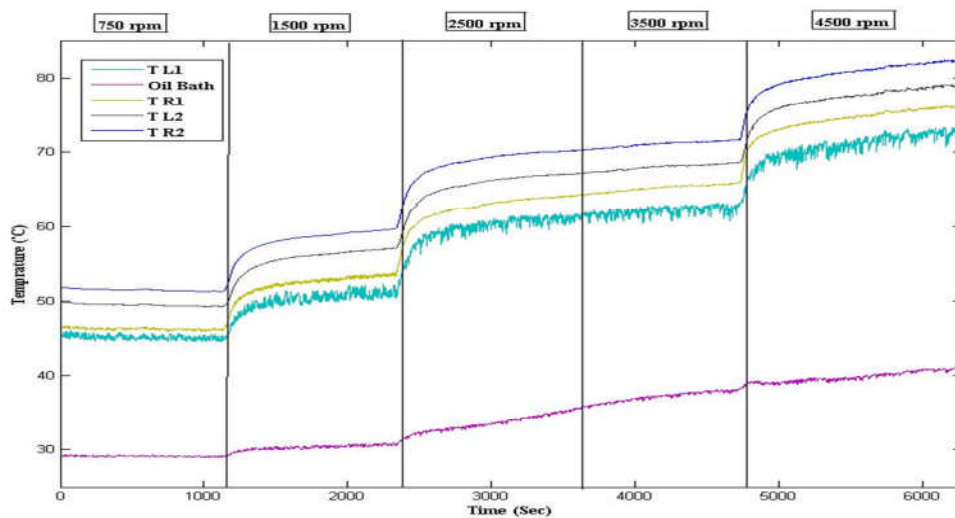


Figure 6-2 Temperature for each thermocouples and effect of rises due to speed at 10kN load condition

6.1 Operational Experimental Results

From these results and the given attenuation rates already ascertained, measurements of acoustic emission on the hydrodynamic bearing will be devoid of any contributions from the load carrying rolling element bearing, see section 4.1.5. For instance, AE measurements from ch-3 at 4500 rpm were noted at 0.14V RMS; given the attenuation across the plastic strip to the sensors on the test bearing, it would be expected that channels 1 or 2 would record AE levels in the order of 0.003V RMS on the bearing. However, AE levels on the test bearing were between 10 and 20-fold higher than at 0.003V RMS, confirming that the measured AE on the bearing could be attributed to emissions from the test bearing only.

6.1.1 Power Losses Results

The most important item to note in figure 6-3 and figure 6-4 is the relationship between rotational speed and theoretical power losses of the hydrodynamic bearing (in the region of the minimum film thickness) where it is evident that power losses of the bearing do not affect in theory the film thickness and friction factor. The increment of the load is approximately linear with speed. The load only has a slight effect on the power losses, speed being the main parameter that characterises the friction losses, see equation 2-2.

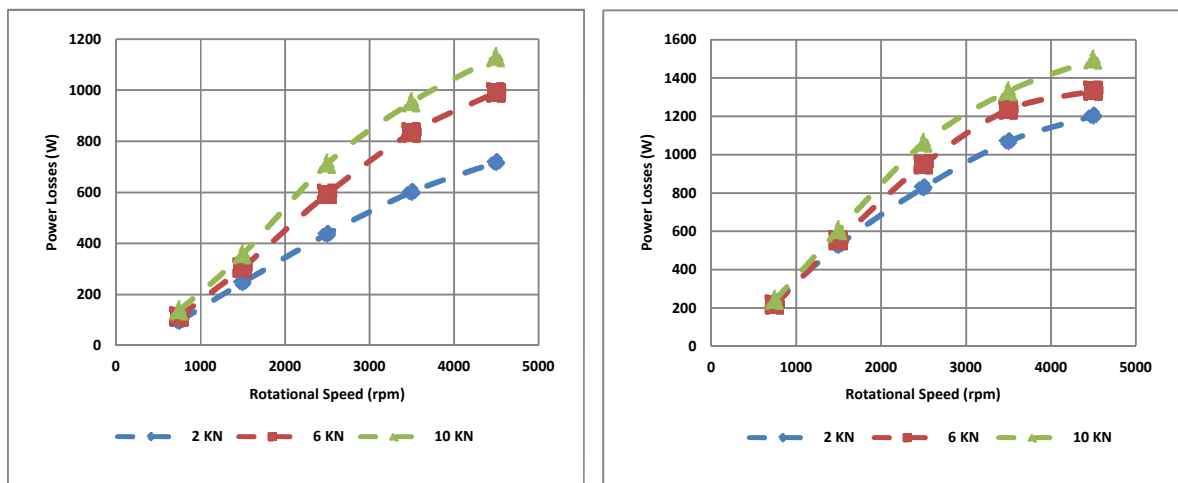


Figure 6-3 Power Losses vs rotational speed at SP 68 and SP220

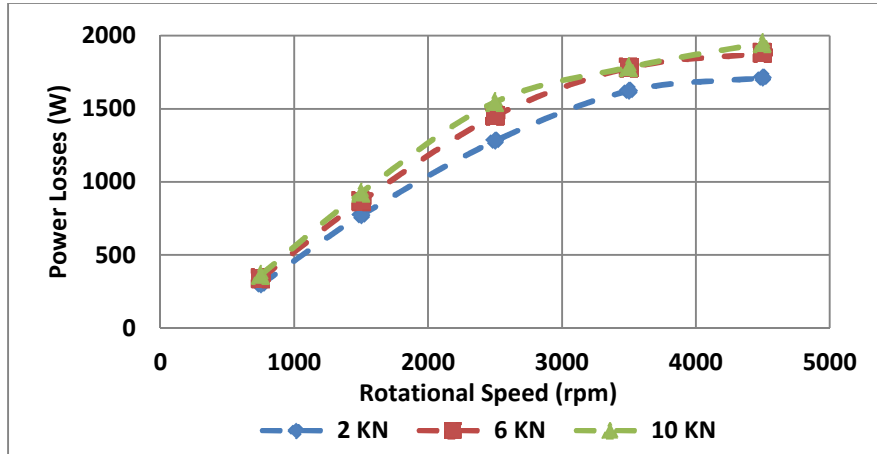


Figure 6-4 Power Losses vs rotational speed at SP 460

From the figures 6-3 and figure 6-4, it is clear that power losses have an important relationship with the oil type, i.e. the more viscosity, the more power lost by friction. The high effect of the oil type in the power losses outshines the effect that a higher load should have. Therefore, power losses are mainly affected by speed and density, and the load effect is not relevant – see figure 6-5 in which the effect of the density and the viscosity of the losses due to friction is shown. For instance, oil type SP460 has a density of 900 kg/m^3 while SP220 has a density of 895 kg/m^3 and SP68 has a density of 880 kg/m^3 , so apparently differences in density and viscosity slightly lead to much higher power losses in the denser oil.

The viscosity has an indirect effect on the friction losses, but it is not going to be considered because the main effect that leads viscosity to be lower and the power losses to be higher is the velocity,

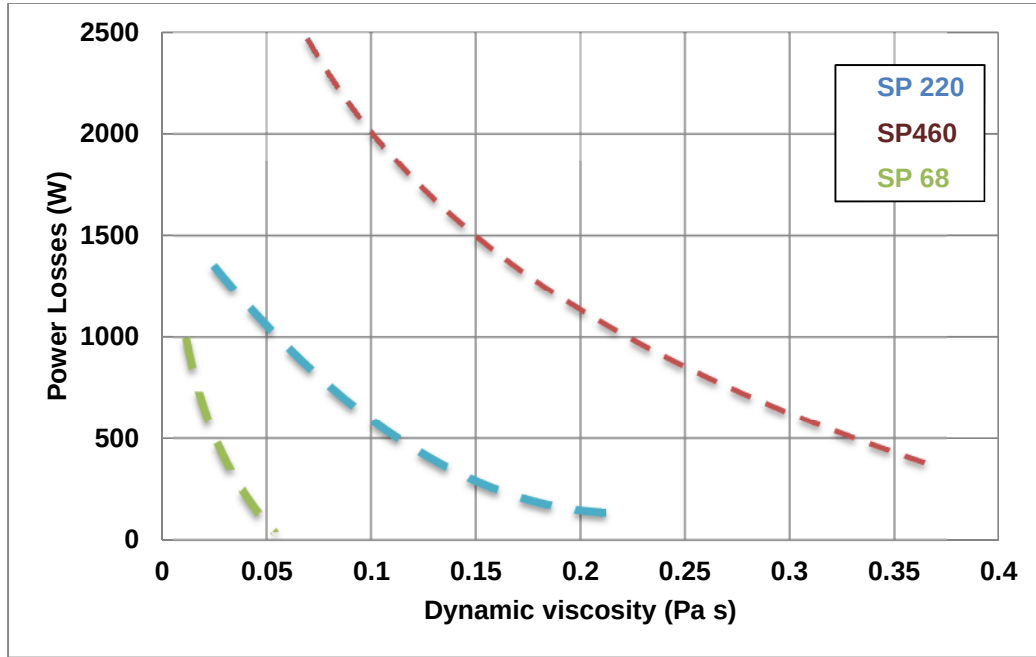


Figure 6-5 Power Losses vs Viscosity for Different type of Oils

6.1.2 Film Thickness Results

The minimum film thickness is another dynamic factor to establish the relationships of the shear stresses because the shear stresses are inversely proportional to the square of film thickness, see equation 2-34.

The maximum film thickness does not necessarily match the edge values of speed but it will match for medium values. According to the mathematical model, the minimum film thickness is read in a Raimondi-Boyd [136] or Bannister [138] graph; it is therefore dependent on the Sommerfeld number, see equation 2-4; for a given $\mathcal{S}$ there is a corresponding constant $\phi_{\min}/c$ being the radial clearance, a geometrical value, which is constant.

Since the constant $\phi_{\min}/c$ increases as $\mathcal{S}$ grows, and since c is constant, the film thickness $\phi_{\min}$ grows as the Sommerfeld number grows. In order to refresh, the Sommerfeld number is a characteristic value for a journal bearing obtained through the equation 2-4

So, for a constant load, there are two contradictory effects that affect the Sommerfeld number behaviour, see equation 2-4.

- The speed effect: as the speed increases, Sommerfeld number increases
- The viscosity effect: the Sommerfeld number is reduced as the oil viscosity decreases, the oil viscosity decreases as the temperature increases, and the temperature increases as the velocity increases.

Then, as can be seen in figure 6-6, the relationship between the Sommerfeld number and the minimum lubricant film thickness is practically exponential, where oil type; load or velocity had no effect.

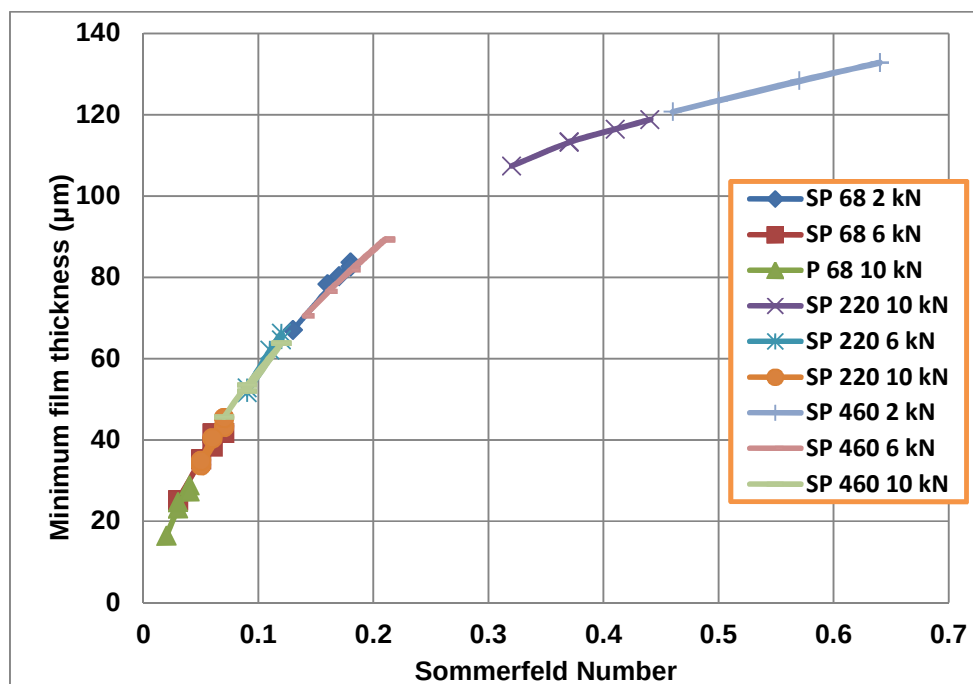


Figure 6-6 Sommerfeld Number vs Film Thickness at all viscosities

In figures 6-7 and 6-8 the minimum film thickness ($h_{\min}$) is plotted against the rotational speed (ω) base on Raimondi-Boyd charts. As the load is increased the journal displaces downward; consequently, the film thickness will be smaller due to the compression of the shaft; the limiting position is reached when $h_{\min}=0$ and $\epsilon=c$, that is, when the journal touches the bearing. The oil type does have a slight influence, making the film thickness indistinctly higher in the more compressed oil.

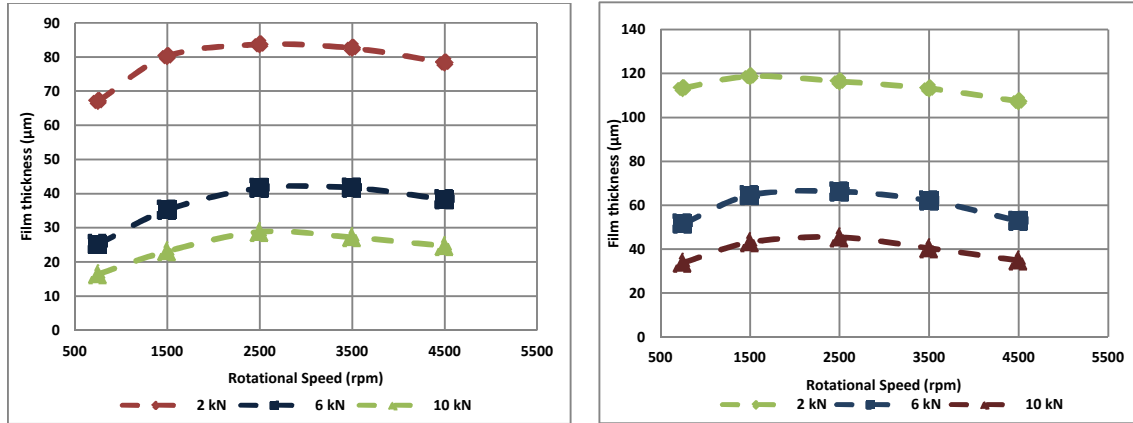


Figure 6-7 Film thickness vs rotational speed at SP68 & SP220 oil

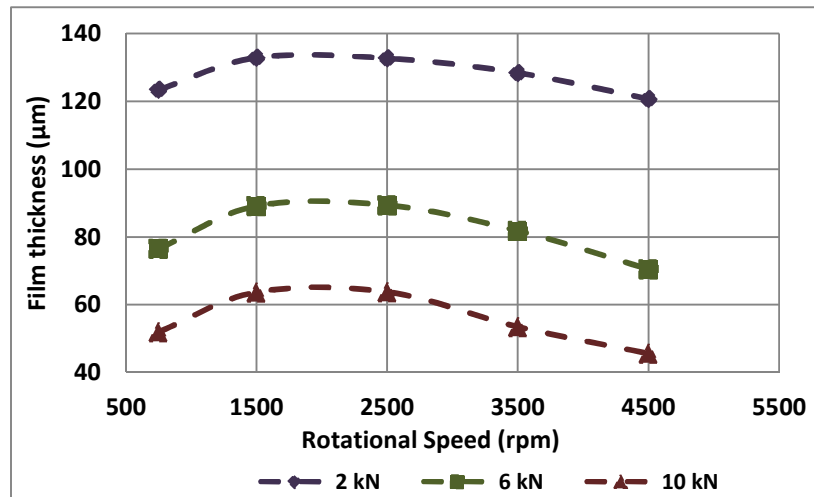


Figure 6-8 Film thickness vs rotational speed at SP460 oil

As can be seen in figures 6-7 and figure 6-8, the most critical values for film thickness (the minimum) are located in the edge values of speed, and the maximum value is happening at an intermediate site.

6.1.3 Shear Stresses Results

Equation 2-34 was used to analyse the behaviour of the shear stresses with the rotational speed at different viscosities, speeds and load conditions in figures 6-9 and figure 6-10.

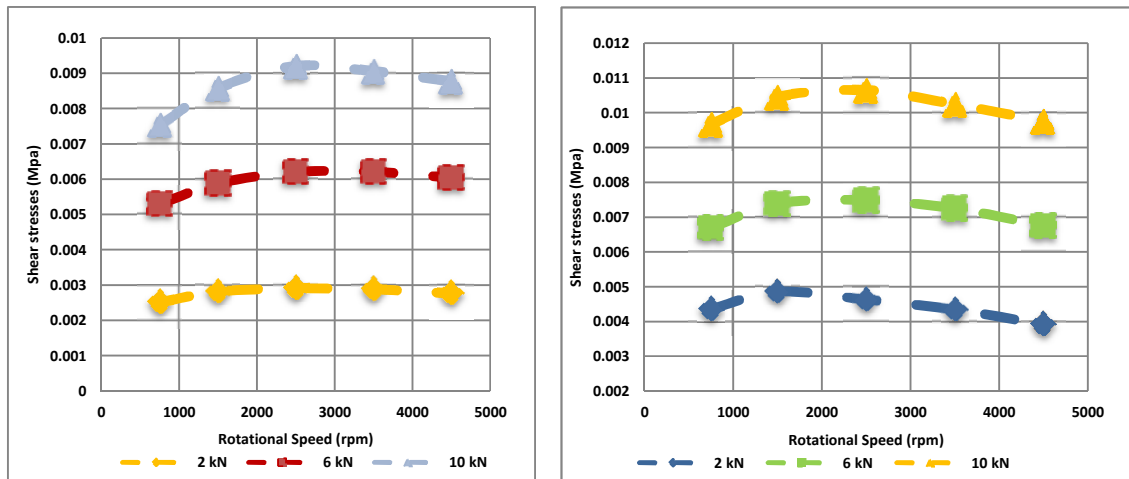


Figure 6-9 Shear stress vs rotational speed at SP68 & SP220 oil

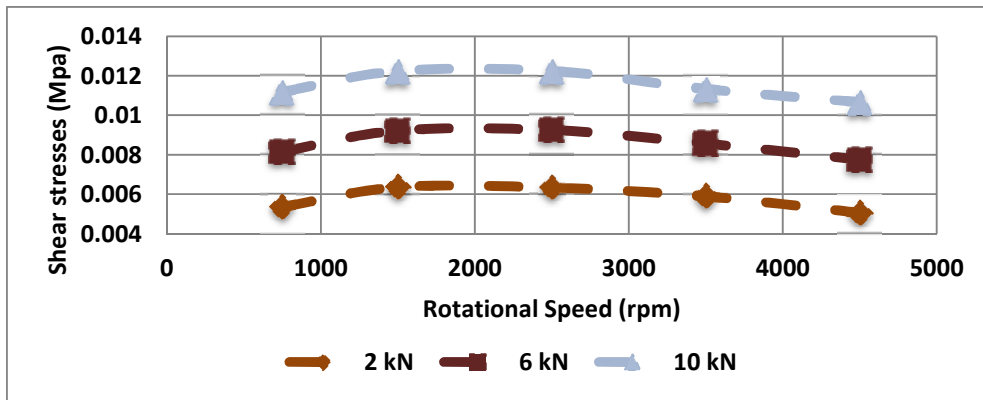


Figure 6-10 Shear stress vs rotational speed at SP460 oil

There are three variables which define the maximum value of the shear stresses; rotational speed (ω), viscosity (μ) and film thickness ($h_{\min}$):

- ✚ The shear stresses increase when the rotational speed increases, see equation 2-34.
- ✚ The shear stresses decrease as the oil viscosity decreases, the oil viscosity decreases as the temperature increases, and the temperature increases as the velocity increases.
- ✚ The maximum shear stress value is inversely proportional to the square of minimum film thickness ($\tau_{r\theta} \propto \frac{1}{h_{\min}^2}$)

So, there are three contradictory effects that lead to the value of the shear stresses. As can be seen in figure 6-9 to figure 6-10, there are three zones in the shear stresses curve:

- Zone 1 at low speeds: in this zone, the effect of the speed is low. The main effect that provokes shear stresses is high viscosity.
- Zone 2 at medium speeds: this is the most critical zone for shear stresses. The maximum values are found in this zone. The effect of both speed and viscosity is added to reach the largest values.
- Zone 3 at high speeds: in this zone, the maximum value for the shear stresses decreases. The main effect in this zone is the speed effect, while viscosity loses importance.

6.1.4 Acoustic Emission (AE) Results

The typical results of acoustic voltage RMS for the applied load condition of 10kN are plotted in figure 6-11 and table 6-1.

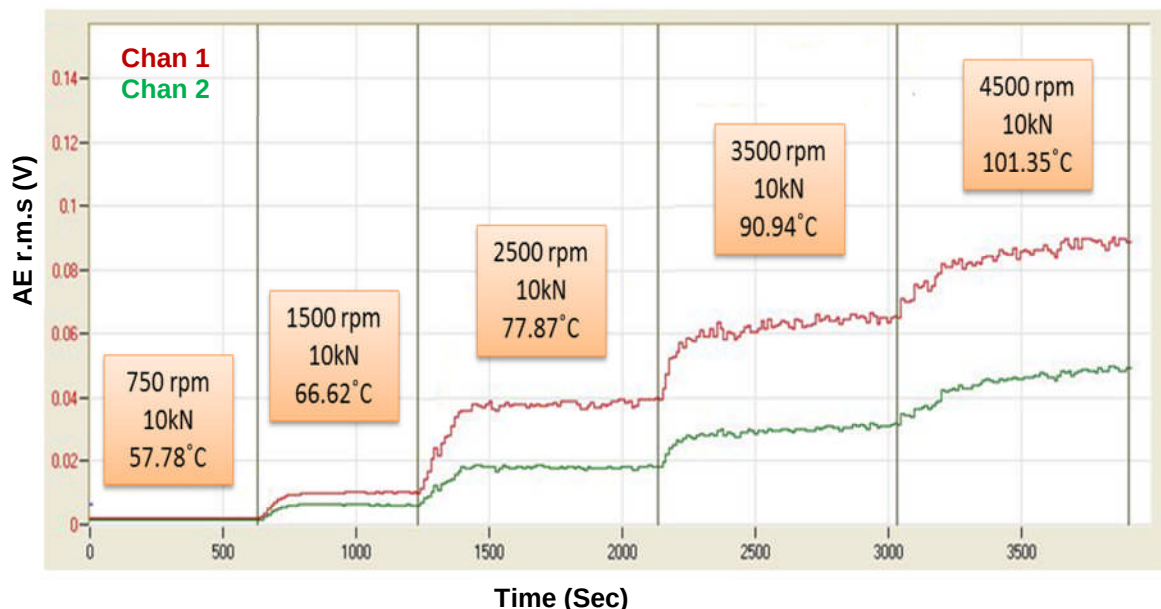


Figure 6-11 Typical test sequence for a load of 10kN

The two channels were connected to the following transducers for recording acoustic emission RMS from hydrodynamic bearing test rig:

- a) Channel 1 i bronze metal, hydrodynamic bearing at right axial end;

b) Channel 2 i bronze metal, hydrodynamic bearing at left axial end;

In the next step, the behaviour of the AE RMS with the rotational speed has been shown for the different viscosities, rotational speed and load conditions, see figure 6-12 and figure 6-13. AE will increase, as expected, as the rotational speed increases.

AE will decrease slightly as load increases; the key factor is the rigidity of the test rig. Increasing load will make the hydrodynamic bearing test rig more stable and the shaft will absorb the elastic acoustic waves more easily. The effect of both the load and the oil type is of little significance as can be seen in figure 6-12 and figure 6-13. To understand the influence of generating AE on load, three different oil types were employed and are shown here for viscosity variety.

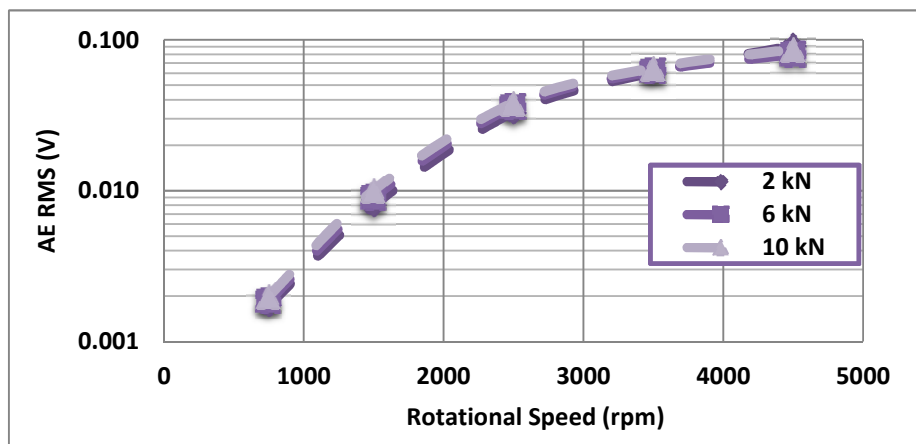


Figure 6-12 Acoustic emission vs rotational speed at SP460 oil

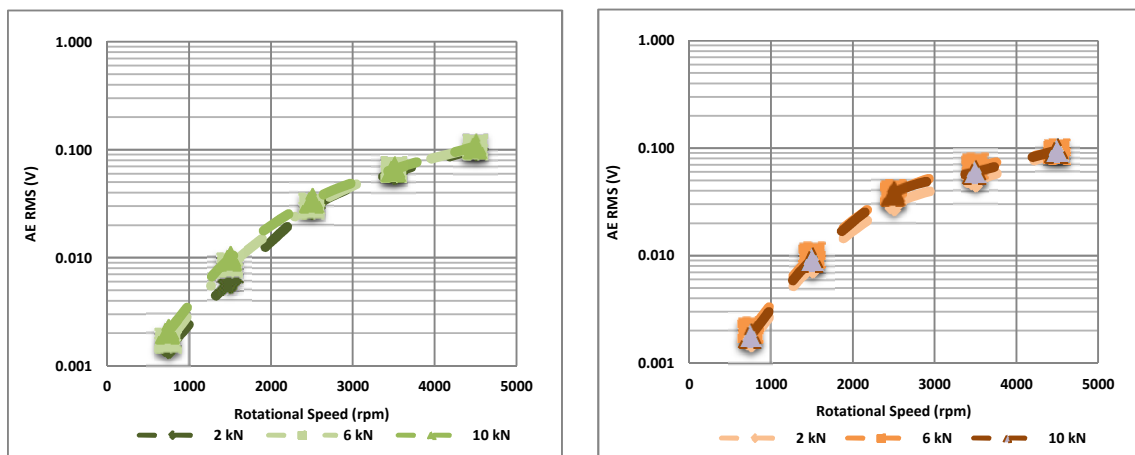


Figure 6-13 Acoustic emission vs rotational speed at SP68 & SP220 oil

6.1.5 Power Losses, AE and Shear Stresses in the Fluid

The performance of the AE RMS against power losses, film thickness and shear stresses in the fluid has been studied to verify the relationship with the operational condition at different oil viscosities in the following sections. Such a relationship will go some way in further developing the applicability of AET in monitoring rotating machinery.

6.1.5.1 Power Losses vs AE

The comparison results of AE RMS against power losses from experimental measurement show that there is a logarithmic relationship between power losses and AE RMS, see figure 6-14 and figure 6-15. Density and load are the key factors in the relationship between power losses, load and varying viscosities, see equation 2-2. An increment in the load does not have influence on the power losses yet the viscosity has a great influence on the power losses. The density of the oil has its effect; the higher the densities, the more power losses, see figure 6-14 and figure 6-15.

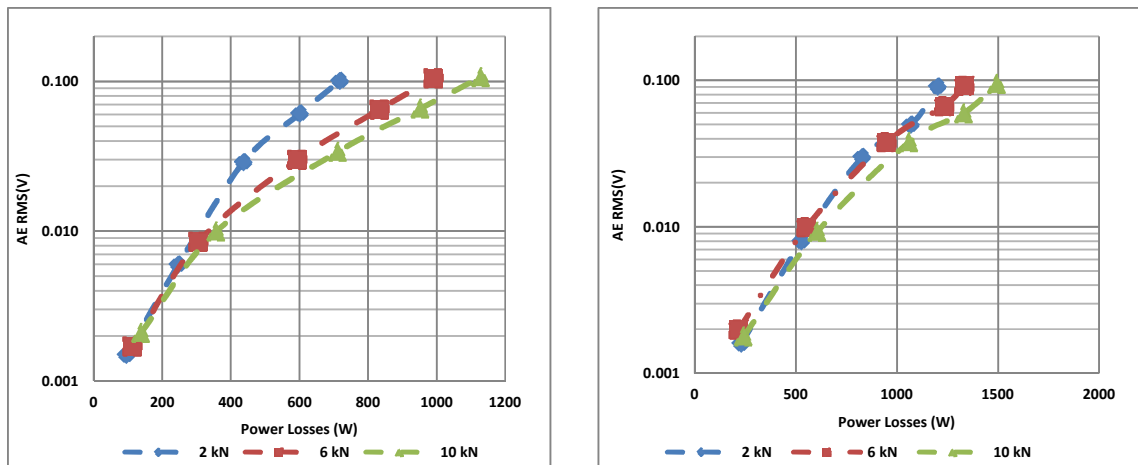


Figure 6-14 Power Losses vs Acoustic Emission at SP68 & SP220 oil

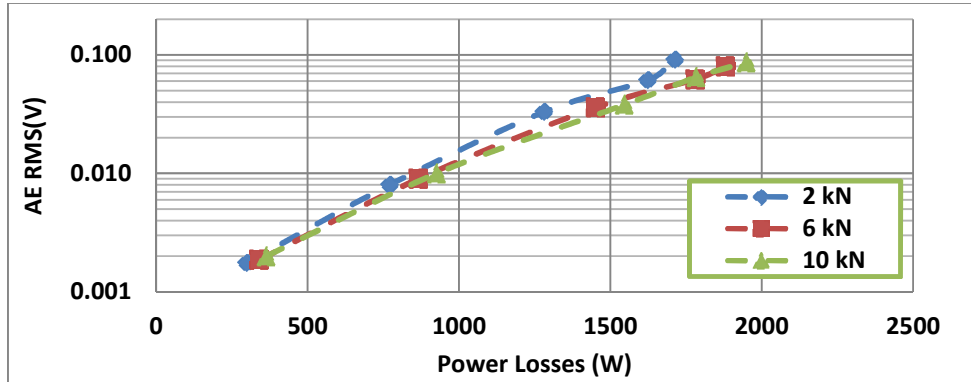


Figure 6-15 Power Losses vs Acoustic Emission at SP460 oil

Observations showed that AE levels increased with power losses, see figure 6-14 and figure 6-15. The averaged data are given in tables 6-2 to 6-4 from over 10,000 recorded files.

Oil Type	P (kN)	Speed (RPM)	Temperature(°C)	$h_{min}(\mu m)$	KW Btm(kW)	Total Power Loss(kW)	RMS (V)
SP68	2	750	42.64	67.05	0.0800	0.0963	0.002
	2	1500	54.21	80.24	0.1996	0.2474	0.006
	2	2500	66.83	83.64	0.3521	0.4374	0.029
	2	3500	77.21	82.6	0.4844	0.6015	0.061
	2	4500	87.17	78.32	0.5800	0.7182	0.101
Oil Type	P (kN)	Speed (RPM)	Temperature(°C)	$h_{min}(\mu m)$	KW Btm(kW)	Total Power Loss(kW)	RMS (V)
SP68	6	750	49.43	25.17	0.0993	0.1140	0.0017
	6	1500	56.7	35.24	0.2581	0.3059	0.009
	6	2500	65.15	41.58	0.4962	0.5942	0.03
	6	3500	74.76	41.7	0.6965	0.8343	0.065
	6	4500	85.25	38.33	0.8317	0.9911	0.105
Oil Type	P (kN)	Speed (RPM)	Temperature(°C)	$h_{min}(\mu m)$	KW Btm(kW)	Total Power Loss(kW)	RMS (V)
SP68	10	750	51.56	16.41	0.1245	0.1386	0.0021
	10	1500	57.95	23.12	0.3114	0.3582	0.01
	10	2500	64.35	28.79	0.6099	0.7130	0.034
	10	3500	76.18	27.23	0.8185	0.9531	0.066
	10	4500	87.25	24.65	0.9778	1.1301	0.107

Table 6-2 Minimum film thickness and power loss at different load condition and speed on bearing for SP 68

It is worth stating that, due to the minimal influence of load on film thickness under these test conditions, the power losses associated with varying loads over the same speed conditions were very similar, see table 6-2 to 6-4. The increase in AE with power loss was not surprising given that an increase in speed results in a greater power loss due to the increased friction in the shearing of the lubricant, see equation 2-2. Reduced minimum film thickness causes increased power losses and such energy losses are noted by an increase in AE RMS levels. Through three experimental tests programmes that were undertaken, a relationship between the power losses and AE RMS levels was determined as the experimental procedure described in the beginning of this chapter.

Oil Type	P (kN)	Speed (RPM)	Temperature(°C)	$h_{min}(\mu m)$	KW Btm(kW)	Total Power Loss(kW)	RMS (V)
SP220	2	750	45.42	113.36	0.1869	0.2303	0.002
	2	1500	58.53	118.74	0.4277	0.5281	0.008
	2	2500	72.6	116.43	0.6718	0.8288	0.03
	2	3500	83.04	113.18	0.8684	1.0697	0.05
	2	4500	92.96	107.37	0.9792	1.2028	0.091
Oil Type	P (kN)	Speed (RPM)	Temperature(°C)	$h_{min}(\mu m)$	KW Btm(kW)	Total Power Loss(kW)	RMS (V)
SP220	6	750	54.15	51.6	0.1822	0.2171	0.002
	6	1500	63.17	64.55	0.4601	0.5529	0.01
	6	2500	74.65	66.24	0.7894	0.9495	0.038
	6	3500	85.21	62.08	1.0286	1.2343	0.067
	6	4500	96.98	52.8	1.1182	1.3337	0.092
Oil Type	P (kN)	Speed (RPM)	Temperature(°C)	$h_{min}(\mu m)$	KW Btm(kW)	Total Power Loss(kW)	RMS (V)
SP220	10	750	55.53	33.76	0.2076	0.2426	0.0018
	10	1500	64.52	43.14	0.5129	0.6066	0.009
	10	2500	75.29	45.39	0.8956	1.0616	0.038
	10	3500	87.25	40.43	1.1293	1.3316	0.06
	10	4500	98.15	34.83	1.2781	1.4960	0.095

Table 6-3 Minimum film thickness and power loss at different load condition and speed on bearing for SP 220

AE RMS (volt), film thickness (mm) and power loss (KW) for varying rotational speeds are shown in these tables (6-2, 6-3 and 6-4); the values of AE levels presented in these tables were obtained by averaging the AE RMS over the entire specific speed conditions. From table 6-2 to 6-4 it can be noted in detail that an increase in rotational speed and temperature results in an increase in film thickness, AE levels, and power loss.

Oil Type	P (kN)	Speed (RPM)	Temperature(°C)	$h_{min}(\mu m)$	KW Btm(kW)	Total Power Loss(kW)	RMS (V)
SP 460	2	750	54.92	123.44	0.2421	0.2991	0.002
	2	1500	64.84	132.87	0.6114	0.7711	0.008
	2	2500	76.22	132.7	1.0152	1.2804	0.033
	2	3500	86	128.38	1.2825	1.6224	0.061
	2	4500	96.29	120.7	1.3509	1.7125	0.091
Oil Type	P (kN)	Speed (RPM)	Temperature(°C)	$h_{min}(\mu m)$	KW Btm(kW)	Total Power Loss(kW)	RMS (V)
SP 460	6	750	56.2	76.51	0.2814	0.3403	0.0019
	6	1500	65.44	89.12	0.6952	0.8666	0.009
	6	2500	76.6	89.4	1.1642	1.4515	0.036
	6	3500	87.7	81.82	1.4344	1.7801	0.062
	6	4500	98.77	70.49	1.5269	1.8809	0.08
Oil Type	P (kN)	Speed (RPM)	Temperature(°C)	$h_{min}(\mu m)$	KW Btm(kW)	Total Power Loss(kW)	RMS (V)
SP 460	10	750	57.78	51.96	0.3057	0.3644	0.002
	10	1500	66.62	63.71	0.7558	0.9264	0.01
	10	2500	77.87	63.8	1.2616	1.5464	0.038
	10	3500	90.94	53.5	1.4685	1.7838	0.065
	10	4500	101.35	45.6	1.6191	1.9496	0.087

Table 6-4 Minimum film thickness and power loss at different load condition and speed on bearing for SP 460

The relationship between the power losses and AE RMS levels was determined for varying speed and constant load conditions. The equation 6-1, holds, based on the assumption that the load has minimal influence on the film thickness, which is the case in this investigation; see section 6.1.6 for further investigation on modelling AE RMS with bearing parameters.

$$AE_{r.m.s} = 0.01e^{2.20P_{Btm}} \quad (6-1)$$

where P_{Btm} = power loss (kW) and AE = AE r.m.s (Volt)

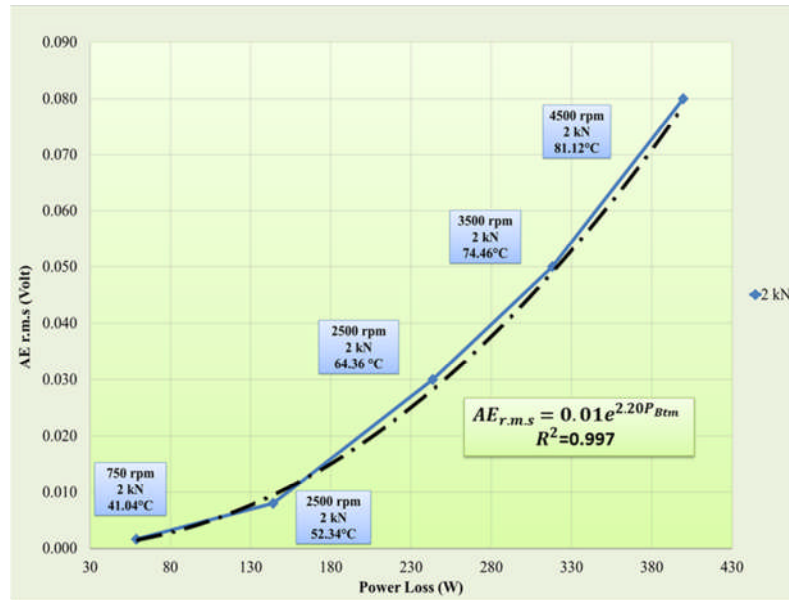


Figure 6-16 AE r.m.s and total power loss for varying speed 2kN

The results of experimental measurement show that, AE levels increased with power losses, see figure 6-16, and averaged data are given in tables 6-2 to 6-4. The comparison plot of AE RMS over power loss under varying rotational speed conditions are presented in figure 6-17 for varying load conditions for viscosity SP 220.

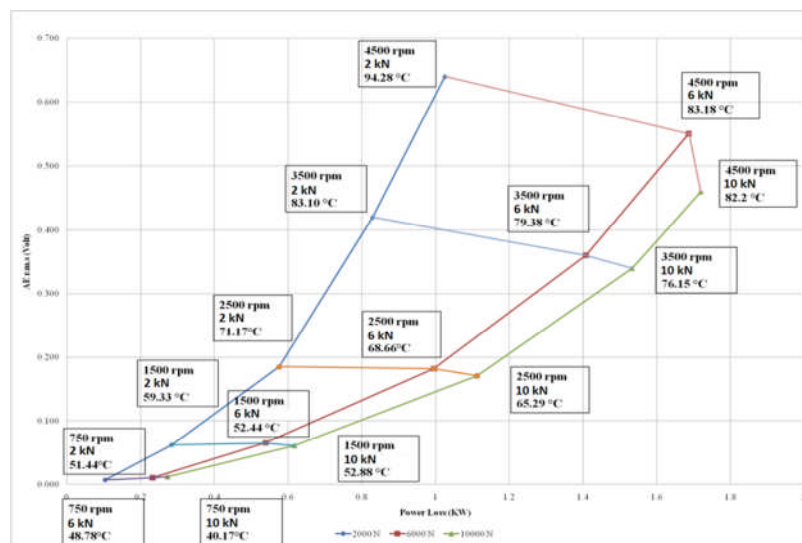


Figure 6-17 Typical averaged AE r.m.s and total power loss for varying speed and constant load conditions

Figure 6-18 shows that AE levels increased with increasing power losses for all three different dynamic viscosities. However, at any given speed and dynamic viscosity, the variation in load caused an increase in AE levels although the levels were relatively small in relation to the increase in AE levels with increasing speed.

Power losses are the friction losses due to metal-to-metal contact and there is a direct relationship with acoustic emission. Increasing rotational speed increases the probability of contact, at the same time, AE and power losses also increase. Power losses are dependent on the speed and lubricant density, based on Raimondi and Boyd [136], see equation 2-2. A denser lubricant will produce greater resistance to shaft rotation, and the shaft would then require extra power to maintain its speed of rotation, as a result, power losses will be higher in denser lubricants, see figure 6-18.

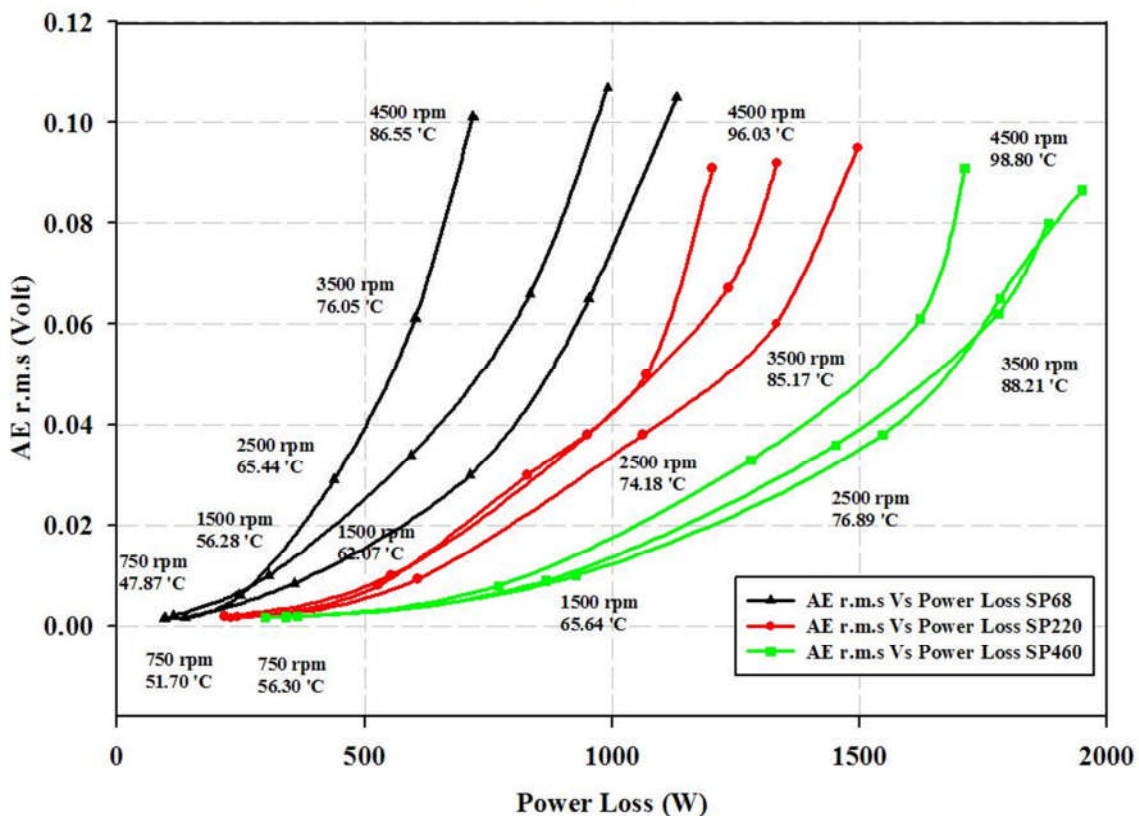


Figure 6-18 AE r.m.s and power loss for varying speed and load conditions (2kN, 6kN, 10kN) at varying dynamic viscosities

6.1.5.2 Power Losses vs Shear Stress

The graphs from the results (figure 6-19 and figure 6-20) show that there is a relationship between power losses and shear stress.

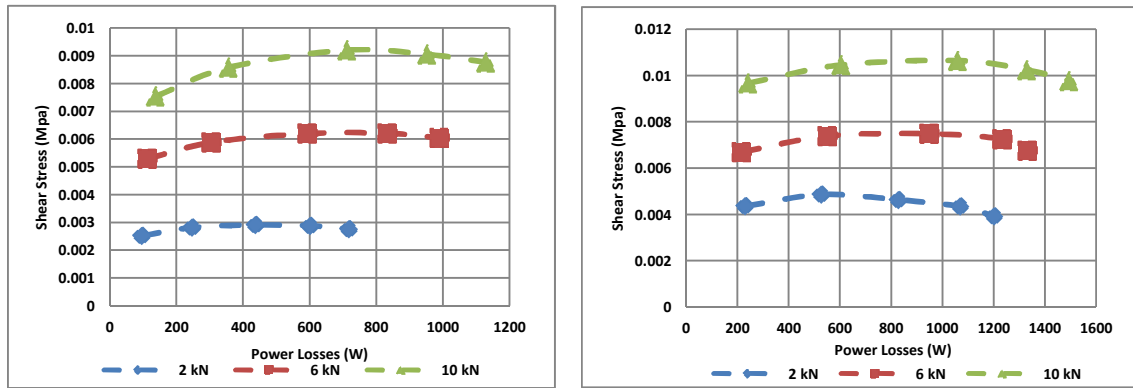


Figure 6-19 Power Losses vs Shear Stress at SP68 & SP220 oil

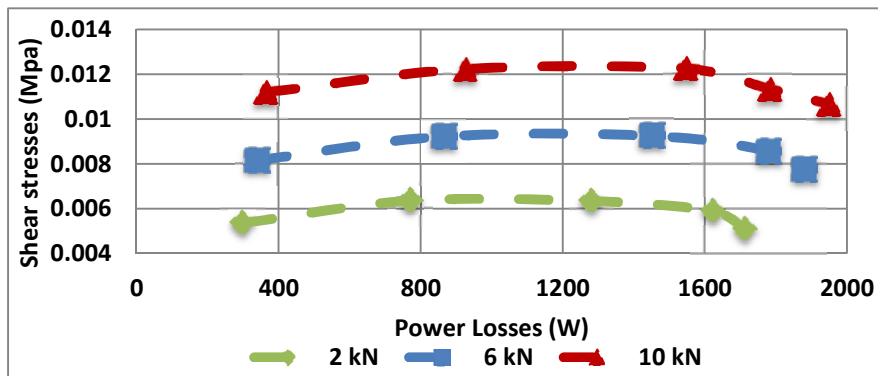


Figure 6-20 Power losses vs Shear Stress at SP460 oil

The shear stresses in the fluid are calculate following the equation:

$$\tau = \mu \frac{du}{dx} \quad (6-2)$$

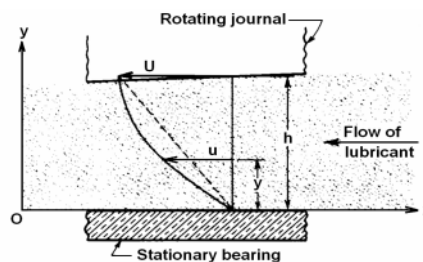


Figure 6-21 Velocity distribution in the oil film [159]

By assuming the oil is an incompressible fluid, and taking a ∂u as it is shown in figure 6-21, the forces acting on it are illustrated as figure 6-22. As shown, the forces acting on a fluid element of height dy , width dx , velocity u , and top to bottom velocity gradient ∂u is considered through Reynolds's Equation, see figure 6-22.

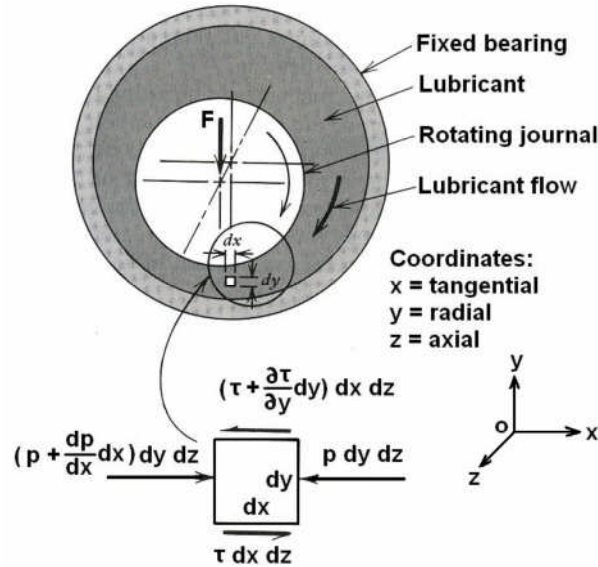


Figure 6-22 Pressure and viscous forces acting on an element of lubricant
[159]

For the equilibrium of forces in the x direction acting on the fluid element acting on the fluid element shown in figure 6-22. It must be noted that Only X components are shown.

$$\sum F_Y = 0$$

$$\begin{aligned} p dx &= \left(p + \frac{\partial p}{\partial x} dx \right) dy + \rho g dx dy \rightarrow 0 \\ &= \frac{\partial p}{\partial x} dx dy + \rho g dx dy \end{aligned} \quad (6-3)$$

$$\frac{\partial p}{\partial x} = \rho g \rightarrow \int p = \rho g * y + f(x) + K_1$$

$$\sum F_X = 0$$

$$p dy + \left(\tau + \frac{\partial \tau}{\partial y} dy \right) dx - \tau dx - \left(p + \frac{\partial p}{\partial x} dx \right) dy = m * a = \quad (6-4)$$

$$\rho dV * \frac{du}{dt}$$

$$\frac{\partial \tau}{\partial y} dxdy - \frac{\partial p}{\partial x} dxdy = \rho dxdy * \frac{du}{dt}$$

$$\frac{\partial \tau}{\partial y} - \frac{\partial p}{\partial x} = \frac{du}{dt} \quad (6-5)$$

But since this is a case of permanent flow $\frac{du}{dt} = 0$ hence, the equation 6-5 is reduced to:

$$\frac{\partial \tau}{\partial y} - \frac{\partial p}{\partial x} = 0 \quad \rightarrow \quad \frac{\partial \tau}{\partial y} = \frac{\partial p}{\partial x} \quad (6-6)$$

Since the shear stresses have the following equation 6-7; where the partial derivatives is used since the velocity u depends upon both x and y . Substituting

$$\tau = \mu \frac{du}{dy} \quad \rightarrow \quad \frac{\partial \tau}{\partial y} = \mu \frac{d^2 u}{dy^2} \quad (6-7)$$

$$\rightarrow \quad \frac{\partial p}{\partial x} = \mu \frac{d^2 u}{dy^2} \quad (6-8)$$

Rearranging the terms in equation 6-8, the result would be:

$$\frac{d^2 u}{dy^2} = \frac{1}{\mu} \frac{\partial p}{\partial x} \quad (6-9)$$

Since $\frac{\partial p}{\partial x} = K$ is constant because the gradient of flow is not considered in Couette's Flow Equation (linear relationship of pressure) as it is shown in the equation 6-5 so by holding x constant and integrating twice with respect to y the results would be:

$$u(y) = \frac{K y^2}{2\mu} + C_1 y + C_2 \quad (6-10)$$

The assumption of no slip between the lubricants and the boundary surfaces gives boundary conditions enabling C_1 and C_2 to be evaluated:

$$u=0 \text{ at } y=0 \text{ and } u=U \text{ at } y=h$$

Hence,

$$C_1 = \frac{U}{h} - \frac{K h}{\mu 2} \quad \text{and} \quad C_2 = 0$$

Velocity distribution of the oil film demonstrates of two terms linear and parabolic, see equation 6-11 and figure 6-23.

$$u = \frac{1}{2\mu} K(y^2 - yh) + \frac{U}{h} y \quad (6-11)$$

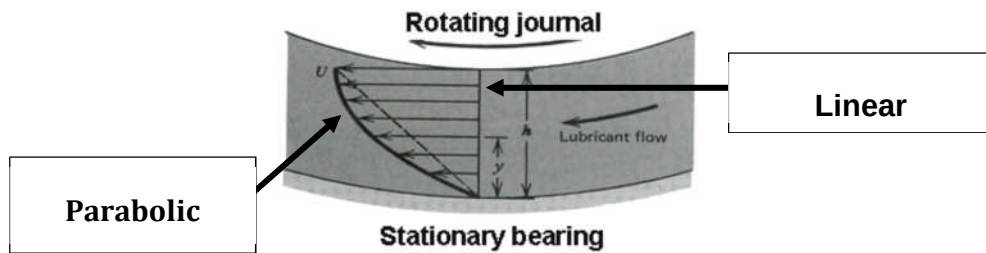


Figure 6-23 Velocity gradient in the oil film [159]

With reference to figure 6-24, an expression for viscous friction drag torque is derived by considering the entire cylindrical oil film as the fluid block acted upon by force F in cartesian coordinate. From Newton's law of Viscosity:

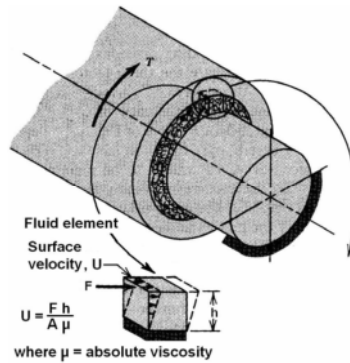


Figure 6-24 Laminar flow of fluid in clearance space [159]

$$F = \text{friction torque/shaft radius} = 2T_f/d$$

$$A = \pi dl, \quad h = c \text{ (where } c = \text{radial clearance} = 0.5(D-d)) \text{ , } r = d/2$$

$$H = T_f \omega \quad \text{so} \quad (6-12)$$

$$T_f = H/\omega$$

Therefore:

$$F = \frac{2H}{\omega d} \quad \text{and} \quad F = \mu \frac{Au}{h} \quad (6-13)$$

$$\frac{2H}{\omega d} = \mu \frac{Au}{h}$$

Therefore from equation 6-13:

$$\frac{2H}{\omega d} = \mu \frac{A}{h} u = \frac{A}{2h} K(y^2 - yh) + \mu \frac{AU}{h^2} y \quad (6-14)$$

$$H = \frac{\omega^2 d}{2} \left(\frac{A}{2\omega h} K(y^2 - yh) + \mu \frac{AU\omega}{h^2} y \right) \quad (6-15)$$

By rearranging the equation 6-15 base on rotational speed (ω) and substituting into shear stress equation 2-34; the result would be:

$$\tau^2 = \left(\mu \frac{4R_1^2}{1 - \left(\frac{R_1}{R_2} \right)^2} \right)^2 * \frac{1}{h_0^2} \frac{H}{\left(\frac{A}{2h} K(y^2 - yh) + \mu \frac{AU}{h^2} y \right)} \quad (6-16)$$

The final equation is showing, the relation between shear stress τ and power loss following parabolic equations.

The behavior of the shear stresses is defined by the effect of the viscosity at high rotational speeds (and low power losses) and the effect of the rotational speed (and high-power losses) being the maximum value for intermediate values (see equation 6-18), which is illustrated in figure 6-19 and figure 6-20. Load has an important influence in shear stresses. Increasing load leads to a higher number of stresses in the system.

Density and load are the key factors in the relationship between shear stresses and power losses for varying viscosities, see figures 6-19 and 6-20. For the same given conditions of speed, a denser environment makes both the maximum shear stress and the load increase. An increment in the load does not

have influence on the power losses, but the load has a great influence on the shear stresses. Furthermore, the density of the oil has its effect; the higher the density, the more shear stresses and power losses, see equation 6-18.

6.1.5.3 Power Losses Vs Min Film Thicknesses

The relationship between power losses and minimum film thickness through the equation 6-17, present the power loss increase when the rotational speed increases but, when the oil viscosity decreases the minimum film thickness decreases as well. The oil viscosity decreases as the temperature increases. Finally, the temperature increases as the velocity increases, see figure 6-25 and figure 6-26.

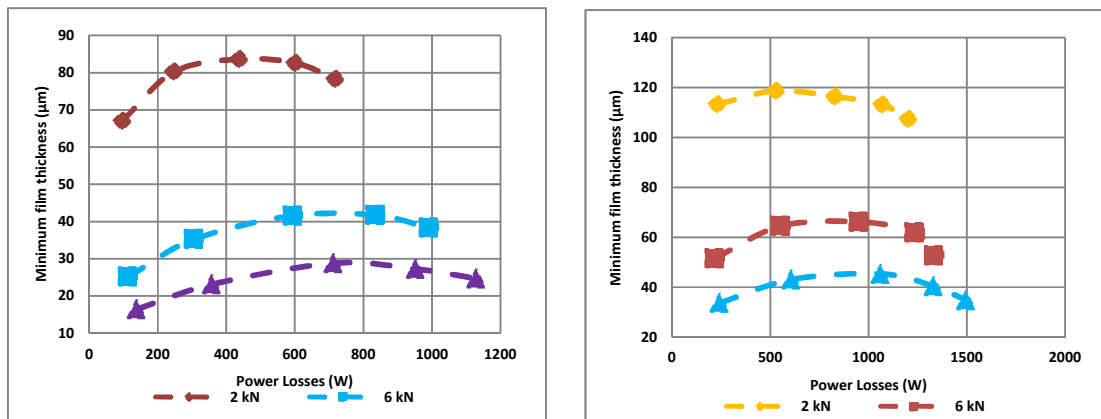


Figure 6-25 Power losses vs film thicknesses at SP68 & SP220 oil

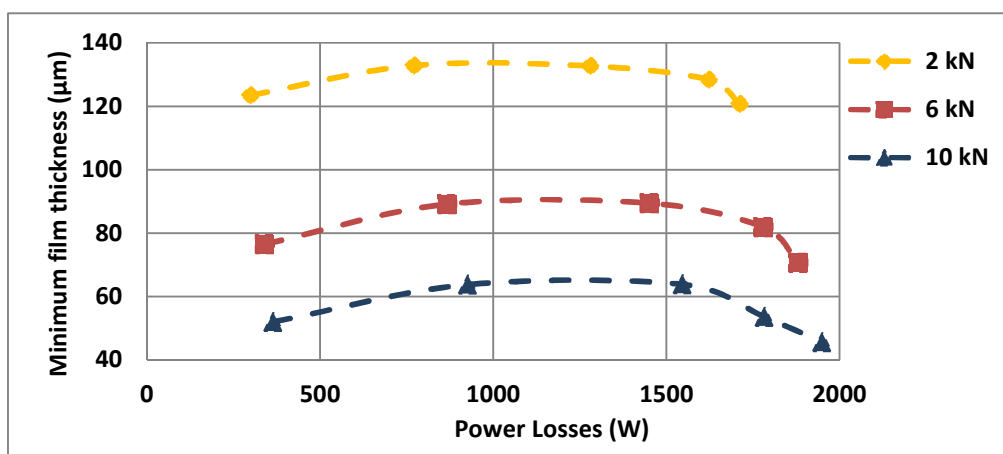


Figure 6-26 Power losses vs film thicknesses at SP460 oil

As shown in figure 6-6, the relationship between the Sommerfeld number and the minimum lubricant film thickness is practically exponential, where oil type, load or velocity was not affected. Consequently, as there is a direct relationship between S and $h_{\min}$, the film thickness behaves like the model in figure 6-25 and figure 6-26. Noticeably, the film thickness will decrease against power losses, when the load is increased, due to compression of the shaft. The oil viscosities do have a slight influence, making the film thickness to be marginally higher in denser oil see figure 6-27.

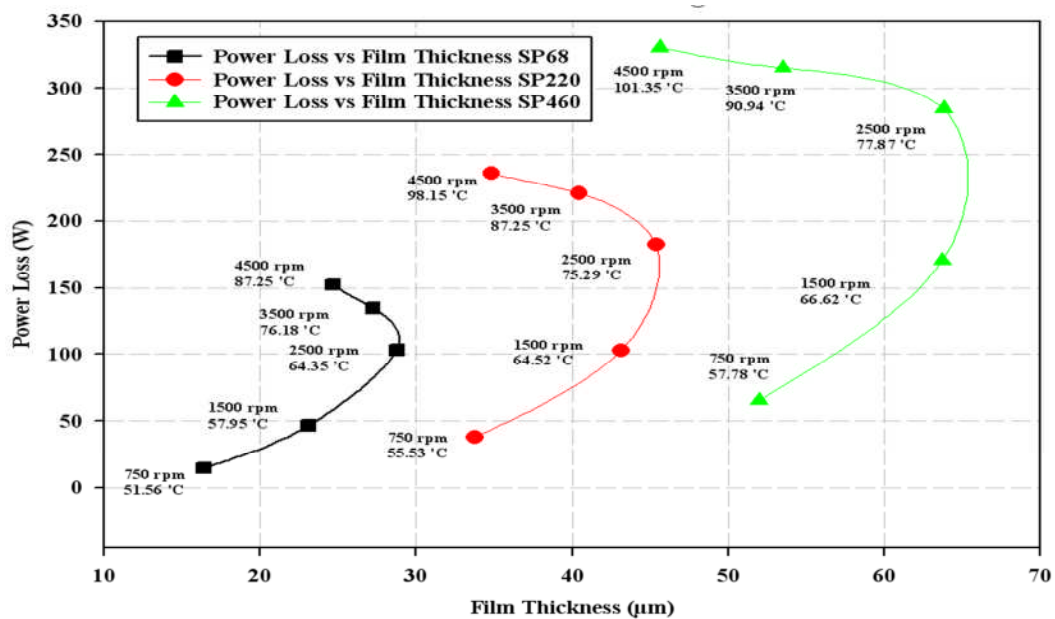


Figure 6-27 Influence of Minimum film thicknesses on Power Loss for 10kN load condition

The calculation of the shear stresses in the fluid, for straight and parallel flow is formulated in section 2.14 and equation 6-18. The film thickness behaviour is similar to the behaviour of the shear stresses; therefore, the relationship between film thickness and the power losses will be following the same relationship between power losses, and shear stresses.

6.1.5.4 Acoustic Emission vs Minimum Film Thicknesses

Observations of a theoretical minimum film thickness ($h_{\min}$) against AE RMS measured on the bearing for a fixed radial load conditions of 2kN, 6kN and 10kN for varying viscosities (SP68, SP220 and SP460) at five rotational speeds

(750 rpm, 1500 rpm, 2500 rpm, 3500 rpm and 4500 rpm) have been highlighted in figure 6-28 and figure 6-29. It is evident that an increase in speed resulted in an increase in minimum film thickness and a corresponding increase in measured AE RMS values for varying viscosities. This highlighted the sensitivity of the AE technology in discriminating film thickness. Furthermore, it must be noted that during these test conditions, the temperature varied by a maximum of approximately 3°C.

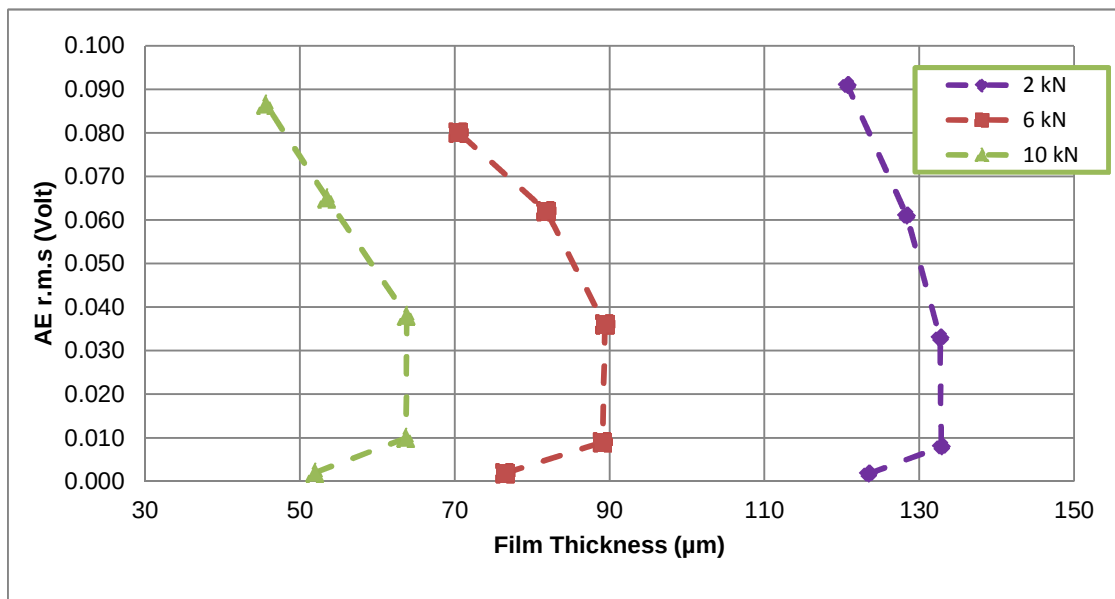


Figure 6-28 Acoustic emission vs film thicknesses at SP460 oil by increasing the speed [750rpm to 4500rpm]

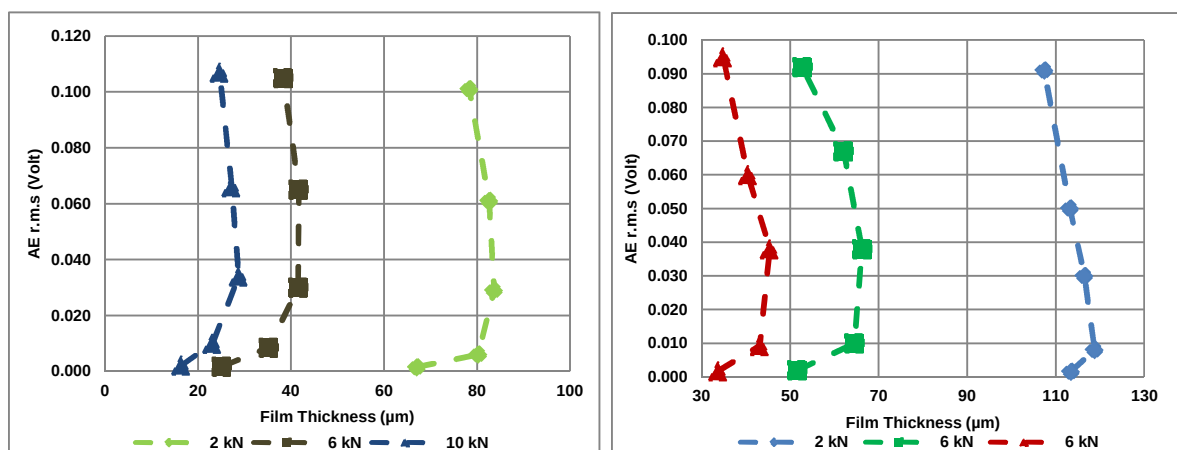


Figure 6-29 Acoustic emission vs film thicknesses at SP68 and SP220 oil by increasing the speed [750rpm to 4500rpm]

All calculations of minimum film thickness and power losses were based on well established procedures (Raimondi-Boyd [136]) with the main influencing factors including oil kinematic viscosity, operating temperature, bearing geometry, Sommerfeld number and shaft eccentricity, see equation 2-4.

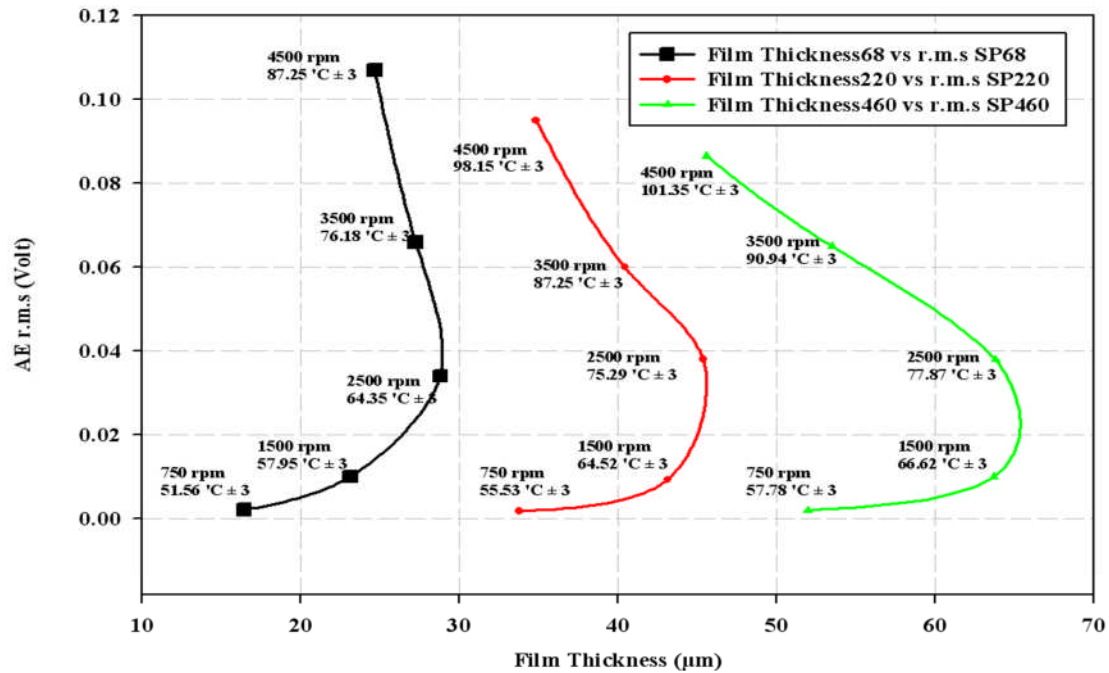


Figure 6-30 AE r.m.s levels at an applied load 10kN for varying viscosities

It was evident that an increase in speed resulted in an increase in measured AE even though, for the same test conditions, there was a decrease in theoretical film thickness, see figure 6-30. For instance, at a speed of 3500 rpm and a corresponding bearing temperature of 87°C, the oil film thickness was less than at the lower speed of 2500 rpm (75°C) due to the reduction in viscosity of the oil. Irrespective of the change in theoretical film thickness, the levels of AE increased with increasing rotational speed. Changes in rotational speed resulted in increased AE levels of the order of 200% on average, although the actual predicted change in minimum film thickness with increased speed was of the order of approximately 10% on average for all tests. Standard deviations (δ) of calculated minimum film thickness values and measured temperature values are detailed in appendix D, and the deviations were deemed low and acceptable. As with hydrodynamic bearings, there is no asperity contact and, in the author's view, the corresponding increase in AE levels is attributable to the

friction associated with the shearing effect of the lubricant and associated frictional/power losses. The author considered it prudent to establish/confirm such a phenomenon. It should also be stated that the minimum film thickness of all test conditions was an order of magnitude higher than the bearing surface.

6.1.5.5 Acoustic Emission vs Shear Stress

The result of experimental measurement for studying the behaviour and relationship between shear stress and AE RMS has been presented in figure 6-31 and figure 6-32. The results shown, the shear stress, see equation 2-34, increases in levels between 750 and 1500 rpm after which levels remained relatively steady (1500 to 2500 rpm). While, by increasing rotational speed from 2500 to 4500 rpm, the levels of shear stress steadily decreased with increased speed. At the same time, the level of AE RMS has been increasing as rotational speed increased during the measurement. The observations presented, have confirmed that in a properly maintained hydrodynamic lubrication regime a principal source of AE is the friction in the shearing of the lubricant. It has also been shown, see figure 6-31 and figure 6-32 that an increase in running speed generates higher AE activity in comparison to an increase in bearing load, see paper *OBSERVATIONS OF ACOUSTIC EMISSION IN A HYDRODYNAMIC BEARING* by Arthur.

Shear stresses have a direct relationship with rotational speed, density, viscosity, load, and an inverse relationship with film thickness, see equation 2-34. Consequently, the only link between the AE and shear stresses is the rotational speed. The rotational speed would affect the maximum value of shear stresses and will appear as source of acoustic emission signals, but will not correspond to the maximum of AE RMS. The most critical point, where the shear stresses are the maximum, occurs at medium velocities. At low rotational speeds shear stresses are lower because the only impediment for a soft motion is a high viscosity.

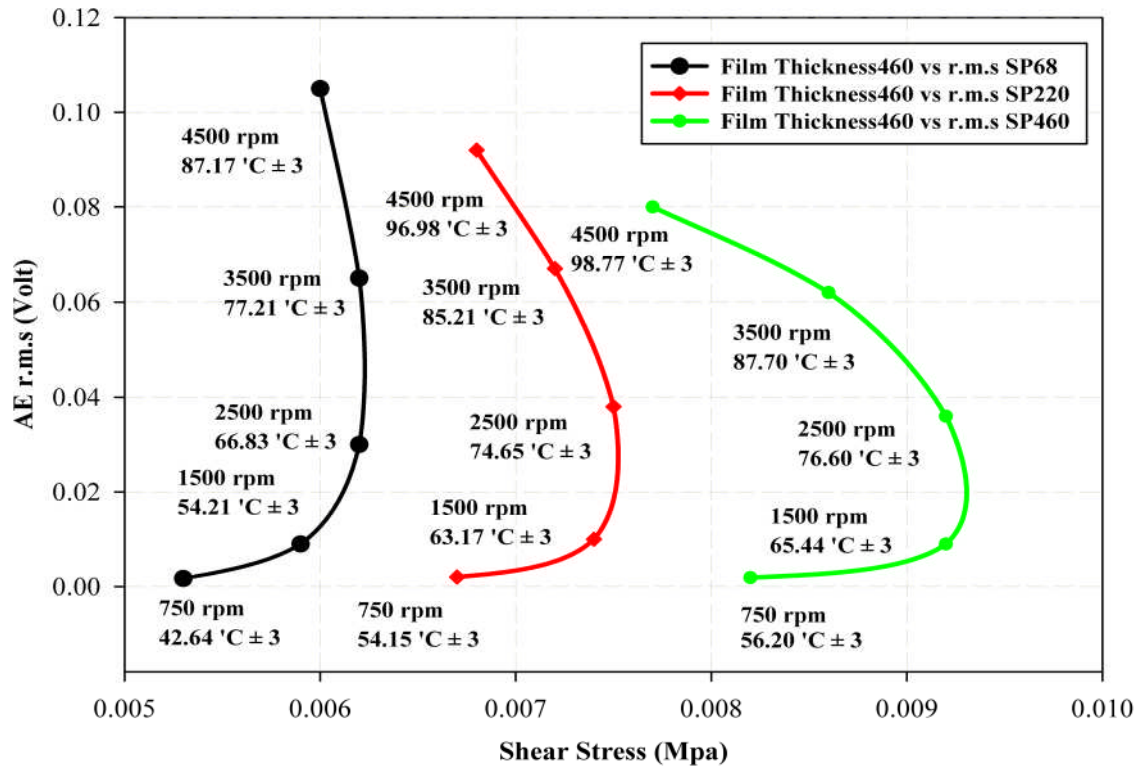


Figure 6-31 AE r.m.s and shear stress for varying speed and constant load conditions of 6kN

On the other hand, at high speeds, the viscosity is not an obstacle anymore because the effect has reduced considerably, due to the increment of the temperature; hence, the only source of shear stresses is the high rotational speed. At medium speeds these two factors are added, causing the shear stresses to maximums at a certain point. Density and load are constant during the rotational speed changes; they only affect the magnitude of the peak point of the shears stresses.

Figure 6-31 and figure 6-32 confirmed that, the load has a direct effect on the shear stresses but does not affect the value of RMS. The density of the lubricant has an effect on the shear stresses as well.

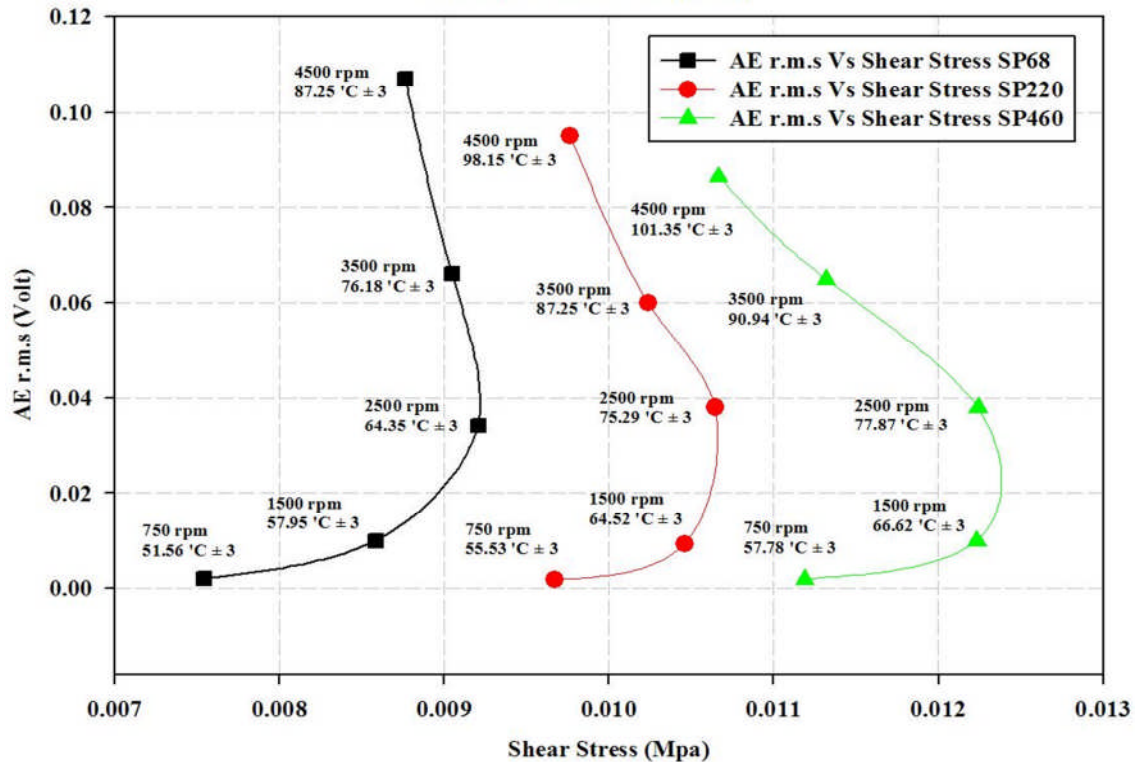


Figure 6-32 AE r.m.s and shear stresses for varying speed and constant load conditions of 10kN and different dynamic viscosities

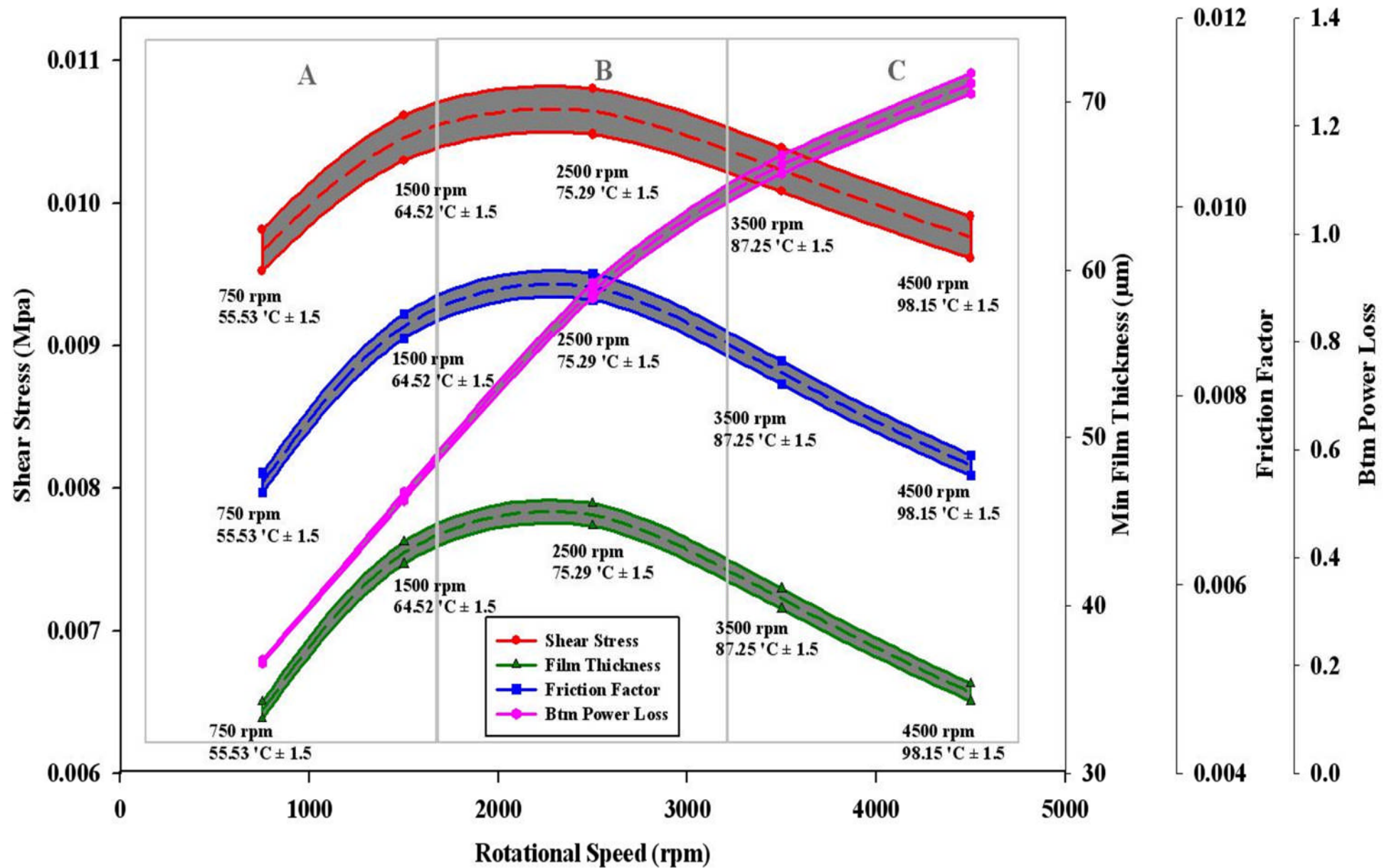
In an attempt to understand more about the relationship between AE and shear stress the correlation between shear stress, film thickness and rotational speed is presented in figure 6-33 in which three regions are detailed: *A*, *B* and *C*. It is evident that in region *A* an increase in speed resulted in an increase in the theoretical minimum film thicknesses and a corresponding increase in estimated shear stress values. In region *B* both parameters of dynamic viscosity and rotational speed had a direct influence on reaching shear stress levels and their calculations with regard to shear stress levels are assumed to be equal. Finally, in region *C* where increasing the rotational speed caused an increase in the oil temperatures and a reduction in the dynamic viscosity, the theoretical minimum film thicknesses decreased and a corresponding decrease in the estimated shear stress levels was noted. In this region, the most significant parameters are low viscosity and high rotational speed, see figure 6-33.

These relationships are attributed to the fact that the shear stresses increase when the rotational speed increases but the increase in speed causes an

increase in oil temperature, thereby decreasing the shear stresses as the oil viscosity decreases. The shaded area in figure 6-33 shows the range of theoretically determined shear stress and film thickness values based on experimental measurements and sensor accuracies. In addition, the associated friction factor values are presented in figure 6-33, the values of which mirror trends in shear stress and film thickness. The friction factor is a dimensionless parameter which is determined from the eccentricity ratio and Sommerfeld number of the bearing, and is employed to calculate the power losses of the bearing.

The results presented in figure 6-33 are similar for all three different viscosities investigated. Of significant note in this figure is the relationship between rotational speed and theoretical power losses on the bottom (P_{Btm}) of the journal bearing (in the region of minimum film thickness) where it is evident that power losses do not mirror the changes in theoretical film thickness, shear stress and friction factor. Even though the friction factor reduces in value with increasing speed, the power losses increase with increasing speed. This is because the estimation of power loss has a direct relationship with speed and the levels of reduction in the friction factor are significantly outweighed by the increase in speed.

Figure 6-33 Shear stresses, rotational speed, dynamic viscosity (SP220), power loss (bottom of bearing) and minimum film thickness for load condition 10kN



This is noted in figure 6-34 where AE levels are seen to increase with total power losses and the losses attributed to the bottom of the bearing. This was not surprising, it was expected that AE would be generated from not only shear stress in the region of minimum film thickness but also from the splashing of the oil and its motion within the top of the bearing. Therefore, it is the total power loss and not the film thickness, shear stress or friction factor that is most directly correlated to the measured AE levels; a finding which hitherto has not been reported.

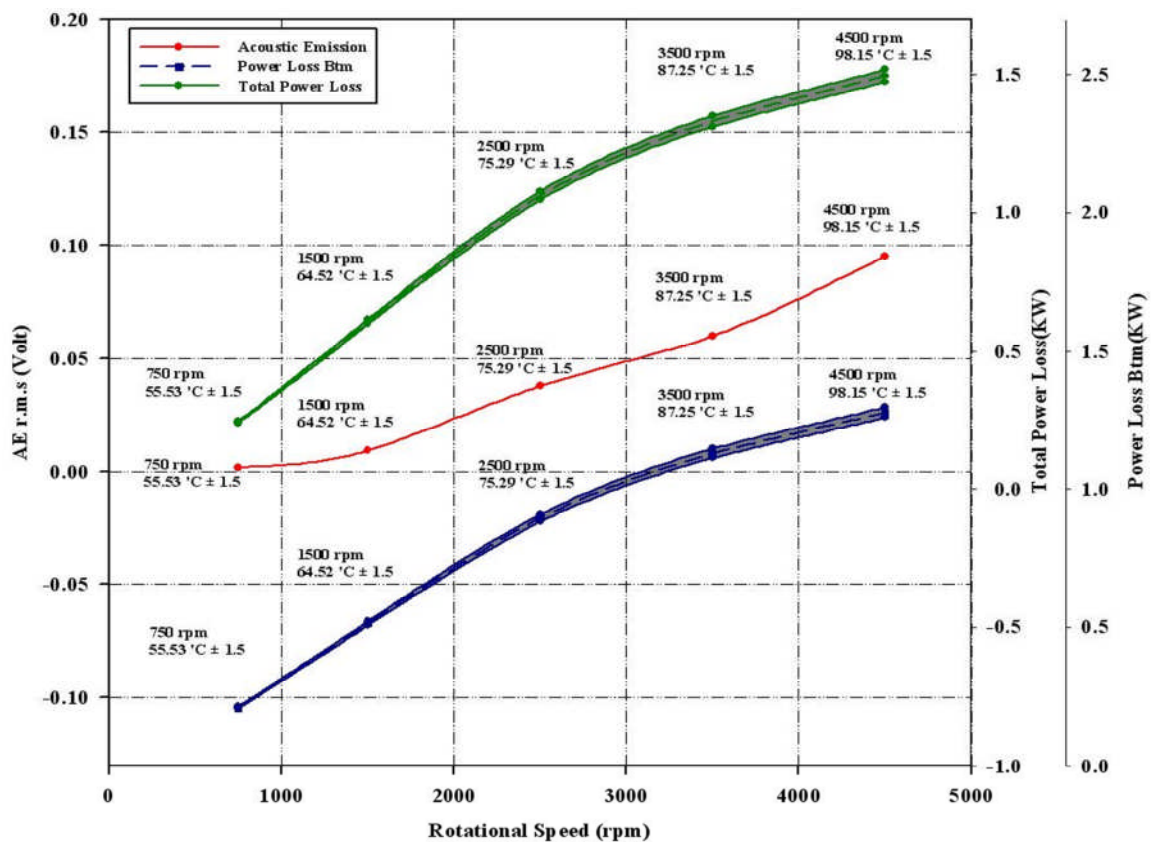


Figure 6-34 AE r.m.s, rotational speed, dynamic viscosity (SP220), bottom power losses and total loss for load condition 10kN

It is evident that the Acoustic Emission technology (AET) has a great working scope in its application to hydrodynamic bearings. The observations demonstrated that in a properly maintained hydrodynamic lubrication regime, the principal source of AE is the friction in the shearing of the lubricant in addition to the splashing and motion of the oil within the bearing. Such a

relationship will go some way to further develop the applicability of AET in monitoring rotating machinery.

6.1.6 Modelling AE with bearing parameters

The relationship between the friction and the AE RMS for different load and rotational speed conditions was measured for 2 minutes and plotted in figure 6-18 at varying viscosities. Power losses are the friction losses due to metal-to-metal contact; therefore, there is a direct link with acoustic emission. Increasing rotational speed causes, more probabilities of contact, at the same time AE and power losses increase. Power losses are dependent on the speed and lubricant density, based on Raimondi and Boyd [136], see equation 2-2. Higher viscosity will reduce the shaft rotation speed, and the shaft would then require an extra power to maintain its speed, as a result, power losses are higher in denser lubricants, see figure 6-18.

The results revealed a good correlation, see figure 6-34 and tables 6-2 to 6-4, between the power losses attributed to the bottom (P_{Btm}) of the bearing (which is proportional to the friction factor) and AE RMS for the hydrodynamic bearing test rig for all three different dynamic viscosities. Such an association may allow the prediction of a reasonable power loss (P_{Btm}) from an AE signal. It was also determined that the friction work correlated well with the corresponding integrated AE RMS over time. Proportionality between the theoretically determined oil minimum film thickness and the AE RMS was observed and discussed, in previous sections.

Observation from the measurement results presented, see figure 6-35 and tables 6-2 to 6-4, demonstrated that AE levels increased with increasing power losses for all three different dynamic viscosities. However, at any given rotational speed and dynamic viscosity, the variation in load caused an increase in AE levels although these levels were relatively small in relation to the increase in AE levels with increasing speed. All different rotational speeds, loads and viscosities were measured and averaged, following the same pattern, see figure 6-18 using an exponential law.

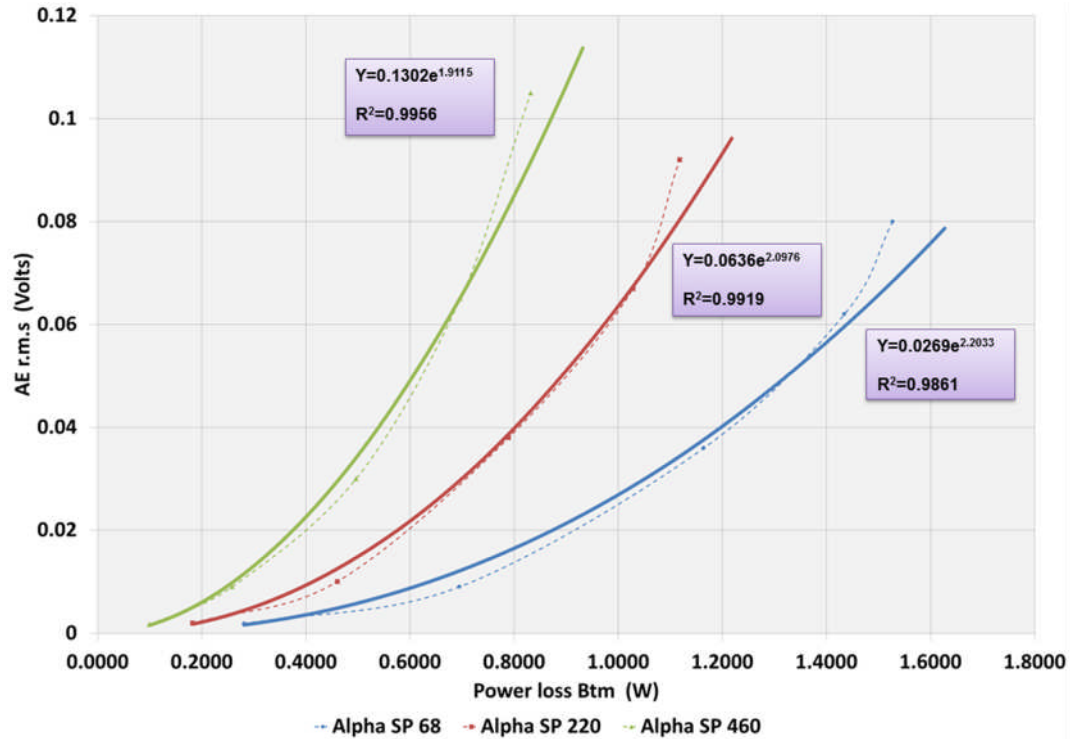


Figure 6-35 AE r.m.s and Power loss bottom (P_{Btm}) for varying dynamic viscosity

The power loss bottom (P_{Btm}) in a bearing as a result of bearing friction can be obtained using equation 6-17:

$$P_{Btm} = f \frac{R CPLNR}{C 79520} \quad (6-17)$$

where:

R (mm) Bearing Radius

P (MN/m²) Load

L (mm) Bearing length

N (R.P.S) Rotational speed

f Friction Factor

C (mm) Clearance

The correlation between power loss bottom (P_{Btm}) and the AE RMS, developed from data obtained at different sliding speeds, see figure 6-35, can be reasonably quantified by an exponential law as shown in this figure and expressed by equation 6-18.

$$AE_{r.m.s} = 0.01e^{2.20P_{Btm}} \quad (6-18)$$

Although a linear form can also approximate these results, the proposed exponential law more accurately expresses the trend. By substituting equation 6-17 into the equation 6-18 for power loss bottom (P_{Btm}), the relation can be written as:

$$AE_{r.m.s} = 0.01e^{2.20*f\frac{R}{C}*CPLNR} \quad (6-19)$$

Or
$$AE_{r.m.s} = 0.01e^{f\frac{R}{C*\frac{2.20}{79520}}*CPLNR} \quad (6-20)$$

After considerable basic algebra, the final relationship between the power losses attributed to the bottom (P_{Btm}) of the bearing (which is proportional to the friction factor) and AE RMS for the hydrodynamic bearing test rig for different dynamic viscosities can be written as:

$$AE_{r.m.s} = 0.01\Lambda e^{0.00003f\frac{R}{C}*CPLNR} \quad (6-21)$$

where;

P_{Btm} ; (kW) Power loss

N ; (R.P.S) Rotational speed

AE; (Volt) Acoustic emission r.m.s

f ; Friction Factor

R ; (mm) Bearing radius

C ; (mm) Clearance

P ; (MN/m^2) Load

Λ ; Viscosity factor

L ; (mm) Bearing length

Different viscosity factor (Λ) for varying grades (Castrol Alpha SP), can be summarized as, see table 6-6:

Table 6-5 The Castrol Alpha SP oil range of high quality lubricants at different grade (Λ) with viscosity factor

ISO	Alpha	Alpha	Alpha	Alpha	Alpha	Alpha	Alpha	Alpha	Alpha
Viscosity	SP32	SP68	SP100	SP150	SP220	SP320	SP460	SP680	SP1000
Grade	32	68	100	150	220	320	460	680	1000
Λ	1	2	3	4.5	6	8.5	13	19	28

The development of such an important relationship allows us, in the Sutton test rig, to conclude that an AE RMS signal can be used to predict, with reasonable accuracy (on average plus or minus 5% for different load conditions), the power loss bottom (P_{Btm}) for tribological systems without prior knowledge of the relative rotational speeds or load conditions. An exponential law is proposed to predict AE RMS from the measured power loss bottom (P_{Btm}) or vice versa. To date this is the first attempt at correlating AE with fluid friction and power losses in a hydrodynamic bearing. It is clear that more experiments are needed to be able to generalise this conclusion and a calibration process has to be established with the intention of adjusting equation 6-21 to different situations in the Sutton hydrodynamic bearing test rig.

7 Incipient Hydrodynamic Bearing Failure Detection

In this chapter the experimental measurement results identified the correlation between AE activity, displacement and vibration for the rubbing simulation test. It was considered prudent to assess the sensitivity of the AE measurements at the onset of contact between the shaft and the hydrodynamic bearing. The AE RMS, temperatures, vibration signals and displacement have been recorded, and the following analyses have been produced.

As there is no asperity contact between the rotating shaft and the hydrodynamic bearing in a properly maintained hydrodynamic lubrication regime, it was thought prudent to assess the sensitivity of the AE measurements to the onset of contact between the shaft and the journal bearing. In the rubbing tests, the shaft was forced to rub against the journal. This was achieved by increasing the load on the ball bearing at the drive end only, the position of channel 2, see figure 7-1; the loading of the bearing was changed incrementally by 1kN to a maximum load of 6kN.

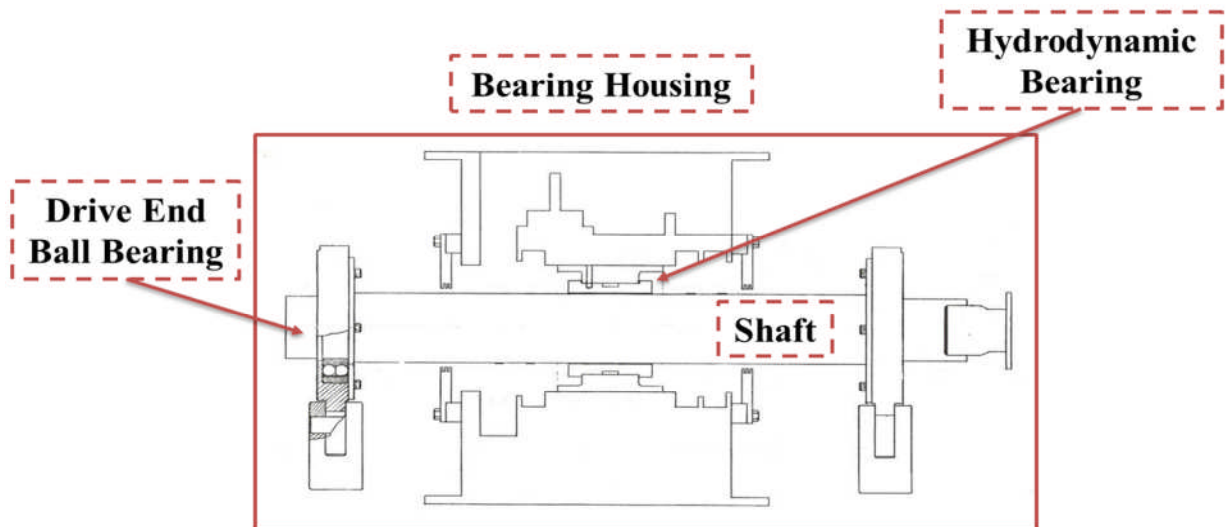


Figure 7-1 Load position on ball bearing at the drive end

Prior the rubbing simulation, at the synchronous load condition (equal load at both side of hydrodynamic bearing); AE RMS (volt), for 1500 rpm rotational speed at 3kN, 4kN and 6kN are presented in table 7-1; the values of AE levels presented in these tables were obtained by averaging the AE RMS over the

entire specific rotational speed and load conditions. From table 7-1 it can be noted in detail that an increase in load conditions, results in an increase in AE levels.

Table 7-1 AE activities at symmetric load condition for SP 460

Oil Type	P (kN)	Speed (RPM)	Temperature(°C)	AE RMS (V)
SP 460	3	1500	64.48	0.007
	4	1500	66.45	0.008
	6	1500	67.54	0.009

Initially, in the rubbing simulation, the shaft ran at 1500rpm (25 Hz) for two hours, after which 1kN increments were made within a two-minute time frame to the maximum load of 4kN and 6kN at the drive end only. The typical result of acoustic emission RMS for misalignment condition is plotted in figure 7-2. It is quite clear from figure 7-2 and table 7-1, that the average level of AE activities in rubbing simulation are significantly higher than in the synchronous load condition.

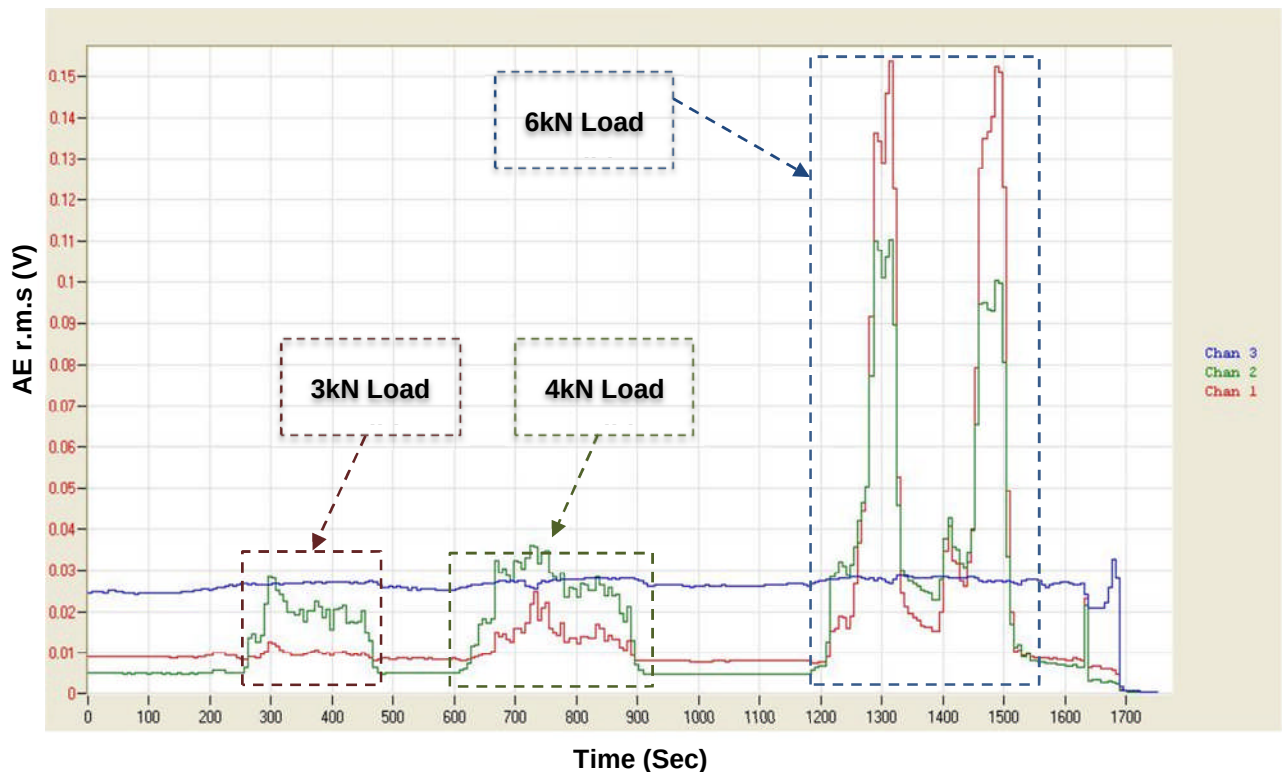


Figure 7-2 Typical AE test sequence at 1500 rpm

At the symmetric load condition of 6kN, the AE RMS level is 0.009V, while in rubbing condition the average AE RMS level is 0.125, showing the astonishingly increase of 1288% over the symmetric load conditions of 6kN, see table 7-2.

Table 7-2 The Relative percentage changes in AE RMS for varying Load between Rubbing and Symmetric load conditions

Oil Type	P (kN)	Speed (RPM)	AE RMS (V) at Symmetric Load	AE RMS (V) at Symmetric Load	$\Delta X\%$
SP 460	3	1500	0.018	0.007	157.1
	4	1500	0.025	0.008	212.5
	6	1500	0.125	0.009	1288

The three channels were connected to the following transducers for recording acoustic emission RMS from hydrodynamic bearing test rig, see figure 7-3:

- a) Channel 1 ĩ hydrodynamic bearing at right axial end;
- b) Channel 2 ĩ hydrodynamic bearing at left axial end;
- c) Channel 3 ĩ the loading arm, drive end.

Figure 7-2 demonstrates that, AE levels increased with increasing load from 3kN to 6kN, while the AE RMS level in channel 2 is higher than channel 1. This can be attributed to the attenuation test which determines signal strength as a function of distance from the emission source. In this case, channel 2 was located closer in order to detect rubbing for 3kN and 4kN loads at the drive end, see figure 7-3 and figure 7-4. On the other hand, at 6kN, the AE level increases even more, but remarkably, the AE level of channel 1, showed higher RMS values than channel 2, which we will investigate further later in this chapter. Figure 7-2; represented channel 2 with average AE RMS level of 0.11V and channel 1 with an average AE RMS level of 0.14V at 6kN load, while at 4kN load, channel 2 had average AE RMS levels of 0.03V and channel 1 had an average AE RMS level of 0.015V.

The higher sensitivity of channel 1 at 6kN than channel 2, can be attributed to the start of a second rubbing on the other side of hydrodynamic bearing, see

figure 7-3 and figure 7-4, which generated a higher level of AE RMS values, close to channel 1. Further signal processing investigation would require captured AE waveforms to demonstrate the effect of 6kN and 4kN on the rotational shaft at 25Hz.

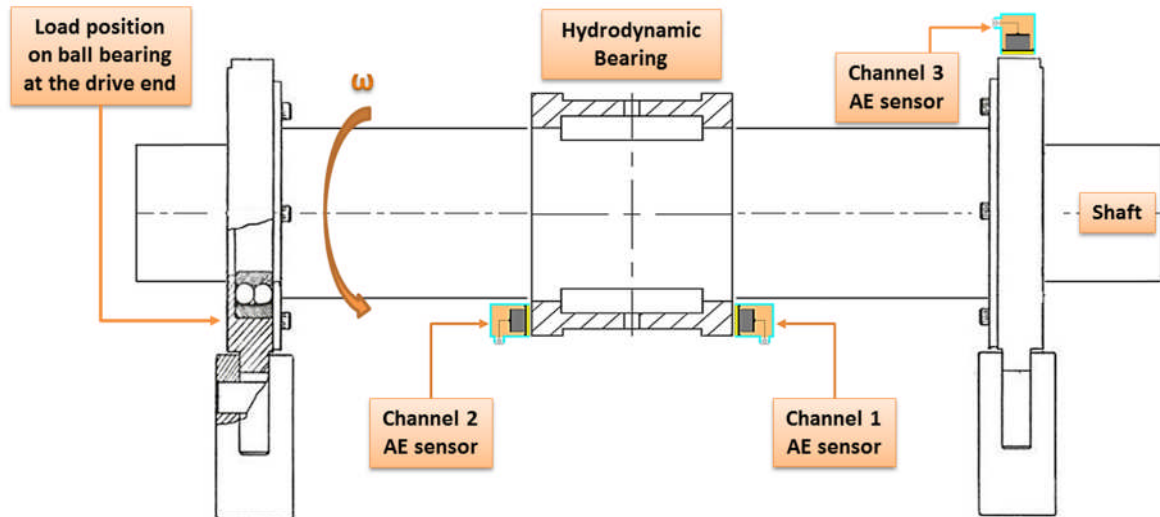


Figure 7-3 AE sensors position

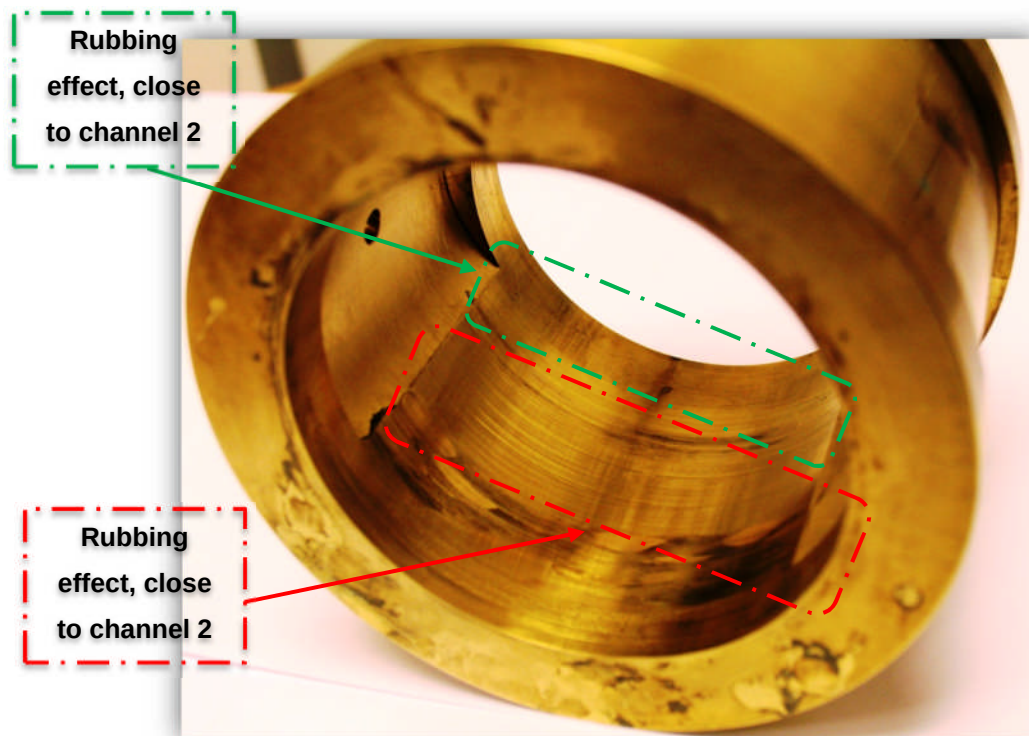


Figure: 7-4 Influence of rubbing on hydrodynamic bearing and shaft

A total of 120 AE data files, sampled at 2 MHz, were recorded at each of the loads and rotational speed conditions for a kinematic viscosity of 460cSt. In the following sections, AE amplitude modulation has been studied and related to the periodicity of the shaft and bearing. Results from the frequency domain, in relation to the periodicity of the unit, were reached following observations on all recorded AE's data. AE parameter RMS, is derived for a better understanding of this experimental research. Results were included for background noise.

The origin of AE activity is microscopic making it a sensitive tool for incipient damage detection. Appendix F, compares different nondestructive testing technologies based on their operational frequency range. The typical frequency range associated with AE signature is between 100 kHz to 1 MHz. Furthermore, due to the high-frequency content of AE and given that the structural dynamics operate below 1 kHz, the captured AE signatures are less likely to be influenced by background noise and structural resonances [108].

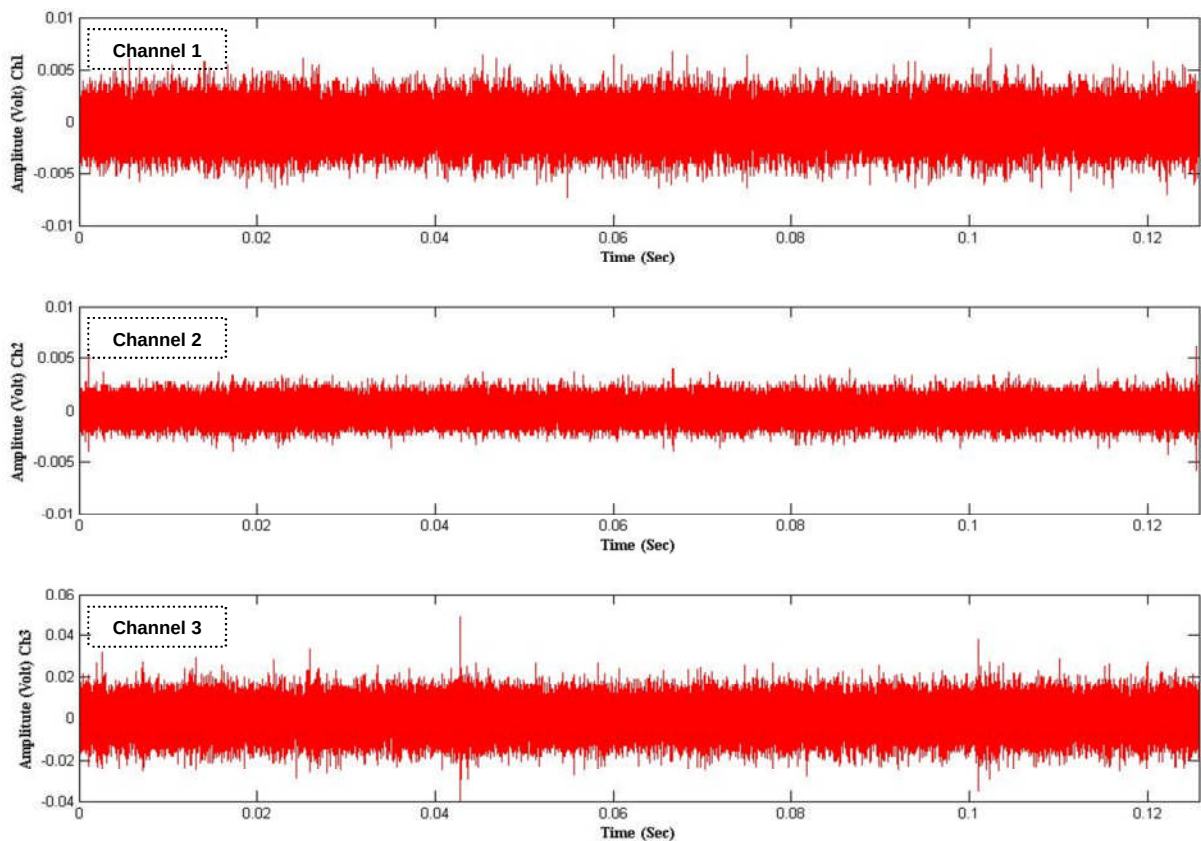


Figure 7-5: Typical AE time traces background noise; 1500rpm and no applied load

AE time traced for background noise has been illustrated in figure 7-5. The acoustic emission RMS value for background noise at the 1500rpm was 0.005V (channel 1 and 2) and 0.02V (channel3). These results suggest that background noise did not interfere with signals during tests and analysis.

7.1.1 Observation of the signals in time and frequency domains

Experimental measurement's results from AE and vibration sensors are presented in analogue format. They appear as a time varying electric voltage, see figure 7-6 and figure 7-7. Data's acquisition systems are employed to convert the acquired analogue signal into digital. In this section signal processing methods have been used to interpret and identify the source of the recorded AE and vibration signals. These signals are presented and analysed in time, frequency or time-frequency domains.

7.1.1.1 Time Domain

The rub simulation tests are performed to assess the sensitivity of the AE monitoring techniques at the onset of contact between the shaft and the hydrodynamic bearing. Throughout the test, both vibration (see section 7-2) and AE signals were captured and processed using the time domain (the vertical axis is amplitude, and the horizontal axis is time) features of the root-mean-square, see figure 7-6 and figure 7-7. Figure 7-6; channel 2 represented an average AE RMS level of 0.11V and channel 1 an average AE RMS level of 0.14V, while figure 7-7 illustrated channel 2 with an average AE RMS level of 0.03V and channel 1 with an average AE RMS level of 0.015V, as discussed earlier in figure 7-2 and figure 7-4. The analysis of the time domain was expected to reveal the overall signal amplitude, and cyclic features [85]. As it discussed, the AE level has been increased by increasing load at the drive end, by 4kN and 6kN at 1500rpm.

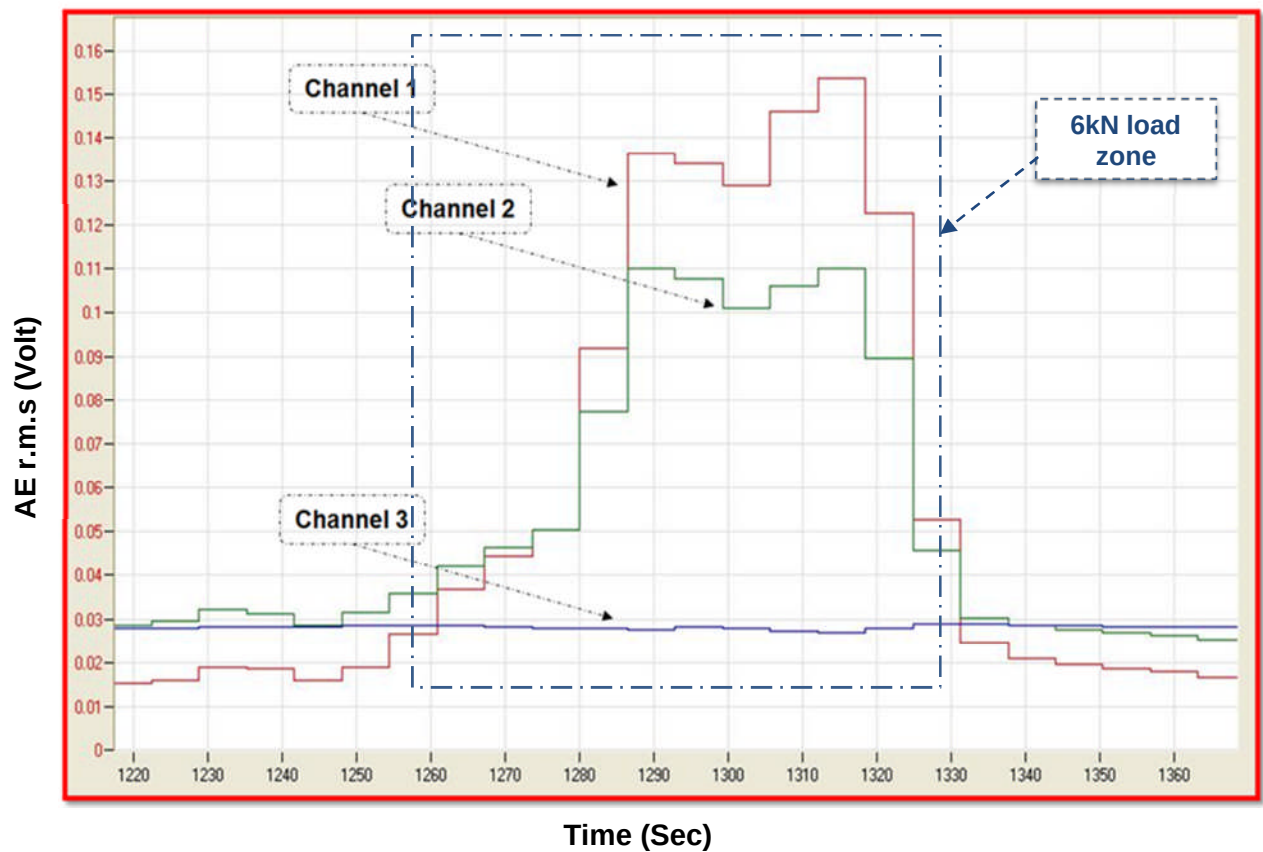


Figure 7-6 AE RMS levels during rub tests at load cycle 6 kN

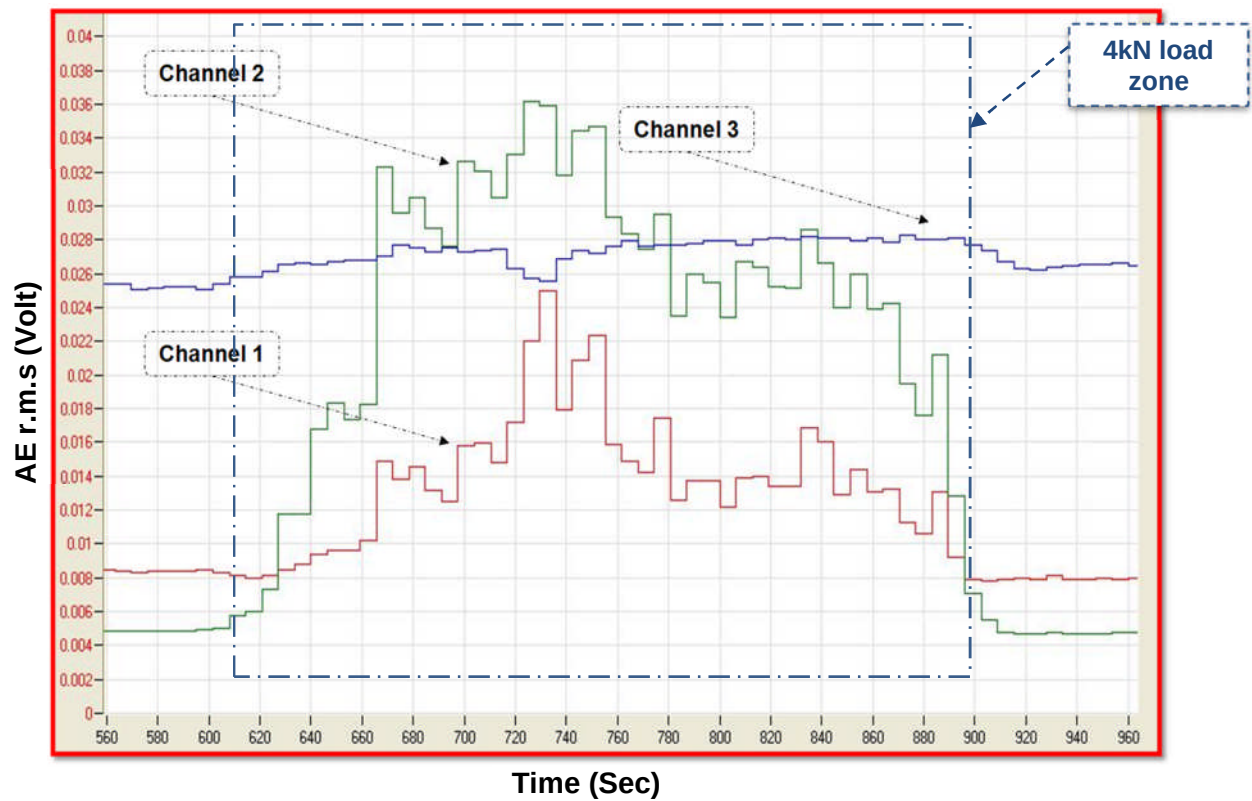


Figure: 7-7 AE RMS levels during rub tests at load cycle 4 kN

The RMS values of AE were compared for increasing loads from no load applied to 4kN and 6kN at a fixed rotational speed of 1500rpm, see figure 7-6 and figure 7-7. The AE RMS values, sampled at 2 MHz, showed considerable increase in AE characteristics as a function of time by increasing load conditions in channel 1 and channel 2, see table 7-2.

Prior to performing the rubbing tests, the temperature of the test bearing was raised to approximately 50°C; this was achieved by running the test-rig in excess of 1500 rpm (25Hz) at no load applied condition for two hours. As soon as the temperature stabilised the test sequence began. Initially, the experimental measurement of the rubbing test was performed, after which 1kN load increases were made within a two-minute time frame to the maximum load of 6kN and 4kN at the drive end only.

A comparison took place when loads were equally located at both sides of the hydrodynamic bearing (balance) and the rubbing simulation (which maximum loads were located at the drive end only). Figures 7-8 and 7-9 highlight a few AE waveforms showing varying degrees of amplitude of the modulated waveform at 6kN and 1500rpm in the absence of rubbing.

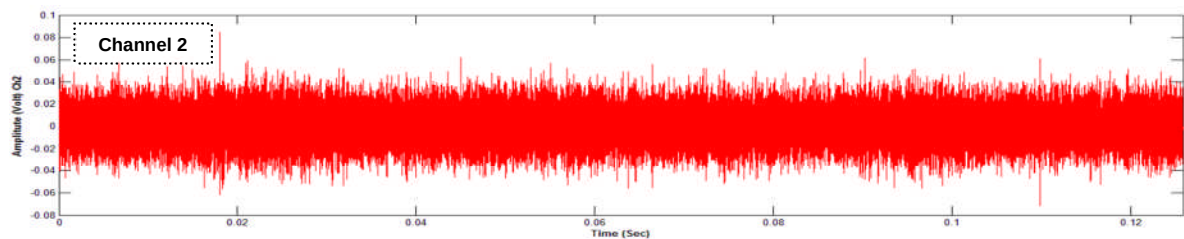


Figure: 7-8 Typical AE waveform for 4kN and 1500rpm balanced loads

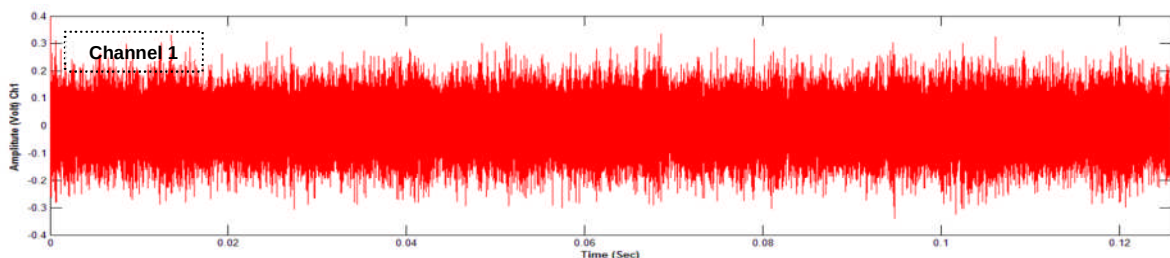


Figure: 7-9 Typical AE waveform for 6kN and 1500rpm balanced loads

In comparison; figures 7-10 and 7-11 highlight a few AE waveforms showing varying degrees of amplitude of the modulated waveform which proved to be a

function of the intensity of the rub. Such observations validate the results of others [5, 6] as indicative of a rubbing contact within the journal.

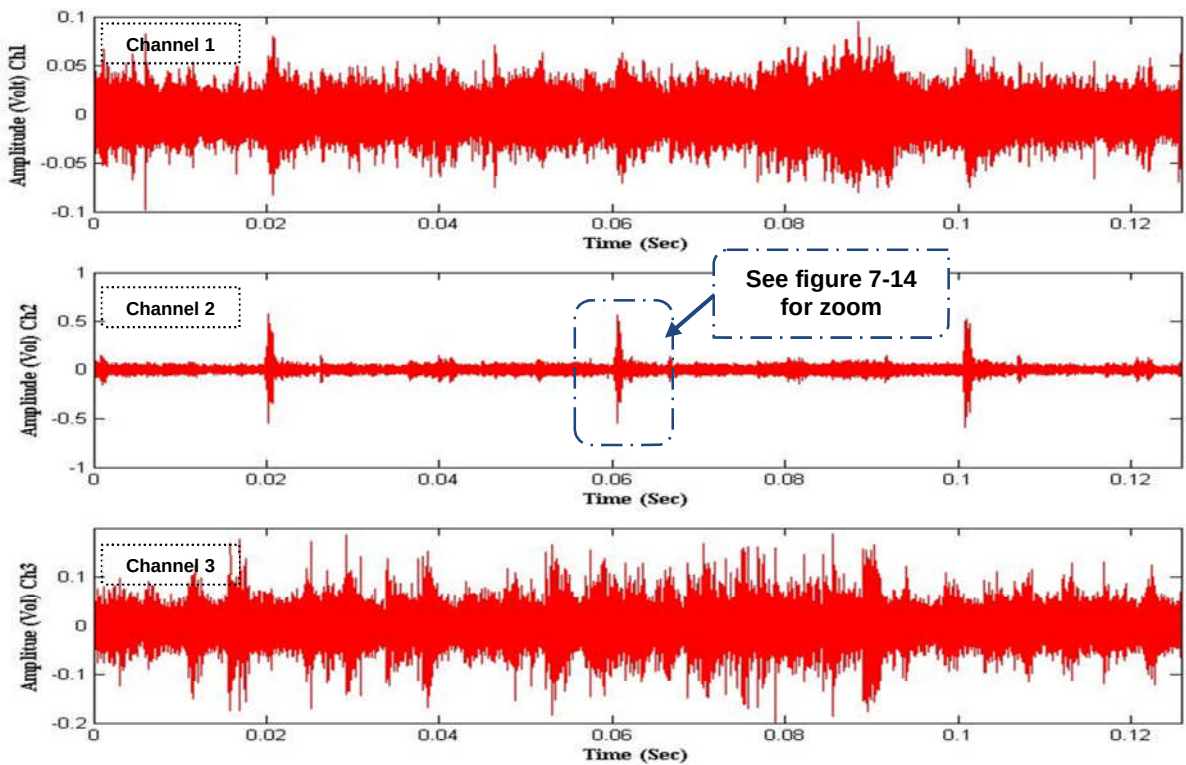


Figure: 7-10 Modulated AE waveforms due to rubbing for 4kN load

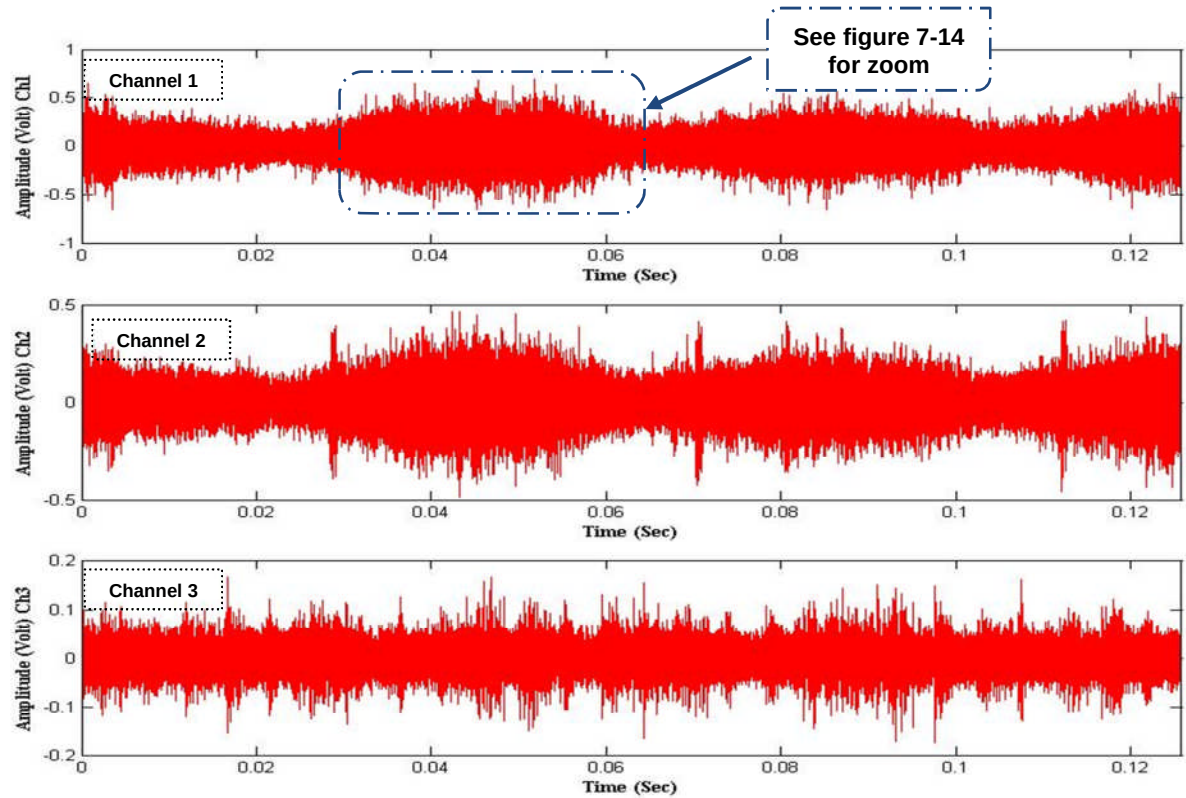


Figure: 7-11 Modulated AE waveforms due to rubbing for 6kN load

Different features can be extracted from the time domain signals. These features have been used to compare the change in signal characteristics over the specified period of time, 0.125 seconds. It must be noted that at the 1500rpm (25Hz), one period of shaft rotation corresponded to 0.04 seconds. In hydrodynamic bearing investigation, rubbing simulation a thrice-rotational speed modulation, particularly at 4kN load condition, was observed; see figure 7-10 and figure 7-11, which illustrated that the periodicity of all AE transient events were three times the rotational speed. This observation was validated by the results of others [5, 6] and indicates that shaft is rubbing on the hydrodynamic bearing.

It is particularly interesting, from the time domain analysis, to note that the waveform observed in the operations showed a periodicity of AE transient bursts; this periodicity is associated with the defect of the hydrodynamic bearing, and it is a clear indication of bearing wear and damage (see figure 7-8 and figure 7-9). The presence of these periodic AE transients bursts in the base acoustic emission classification is clearer in channels 1 and 2 because of the higher noise signal ratio.

An acoustic emission signature modulated at the rotational speed of the unit is illustrative of a continuous rub source, see figure 7-10 and figure 7-11. A continuous rub implies a sustained contact between the shaft and the hydrodynamic bearing generating AE levels above operational background noise levels. The misaligned condition of the shaft demonstrated an increase in contact pressure at a periodicity equivalent to the rotational speed of the unit. The mechanism of a continuous rub established an increasing acoustic emission energy and amplitude levels as a function of contact pressure and rotational speed. Time domain analysis is a valuable tool when the periodicity clearly exists in a signal.

Inspection of the hydrodynamic bearing, close to the drive end (channel 2), revealed rubbing inside the bearing, see figure 7-12. Area A, highlighted in figure 7-12 shows uneven rubbing inside the hydrodynamic bearing, the areas indicated by the orange arrows, producing metal-to-metal rubbing and thus

generating elastic wave activities, see channel 2 in figure 7-10 and figure 7-11 to be detectable by DAQ. This can be attributed as misalignment for the 4kN load condition, while the bearing surface has been eroded smoothly.



Figure: 7-12 Influence of rubbing on hydrodynamic bearing close to channel 2

Inspection of the hydrodynamic bearing, close to the motor (channel 1), revealed the start of a second area of rubbing inside the bearing, see figure 7-13. The area B, highlighted in figure 7-13 shows uneven rubbing inside the hydrodynamic bearing, highlighted by the orange arrow. This was produced by metal-to-metal rubbing and thus generated elastic wave activities, see channel 1 in figure 7-10 and figure 7-11 for acoustic emission signals, detectable by DAQ. This can be attributed to misalignment in the 6kN load condition in which the bearing surface has started to wipe. Finally, area C represents an area of rubbing at the top of hydrodynamic bearing due to misalignment and bending of the shaft.

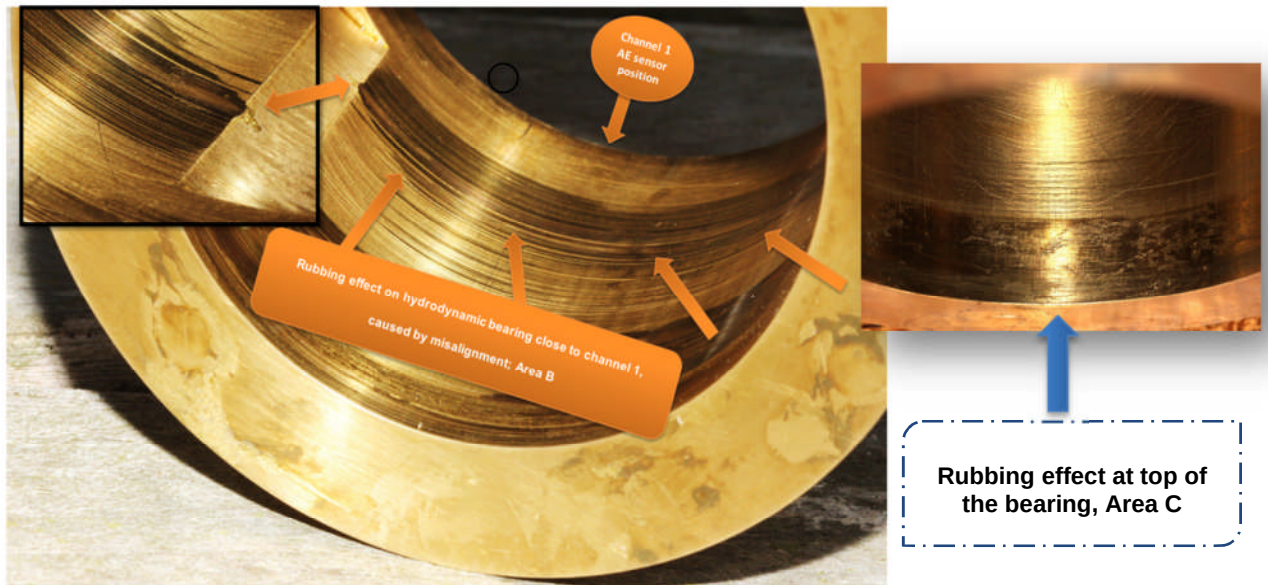


Figure: 7-13 Influence of rubbing on hydrodynamic bearing close to channel 1

AE waveforms have a high data count. This means that the chances of seeing the acoustic emission activities in more detail were slim. Consequently, the data shown in the frequency spectrum will be reduced until it is slightly higher than a single burst but lower than the whole AE waveform. In theory a single burst. The data collected at the same time should have the same frequency; therefore, the result is the same, just with fewer points. [167] Figure 7-14 is a close-up view of a selected portion (within the dashed frame box) of figure 7-10 (the first diagram) and figure 7-11 (the middle diagram) for one revolution of the shaft, also provides evidence that the transient bursts, part of the AE signatures, are not attributable to spurious electronic noise [83].

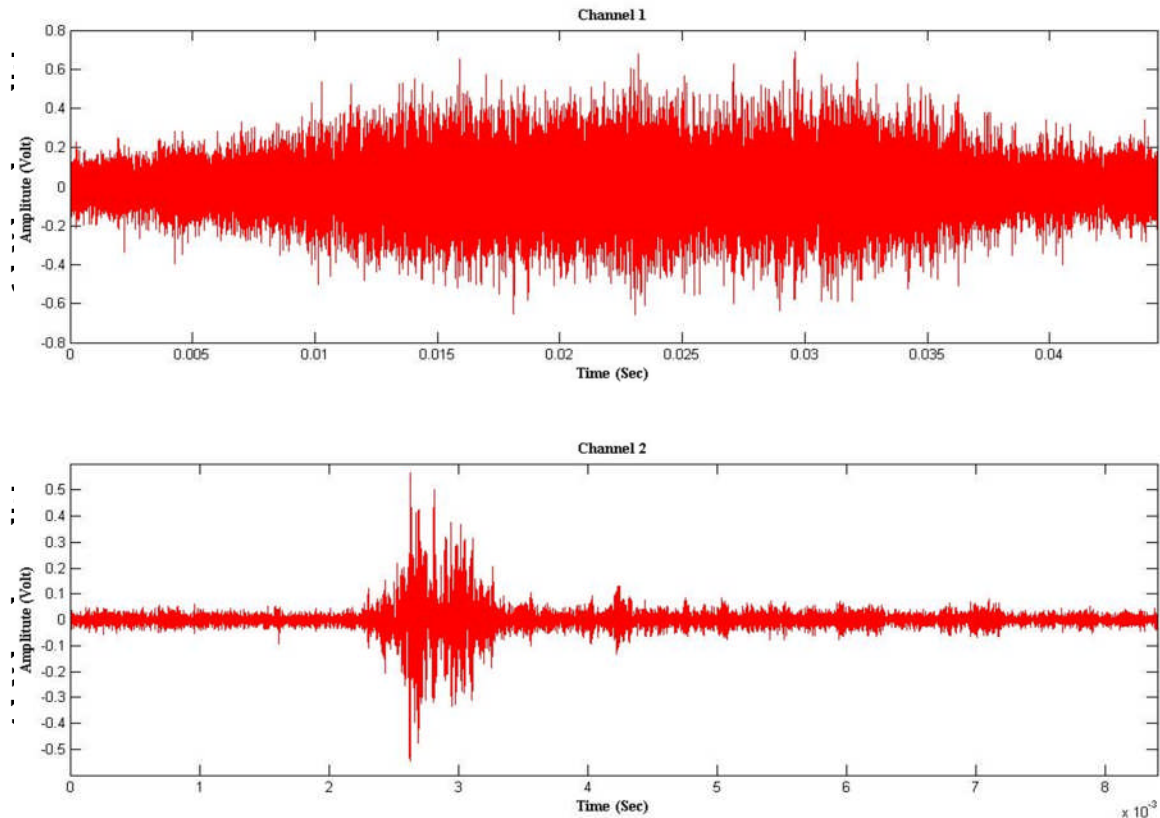


Figure: 7-14 Close-up view of a selected region in figures 7-11 and 7-12

7.1.1.2 Frequency Domain

It is important to study the frequency content of the AE signals, in addition to the time domain analysis. In condition monitoring of rotating machines the recorded signals are diagnosed for a specific frequency band associated with machine elements such as shafts and bearings [79]. This is based on each machine element having a characteristic rotational frequency. A defect on any of the bearing components is manifested as an increase in energy of the particular frequency band associated with the defective component [79]. Throughout this experimental research, the concept of fast fourier transform (FFT) has been widely studied to generate the frequency spectrum of the acquired AE waveforms.

As mentioned earlier, and to proof that the frequency spectra are the same through the waveform, an investigation has been done for two different places with the same amount of data and analysed with FFT spectrum.

The splits are being shown in figure 7-15 where it can be illustrated that there are no relationships except they are taken from the same waveform. The data was taken from the 6kN load condition test. The blue plot (distance A) was taken from the point 44829 to 133600, and the red plot (distance B) was taken from 133600 to 222371 as showed in figure 7-15 and figure 7-16.

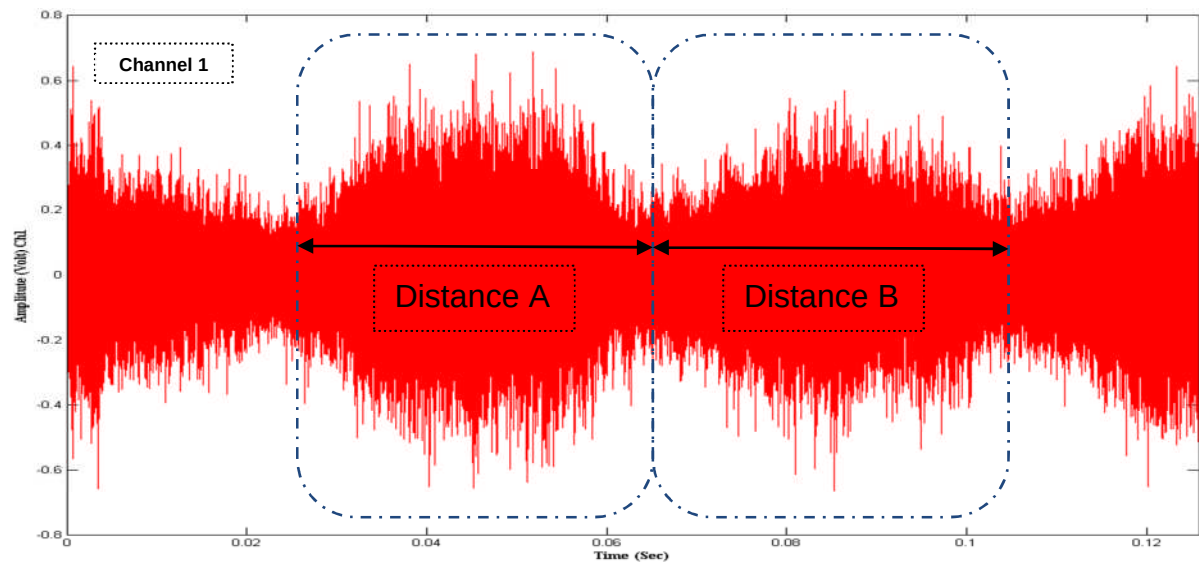


Figure 7-15: different time compression graph

The same investigation took place for 4kN load condition; the blue plot (distance A) was taken from the point 95260 to 164000, and the red plot (distance B) was taken from 164000 to 232740 as showed in figure 7-16.

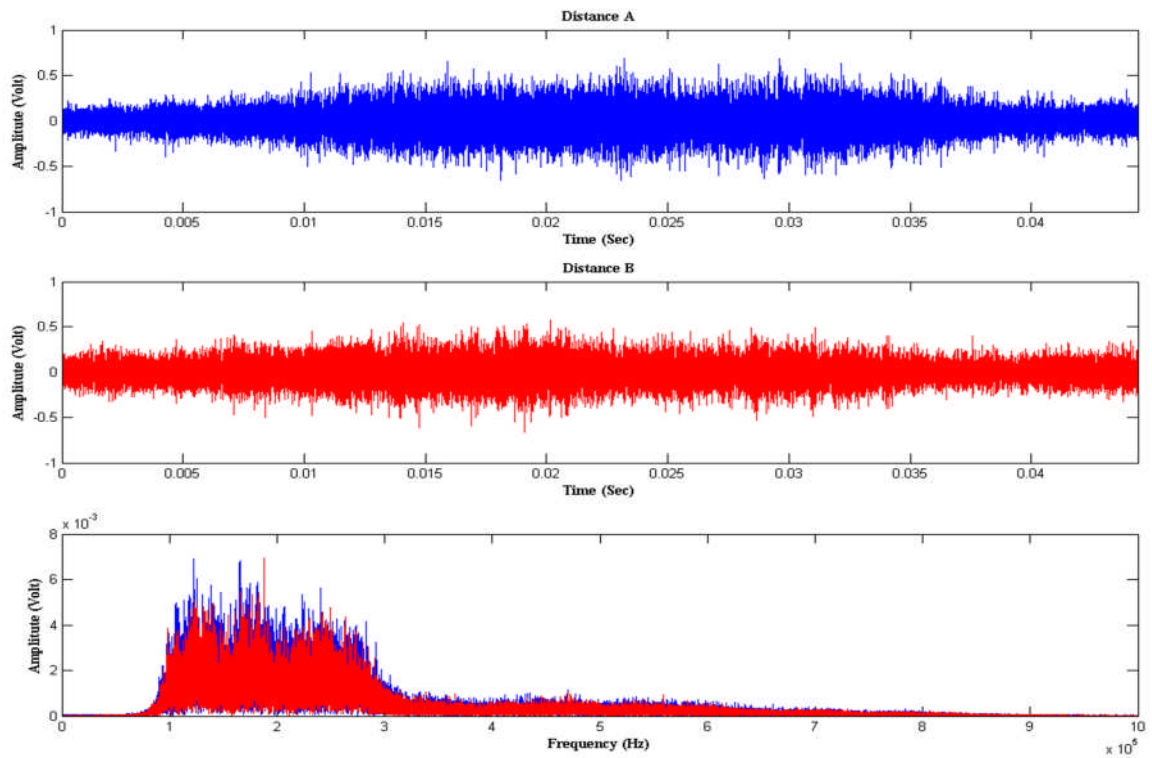


Figure 7-16: Two FFT at different time at 6kN

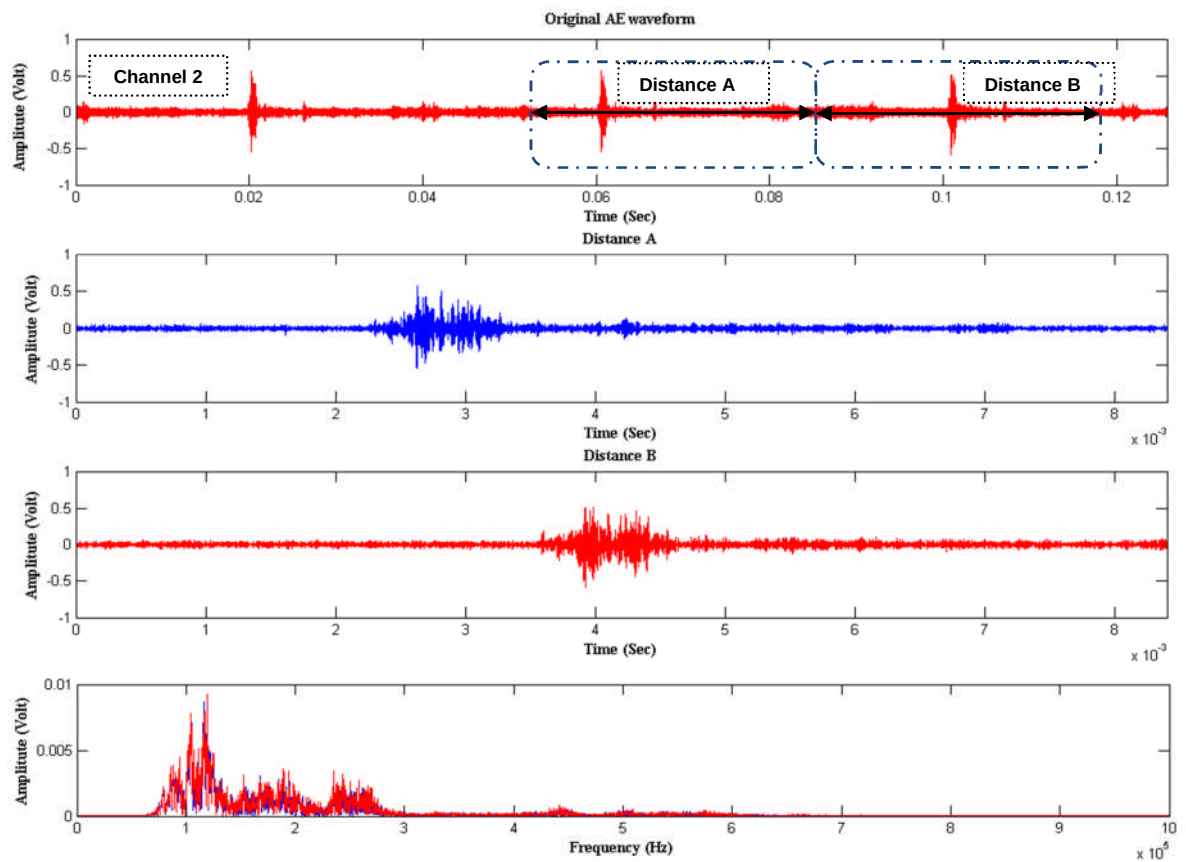


Figure 7-17: Two FFT at different time at 4kN

The FFT analysis is presented in figure 7-16 (6kN) and figure 7-17 (4kN), The results follow the same pattern; so there is no interference when focusing on one shaft revolution but still we need to mention that the data has to extend the points from a single burst.

The frequency spectrum is applied to all AE data for every test condition. A time signature with corresponding frequency spectrum of a typical AE burst signature associated with a 4kN and 6kN load conditions at rotational speed of 1500rpm can be seen in figure 7-18 and figure 7-19. The frequency distribution of the signal obtained when the shaft was rubbing the hydrodynamic bearing shows the wide frequency bandwidth of the vibration response produced by the chaotic behavior of the rubbing phenomenon.

The frequency spectrum of the AE signal shown, in figure 7-18 for 4kN load condition, shows AE signatures of a broad nature, mostly concentrated between 80 kHz and 600 kHz at both load conditions. In addition, AE signatures of relatively lower amplitudes and energies were evident between the higher amplitude bursts. This phenomenon was observed at 4kN load and 1500 rpm rotational speed conditions though at lower AE energy levels.

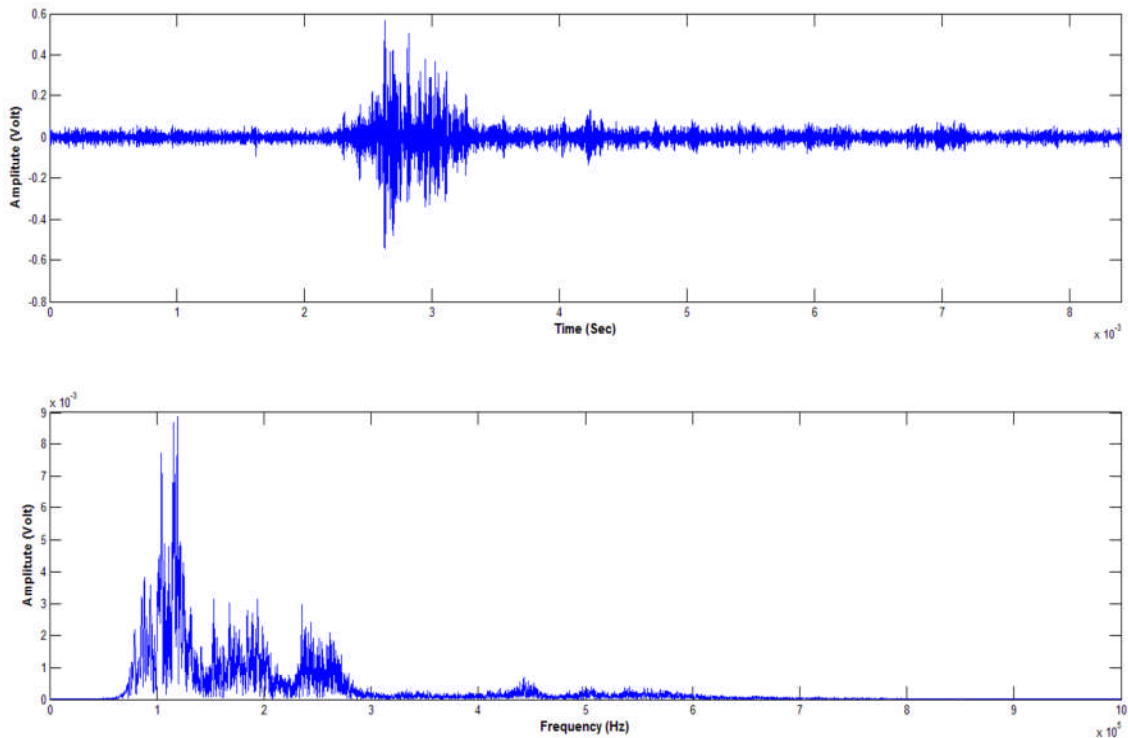


Figure 7-18: AE signature with corresponding frequency spectrum at 4kN

The frequency spectrum of the AE signal is shown in figure 7-19 for 6kN load condition, the AE signatures are of a broad bandwidth, mostly concentrated between 80 kHz and 900 kHz at both load conditions. The high amplitude of AE burst signatures detected at 6kN was generated at 0.04 second intervals, equivalent to the rotational speed of the unit, i.e. 25Hz.

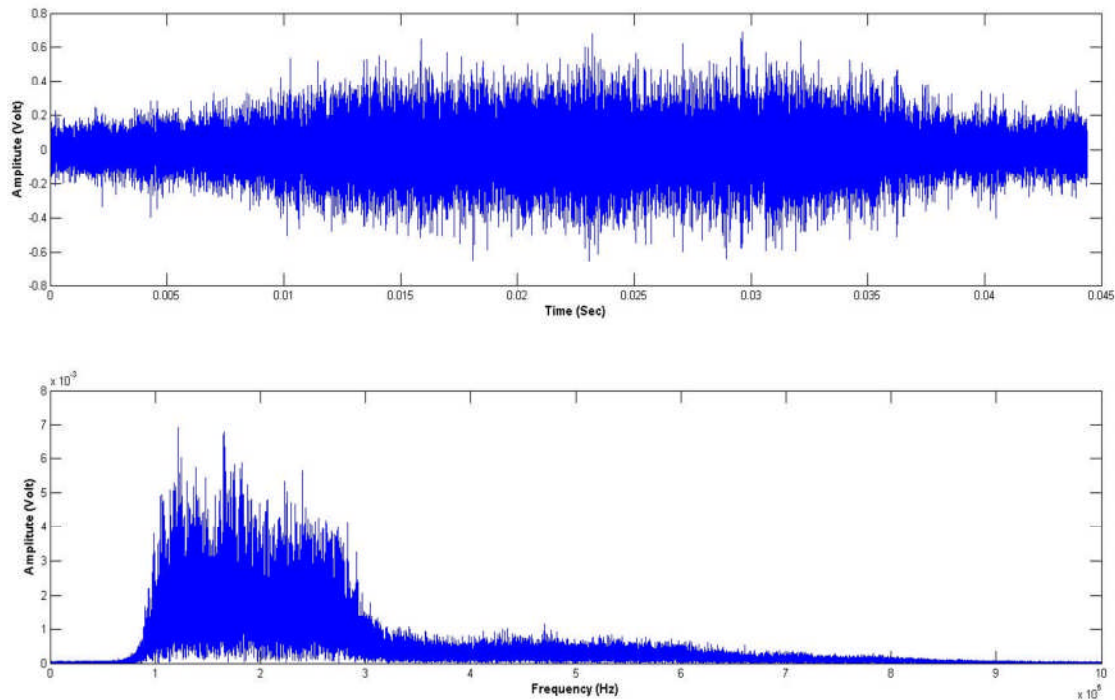


Figure 7-19: AE signature with corresponding frequency spectrum at 6kN

A continuous rub implies a sustained contact between the shaft and the hydrodynamic bearing generating AE levels above operational background noise levels. The misaligned condition of the shaft demonstrated an increase in contact pressure at a frequency equivalent to the rotational speed of the unit. The mechanism of a continuous rub established an increasing acoustic emission energy and amplitude level as a function of contact pressure and rotational speed, creating the modulated high-frequency AE signature [10, 68, and 162]. Modulated high frequency AE waves at the rotational speed of an operational turbine unit had been observed by Sato [5]. It was concluded that the modulation was attributed to the high frequency AE signature.

7.1.1.3 Time - Frequency Domain

The applications of time domain or frequency domain analyses are limited to non-stationary signals. Since rotating machines are often subjected to variation in operational conditions or dynamic fluctuation, the resultant output signals may possess some element of non-stationary behavior. These non-stationary features may contain important information which can provide advantages in diagnosis; hence the application of time-frequency analysis has increased interest as an effective tool in analyzing non-stationary signals [54].

Acoustic emission signals are transient, with spectral content changing as a function of time. Fourier analysis describes signals in the frequency domain and consequently has no time resolution. In the case of non-stationary signals such as acoustic emission signals whose frequency content varies, see figure 7-18 and figure 7-19, it is convenient to describe them in time and frequency domains to know when the different frequencies are presented the rotating machines.

From figure 7-20, different features can be extracted from time-frequency domain signals. These features have been used to compare the change in signal characteristics over the specified period of time, 0.125 seconds. It must be noted that at 1500rpm (25Hz), one period of shaft rotation corresponded to 0.04 seconds. As demonstrated in figure 7-20, in the hydrodynamic bearing investigation rubbing simulation, a thrice-rotational speed modulation, particularly at 4kN and 6kN load conditions, was observed. This illustrated that the periodicity of all AE transient events was three times the rotational speed. In the frequency domain section the spectrum of the AE signal showed the AE activity has of a broad bandwidth and is mostly concentrated between 100 kHz and 700 kHz. However, as shown in the spectrogram in greater detail, AE rub for full waveforms is concentrated between 100 kHz and 300 kHz with distinct features evident between the healthy and damaged shaft and bearing, see figure 7-20.

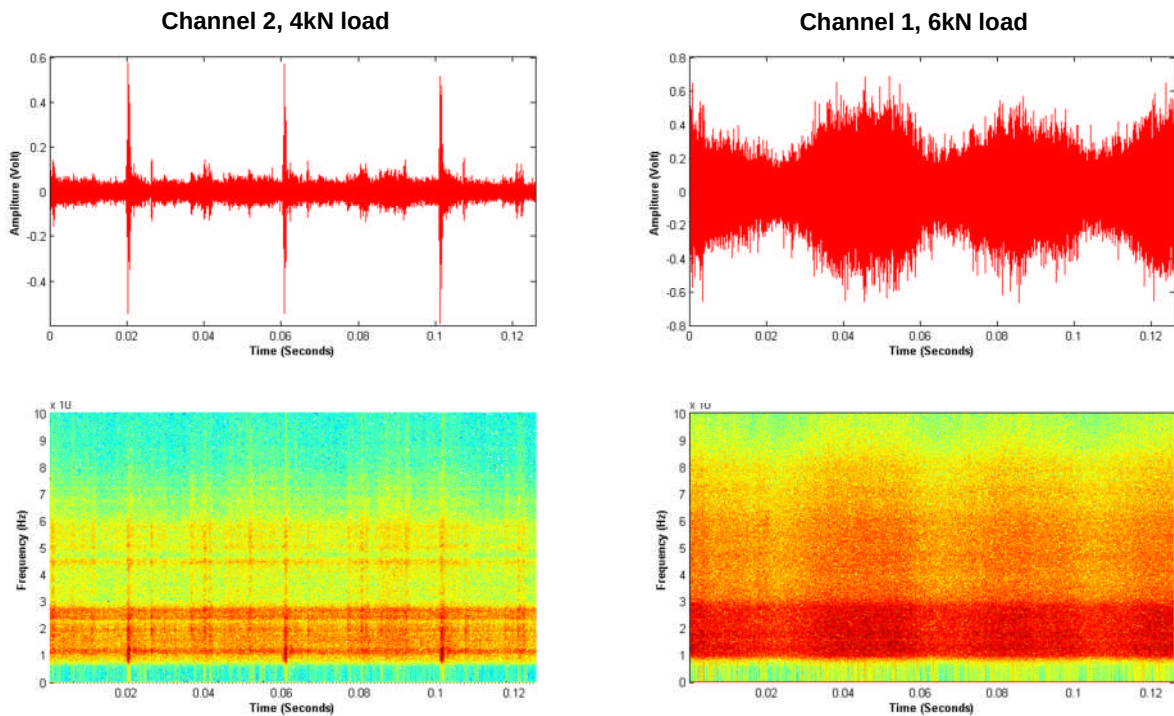


Figure 7-20 Spectrogram plot associated with 4kN and 6kN load condition

Figure 7-20, the color coding is green for low AE amplitudes throughout, and red for high AE amplitudes. It is clear that the intermittent nature of the rubbing and, in concordance with figure 7-18 and figure 7-19, that the strong energy of the acoustic emissions are limited to frequencies between 100 kHz to 300 kHz. This is attributed to metal wipe, or tearing and breaking of surface asperities during the sliding/rubbing simulation. This indicates a severe form of rubbing, as seen in figure 7-4, figure 7-12 and figure 7-13. The time-frequency spectrogram showed again the wide spectral response and differences between light (4kN) and severe (6kN) shaft rubbing against the hydrodynamic bearing and location in time of the rub-impacts. It can be concluded that the rubbing condition was related to misalignment.

It can be seen that as the experiment progresses the contact between shaft and hydrodynamic bearing increases and higher frequency components appear. These components get their peak values near the middle of the time axis where the maximum contact is reached. They then begin to fade showing a trend for the higher frequencies to disappear as the contact decreases. These figures confirm the wide spectral bandwidth of the rubbing phenomena; see figure 7-21 for 4kN and figure 7-22 for 6kN load conditions. The time-frequency

distribution of the signal obtained when the shaft was rubbing the hydrodynamic bearing shows the wide frequency bandwidth in the acoustic emission response produced by the chaotic behavior of the rubbing phenomenon.

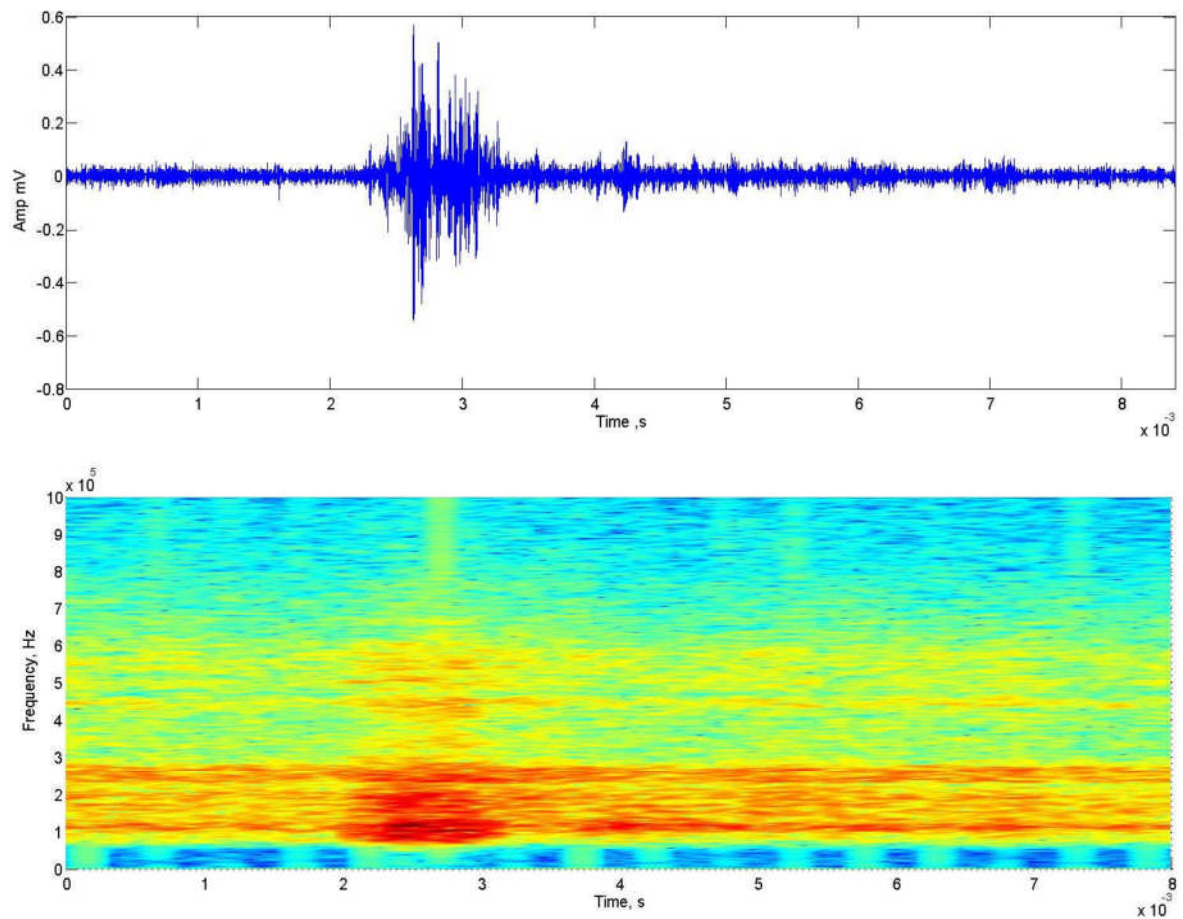


Figure 7-21 Spectrogram analyses for stimulated rub at 4kN load condition

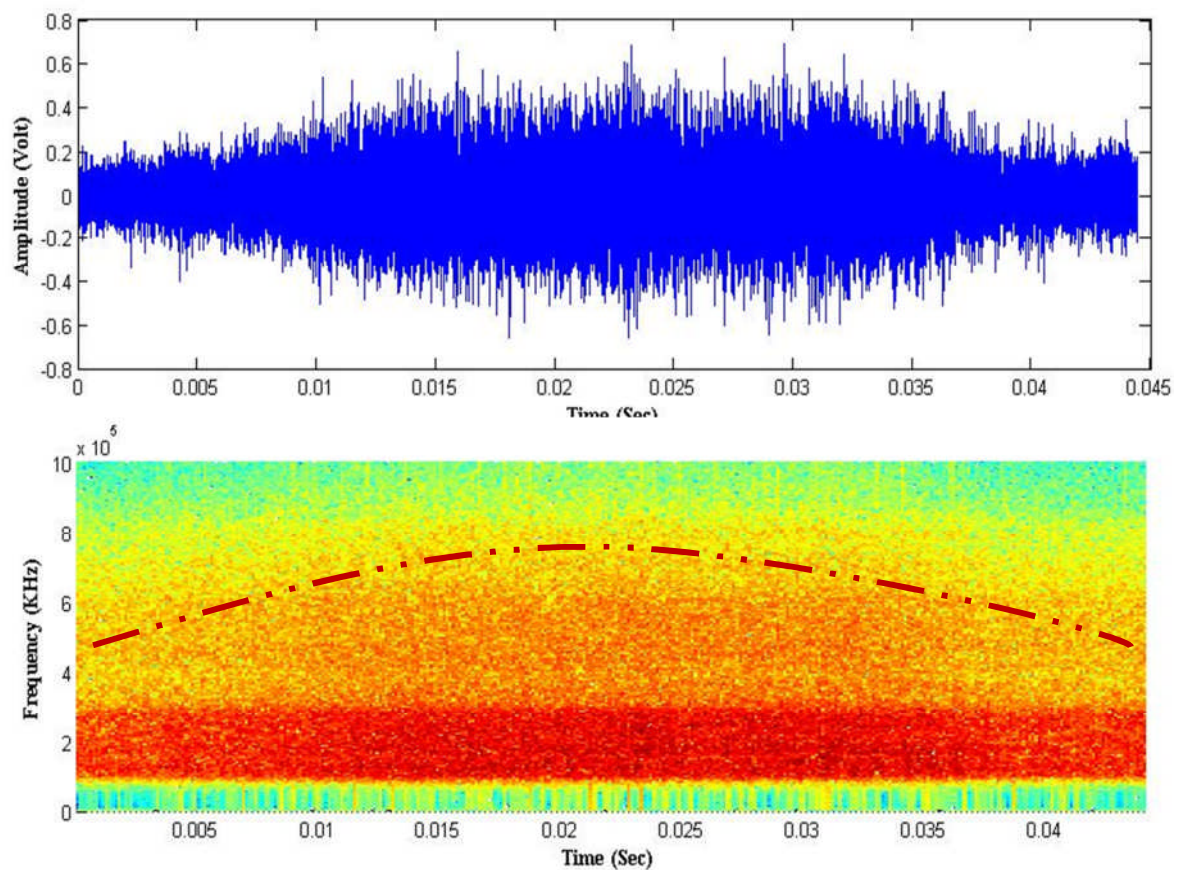


Figure 7-22 Spectrogram analyses for stimulated rub at 6kN load condition

It has been established in this study, that when the hydrodynamic bearing is subject to a misaligned condition; metal wipe, tearing of the molecules and metal to metal contacts can occur between the hydrodynamic bearing and the shaft producing high-energy acoustic signatures, see figure 7-21 and figure 7-22.

7.2 Vibration Results

In rotating machines the high-frequency components have been used for early detection of defects. Vibration analysis is used to investigate the overall value for the energy in the high-frequency analysis which is then compared with reference values and correlated to the severity of the defect. [163, 38, 45] The high-frequency analysis techniques are associated with the excitation of natural frequencies that have been applied in this experimental research, for enveloping, demodulating or PeakVue and peak value analysis; see section 5-2.

PeakVue analysis is the technique described in this section and compared with demodulation for rubbing simulation. As mentioned in section 5-2, in the high-frequency analysis, there are two main difficulties; the first is the low amplitude of the high-frequency bursts compared with the other components of low frequency and secondly is the difficult of identifying the characteristic defect frequencies of the bearings due to the noise and the poor spectral resolution [114, 122].

7.2.1 Case Study: Demodulation and PeakVue

This case study would compare the advantage of PeakVue over demodulation which has been investigated at 1000 rpm (16.6 Hz) rotational speed and the 1kN load condition when the load is equally separated at both sides of the hydrodynamic bearing (balance).

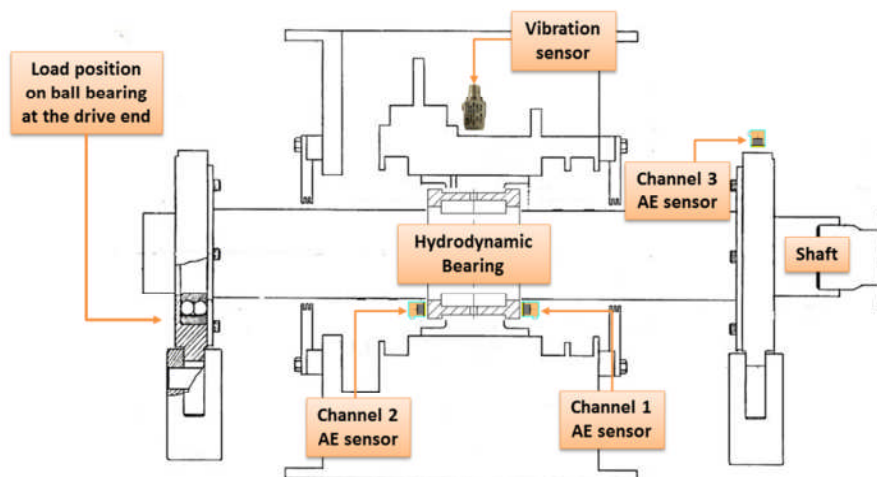


Figure 7-23 Vibration and AE sensors position

The vibration signals were measured with accelerometers placed on the bearing pedestal, Figure 7-23 illustrates a schematic view of the setup employed to acquire the vibration signal from the hydrodynamic bearing test rig.

Peak values were captured over sequential, discrete, time intervals. Plotting these peak values over a time interval and generating a frequency based spectral plot produces a waveform that can then be analysed, see figure 7-24. The signal processing is aimed at monitoring stress waves with short durations.

A frequency spectrum of the vibration waveform at 1000 rpm (16.6 Hz) has been illustrated in figure 7-24, the important features of frequency content, including shaft frequencies, and second harmonic of shaft frequency, are clear throughout the frequency spectrum.

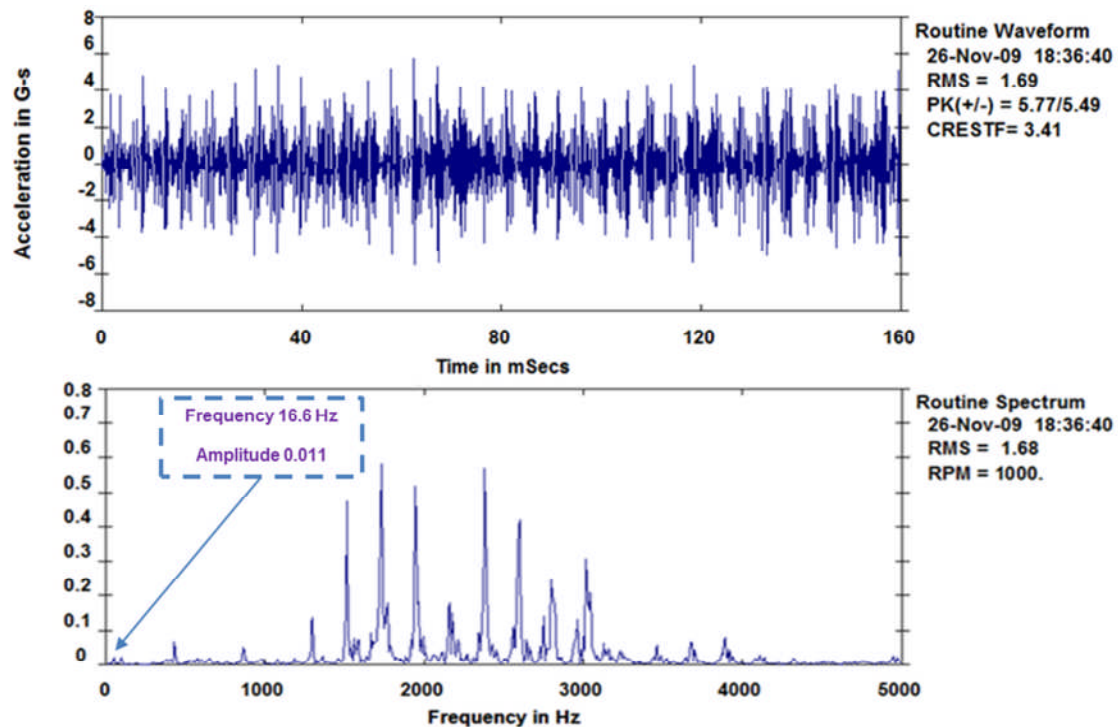


Figure 7-24 Vibration waveform at 1000 rpm with FFT frequency 16.67Hz

It needs to be mentioned that the peak values were measured in g-s. In order to obtain a better time frequency resolution on the frequency content of the vibration signals in the rubbing test, a further investigation has been proposed on the entire vibration data, as described earlier; PeakVue and demodulation.

The frequency spectrum analyses of the experimental measurement are presented in figure 7-25 and figure 7-26. Due to the low rotational speed, see pattern the US-7493220-B2, PeakVue and demodulation filters were both set to 200 Hz low-pass using 3200 lines of resolution for more accurate results. During the signal processing, a defect could be indicated by excessive activity when compared to the previous condition of the part or to other similar parts at known discrete fault frequencies.

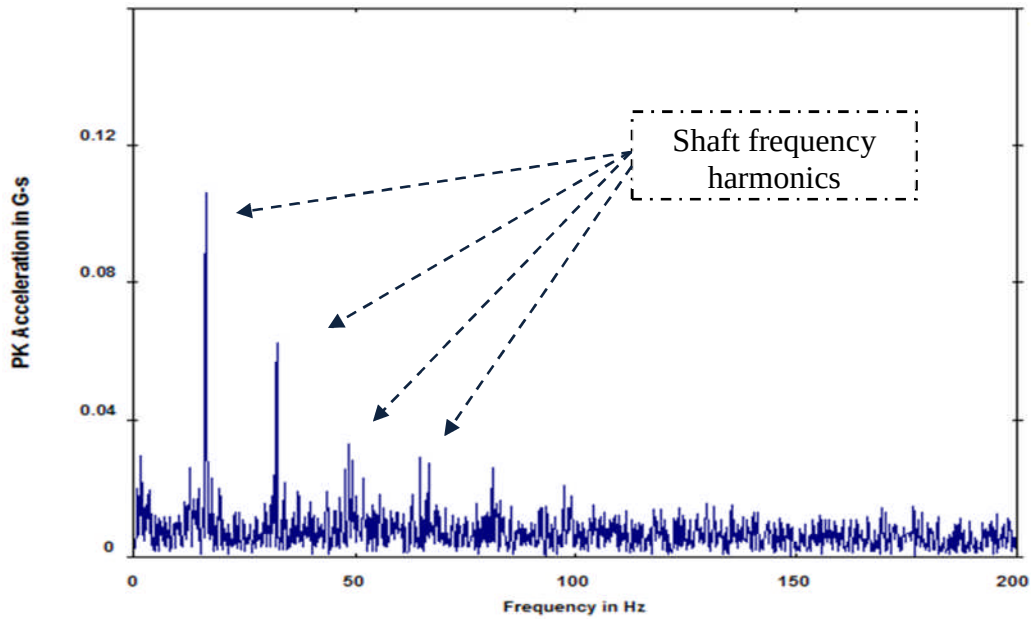


Figure 7-25 Demodulation spectrum at 1000 rpm and 1kN

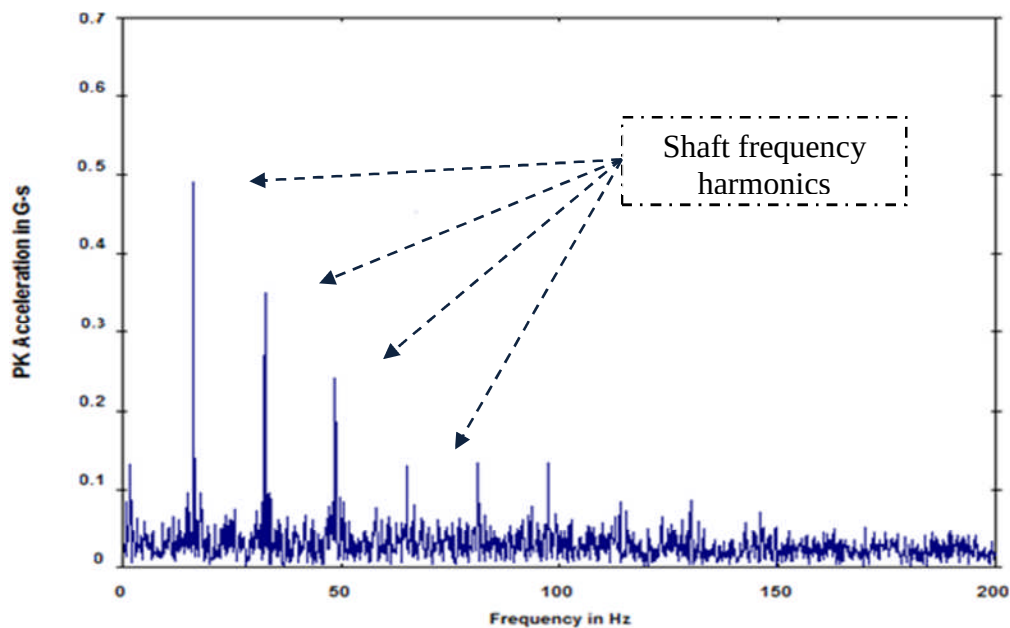


Figure 7-26 PeakVue Spectrum at 1000 rpm (16.67 Hz) and 1kN

If a bearing defect is identified in the PeakVue and normal velocity spectral data then it is expected for both methods to be used for trending the signal values to identify the severity of the defect, see figure 7-27 for further comparison between demodulation and PeakVue spectrum. The results demonstrated from both techniques were similar. Nevertheless, the amplitude obtained using PeakVue techniques tends to be higher than when using demodulation. This makes it easier to detect faults.

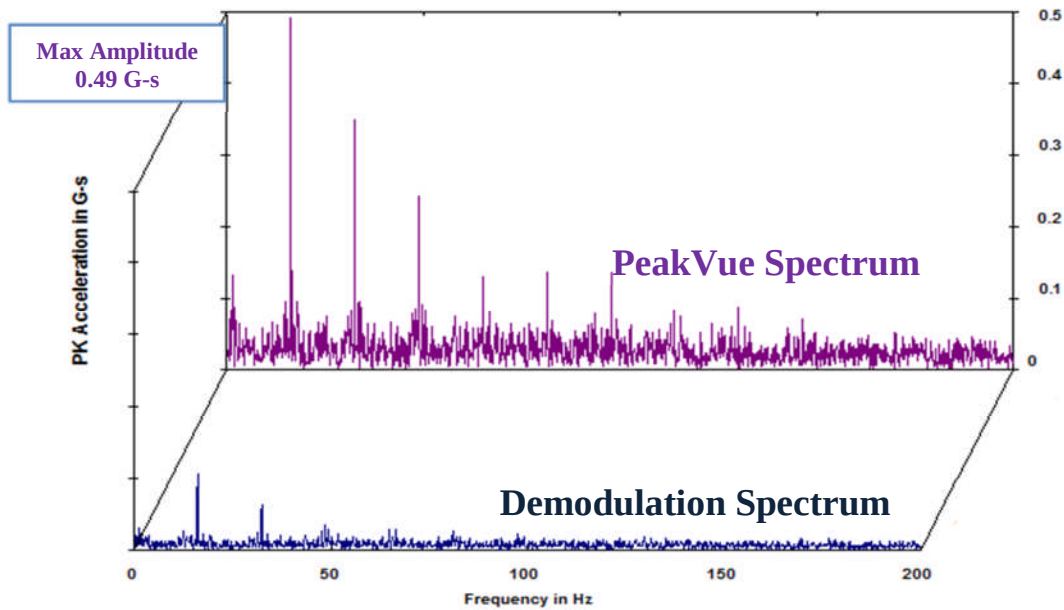


Figure 7-27 Comparison between the Demodulation and PeakVue

The final comparison has proved that the demodulation and PeakVue techniques are more useful than the normal FFT spectrum analysis for condition monitoring of rotating machine with high rotational speeds.

7.2.2 PeakVue Results for Rub Simulations

A total of 40 vibration data files, sampled at 10 kHz, were recorded through an accelerometer placed on the bearing housing close to the hydrodynamic bearing, see figure 7-3. The rotation speed for the test is 1500 rpm and the load was maintained at 4kN and 6kN for kinematic viscosity 460cSt. The methodology described in the section 7-1, was applied and experimental runs were carried out with the aid of the test rig to obtain a deeper comprehension of the rubbing phenomenon using vibration analysis.

In the next step, the vibration spectrum figures proved to be a valuable tool in conjunction with routine vibration data to provide a more complete picture of condition monitoring for the rotating machine. The overall rubbed trends of vibration of the shaft against the bearing are depicted in figure 7-28. When a misalignment of the hydrodynamic bearing is introduced with higher radial load, from no applied load to 4kN and 6kN, the peak amplitude of the accelerometer

signatures increased from 2.08V to 2.17V and 2.24 at present of 4kN and 6kN load conditions, respectively.

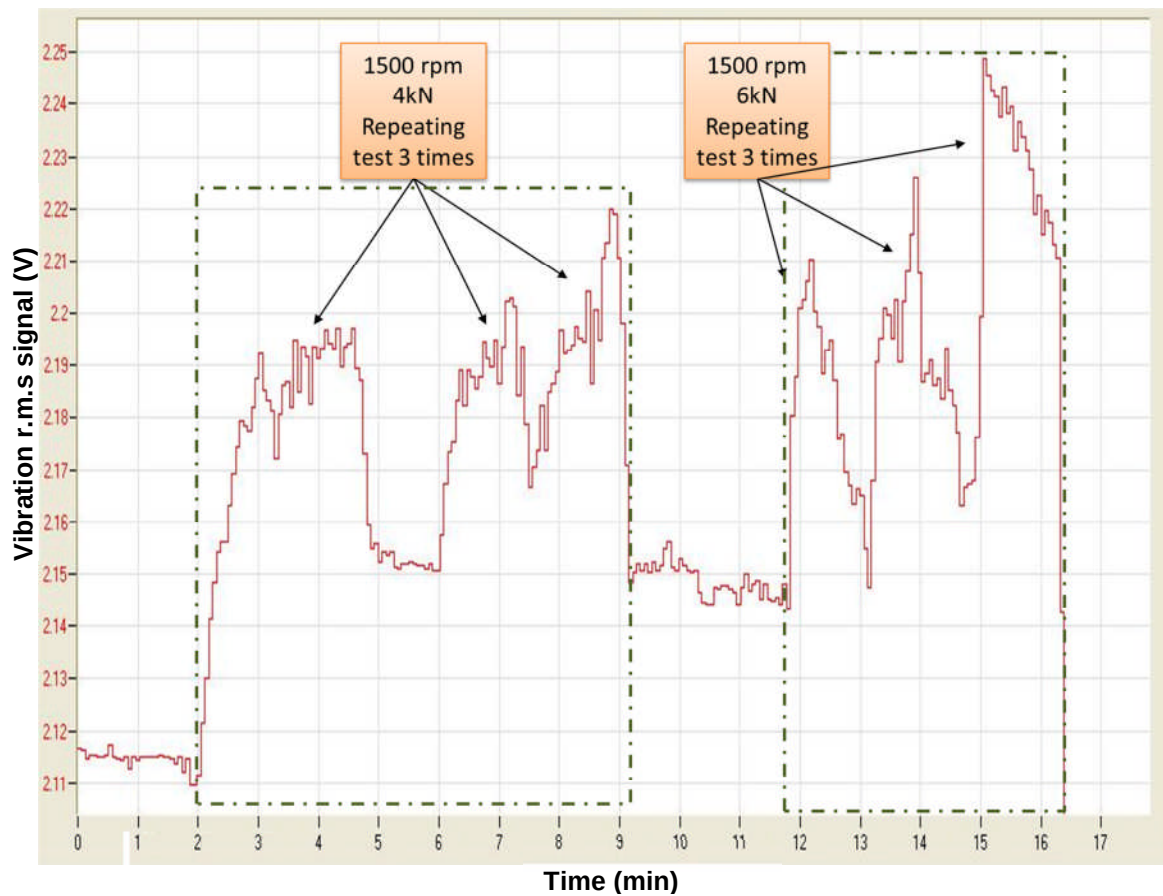


Figure 7-28 Vibration levels during rub tests at 4kN and 6kN condition

The RMS values of vibration signatures for all load conditions were compared for increasing loads at a fixed speed, see figure 7-29. From this figure, it was noted that the change in loads (4kN and 6kN) had a small influence on the level of vibration activity for 1500rpm rotational speed condition, around a 25% increase in RMS value. However, the change in load conditions from “no load applied” to 4kN or 6kN, increased vibration levels significantly. The corresponding increase in vibration levels from changes in loads were in the order of over 42% on average, see figure 7-29.

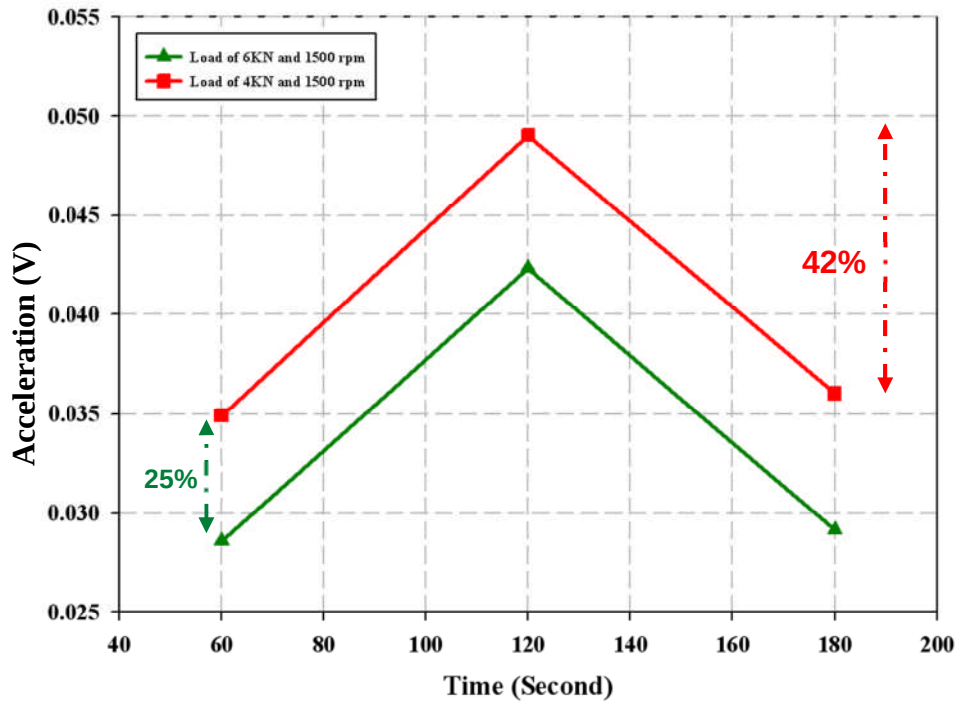


Figure 7-29 Acceleration compression for load conditions

To fully understand the behaviour of the hydrodynamic bearing under rub simulation it is important to know that this research needed to be carried out under controlled noise and vibration conditions. However, in industry more difficulties can be found especially at high rotation speeds due to the various other noise sources, from other machines (motor, belt drive, coupling, ball bearing, etc.) in addition to the rotational speed and load variation data collected during the experimental measurements. Therefore, the traditional spectral analysis must be complemented with the use of other techniques, which are available for vibration monitoring analysis as described earlier.

In this section, the concept of fast fourier transform (FFT) has been studied using hamming window as it provides a good compromise on both frequency and amplitude resolution to generate the frequency spectrum of the acquired vibration waveforms. A frequency spectrum of the vibration waveforms recorded throughout this investigation has been presented at various load conditions, see figure 7-30. This figure depicts important features of frequency content include the shaft frequency and the second harmonic of shaft frequency of the vibration signal at 1500rpm (25 Hz) at different load conditions. As the shaft is rubbing

the hydrodynamic bearing on this research the FFT frequency spectrum recognized shaft harmonic frequencies and there is no clear sign of rubbing in the frequency spectrum, see figure 7-30. Furthermore another observation from figure 7-30 showed that the amplitude of the frequency spectrum has been increased by increasing loads from the synchronized load to the misalignment conditions (4kN and 6kN).

In order to obtain a better time frequency resolution on the frequency content of the vibration signals in the rubbing test a further analysis (PeakVue and demodulation) has been demonstrated on the entire vibration data, as proposed earlier for high-frequency components.

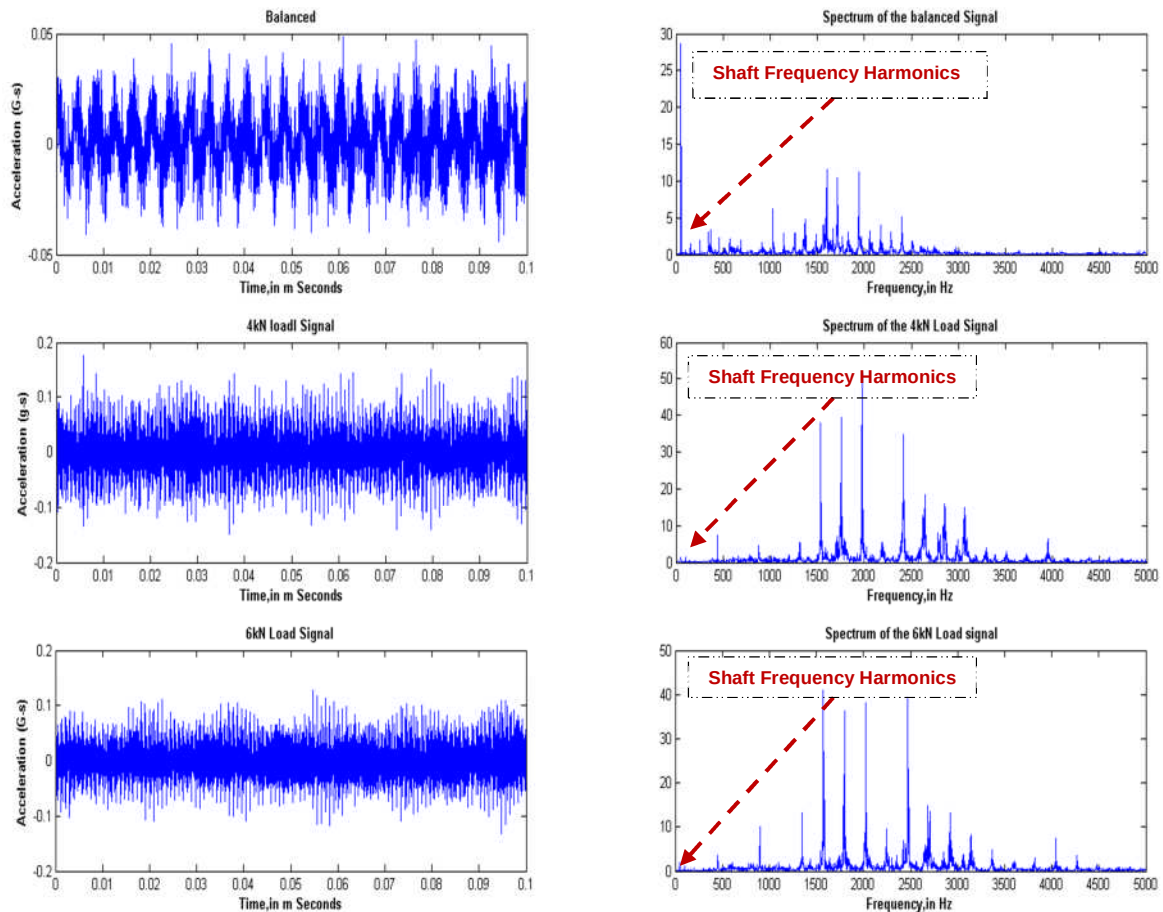


Figure 7-30 FFT spectrum with balanced, 4kN and 6kN load conditions

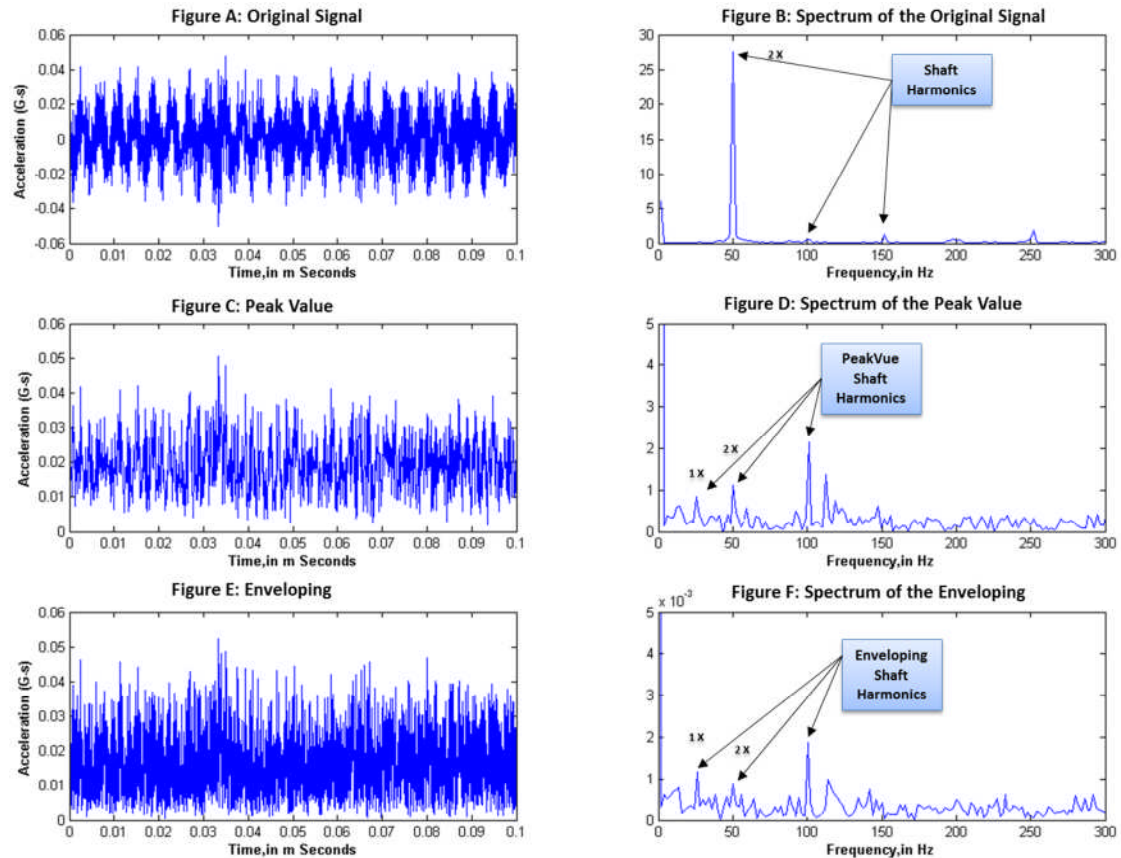


Figure 7-31 Comparison frequency spectrum at balanced condition with different techniques

Figures 7-31 presented the comparison frequency spectrum at the balanced condition (symmetric load condition of 6kN) with different techniques (FFT, PeakVue and Enveloping). This figure depicts important features of frequency content including the shaft frequency and second harmonic of the shaft frequency of the vibration signal at 1500rpm (25 Hz). There is no indication of rubbing in the frequency spectrum.

Figure 7-32 and figure 7-33 illustrate the comparison frequency spectrum at the 6kN and 4kN load condition with different techniques (FFT, PeakVue and Enveloping). It should be mentioned that radial force acts from the centre of the shaft; the magnitude of the radial force is related to inertia and rotational speed at the moment of contact between shaft and hydrodynamic bearing. Due to higher stiffness of contact and structure, the higher level of pressure can be

created. This accelerates the shaft causing it to rebound from the surface at 1X (25Hz) vibration in frequency spectrums.

Figure 7-32 to 7-35 contains the results of using the PeakVue, Enveloping and FFT methods. In these figures, the original signal has been labelled as figure A. In figure B a spectrum of the original signal has been depicted. This spectrum is obtained simply by the standard fast Fourier transform (FFT) method. Figure C shows the peak value of the signal that was obtained by the method described above. The FFT of this signal has been depicted in figure D. Figure E shows the enveloping value of the signal and the FFT of the enveloping signal has been depicted in figure F. Comparing the spectra in figures 7-32 to 7-35, it can be found that the amplitudes of the friction frequency and its harmonics increase with the increasing synchronized load to misalignment conditions (4kN and 6kN).

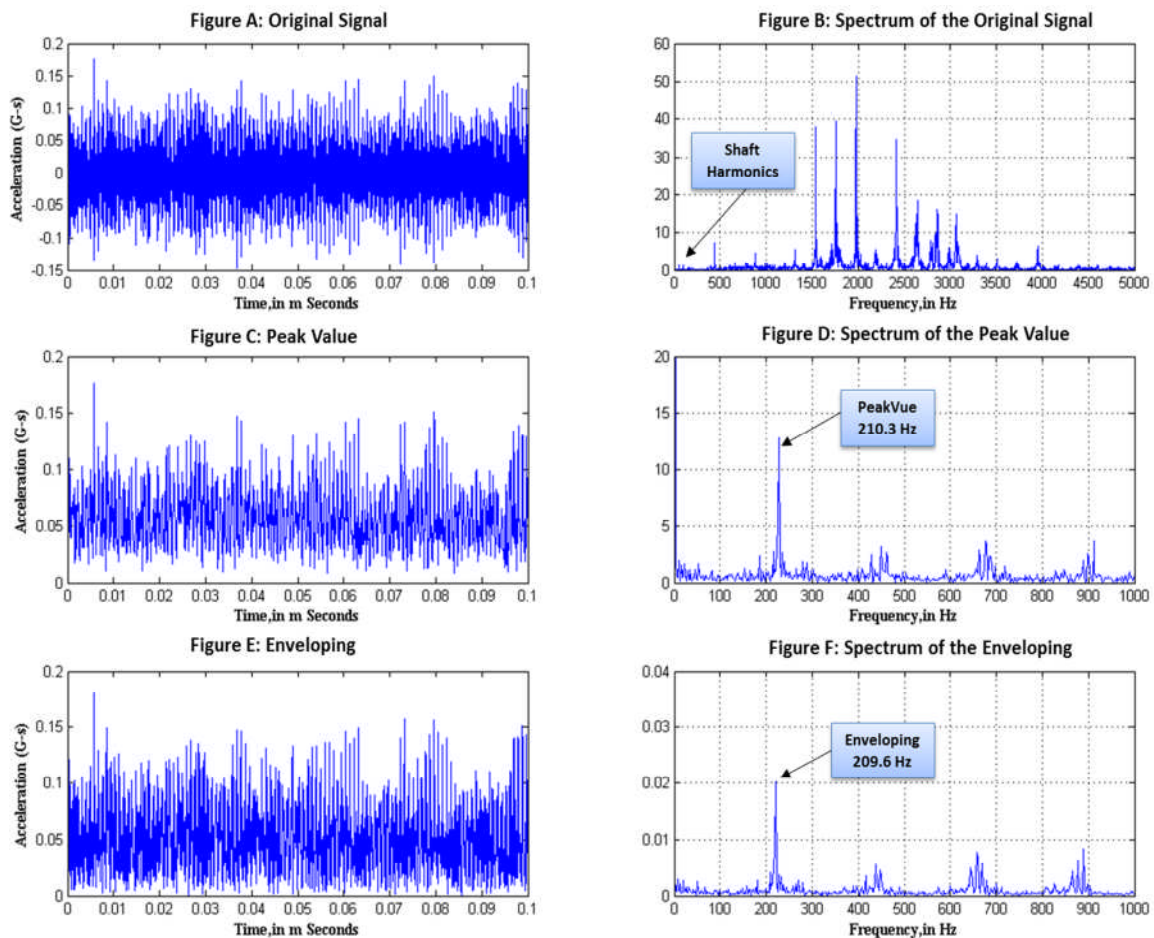


Figure 7-32 Comparison frequency spectrum at 4kN load condition with different techniques

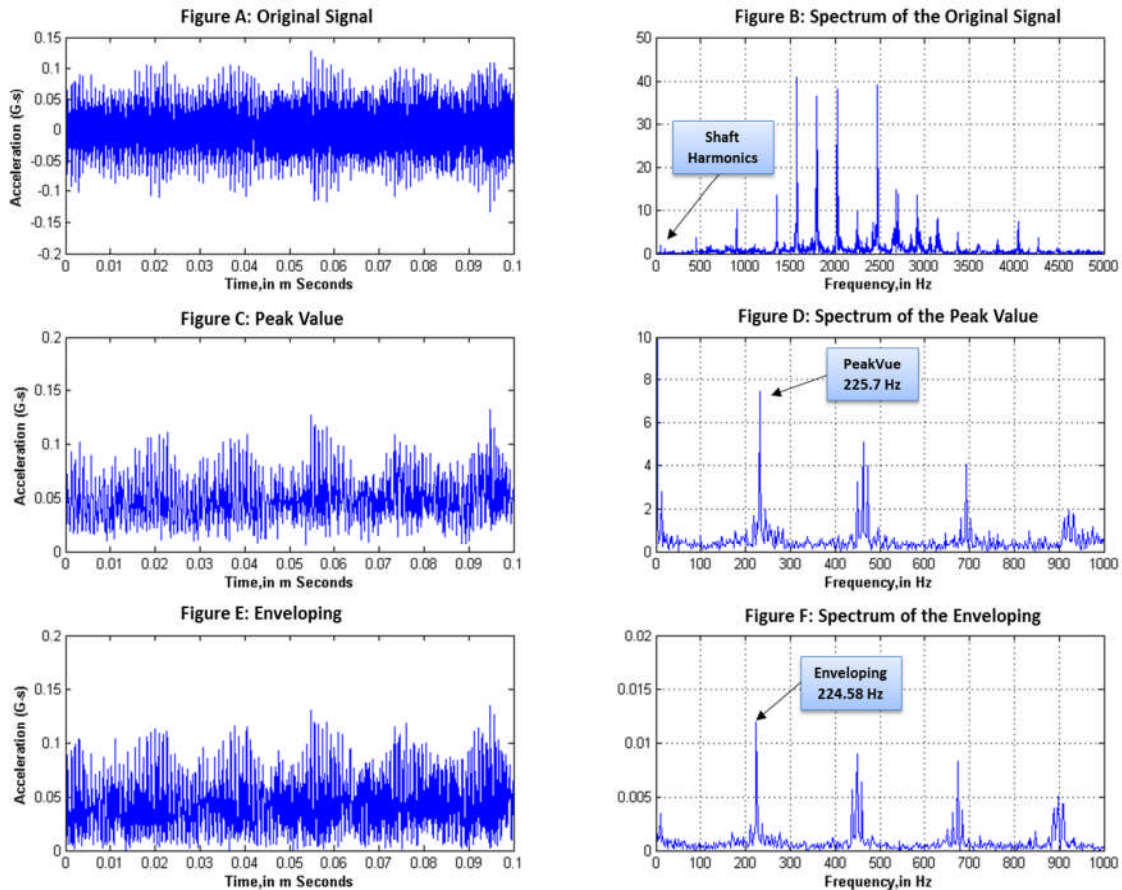


Figure 7-33 Comparison frequency spectrum at 6kN load condition with different techniques

Rubbing can cause strong harmonics of the shaft rotational speed in vibration analysis. In more severe cases, half order and one- third order harmonics can be presented in the FFT frequency spectrum, see figure 7-32 and figure 7-33; of the original signal. In rubbing simulation once the shaft and hydrodynamic bearing are in contact the stiffness of the test rig increases. This behaviour is due to a partial rub which produces impulses which excite system resonance and in addition create harmonics. After conducting the rubbing simulation, see figure 7-32 and figure 7-33, it was found that the dominant frequency of the vibration signals for 4kN and 6kN load conditions are registered to be approximately 210Hz and 225Hz, respectively.

Non-synchronous peaks on the frequency spectrum are an indication of bearing wear. Monitoring the increase in amplitude of these frequencies indicates the rate of wear. As t wear increases, the frequency of these peaks develop

harmonics, indicative of impacting, see figure 7-34 and figure 7-35. The FFT spectrum represents the strong harmonics of the shaft speed while PeakVue and Enveloping represents the frequency of rubbing.

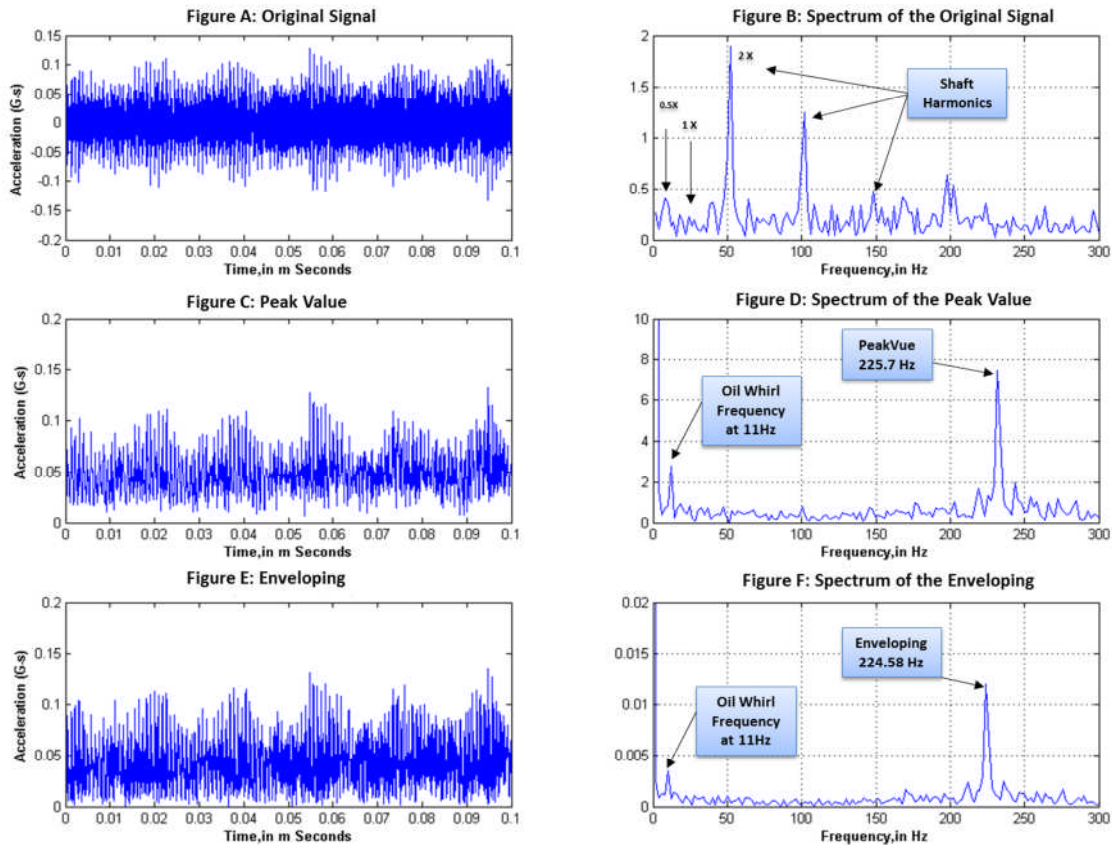


Figure 7-34 Comparison frequency spectrum at 6kN load condition with different techniques up to 300Hz

By increasing the radial load on the shaft, the frequency increased, to the high-frequency side of the spectrum. This shows that the system is hardening as a result of the stiffness increase produced by the contact of the shaft with the hydrodynamic bearing. The amplitude of the frequency also increases as a product of the higher-energy content of the rub-impacts; see figure 7-34 and figure 7-35. The observation between Enveloping and PeakVue analysis demonstrate that the dominant frequency of the vibration signals for 4kN and 6kN load conditions are registered to be approximately 210Hz and 225Hz, respectively at rubbing simulation.

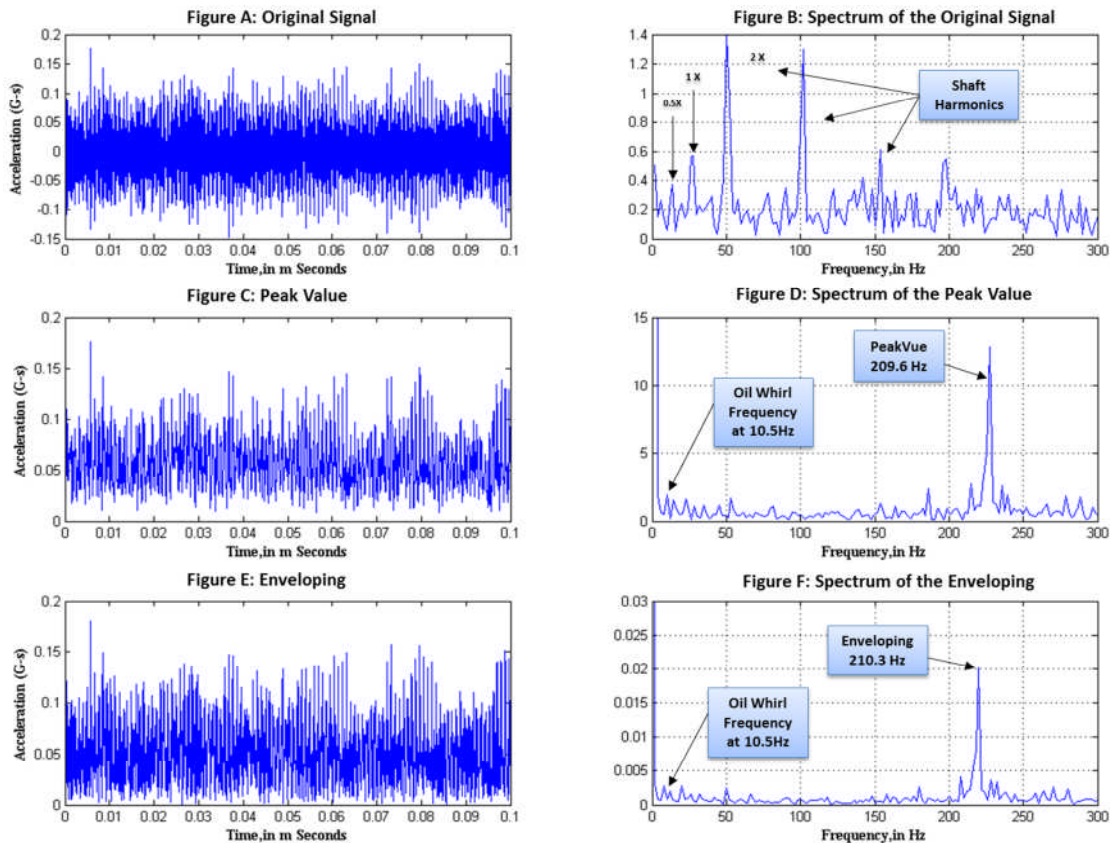


Figure 7-35 Comparison frequency spectrum at 4kN load condition with different techniques up to 300Hz

In another observation in figure 7-34 and figure 7-35, represents an instability caused by oil whirl. Oil whirl is caused by excessive clearance and radial load which results in the oil film building up and forcing the hydrodynamic bearing to migrate around the bearing at less than half the shaft rotational speed. Oil whirl is a condition in which strong vibration occurs at between 0.389X to 0.48X (9.725 Hz to 12Hz) It never appears at precisely 0.5X, but always a little lower. Oil whirl frequencies at 6kN and 4kN radial load conditions are 11Hz and 10.5Hz, respectively.

FFT spectrum was not capable of detecting these results while PeakVue and Enveloping demonstrate their advantage in detecting oil whirl frequency. Oil whirl is a serious condition in rotating machines and needs to be corrected. It can deteriorate and cause metal –metal contact in hydrodynamic bearings. This will Increase the radial load, decrease eccentricity, increase oil temperature and

reduce viscosity. We can correct oil whirl in hydrodynamic bearings. See appendix D.2 for further investigation on vibration signals.

The results delivered from the demodulation and PeakVue techniques were similar in the rubbing simulation at 1500 rpm with different load conditions. In fact, some defects on the hydrodynamic bearing at 1500rpm would not have been seen in a timely manner without the PeakVue data. Nevertheless, the amplitude obtained using PeakVue were higher than when using demodulation. In high rotational speed machines it is more advantageous to use these techniques rather than the traditional ones because the results obtained were not capable of detecting rub in the hydrodynamic bearing test rig. The PeakVue figures also proved to be a valuable tool in conjunction with routine vibration data to provide a more complete picture of the health of the rotating machinery. However, it also has been confirmed that AE was more effective in the early diagnosis of bearing faults.

In the most complex metal structure, see figure 7-36, if there is any excitation, it will move around structure. Sometimes if there are bolted joints, it is possible to achieve some damping but it is unlikely that complete isolation of the excitation in the research component will be realised. The most common source of excitation recognized in the hydrodynamic bearing test rig are; the universal coupling, the eccentricity of shaft, the eccentricity of pulleys, the toothed belt pitch variation (causing angular acceleration\ deceleration), the AC electronic motor, the ball bearings, etc. It is important to retain t for future research on this test rig every component needed to reproduce controlled noise and vibration conditions.

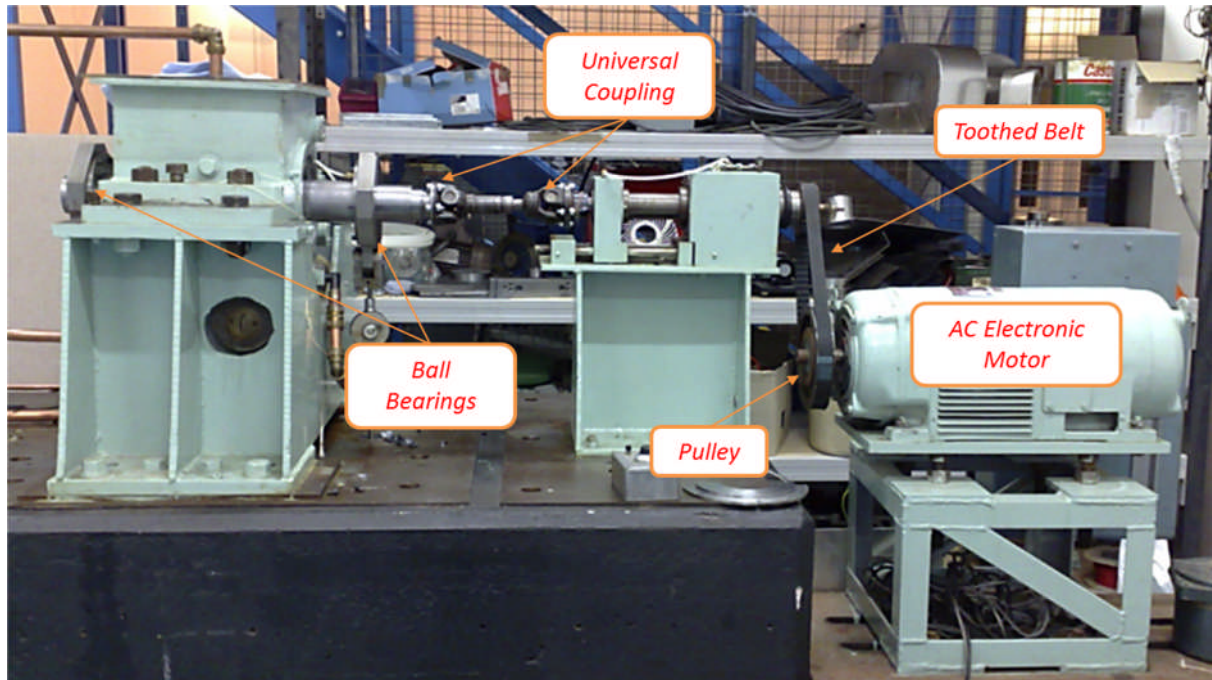


Figure 7-36 Test rig layout

7.3 Displacement Results

The installation of a pair of eddy current displacement sensors is shown in figure 7-37.

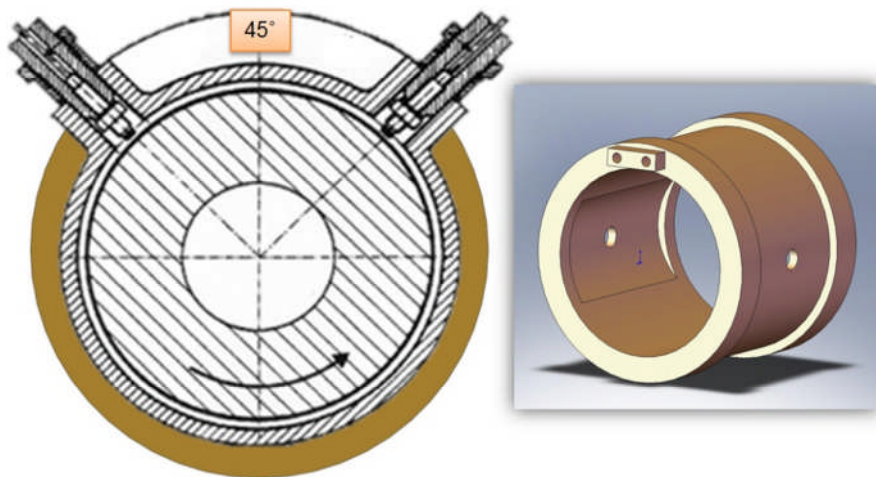


Figure 7-37: Typical eddy current displacement sensor installation

After mounting the eddy current sensors on the shaft, as the eddy current testing is essentially a near surface technique, the results of the speed test

were plotted as shown in figure 7-37 against operational time for channel 4. Over 150 displacement signals and waveforms have been captured continuously while the rotational speed test involved running the journal bearing at five rotational speeds of 750 rpm, 1500 rpm, 2500 rpm, 3500 rpm and 4500 rpm, with a constant radial load of 2kN, 6kN and 10kN, starting at 750 rpm, and increasing the rotational speed increments every 15 minutes to a maximum of 4500 rpm at one minute intervals.

The typical displacement signal with changing speeds (750 rpm to 4500 rpm) and constant load of 2kN at one minute intervals is presented in figure 7-38. This figure also presents the sample waveforms at each rotational speed level. These displacement waveforms show the relative changes in Peak-Peak displacement of the rotation shaft during running tests. As already demonstrated, the maximum change in shaft displacement is $2\mu\text{m}$ which is approximately equal to the theoretical minimum film thickness.

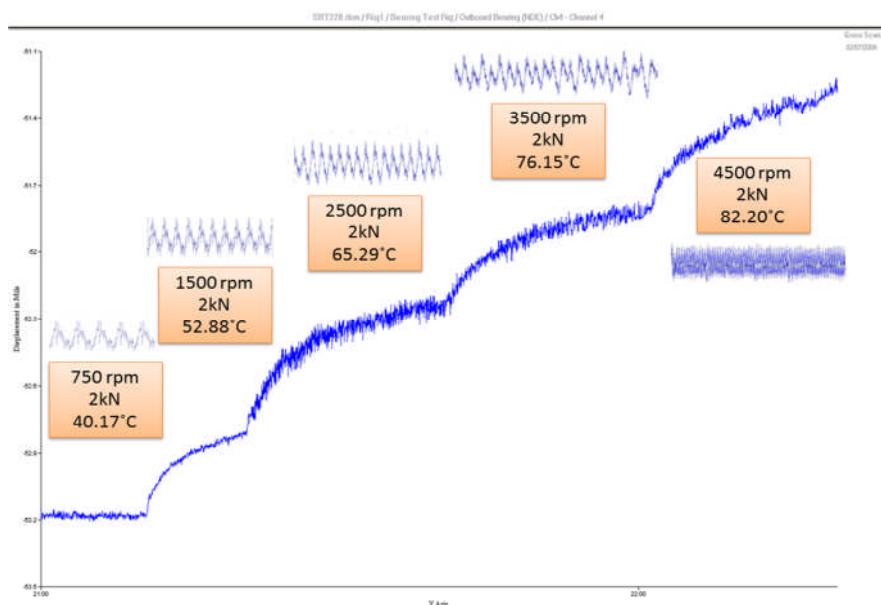
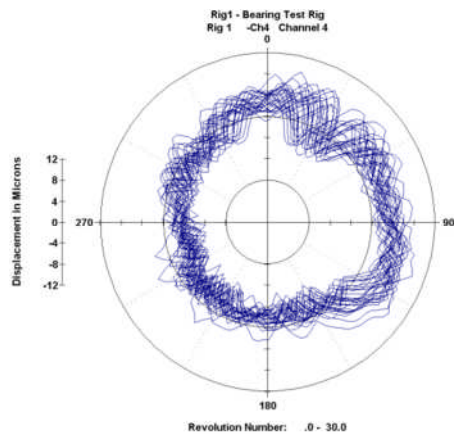


Figure 7-38 Typical displacement waveform at 2kN

Furthermore, it must be mentioned that one of the fundamental reasons for testing under very lightly loaded conditions (10kN) was to ensure that a minimum film thickness of the test bearings was much larger than the surface roughness of the bearing ($3\mu\text{m Ra}$). The next step was to set up an orbit plot.

This plot demonstrates, see figure 7-39, the actual location of the eddy current probes relative to the machine's geometry and is used as a reference to compare to the real machine's vertical orientation. In this case, it shows a prevalent arrangement with the probes at 225° and 135° on either side of the shaft plane.

All revolution at 4500 rpm and 2kN



one revolution at 4500 rpm and 2kN

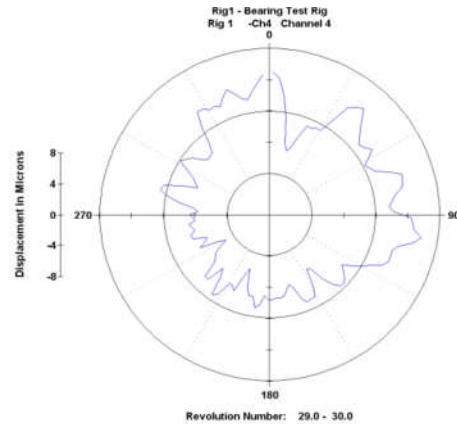


Figure 7-39 Typical Orbit graphs at 750 rpm

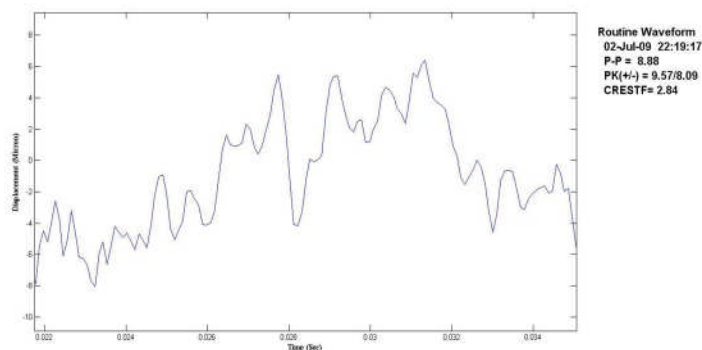


Figure 7-40 One revolution at 4500 rpm and 2kN

AMS Machinery Manager CSI 4500 can present live plots for shaft displacement simultaneously. Figure 7-39 shows the orbits plotted for visual comparison provided at 4500 rpm, 2kN and figure 7-40 presents one revolution in a normal plot. Each waveform, in order to obtain a better time and frequency resolution of displacement signals, was defined to record frequencies up to 4 kHz, see figure 7-41.

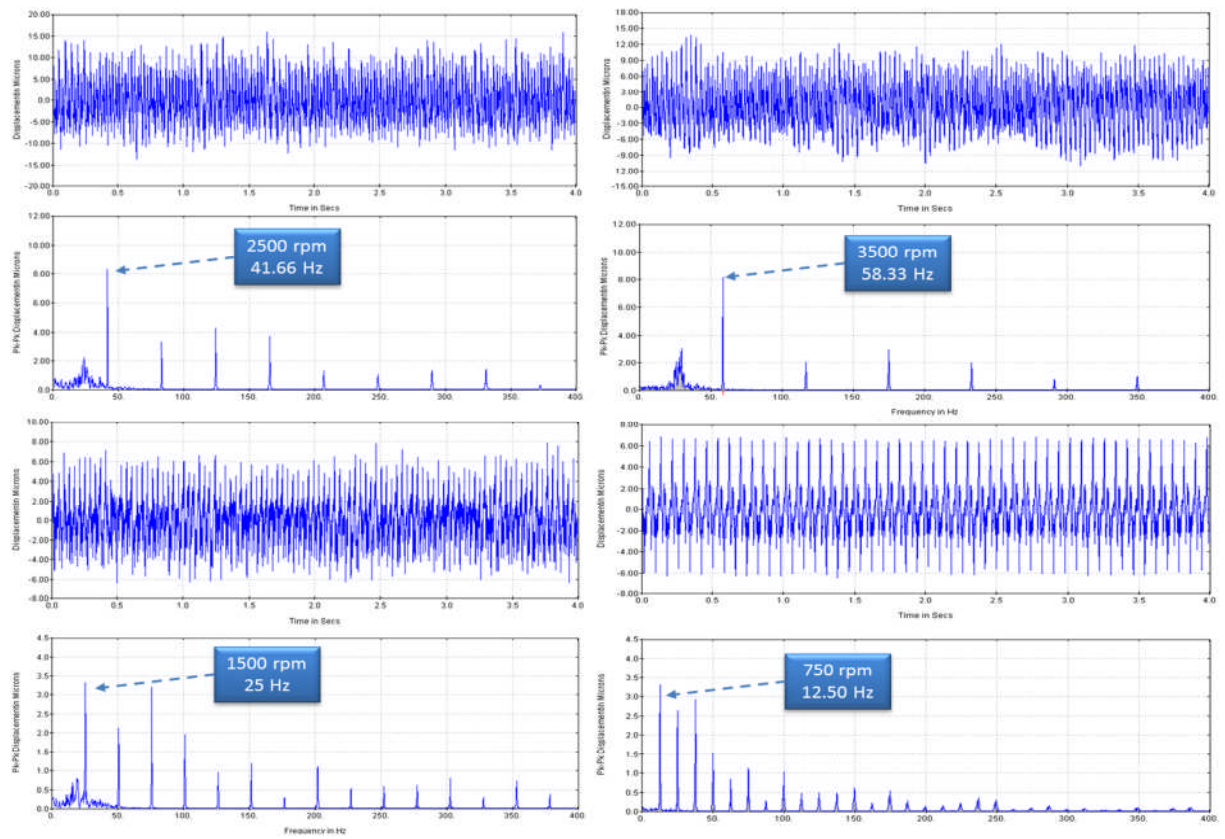


Figure 7-41 Displacement waveforms and their frequency domains at 2kN

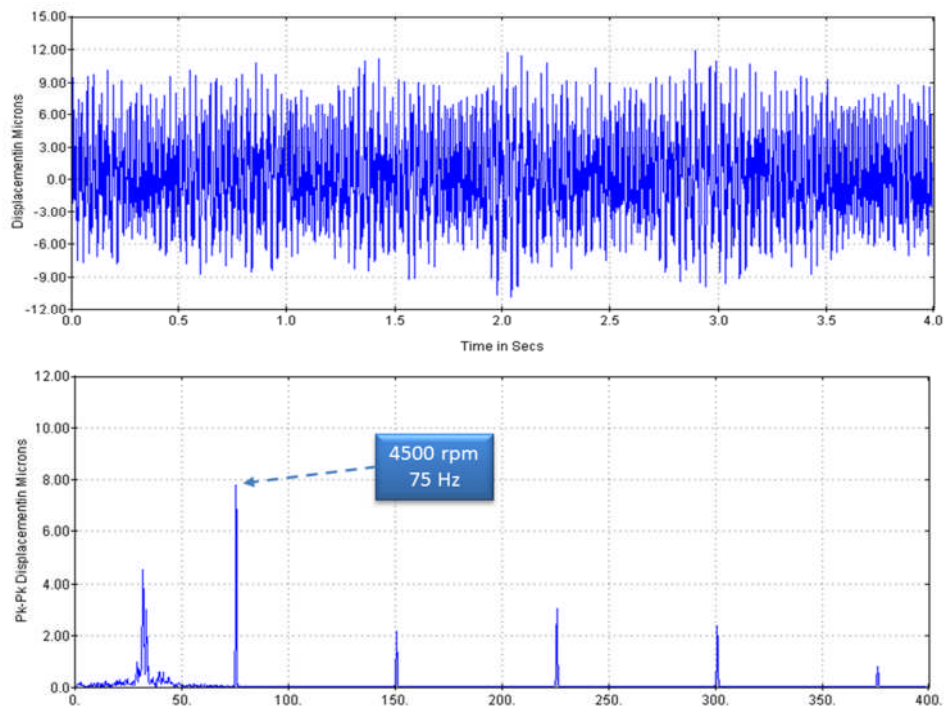


Figure 7-42 Displacement waveform and frequency domain at 4500rpm for 2kN load

150 sets of data for each defect at different conditions based on a 4 kHz sampling rate were captured and synchronously averaged. Each data set acquired was associated with a time window. Displacement P-P values for each test condition are illustrated in Figure 7-42. The displacement P-P level at the 4500 rpm condition was relatively higher than all other conditions for the reasons discussed earlier. In a typical displacement waveform with a frequency domain at 4500 rpm in figure 7-42, displacement waveform and the important features of frequency content and shaft frequency are clearly shown throughout the frequency spectrum. It is recommended by the author, in order to obtain further results and analysis, that one revolution is plotted at each level of speed or load located at 135° against the one at 225° on each side of the hydrodynamic bearing in order to show the eccentricity, relative shaft displacement and vibration direction and orientation. The normal plot to demonstrate the machine P-P displacement for channel 3 and channel 4 is shown in figure 7-43.

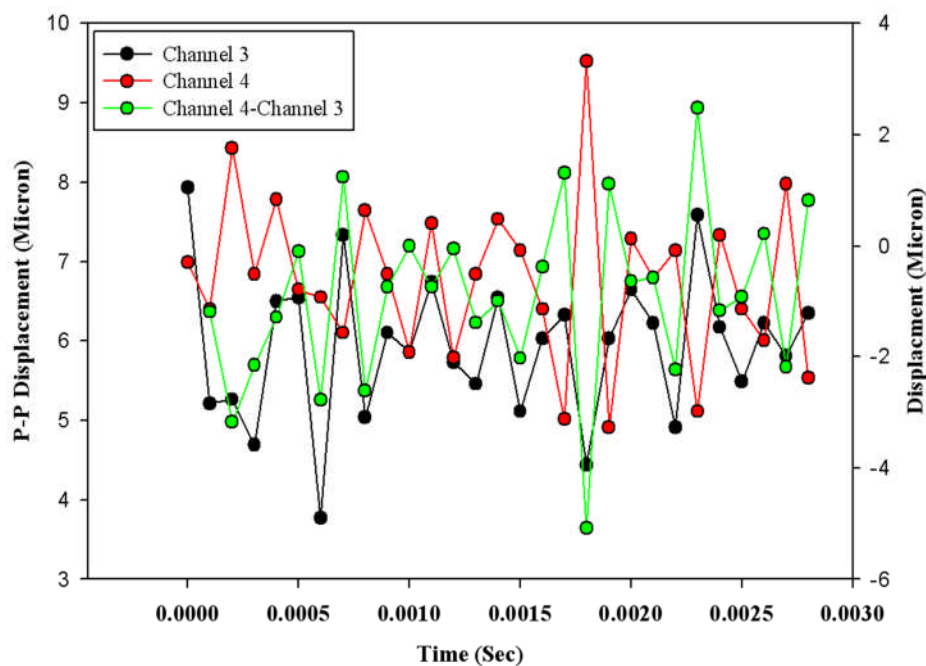


Figure 7-43 P-P Displacement for channel 3 and Channel 4 and $2\mu\text{m}$ change in shaft displacement at 4500 rpm and 10kN

As demonstrated in figure 7-43, the maximum change in shaft displacement is $2\mu\text{m}$ which is approximately equal to theoretical minimum film thickness.

8 Conclusions

In the past, machines were simply utilised until failure and then maintained. Nevertheless, run-to failure has become less cost-effective as machine sizes increase and their rotating components (e.g. shafts and bearings) become more costly. These factors influence operating and maintenance economics. Hence, this research investigated the qualities of AE and vibration within the operational parameters of a journal bearing to find a reliable diagnosis method that can be used in rotating applications. A series of tests was conducted through which various types of parameter were monitored. The outcomes are explained and the results, as presented before, will be summarised, in this section.

In order to have a complete understanding of the influence of operational variables on the characteristics of AE, an inclusive operating range of parameters, such as oil viscosities, rotational speeds and radial loads effects, has been demonstrated. The observations of AE RMS that are presented have revealed that in a properly maintained hydrodynamic lubrication regime the principal source of AE is friction in the shearing of the lubricant in addition to the splashing and motion of the oil within the bearings. It has also been shown that an increase in running speed generates higher AE activity in comparison to an increase in bearing loads. This is attributed to the power losses as a direct result of the shearing of the lubricant film. To date this is the first attempt at correlating AE with fluid friction and power losses in a hydrodynamic bearing.

A mathematical model was presented at the end, constituting a methodical basis for the control and diagnostics of AE activities within the operational parameters of hydrodynamic bearings on the Sutton test rig. AE RMS signal can be used in this model to predict, with reasonable accuracy (average plus or minus 5% for different load condition); the power loss bottom (P_{Btm}) for tribological systems without prior knowledge of the relative rotational speeds or load conditions, which hitherto has not been explored.

This experimental investigation conducted a comparative study between AE and Vibration to diagnose the rubbing at high rotational speeds in the hydrodynamic bearing. As it is the first known attempt in hydrodynamic bearings.

It was prudently proposed to assess the sensitivity of the AE measurements at the onset of contact between the shaft and the hydrodynamic bearing. Therefore, rubbing investigations for hydrodynamic bearing application with AE techniques for condition monitoring of high speed shafts, have been presented. It has been concluded, that AE parameters such as amplitude, can perform as a reliable and sensitive tool for the detection of rubbing between surfaces of a hydrodynamic bearing and high speed shaft.

The application of vibration (PeakVue) analysis was studied and compared with demodulation. As it is the first known attempt in a hydrodynamic bearing. The results observed from the demodulation and PeakVue techniques were similar in the rubbing simulation test. In fact, some defects on hydrodynamic bearings would not have been seen in a timely manner without the PeakVue data. Nevertheless, the amplitude obtained using PeakVue showed higher than when using demodulation. In high rotational speed machines it is more advantageous to use these techniques rather than the traditional ones when the results obtained were not capable of detecting rub in hydrodynamic bearing test rig. The PeakVue figures also proved to be a valuable tool in conjunction with routine vibration data to provide a more complete picture of the health of the rotating machinery.

Comparing AE and vibration as CMTs is always a tough task. According to the results of this research, the applicability of both AE and vibration technologies were studied for diagnosis of the stimulated rub on bearings, and this led to a high sensitivity when identifying the presence of hydrodynamic bearing defects. The results of time-frequency domain analysis could show a clear periodicity of AE activity associated with the rubbing of the shaft to the hydrodynamic bearing.

The AE showed better sensitivity toward early diagnosis and propagation of damage than vibration. The AE monitoring was capable of detecting incipient damage, far in advance of the vibration levels began to raise; it also has been confirmed that AE was more effective in the early diagnosis of bearing faults.

The main points are summarised as follows:

- i. The background noise level was low in comparison to AE test results.
- ii. The trend of AE, vibration and displacement are similar in all tests, which confirms the repeatability of the data.
- iii. AE provides a better alternative to the traditional vibration bearing condition monitoring schemes.

In addition, the application of advanced signal processing and statistical methods was established to extract useful diagnostic features from the acquired AE signals in both time and frequency domain. It was also concluded that the use of different signal processing methods is often necessary to achieve meaningful diagnostic information from the signals. The outcome contributed to the development of effective intelligent condition monitoring systems which can significantly reduce the cost of plant maintenance.

Finally, the use of AE parameters such as RMS has been validated as an online condition monitoring technique (CMT) for understanding the behaviour of hydrodynamic bearings in different situations. It was evident through this experimental research that the AE technology (AET) has a great working scope in the application of hydrodynamic bearings.

8.1 Recommendations for Future Work

There is scope for further of monitoring hydrodynamic bearings with Acoustic Emission and vibration analysis in view of the field requirements for the smooth operation of critical machinery and equipment. Several aspects have been identified, whose effectiveness, both technically and economically, has to be proved with further investigations. The approach of the current research work can be extended further to consider the following:

- ❖ This research has defined a new method for detecting failure in hydrodynamic bearings. The possibility of undertaking further tests with range of different materials, shafts etc., is essential, and it will be dependent on the adoption of this method by industry, and here lies the basic premise of the research.
- ❖ The author sees the need to performing further tests focusing on shaft diagnosis, different shaft specimens with cracks at different locations and depths to find the consequence on acoustic emission and vibration signals from hydrodynamic bearings.
- ❖ The complex influence of vibration and noise interference on the hydrodynamic bearing from the grease lubrication ball bearings were mentioned in Chapter 4.1.4 of this thesis. As an investigation area for future work this would include the determination of the vibration properties of the lubrication grease in grease film formation and the influence of the damping properties of the grease on the vibration of the bearing components.
- ❖ The natural frequencies of different components in the hydrodynamic test rig such as; electronic motor, belt, coupling, ball bearing, bearing housing, etc, need to be measured. This will provide information on the approximate frequency range we should be looking for analysing hydrodynamic bearings with vibration and acoustic emission.
- ❖ To verify metal-to-metal contact, wear and damage were occurring during rubbing simulation, an oil wear debris analysis needs to carry out on the oil taken out of the hydrodynamic bearing at different times.
- ❖ It has been shown that an increase in running speed generates higher AE activity in comparison to an increase in bearing load. Further research is needed to verify the results of rubbing condition with acoustic emission and PeakVue at different rotational speeds.

- ❖ This study indicated the necessity to perform further advanced signal processing for AE hydrodynamic analysis using Spectral Kurtogram on the shaft and bearing to assess the effectiveness of Spectral Kurtogram methods in detecting defects for rotating machines. This will provide a reasonable opportunity to compare the effectiveness of both vibration and AE technologies.
- ❖ Performing simulation and focusing on developing a finite element model by using FEM softwares, to simulate the dynamic properties of operational variables over a much broader operating range. These simulations are likely to offer unique opportunities to study more deeply the correlation between the operational variables and the different signal features.

REFERENCES

- 1) Al-Ghamdi. A, Cole. P, Such. R, 2004, "Estimation of Bearing Defect Size with Acoustic Emission," *Insight: Non-Destructive Testing and Condition Monitoring*, 46(12) pp. 758-761.
- 2) Albers, A. et al. 2006. Monitoring Lubrication Regimes in Sliding Bearings: Using Acoustic Emission Analysis.[Online] Available at: http://www.ipek.uni-karlsruhe.de/medien/veroeffentlichungen/060515_Monitoring_Lubrication_Regimes_in_Sliding_Bearings.pdf
- 3) Miller R.K, McIntire P. *Non-destructive Testing Handbook*. American Society for Non-destructive Testing, USA, 1987.
- 4) Mba, D, Rao, R.B.K.N., 2006, "Development of Acoustic Emission Technology for Condition Monitoring and Diagnosis of Rotating Machines; Bearings, Pumps, Gearboxes, Engines and Rotating Structure," *The Shock and Vibration Digest*, 38/1, pp. 3-16.
- 5) Sato. I, (1990) "Rotating Machinery Diagnosis with Acoustic Emission Techniques", *Electrical Engineering in Japan*, Vol. 110, No. 2. Pp. 145-152
- 6) Leahy. M, Mba. D, Cooper. P, Montgomery. A, Owen. D, An experimental investigation into the capabilities of Acoustic Emission for the detection of shaft-to-seal rubbing in large power generation turbines, *Journal of Engineering Tribology, Proc. Instn Mech. Engrs, Part J*. 220(J7), PP 607 – 615, 2006.
- 7) Leahy. M, Mba. D, Detecting shaft-to-seal rubbing in power generation turbines with the Acoustic Emission technology, *Journal of Vibration and Acoustics, ASME*, Volume 128, Issue 6, pp. 798-800, 2006.
- 8) Mba, D, Cooke, A, Roby, D, Hewitt, G, (2004) "Detection of shaft-seal rubbing in large scale power generation turbines with Acoustic Emissions; Case study", *Journal of Power and Energy - Part A, I Mech E*, Vol. 218, No. 2, pp. 71-82.

- 9) Hall L. D, Mba, D, Acoustic emissions diagnosis of rotor-stator rubs using the KS statistic, Mechanical Systems and Signal Processing, Volume 18, Issue 4, pp. 849-868, 2004.
- 10) Wang. Q, Chu. F, Experimental determination of the rubbing location by means of acoustic emission and wavelet transform, Journal of Sound and Vibration, vol. 248, no. 1, pp. 91-103, 2001
- 11) Leahy. M, Mba. D, Cooper. P, Montgomery. A, Owen. D, Acoustic Emission technology for detection of shaft-to-seal rubbing on power generation turbines; a qualitative verification., INSIGHT, Vol. 48, no. 12, pp. 754-755, Dec 2006.
- 12) Al Shaikh Mubarak. M, Mba. D, Cooper. P, Opportunities offered by acoustic emission for seal and blade rub detection on large scale power generation turbines. 16th International congress on Condition monitoring and Diagnostic engineering management (COMADEM'2003) Vaxjo, Sweden, August 27th – 29th2003, pp. 513, ISBN 91-7636-376-7.
- 13) Someya. T, Mitsui. J, Esaki. J, Journal-bearing data book ,Springer-Verlag, 1989. ISBN: 978-3540170747. Springer-Verlag Berlin and Heidelberg GmbH & Co. K (October 1989).
- 14) Prieto, S. 2006. A Bearing Test-Rig for Extreme Operating Conditions. MSc Thesis Cranfield University, School of Mechanical Engineering. <http://dspace.lib.cranfield.ac.uk:8080/bitstream/1826/1762/1/Prieto-2006.pdf>
- 15) Kingsbury, Inc. n.d. A General Guide to the Principles, Operations, and Trouble Shooting of Hydrodynamic Bearings. http://www.kingsbury.com/pdfs/universe_brochure.pdf
- 16) Neale Consulting Engineers, n.d. Plain Bearing Damage access on: http://www.tribology.co.uk/services/investigate/p_bearing.htm

- 17) Bearings 101 2007. Bearings and Lubricants; access on:
http://www.bearings.machinedesign.com/guiEdits/Content/BDE_6_1/bdemech6_21.aspx

- 18) eFunda, Engineering Fundamental. Properties of Steel. access on
http://www.efunda.com/Materials/alloys/alloy_home/steels_properties.cfm

- 19) Francisco Valero, "Diseño de Máquinas" class notes. Universidad Politecnica Valencia (Spain). 2008

- 20) ABMA. American Bearing Manufacturers Association. "Bearing Industry Timeline" access on: <http://www.abma-dc.org/BearingIndustry/BearingTimeline/tabid/84/Default.aspx>

- 21) Beardwood. M.R "An Expert System for Diagnosing Failure Mechanisms in Journal Bearings" PhD thesis. School of Mechanical Engineering. Cranfield University. UK 1985

- 22) Bhaumik, S.K, Upadhyaya. G. S, Vaidya. M. L 2002; 168 American Society for metals.

- 23) Shigley, Edward. J, Brown. C.R, Thomas. H (2004), Standard handbook of machine design (3rd ed.), McGraw-Hill Professional, pp. 640, ISBN 9780071441643.

- 24) Berndt. F, Bennekom van. A, Pump shaft failure—a compendium of case studies. Eng. Fail Anal. 8 2 (2001), pp. 135–144;

- 25) ESDU 8403, Calculation methods for steadily loaded axial groove hydrodynamic journal bearings; 1Endorsed by The Institution of Mechanical Engineers Calculation methods for steadily loaded axial groove hydrodynamic journal bearings Issued November 1984 With Amendment A September 1991 Re-issued November 2008

- 26) The Engineering Toolbox "Dynamic, Absolute and Kinematic Viscosity"
access on http://www.engineeringtoolbox.com/dynamic-absolute-kinematicviscosity_d_412.html
- 27) Math. Z, A. Sommerfeld, "Zur Hdrodynamischen Theorie der Schmiermittel-Reibung". Physik, vol. 50, 1904.
- 28) Griffin. W. S, Richardson. H. H, Yamanami .S, ``A Study of Fluid Squeeze-Film Damping," Journal of Basic Engineering, Trans. ASME. 1966.
- 29) Richter. J.R, "Acoustic Emission Testing: A Composite Manufacturer's Experience", by the American Society for Nondestructive Testing, 1999
<https://www.asnt.org/publications/materialseval/basics/may99basics/may99basics.htm>
- 30) Henderson. Vaneshette T, "Acoustic Emission", 2003 access on;
http://msewww.engin.umich.edu:81/people/smy/mse465_03/HENDERSON/Acoustic%20Emission2.htm
- 31) Envirocoustics s.a.com, advanced acoustic emission data analysis pattern recognition and neural network software; access on;
<http://www.envirocoustics.gr/pdf/Noesis%20Applications%205.0.pdf>
- 32) Hall.L. D, Mba. D, Bannister. R.H, "Acoustic Emission Signal Classification in Condition Monitoring", School of Mechanical Engineering, Cranfield University, Bedfordshire, 2002
<https://dspace.lib.cranfield.ac.uk/bitstream/1826/1092/1/Acoustic+Emission+Signal+Classification.pdf>.
- 33) Rao. B.P.C, "An Introduction to Acoustic Emission" access on;
<http://www.geocities.com/raobpc/index.html>
- 34) Yoon. D.J, Kwon. O.Y, Chung. M.H, Kim. K.W, "Early detection of damages in journal bearings by acoustic emission monitoring" International Acoustic Emission Symposium No12, Sapporo , Japan, vol. 13, no 1-2, pp. S1-S59 (16 ref.), pp. 1-10, 1995

- 35) Zykova. L, Mazal. P, Pazdera. L, "Identification of Contact Fatigue Stages with Acoustic Emission Method" Brno University of Technology, Brno, Czech Republic, ECNDT Poster 4, 2006 access on; <http://www.ndt.net/article/ecndt2006/doc/P4.pdf>
- 36) Al-Ghamdi. A.M, Zhechkov. Mba. D, "The use of Acoustic Emission for bearing defect identification and estimation of defect size" access on; <http://www.ndt.net/article/ewgae2004/html/htmltxt/l38al-ghamdi.htm>
- 37) Kawamoto. S, Sam William. R, "Acoustic Emission and Acousto-Ultrasonic Techniques for Wood and Wood-Based Composites" A Review by United States Department of Agriculture Forest Service, Technical Report FPL-GTR-134
- 38) Mba. D, Raj Rao B. K. N., " Development of Acoustic Emission Technology for Condition Monitoring and Diagnosis of Rotating Machines: Bearings, Pumps, Gearboxes, Engines, and Rotating Structures", The Shock and Vibration Digest, Vol. 38, No. 1, January 2006 pp. 3-16, SAGE Publications, 2006
- 39) Hall. L.D, Mba. D, "Diagnosis of continuous rotor-stator rubbing in large scale turbine units using acoustic emissions" School of Mechanical Engineering, Cranfield University, 2003
- 40) James Li. C, Li. S.Y., "Acoustic Emission Analysis For Bearing Condition Monitoring", Dept. of Mechanical Engineering, Aeronautical Engineering and Mechanics, 2000
- 41) Sturm. R A, Uhlemann. D. S, "Diagnosis of plain bearings by acoustic emission analysis" Department of Power Station Equipment and Energy Conversion, Faculty Maintenance, University of Engineering, Zittau, GDR, Measurement Vol 3 No 4, Oct-Dec 1985

- 42) Baranov. V.M, Kudryavtsev. E.M Sarychev. G.A, "Modelling of the parameters of acoustic emission under sliding friction of solids" Wear 2002 (1997~ 125-133), 1996
- 43) Lingard. S. K, "An Investigation Of Acoustic Emission In Sliding Friction And Wear Of Metals" Wear, 130 (1989) pp. 361 - 379 367, 1988
- 44) Choudhury. A. Tandon. N. (2004), "Application of acoustic emission technique for the detection of defects in rolling element bearings", Tribology International, Vol. 33, pp. 39-45.
- 45) Yoshioka, T. Fujiwara, T. (1982), "New acoustic emission source locating system for the study of rolling contact fatigue", Wear, Vol. 81 No. 1, pp. 183-186.
- 46) Hawman. M.H. Galinaitis. W.S. In: 1988 Ultrasonic Symp (1988), pp. 885–893.
- 47) Igarashi. A, Noda. B, Matsushima E. Jap. J. Soc. Lub. Engrs 24-2 (1979), pp. 122–129 (in Japanese)
- 48) Rogers. L.M. (1979), "The application of vibration analysis and acoustic emission source location to on-line condition monitoring of anti-friction bearings", Tribology International, Vol. 12 No. 2, pp. 51-59.
- 49) Holroyd. T. Condition monitoring of very slowly rotating machinery using AE techniques. 14th International congress on Condition monitoring and Diagnostic engineering management (COMADEM'2001), Manchester, UK, 4-6 September 2001, 29, ISBN 0080440363
- 50) Holroyd. T.J. & Randall. N. (1993). Field Application of Acoustic Emission to Machinery Condition Monitoring. Proceedings of COMADEM 1993. University of West of England, Bristol. pp. 217-222. 1 – 871056 – 72 – 1
- 51) Bagnoli. S, Capitani. R. Citti. P. Comparison of accelerometer and acoustic emission signals as diagnostic tools in assessing bearing. Proceedings of

2nd International Conference on Condition Monitoring, London, UK, May 1988, 117-125.

- 52) Tandon. N. Nakra. B.C, Defect Detection of Rolling Element Bearings by Acoustic Emission Method, Journal of Acoustic Emission, 1990; 9(1) 25-28
- 53) Huguet. S, Godin. N, Gaertner. R, Salmon. L, Villard. D. Use of acoustic emission to identify damage modes in glass fibre reinforced polyester. Composites Science and Technology, 2002, 62, pp. 1433-1444.
- 54) Kakishima. H, Nagatomo. T, Ikeda. H, Yoshioka. T, Korenaga. A, Measurement of Acoustic Emission and vibration of rolling bearings with an artificial defect, QR of RTRI, Vol 41, No. 3, pp. 127-130, 2000.
- 55) Morhain. A, Mba. D, (2002). Defect Diagnosis of Lightly Loaded Bearings with Acoustic Emissions. Proceedings of COMADEM 2002. COMADEM International, Birmingham. 492-502. ISBN 0 – 9541307 – 0 – 7.
- 56) Kaewkongka. T, Au. Y. H. J, application of Acoustic Emission to condition monitoring of rolling element bearings. Measurement and Control, Vol 34, no. 8, pp. 254-247, 2001
- 57) Mba. D Acoustic Emissions and monitoring bearing health Tribology Transactions, 46 (3), pp. 447-451, 2003; access on;
<https://dspace.lib.cranfield.ac.uk/bitstream/1826/1781/1/Acoustic%20emissions%20and%20monitoring%20bearing%20health.pdf>
- 58) Shepard. W. S, Kenneth A. Cunefare, The influence of substructure modelling on the predicted vibroacoustic response of a fluid-loaded plate system J. Acoust. Soc. Am. Volume 107, Issue 5, pp. 2884-2884 (2000)
- 59) Schlieper. G, Huppmann. W. J, Kozuch. A, "Nondestructive Determination of Sectional Densities by the Gamma Densomat", Progress in Powder Metallurgy, Vol. 43, 1987, pp. 351

- 60) Schwalbe, H.J, Schallemissionsanalyse an Gleitlagern zur Schadensfrüherkennung, Fachtagung Instandhaltung, Wiesbaden (1978).
- 61) Brdicka et al, Brdicka. M, Samek. L, Taraba, Kavitace. O, SNTL, Praha (1981).
- 62) Eisenblätter. J, Fanninger. G, Zur Anwendung der Schallemissionsanalyse Forschung und Technik, Metall 1 and 2 (31) (1977).
- 63) W. Gnille, Papendick. G, 1981, Theorie der Mischreibung und ihre Anwendung auf Gleitlager, Freiburger Forschungsheft A 648, VEB Deutscher Verlag für Grundstoffindustrie (1981).
- 64) Hippmann. N, Uhlemann. S, (1985) Beispiele zur Anwendung der Schallemissionsanalyse für die Zustandsbestimmung an einer Hauptumwälzpumpe, Energietechnik 8 (35), pp. 291–295.
- 65) Musa. K. J, Lemaster .R, Frank. C, Beal Pages 83-89 Study of Acoustic Emission Generation in Sliding Motion ; J. of Acoustic Emission, Volumes 1 - 27, 1982 – 2000
- 66) Leahy. M, Mba. D, Cooper. P, Montgomery. A, Owen. D. 2006. Acoustic emission for the detection of shaft-to-seal rubbing in large power generation turbines. Advanced Materials Research, Vol. 13-14, pp. 433-438.
- 67) Leahy. M, Mba. D, Cooper. P, Montgomery. A, Owen. D. 2006b. Detecting shaft-to-seal rubbing in power generation turbines with the acoustic emission technology. Journal of Vibration and Acoustics, Vol. 128, pp. 798-800.
- 68) Mba, D, Cooke. A, Roby. D, Hewitt. G. 2004. Detection of shaft-seal rubbing in large-scale power generation turbines with acoustic emissions; case study. Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy, Vol. 218, No. 2, pp. 71-81.

- 69) Board. C.B, Stress wave analysis of turbine engine faults, Aerospace Conference, IEEE Proceedings (Cat. No.00TH8484) 6 (2000) 79–93.
- 70) Prieto, S. 2006. A Bearing Test-Rig for Extreme Operating Conditions. MSc Thesis Cranfield University, School of Mechanical Engineering.
- 71) Mukhtar, E. A.; Acoustic emission of journal bearings / (M. Sc.) - Cranfield Institute of Technology, School of Mechanical Engineering, 1986.
- 72) ASTM E 976, Standard Guide for Determining the Reproducibility of Acoustic Emission Sensor Response, American Society for Testing and Materials, West Conshohocken, USA. 1995.
- 73) Grosse. U. C, Hsu-Nielsen Source, Non-destructive Testing Encyclopaedia (NDT.net) Publications, Acoustic Emission (AE), Site of Non-destructive Testing Encyclopaedia (NDT.net), access on; <http://www.ndt.net/ndtaz/ndtaz.php>. 2007.
- 74) Chen. X, Zeng. H, Dietmar. W. In-Process Tool Monitoring through Acoustic Emission Sensing, Publications of Automated Material Processing Group, Automation Technology Division. 2001.
- 75) Scruby. C.B, An Introduction to Acoustic-Emission. Journal of Physics Scientific Instruments 1987; 20: 947-953.
- 76) Dyer. D, Stewart. R.M, Detection of rolling element bearing damage by statistical vibration analysis. Trans. ASME, J. Mech. Design 100 2 (1978), pp. 229–23
- 77) Rush A.A. Kurtosis — a crystal ball for maintenance engineers. Iron Steel Int February 1979:pp. 23–7.
- 78) Weaver. D. K, Shuman .D, Mankin. R. W. Optimizing and assessing the performance of an algorithm that cross-correlates acquired acoustic emissions from internally feeding larvae to count infested wheat kernels in

grain samples Original Research Article Applied Acoustics, Volume 50, Issue 4, April 1997, Pages 297-308

- 79) Tandon, N., and Choudhury, A., 1999, "Review of Vibration and Acoustic Measurement Methods for the Detection of Defects in Rolling Element Bearings," Tribology International, 32(8) pp. 469-480.
- 80) Martin. R.J, Autoregressive and Cepstrum-domain filtering, Signal Processing Journal 76 (1999), pp. 93–97
- 81) Meng. Q, Qu .L, "Rotating machinery fault diagnosis using Wigner distribution," Mechanical Systems and Signal Processing, Vol.5, pp.155-166, 1991.
- 82) Cour-Harbo. La, Aalborg .A, University, Department of Control Engineering, Fredrik Bajers Vej 7C, 9220 Aalborg East, Denmark The symmetric Rudin-Shapiro transform - An easy, stable, and fast construction of multiple orthogonal spread spectrum signals 1973- c2001
- 83) Mba. D, Cooke. A, Roby. D, Hewitt. G; Detection of shaft-seal rubbing in large-scale power generation turbines with Acoustic Emissions; Case study; Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy. Vol. 218, No. 2, 2004, pp 71-81.
- 84) Dornfeld. D, Application of acoustic emission techniques in manufacturing. NDT & E International 25 (1999), pp. 259–269.
- 85) Smith, S., 2002, "Digital Signal Processing: A Practical Guide for Engineers and Scientists," Elsevier Newnes, pp. 1-672.
- 86) Rafiee J. M.A, Prause N.M.P, Schoen , a Department of Mechanical, Aerospace and Nuclear Engineering, Rensselaer Polytechnic Institute, Troy, NY 12180, USA" Wavelet basis functions in biomedical signal processing. http://rafiee.us/files/ESWA_2010_biosignal.pdf

- 87) Benbouzid. M.E.H, 2000. A Review of Induction Signature Analysis as a Medium for Faults Detection.
- 88) Dominick.J. PeakVue as Part of Reliability Based Maintenance Program.
access on;
<http://www.compsys.com/drknow/aplpapr.nsf/19fcc759cbee3c3c8525659800528caa/0c644b02e0d67926852565a200634f3b>
- 89) Journal Bearing <http://www.answers.com/topic/journal-bearing-2>
- 90) James. C, Robinson. J.C, Walker.J.W, Berry.J.E. 2001. Description of peakvue and illustration of its wide Array of applications in fault detection and Problem severity assessment
- 91) Banjevic, Dragan, Jardine, Andrew K. S. 2008. Condition monitoring. Encyclopedia of quantitative risk analysis and assessment. John Wiley & Sons.
- 92) Bannister, R. H, Findlay, G. E, Condition monitoring of low-speed rotating machinery using stress waves,part 1. Proceedings of the Institution of Mechanical Engineers, part E: Journal of Process MechanicalEngineering, Vol. 213, No. 3, pp. 153-170.
- 93) Dai. J, Chen. C. L, Xiao-Yan. Xu et al. 2008. Condition monitoring on complex machinery for predictive maintenance and process control. IEEE International Conference on Systems, Man and Cybernetics.
- 94) De Castro, H.F, Cavalca, Lucchesi. K, Rainer. N. 2008. Whirl and whip instabilities in rotor-bearing system considering a nonlinear force model. Journal of Sound and Vibration, Vol. 317, Issues 1-2, pp. 273-293.
- 95) Girdhar, Paresh , Scheffer, Cornelius 2004. Practical machinery vibration analysis and predictive maintenance Elsevier.
- 96) Goldman.S, 1999 Vibration spectrum analysis; a practical approach. Industrial Press Inc.

- 97) Pratesh. J, Wadhwani. A. K, Mulchandani. K. B, 2008. Machine fault signature analysis.
- 98) International Journal of Rotating Machinery, Vol. 2008, Article ID 583982.
- 99) Kuboyama. K, 1997. Development of low speed bearing diagnosis technique. Fukuyama City, Japan.
- 100) Lee. Jay, 1999. Computer-aided maintenance: methodologies and practices. Springer.
- 101) Maxwell. J. H. 1988. Detecting cracked reactor coolant pump shafts with vibration monitoring. Paper of the American Society of Mechanical Engineers, 88-PVP-II.
- 102) Moshage. R, Lyndell. B, Clay. B. et al, 1993. Vibration monitoring for predictive maintenance in central energy plants. Online Information for the Defense Community, accession no. ADA273548.
- 103) Parno. R, Al-Thobiani. F, Gu. Fengshou, Andrew. Ball, 2010. Early failure detection and diagnostics of high speed self aligning journal bearing. COMADEM, June 28-July 2, Nara, Japan.
- 104) diagnostics of high speed self aligning journal bearing. COMADEM, June 28-July 2, Nara, Japan.
- 105) Rajendran, V. 2009. Engineering Physics. Tata McGraw-Hill Education.
- 106) Randall. R.B, 2011. Vibration-based condition monitoring: Industrial, aerospace and automotive applications. John Wiley and Sons.
- 107) Sabnavis, G, Kirk. R. G, Kasarda. M, 2004. Cracked shaft detection and diagnostics: a literature review. Shock and Vibration Digest, Vol. 36, No. 4.
- 108) Williams, R.V., c1980, "Acoustic emission," Hilger, Bristol, pp. 118
- 109) SKF. 2000. Vibration diagnostic guide. SKF Condition Monitoring.

- 110) Taylor. I, Scatchard. D 2008. More monitoring – less downtime. World pumps, Vol. 2008, Issues 507, pp. 36-37.
- 111) Ugural. A. C, 2003. Mechanical design: an integrated approach. McGraw-Hill Professional.
- 112) Wilson, J. S. 2005. Sensor technology handbook. Vol. 1. Newnes.
- 113) Wowk. V, 1991. Machinery vibration: measurement and analysis. McGraw-Hill Professional.
- 114) Estupinan. E, Saavedra. P. N.d. Diagnostic techniques for the vibration analysis of bearings. access on; http://by.genie.uottawa.ca/~liang/Facilities_files/Equipment/IMI%20bearing%20fault%20detector/PeakVu%20and%20other%20IMI%20documents/Peakvue%20compare%20with%20HFR.pdf.
- 115) Girdhar. P, Scheffer. C, 2004. Practical machinery vibration analysis and predictive maintenance. Elsevier.
- 116) Goly. K. 1988. Using PeakVue to detect machinery faults. CSI Reliability Week, Melbourne, Australia, 18-20 March 1988. Siemens. access on ;http://www.sea.siemens.com/us/internet-dms/Internet/IndustrialServicesComm/General/Docs/Detect_Machinery_Faults_by_Using_Peak_Vue.pdf
- 117) Glenn. H, Truempi. C 2009. Installing predictive vibration monitors. Power Engineering, November 2009 edition.
- 118) Nandi, S, Toliyat H. A 1999. Condition monitoring and fault diagnosis of electrical machine—A review. Conference Record, IEEE IAS Annual Meeting, Vol. 1, pp.197.
- 119) Novaspect. 2009. Machine condition monitoring: PeakVue. Technical Data Sheet, Nova-PeakVue.: www.novaspect.com

- 120) Robinson. J.C, 2001. Detection and Severity Assessment of Faults in Gearboxes from Stress Wave Capture and Analysis. Society for Machinery Failure Prevention Technology (MFPT), 2001 Annual Conference.
- 121) Robinson. J.C , Piety, K. R. 1996. New methodology for bearing fault defect: PeakVue analysis access on;
<http://www.compsys.com/drknow/aplpapr.nsf/06b6f5a4de2eae6285256a3f004d9758/d5fbeeef87af72a485256620007144db>
- 122) Robinson. J.C, Berry. J.E. 2001. Description of peakvue and illustration of its wide array of applications in fault detection and problem severity assessment. Emerson Process Management Reliability Conference, 22-25 October 2001.
- 123) Robinson. J.C, Illig. M, Brown. T, Limbureg. R. N.d. The model 682A05 bearing fault detector: A new approach for predicting catastrophic machine;- failure.
http://www.pcb.com/techsupport/docs/imi/TN-14_BFD_682A05_White_Paper.pdf
- 124) Tranter, J, 2012. Detecting bearing faults. Reality access on;
http://reliabilityweb.com/index.php/print/detecting_bearing_faults_part_2
- 125) Xu. Y.B, Han. D.F, Zhu. S.A, 2010. Bearing fault detection based on Peakvue technology. Transducer and Microsystem Technologies, 2010-07. access on;
http://en.cnki.com.cn/Article_en/CJFDTOTAL-CGQJ201007040.htm
- 126) Petri.H Non-Linear Journal Bearing Model for Analysis of Superharmonic Vibrations of Rotor Systems.200 :<https://oa.doria.fi/handle/10024/42673>
- 127) Robinson. J, Illig .M, Brown. T, Limburg .R. The model 682a05 bearing fault detector. A New Approach for Predicting Catastrophic Machine Failure.
- 128) Robinson. J.C, Berry. J.E, 2001. Description of peakvue and illustration of its wide Array of applications in fault detection and Problem severity assessment.

- 129) Robinson. J.C, Piety. K.R, 1996. New Methodology for Bearing Fault Defect: PeakVue access on;
<http://www.compsys.com/drknow/aplpapr.nsf/06b6f5a4de2eae6285256a3f004d9758/d5fbeeef87af72a485256620007144db>
- 130) Shiroshi. J, Liang. Y.L.S, Danyluk. S, Kurfess. T, Vibration Analysis for Bearing Outer Race Condition Diagnostics. (online).
- 131) Badi, M. N. M, Engin, S. N., and Schonfeld, D., 1996, "Fault classification of a model driveline using time domain data," COMADEM 1996, University of Sheffield- UK, pp. 43-50.
- 132) What is the definition of journal bearing? " Bearing Applications 210" 560100-560130 <http://www.toolingu.com/definition-560210-24475-journal-bearing.html>
- 133) Nandi. S, Toliyat. H. A, "Condition monitoring and fault diagnosis of electrical machine—A review", Conf. Rec. IEEE IAS Annu. Meeting, vol. 1, pp.197 , 1999.
- 134) Robinson. J.C, "Detection and Severity Assessment of Faults in Gearboxes from Stress Wave Capture and Analysis", Society for Machinery Failure Prevention Technology (MFPT) 2001 Annual Conference.
- 135) Robinson. J.C, Berry, P.E Description Of Peakvue And Illustration Of Its Wide Array Of Applications In Fault Detection And Problem Severity Assessment, CSI, an Emerson Electric Company , Technical Associates of Charlotte, P.C.
- 136) Raimondi. A,J. Boyd "A Solution for the Finite Journal Bearing and Its Application to Analysis and Design" American Society of Lubrication Engineers, Vol. 1, No. 1, in Lubrication Science and Technology Pergamon, New York, 1958
- 137) Tsuneo. S, (Editor), Journal-Bearing Databook ; ISBN-10: 354017074X (October 1989)

- 138) Bannister. R.H, "Rotor Dynamics". Class notes.
- 139) Fox. R.W, McDonald. A.T, Pritchard. P.J Introduction to fluid mechanics , ISBN: 0471376531 Edition: 6th. Publication info: Hoboken : Wiley, 2004
- 140) Munson. B. R, Young. D. F, Okiishi. T.H, Fundamentals of fluid mechanics 5th. Hoboken NJ: John Wiley, 2006.
- 141) Rooke. D. P, Compounding Stress Intensity Factors: Applications to Engineering Structures Research Reports in Materials Science, Series 1, Vol 1, Parthenon Publishing Group (November 1986), ISBN-13: 978-1850701101
- 142) Hutton. J.F, Jones. B, Bates. T.W, Effects of isotropic pressure on the high temperature high shear viscosity of motor oils, SAE Detroit (1983) Paper No. 30.
- 143) Máleka. J, Rajagopal K.R, Chapter 5 Mathematical Issues Concerning The Navier-Stokes Equations and Some Of Its Generalizations; Mathematical Institute, Faculty of Mathematics and Physics, Charles University, Sokolovská 83, 186 75 Prague 8, Czech Republic
- 144) Navier, C. L. M. H. (1823): Memoire sur les Lois du mouvement des fluides. Mém. Acad. Sci. Paris, vol. 6, pp. 389-416.
- 145) Petrov. N. P. (1883): Friction in Machines and the Effect of the Lubricant. 7 MA. ZA. Fi. Petersburg, vol. 1, pp. 71-140; vol. 2, pp. 227-279; vol. 3, pp. 377-463; vol. 4, pp. 535-564.
- 146) Stokes, G. G. (1845): On the Theories of the Internal Friction of Fluids in Motion, and of the Equilibrium and Motion of Elastic Solids. Trans. Cambridge philos. Soc., vol. 8. pp. 287-341.
- 147) Bernard J. Hamrock; Steven R, Fundamental of Fluid Film Lubrication Second Edition, Schmid 2004 Marcel Dekker, In chapter 6. Viscous Flow

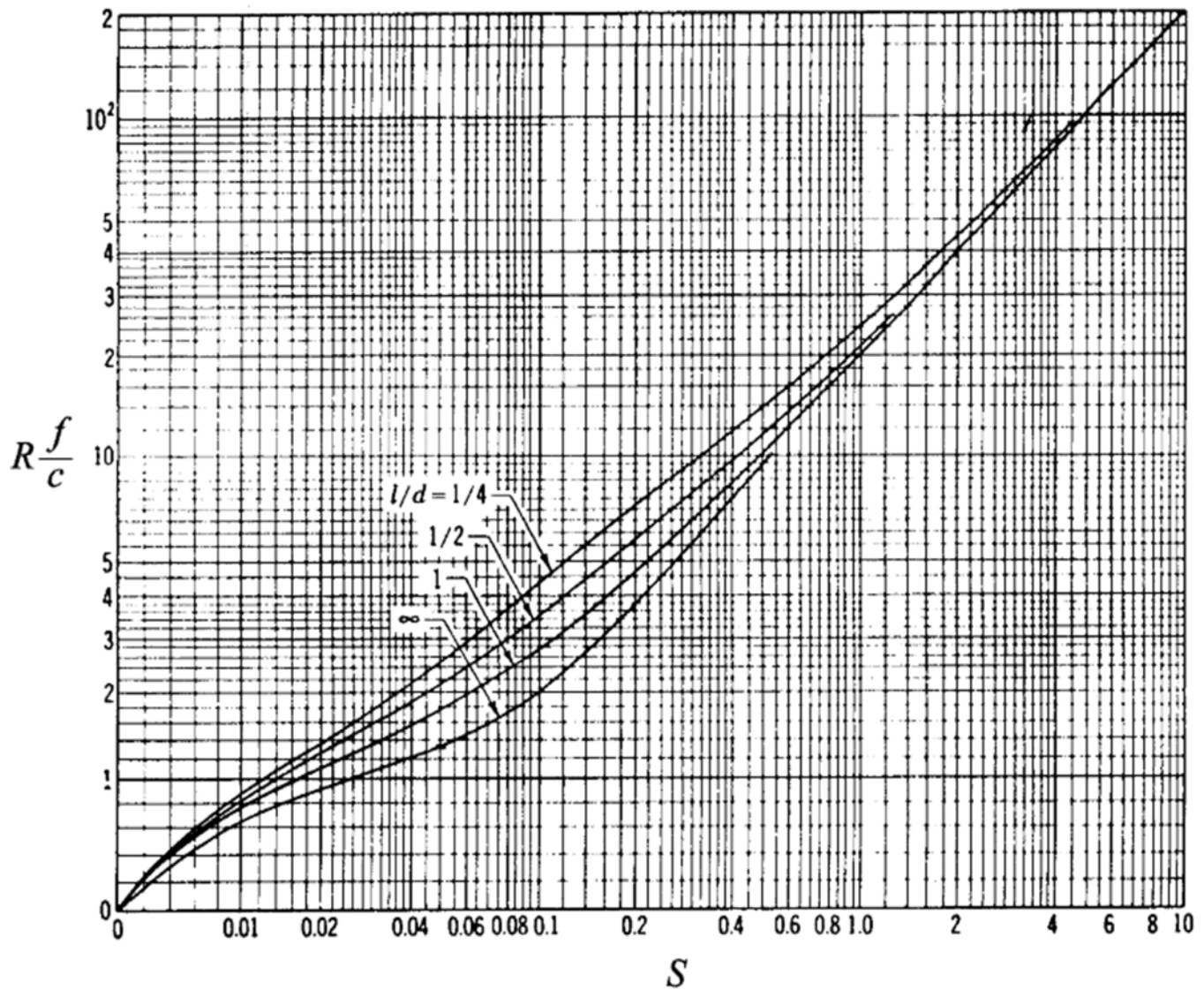
- 148) Hughes W.F, Brighton J.A, Fluid Dynamics, Schaum's Outline Series, McGraw-Hill Book Company (1967).
- 149) NDT Resource Centre "Principles of Eddy Current inspection"
<http://www.ndt-ed.org/>
- 150) Joseph M, Bruckley, formerly in Hocking NDT "An Introduction to Eddy Current Testing, theory and technology"
- 151) Javier G.M, Gómez-Gil.J, Vázquez-Sánchez. E Non-Destructive Techniques Based on Eddy Current Testin, ; ISSN 1424-8220; Sensors 2011, 11, 2525-2565; doi:10.3390/s110302525
- 152) Sadiku, M. N. O. (2007). Elements of Electromagnetics (fourth ed.). New York (USA)/Oxford (UK): Oxford University Press. p. 386. ISBN 0-19-530048-3.
- 153) Ulaby, Fawwaz (2007). Fundamentals of applied electromagnetics (5th ed.). Pearson:Prentice Hall. p. 255. ISBN 0-13-241326-4.
- 154) Michael Faraday, by L. Pearce Williams, p. 182-3
- 155) Martins, De Andrade.R, "Romagnosi ,Volta's Pile: Early Difficulties in the Interpretation of Voltaic Electricity". In Fabio Bevilacqua and Lucio Fregonese (eds). Nuova Voltiana: Studies on Volta and his Times. vol. 3. Università degli Studi di Pavia. pp. 81–102.
- 156) Stern, David P, Peredo. M, (2001-11-25). "Magnetic Fields History". NASA Goddard Space Flight Center.
- 157) Xu Zhe Liu. Chuangwen,Wencui L, Frequency Band Energy Analysis of Vibration Signals for Tool Condition Monitoring; ISBN: 978-0-7695-3583-8; ICMTMA '09 Proceedings of the 2009 International Conference on Measuring Technology and Mechatronics Automation - Volume 01

- 158) Efstathopoulos. E, Patatoukas, G, Pantos.L, Benekos. O, Demosthenes. K, Kelekis. N, Wall shear stress calculation in ascending aorta using phase contrast magnetic resonance imaging. Investigating effective ways to calculate it in clinical practice *Physica Medica*, 2008, 24, 4, 175-181
- 159) Gopinath. K, Mayuram. M; Machine Design II, Hydrodynamic Lubrication Of Journal Bearings Theory And Practice, Module 5 – Sliding Contact Bearings; Indian Institute of Technology Madras
- 160) ThermoNiolet access on; <http://mmrc.caltech.edu/FTIR/FTIRintro.pdf>
- 161) ISO 20096, Condition Monitoring and Diagnostic of Machines, Acoustic Emission General Guidelines, 2007. SNR Group. "Types of Bearings"
- 162) Miettinen, J and Siekkinen, V. Acoustic emission in monitoring sliding contact behaviour. *Wear*, 1995, 181, 897-900
- 163) Tandon, N. / Yadava, G.S. ;A comparison of some condition monitoring techniques for the detection of defect in induction motor ball bearings/ Ramakrishna, K.M., *Mechanical Systems and Signal Processing*, 21 (1), pp.244-256, Jan 2007
- 164) Feng. Y; Novel Acoustic Emission Signal Processing Methods for Bearing Condition Monitoring/ thesis; University of Leicester 2008.
- 165) Sutton. B.A, Design for a dynamic journal bearing. MSc Thesis, 1974, Cranfield University.
- 166) Morton. P.G; Measurement of the dynamic characteristics of a large sleeve bearing; *Journal of lubrication technology ASME* p143, 1971
- 167) K. H. Pedersen, „Condition Monitoring of Gear failure with Acoustic Emission“, MSc Thesis. School of Engineering, Cranfield University, UK, 2005.

APPENDICES

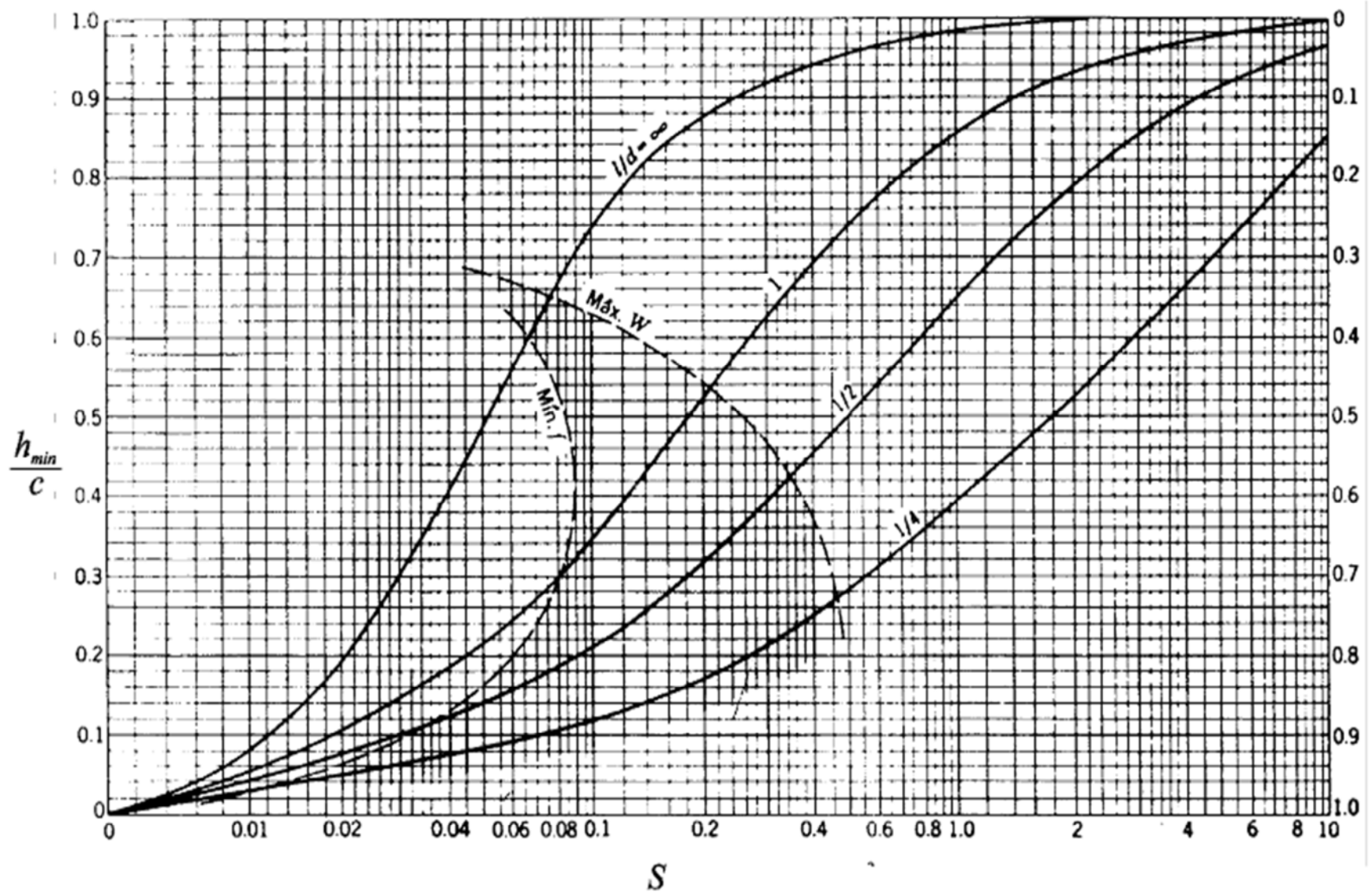
Appendix A

A.1 RAIMONDI BOYD GRAPHS

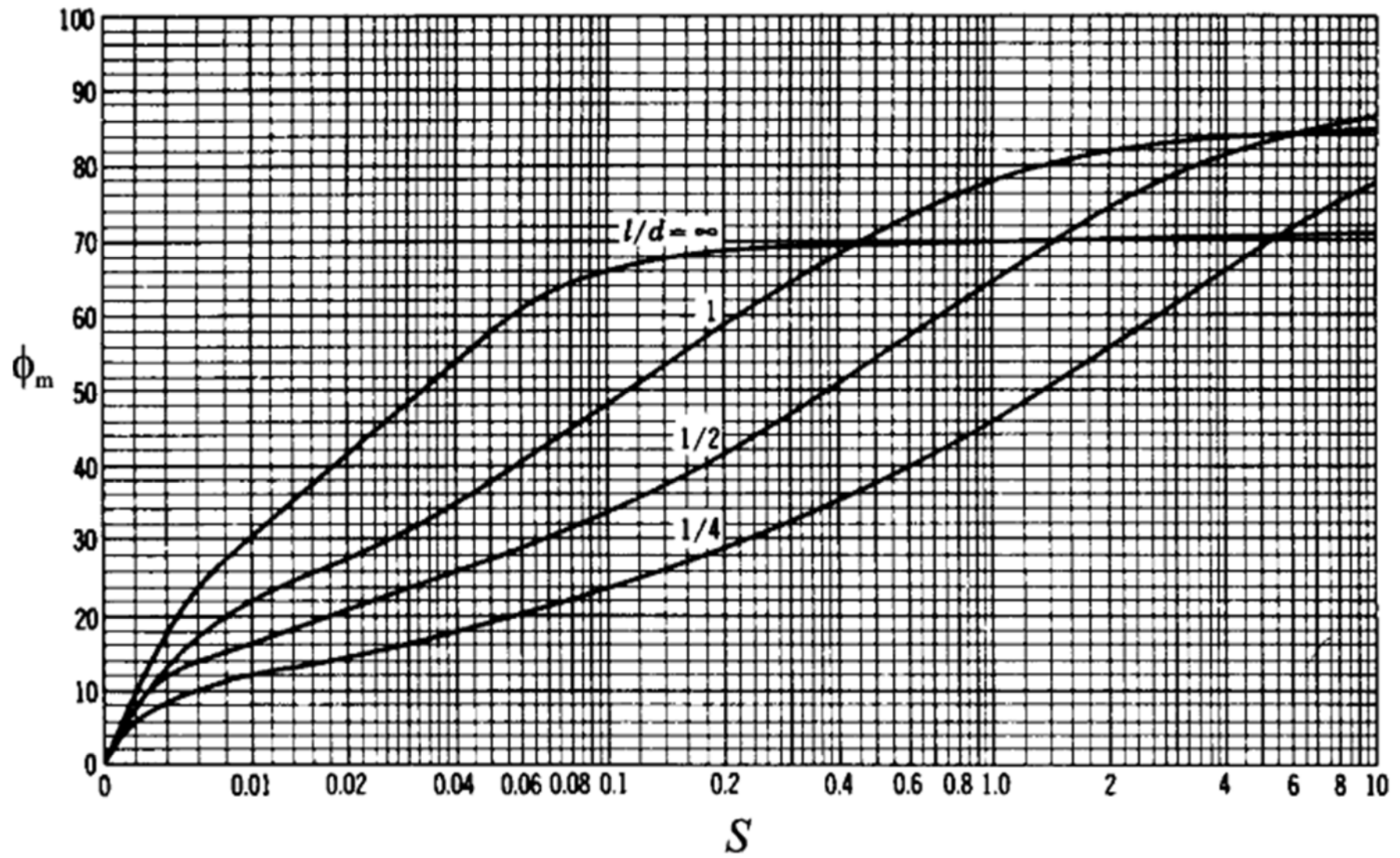


Obtaining the friction factor

Obtaining of the minimum film thickness

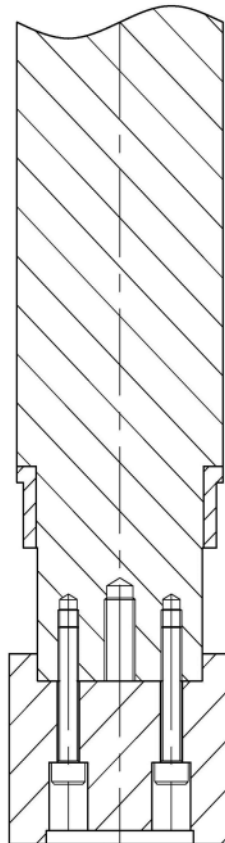
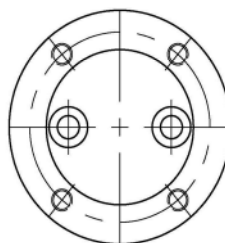


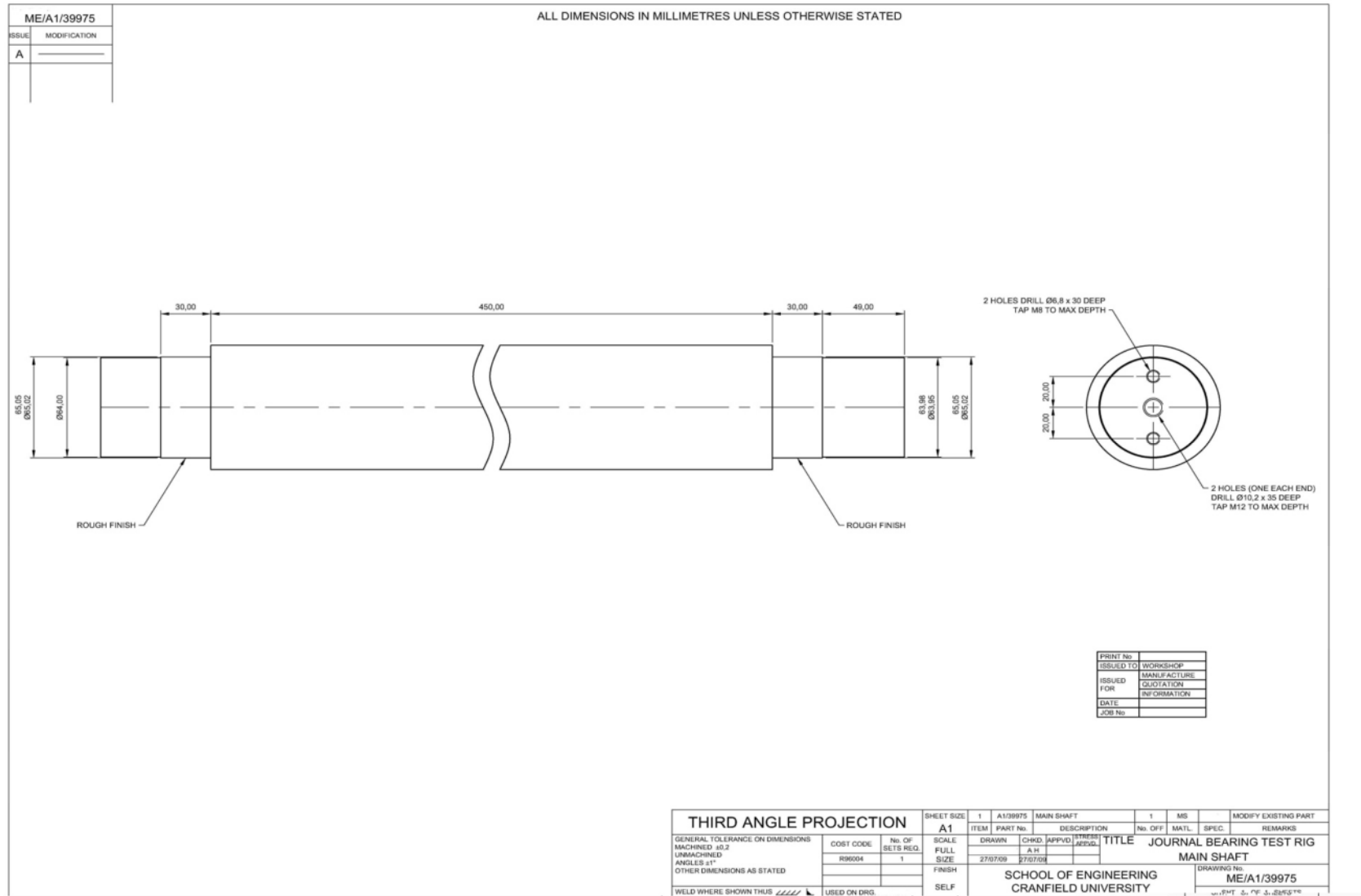
Obtaining of the angle of minimum film thickness location



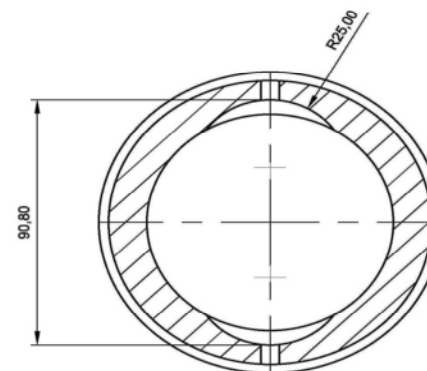
Appendix B

B.1 Shaft Test Rig Design





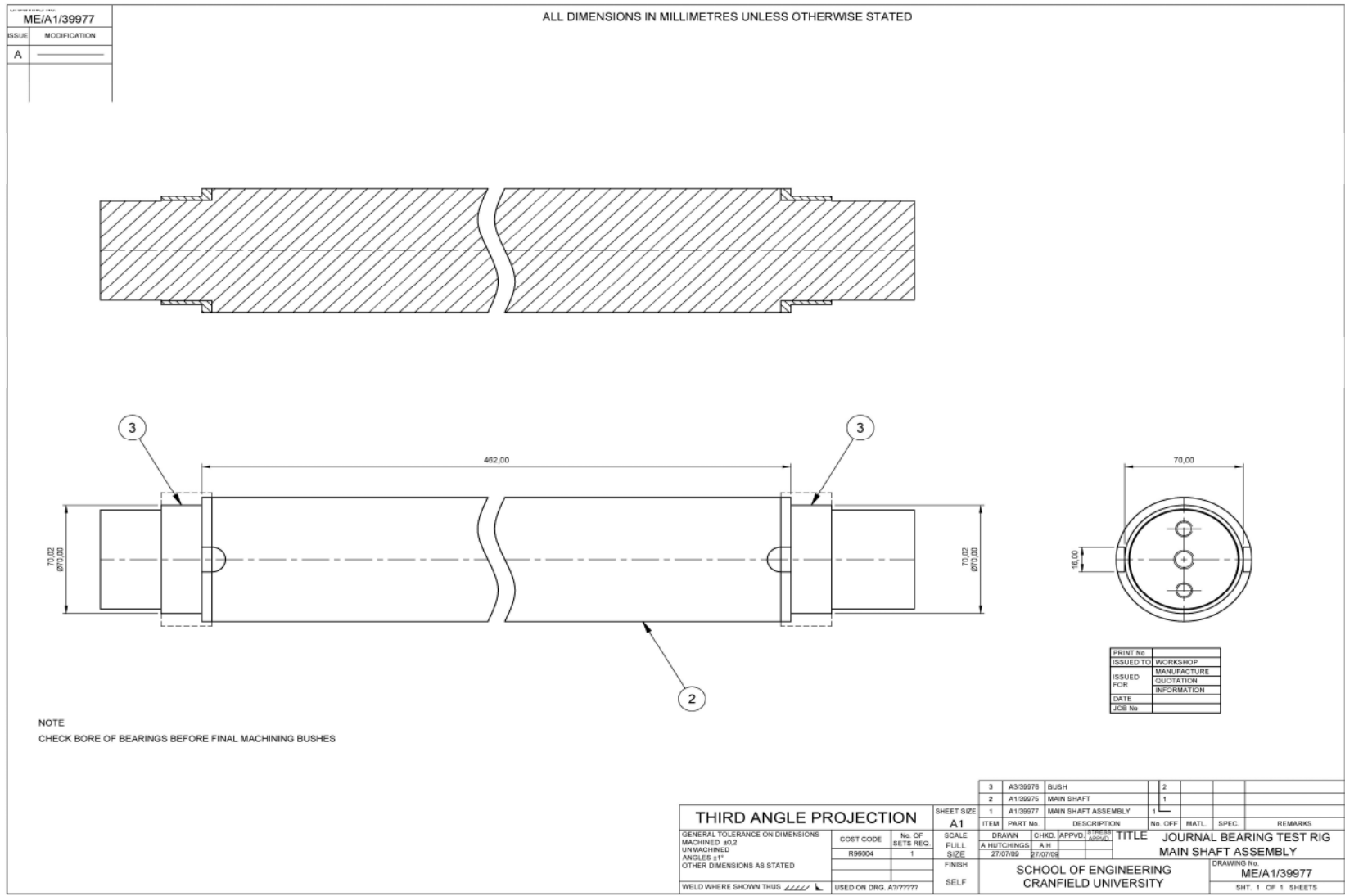
ALL DIMENSIONS IN MILLIMETRES UNLESS OTHERWISE STATED

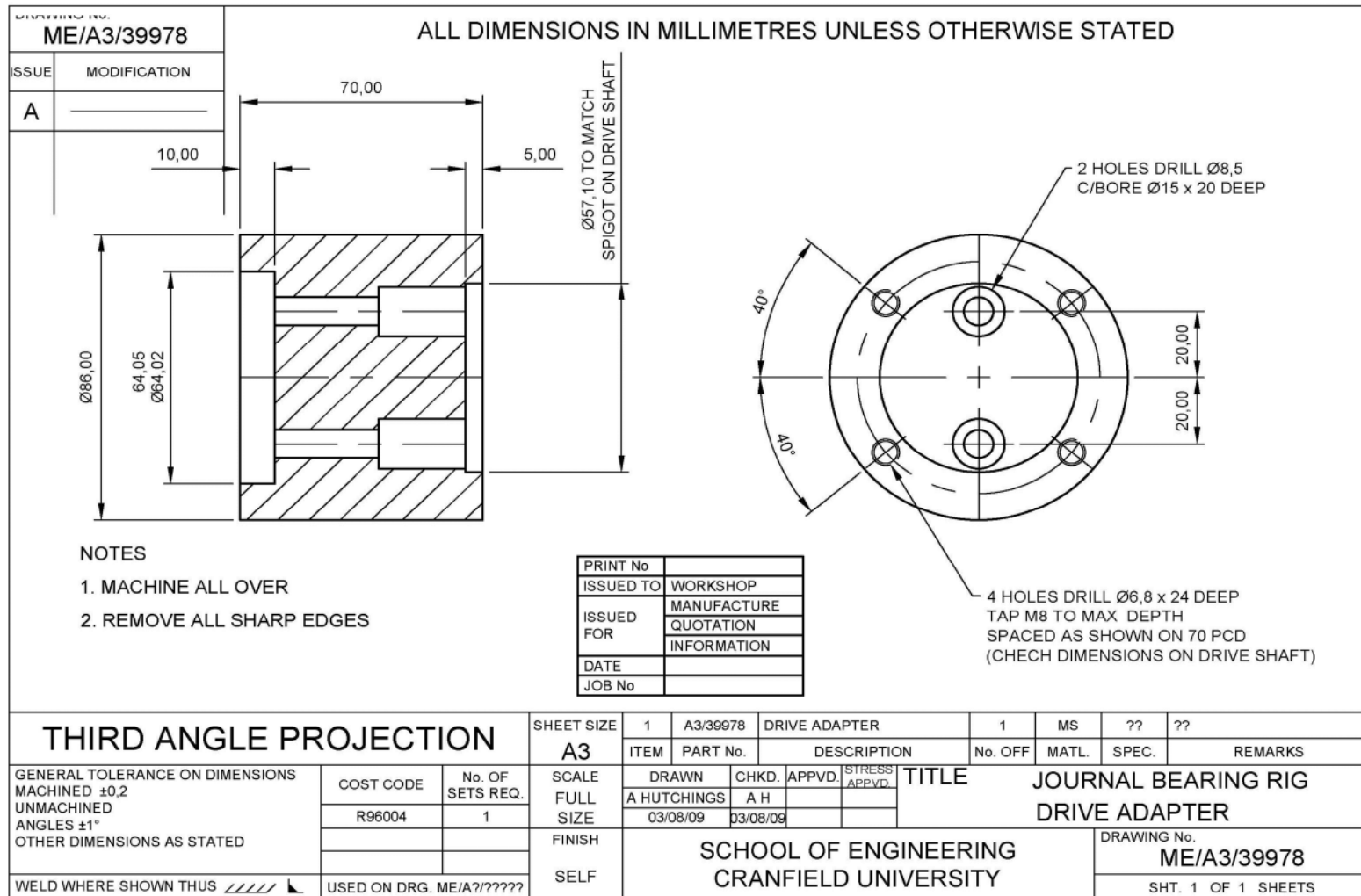


NOTES

1. MANUFACTURE
2 OFF BRASS
2 OFF ALUMINIUM
2. REMOVE ALL SHARP EDGES

THIRD ANGLE PROJECTION			SHEET SIZE	1	A2/39200	JOURNAL BEARING		SEE NOTE 1	SEE NOTE 1		
			A2	ITEM	PART No.	DESCRIPTION		No. OFF	MATL.	SPEC.	REMARKS
GENERAL TOLERANCE ON DIMENSIONS MACHINED ±0.2 UNMACHINED ANGLES ±1° OTHER DIMENSIONS AS STATED	COST CODE	No. OF SETS REQ.	SCALE	DRAWN	CHKD.	APPRVD	STRESS APPRVD.	TITLE			
			FULL SIZE		A H			JOURNAL BEARING TEST RIG			
	R96004	1	SIZE	23/05/08	23/05/08			JOURNAL BEARING			
				FINISH							
			SELF	SCHOOL OF ENGINEERING							DRAWING No.
				CRANFIELD UNIVERSITY							ME/A2/39200
WELD WHERE SHOWN THUS			USED ON DRG.								SHT. 1 OF 1 SHEETS





Appendix C

C.1 Castrol Alpha® SP Range



Description

Castrol Alpha SP Range are formulated with premium solvent-refined, base oils fortified with sulfur-phosphorus. Compounds to impact anti-wear & extreme pressure properties. The combination of base oils and additives is designed to meet all industrial gear oil applications. Alpha SP Range meet AGMA, U.S. Steel and other industrial gear lubricant specifications.

Application

Alpha SP Range are designed for use in all industrial lubricant applications where SP Range are specified by the equipment manufacturers or called for by system requirements. For application while operating or ambient temperatures are higher than normal, please consult your account manager regarding various product operations.

- Alpha SP Range of gear oils are intended for use in all types of enclosed gears with circulation or splash lubrication systems.
- Large and small spur bevel, spiral, helical, herringbone.
- Application where gears and bearings are subject to shock loads.
- Oil mist-lubricated gears and bearings

Advantages

- Excellent load carrying capacity - prevents premature wear of gear surfaces under rigorous operating loads.
- Highly resistant to oxidation - prevents sludging and varnishing, thus providing long service life.
- Excellent corrosion resistance - protects internal surfaces from entrained air and moisture-induced corrosion.
- Highly foam resistant - inhibited to prevent foam under the most rigorous operating loads.
- Increased lubricity/wear protection - the sulfur/phosphorous anti-wear package promotes wear protection and increased lubricity, thereby reducing gear case bulk oil temperatures.

Characteristics

Test Results	Alpha SP 68	Alpha SP 100	Alpha SP 150	Alpha SP 220	Alpha SP 320	Alpha SP 460	Alpha SP 680
AGMA Grade	2 EP	3 EP	4 EP	5 EP	6 EP	7 EP	8 EP
ISO Viscosity Grade	68	100	150	220	320	460	680
Kinematic Viscosity, ASTM D-445, @ 40°C, mm ² /sec @ 100°C, mm ² /sec	67.3 9.05	101.75 10.75	150.0 14.5	220 18.53	318.26 23.68	461 30.63	682.1 36.61
Viscosity Index, ASTM D-2250	98	98	98	96	96	96	95
Flash Point, ASTM D-92, °C	190	226	249	254	226	190	264
Specific Gravity @ 15°C/ 60°F, ASTM D-1298	0.88	0.877	0.89	0.89	0.892	0.897	0.897
Timken OK Load, ASTM D-2670, lb/kgf	65	65	75	75	75	75	75
FZG, Failure Stage	12	12	12	12	12	12	12
Rust, ASTM D-665, A & B	Pass	Pass	Pass	Pass	Pass	Pass	Pass
Copper Strip Corrosion, ASTM D-130	1b	1b	1b	1b	1b	1b	1b
Demulsibility, ASTM D-1401, 15 minutes	40/40/0	40/40/0	40/40/0	40/40/0	40/40/0	40/40/0	40/40/0
Foam, ASTM D-892 Sequence I Sequence II Sequence III	0/0 10/0 0/0	0/0 10/0 0/0	0/0 10/0 0/0	0/0 10/0 0/0	0/0 10/0 0/0	0/0 10/0 0/0	0/0 10/0 0/0

Additional Information

Alpha SP Range are compatible with all oil-resistant seals and metals. These oils can be added to competitive products. However, for optimum performance drain the reservoirs completely and refill with the OEM recommended viscosity grade. Longer gear life can be promoted by maintaining the full level specified on the gear case and by changing the oil as required by operating or ambient conditions.

Castrol Alpha SP Range

03.15.2005, Version Number 2.0

Alpha SP Range and the Castrol logo are trademarks of Castrol Limited.

All reasonable care has been taken to ensure that the information contained in this publication is accurate as of the date of printing. However, such information may, nevertheless, be affected by changes in the blend formulation occurring subsequent to the date of printing. Material Safety Data Sheets are available for all Castrol Ltd products. The MSDS must be consulted for appropriate information regarding storage, safe handling and disposal of a product.

Castrol Industrial Americas

3660 Lakeshore Blvd.

West Toronto

Ontario M8W 1P2

Tel (416) 252 2511

www.castrol.com/Industrial

Appendix D

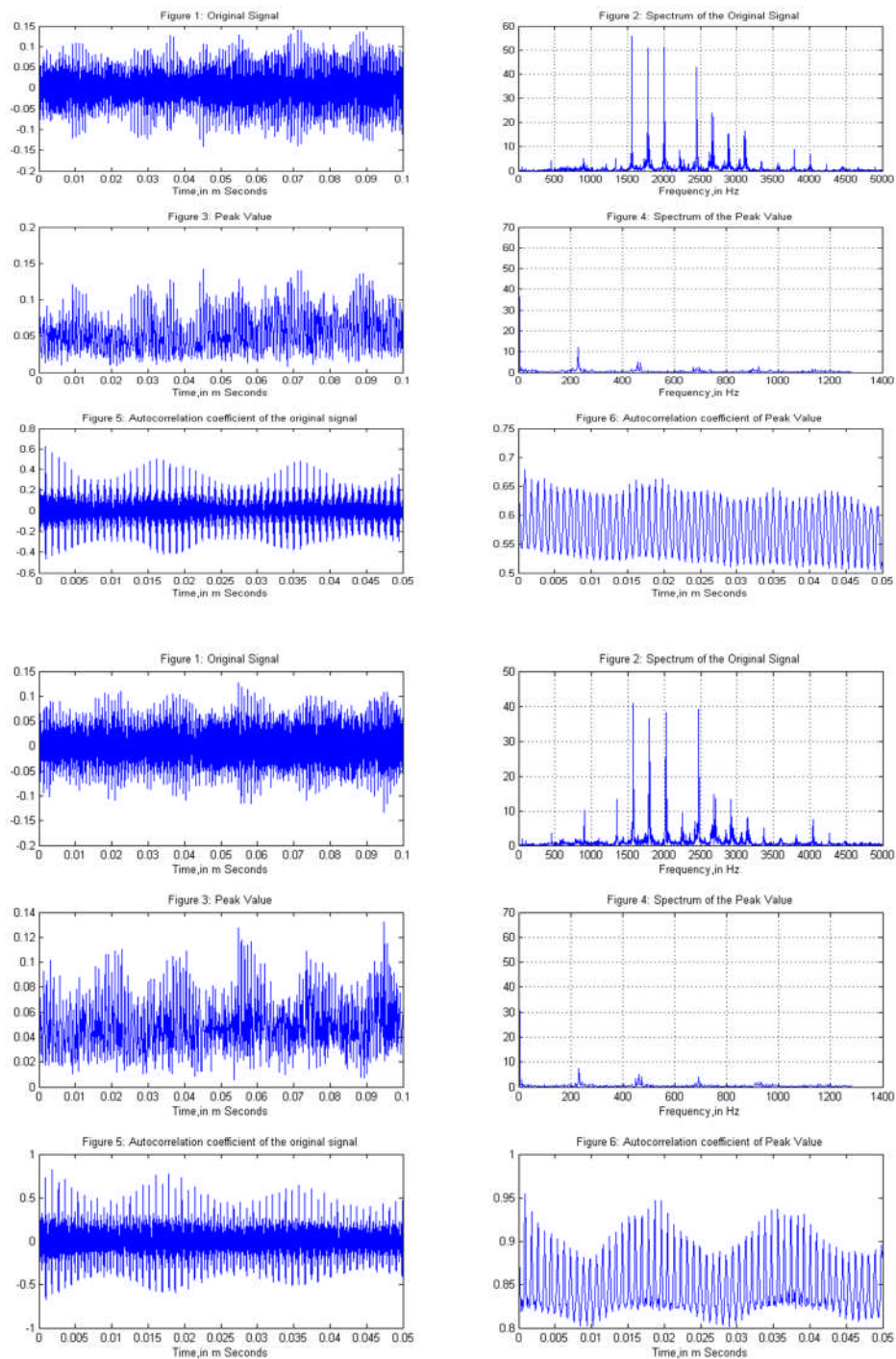
Oil Type	P (kN)	Speed [RPM]	Temperature (°C)	STDV T	V (cSt)	μ (Pa*s)	S	ecc	Hmin (μ m)	STDV hmin	Friction Factor	KW Btm (W)	Total Power Loss(KW)	Shear stresses (MPa)	RMS (V)
SP68															
	2	750	42.64	2.18	62.06	0.05	0.13	0.61	67.05	3.28	0.0136	0.0800	0.0963	0.0025	0.002
	2	1500	54.21	1.99	41.59	0.04	0.17	0.53	80.24	3.15	0.0170	0.1996	0.2474	0.0028	0.006
	2	2500	66.83	1.25	26.88	0.02	0.18	0.51	83.64	1.99	0.0180	0.3521	0.4374	0.0029	0.029
	2	3500	77.21	2.4	18.77	0.02	0.18	0.51	82.6	3.81	0.0177	0.4844	0.6015	0.0029	0.061
	2	4500	87.17	1.62	13.3	0.01	0.16	0.54	78.32	2.57	0.0165	0.5800	0.7182	0.0028	0.101
Oil Type	P (kN)	Speed [RPM]	Temperature (°C)	STDV T	V (cSt)	μ (Pa*s)	S	ecc	Hmin (μ m)	STDV hmin	Friction Factor	KW Btm (W)	Total Power Loss(KW)	Shear stresses (MPa)	RMS (V)
SP68															
	6	750	49.43	2.84	49.07	0.04	0.03	0.85	25.17	1.93	0.0056	0.0993	0.1140	0.0053	0.0017
	6	1500	56.7	2.87	38.16	0.03	0.05	0.79	35.24	2.81	0.0073	0.2581	0.3059	0.0059	0.009
	6	2500	65.15	2.87	28.48	0.02	0.06	0.76	41.58	3.19	0.0085	0.4962	0.5942	0.0062	0.03
	6	3500	74.76	1.77	20.43	0.02	0.07	0.75	41.7	1.96	0.0085	0.6965	0.8343	0.0062	0.065
	6	4500	85.25	2.81	14.21	0.01	0.06	0.77	38.33	2.93	0.0079	0.8317	0.9911	0.006	0.105
Oil Type	P (kN)	Speed [RPM]	Temperature (°C)	STDV T	V (cSt)	μ (Pa*s)	S	ecc	Hmin (μ m)	STDV hmin	Friction Factor	KW Btm (W)	Total Power Loss(KW)	Shear stresses (MPa)	RMS (V)
SP68															
	10	750	51.56	2.42	45.58	0.04	0.02	0.9	16.41	0.97	0.0042	0.1245	0.1386	0.0075	0.0021
	10	1500	57.95	2.63	36.54	0.03	0.03	0.86	23.12	1.65	0.0053	0.3114	0.3582	0.0086	0.01
	10	2500	64.35	2.36	29.28	0.03	0.04	0.83	28.79	1.79	0.0062	0.6099	0.7130	0.0092	0.034
	10	3500	76.18	1.43	19.45	0.02	0.04	0.84	27.23	1	0.0060	0.8185	0.9531	0.009	0.066
	10	4500	87.25	1.66	13.26	0.01	0.03	0.86	24.65	1.08	0.0056	0.9778	1.1301	0.0088	0.107

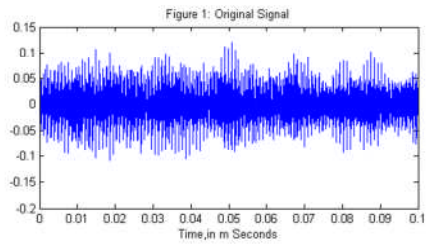
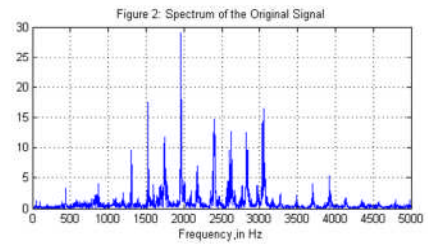
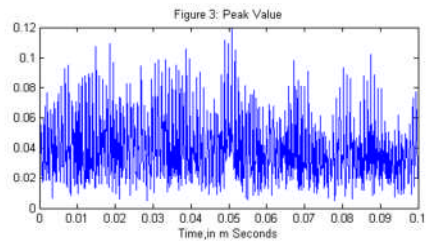
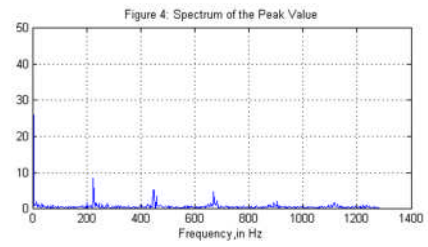
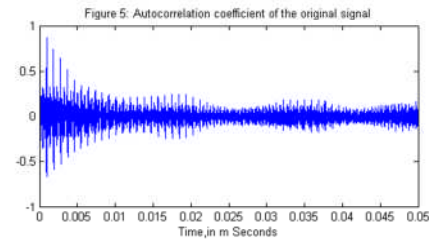
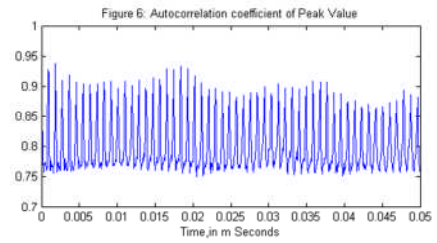
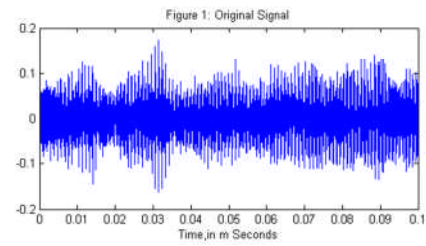
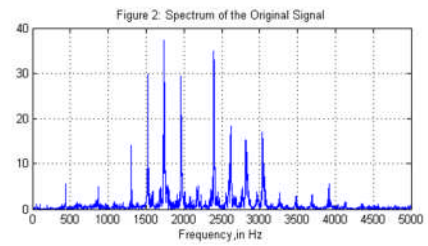
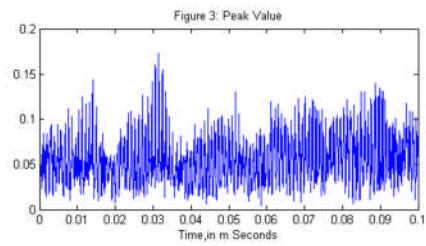
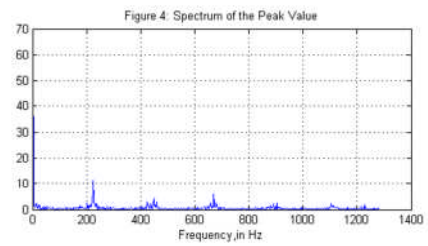
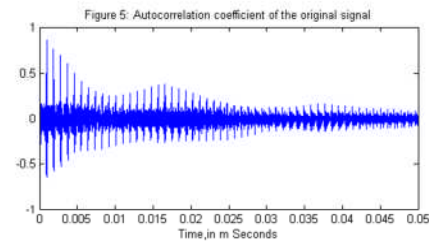
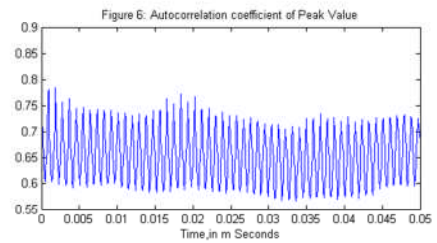
Oil Type	P (kN)	Speed [RPM]	Temperature (°C)	STDV T	V (cSt)	μ (Pa*s)	S	ecc	Hmin (μ m)	STDV hmin	Friction Factor	KW Btm (W)	Total Power Loss(KW)	Shear stresses (MPa)	RMS (V)
SP220															
	2	750	45.42	2.39	176.08	0.16	0.37	0.33	113.36	3.57	0.0318	0.1869	0.2303	0.0044	0.002
	2	1500	58.53	3.59	102.75	0.09	0.44	0.3	118.74	5.14	0.0364	0.4277	0.5281	0.0049	0.008
	2	2500	72.6	2.7	57.64	0.05	0.41	0.32	116.43	3.9	0.0343	0.6718	0.8288	0.0046	0.03
	2	3500	83.04	2.47	37.54	0.03	0.37	0.33	113.18	3.63	0.0317	0.8684	1.0697	0.0043	0.05
	2	4500	92.96	1.75	24.97	0.02	0.32	0.37	107.37	2.75	0.0278	0.9792	1.2028	0.0039	0.091
Oil Type	P (kN)	Speed [RPM]	Temperature (°C)	STDV T	V (cSt)	μ (Pa*s)	S	ecc	Hmin (μ m)	STDV hmin	Friction Factor	KW Btm (W)	Total Power Loss(KW)	Shear stresses (MPa)	RMS (V)
SP220															
	6	750	54.15	2.66	123.01	0.11	0.09	0.7	51.6	4.09	0.0103	0.1822	0.2171	0.0067	0.002
	6	1500	63.17	2.97	84.92	0.08	0.12	0.62	64.55	5.15	0.0131	0.4601	0.5529	0.0074	0.01
	6	2500	74.65	3.82	52.99	0.05	0.12	0.61	66.24	6.74	0.0134	0.7894	0.9495	0.0075	0.038
	6	3500	85.21	1.9	34.34	0.03	0.11	0.63	62.08	3.24	0.0125	1.0286	1.2343	0.0072	0.067
	6	4500	96.98	1.46	21.17	0.02	0.09	0.69	52.8	2.28	0.0106	1.1182	1.3337	0.0068	0.092
Oil Type	P (kN)	Speed [RPM]	Temperature (°C)	STDV T	V (cSt)	μ (Pa*s)	S	ecc	Hmin (μ m)	STDV hmin	Friction Factor	KW Btm (W)	Total Power Loss(KW)	Shear stresses (MPa)	RMS (V)
SP220															
	10	750	55.53	2.28	116.23	0.1	0.05	0.8	33.76	2.49	0.0071	0.2076	0.2426	0.0097	0.0018
	10	1500	64.52	2.21	80.34	0.07	0.07	0.75	43.14	2.86	0.0087	0.5129	0.6066	0.0105	0.009
	10	2500	75.29	2.9	51.61	0.05	0.07	0.73	45.39	4.01	0.0092	0.8956	1.0616	0.0106	0.038
	10	3500	87.25	1.45	31.57	0.03	0.06	0.76	40.43	1.78	0.0082	1.1293	1.3316	0.0102	0.06
	10	4500	98.15	2.42	20.18	0.02	0.05	0.8	34.83	2.59	0.0073	1.2781	1.4960	0.0098	0.095

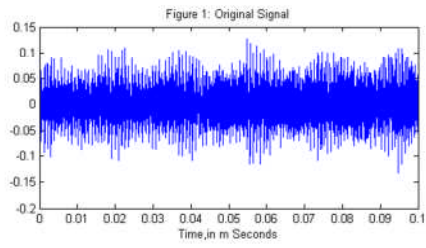
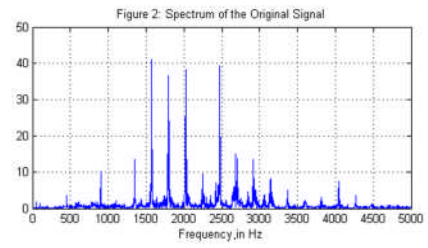
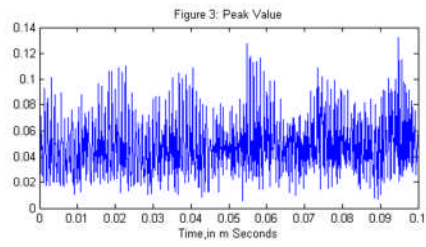
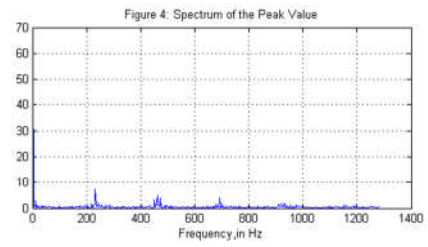
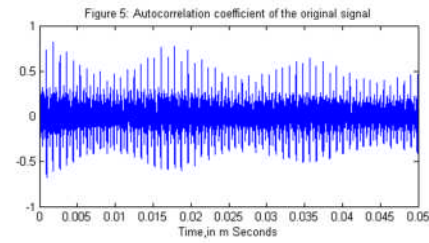
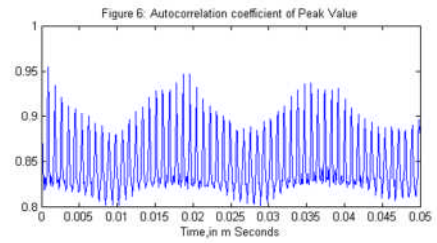
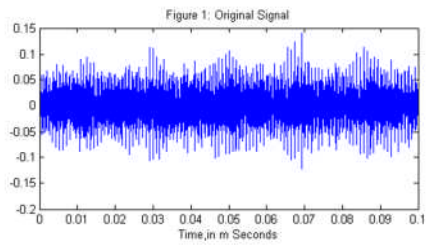
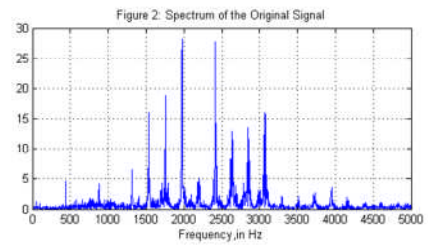
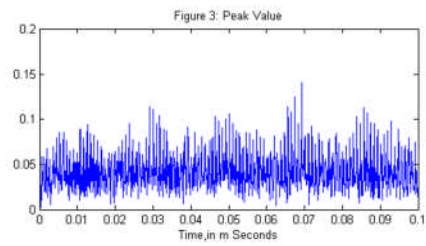
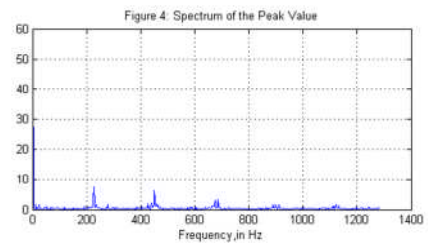
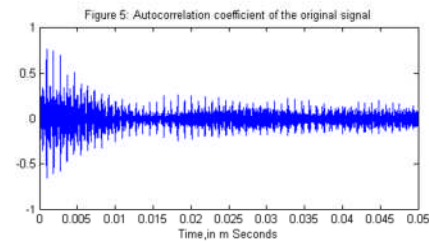
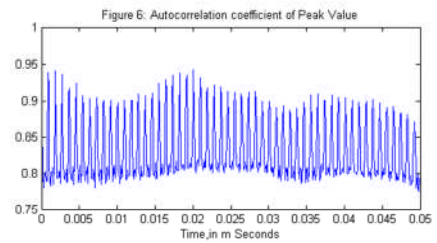
Oil Type	P (kN)	Speed [RPM]	Temperature (°C)	STDV T	V (cSt)	μ (Pa*s)	S	ecc	Hmin (μ m)	STDV hmin	Friction Factor	KW Btm (W)	Total Power Loss(KW)	Shear stresses (MPa)	RMS (V)
SP 460															
	2	750	54.92	1.02	234.27	0.21	0.5	0.27	123.44	1.64	0.0412	0.2421	0.2991	0.0054	0.002
	2	1500	64.84	0.91	149.58	0.13	0.64	0.22	132.87	1.79	0.0521	0.6114	0.7711	0.0064	0.008
	2	2500	76.22	1.7	89.41	0.08	0.64	0.22	132.7	3.38	0.0519	1.0152	1.2804	0.0063	0.033
	2	3500	86	2.66	57.45	0.05	0.57	0.24	128.38	4.37	0.0468	1.2825	1.6224	0.0059	0.061
	2	4500	96.29	2.67	36.07	0.03	0.46	0.29	120.7	4.2	0.0384	1.3509	1.7125	0.0051	0.091
Oil Type	P (kN)	Speed [RPM]	Temperature (°C)	STDV T	V (cSt)	μ (Pa*s)	S	ecc	Hmin (μ m)	STDV hmin	Friction Factor	KW Btm (W)	Total Power Loss(KW)	Shear stresses (MPa)	RMS (V)
SP 460															
	6	750	56.2	0.67	221.09	0.2	0.16	0.55	76.51	1.37	0.0160	0.2814	0.3403	0.0082	0.0019
	6	1500	65.44	1.65	145.58	0.13	0.21	0.48	89.12	3.37	0.0197	0.6952	0.8666	0.0092	0.009
	6	2500	76.6	1.74	87.88	0.08	0.21	0.47	89.4	3.54	0.0198	1.1642	1.4515	0.0092	0.036
	6	3500	87.7	2.3	53.2	0.05	0.18	0.52	81.82	4.77	0.0175	1.4344	1.7801	0.0086	0.062
	6	4500	98.77	1.53	32.24	0.03	0.14	0.59	70.49	3.09	0.0145	1.5269	1.8809	0.0077	0.08
Oil Type	P (kN)	Speed [RPM]	Temperature (°C)	STDV T	V (cSt)	μ (Pa*s)	S	ecc	Hmin (μ m)	STDV hmin	Friction Factor	KW Btm (W)	Total Power Loss(KW)	Shear stresses (MPa)	RMS (V)
SP 460															
	10	750	57.78	0.63	205.85	0.19	0.09	0.69	51.96	1.08	0.0104	0.3057	0.3644	0.0112	0.002
	10	1500	66.62	2.07	138.01	0.12	0.12	0.63	63.71	4.03	0.0129	0.7558	0.9264	0.0122	0.01
	10	2500	77.87	2.28	82.98	0.07	0.12	0.62	63.8	4.44	0.0129	1.2616	1.5464	0.0122	0.038
	10	3500	90.94	2.64	45.95	0.04	0.09	0.69	53.5	4.68	0.0107	1.4685	1.7838	0.0113	0.065
	10	4500	101.35	3.48	28.69	0.03	0.07	0.73	45.6	5.25	0.0092	1.6191	1.9496	0.0107	0.087

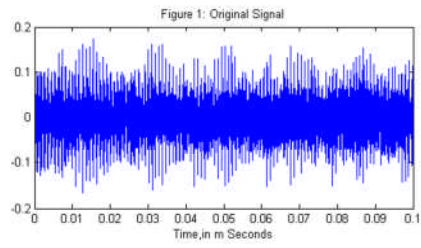
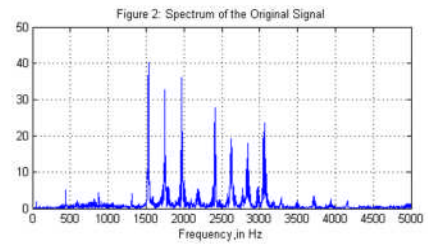
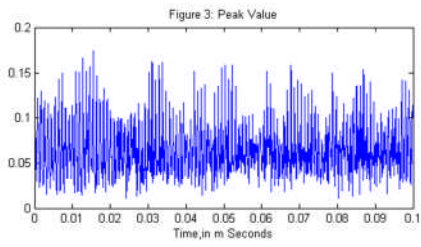
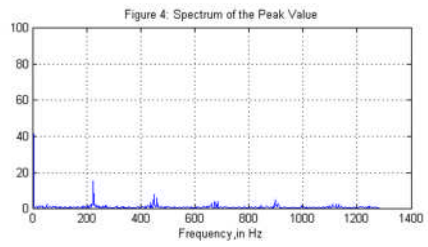
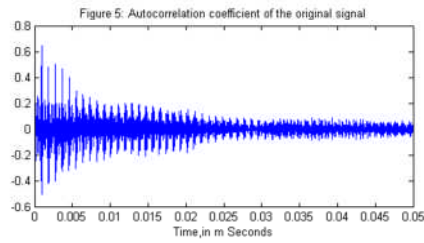
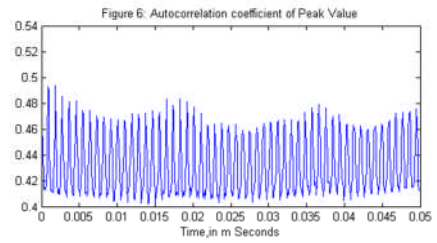
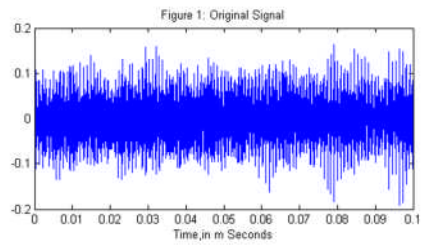
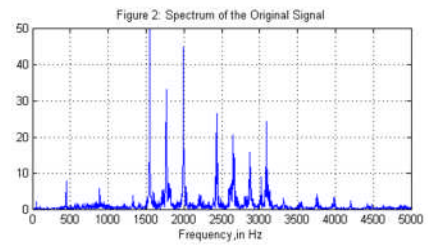
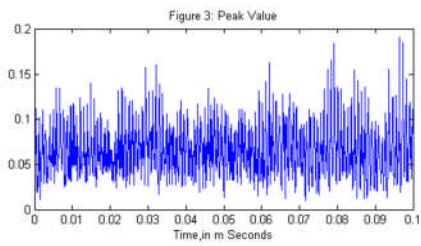
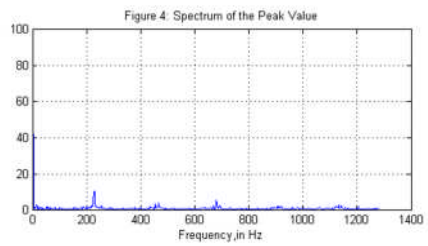
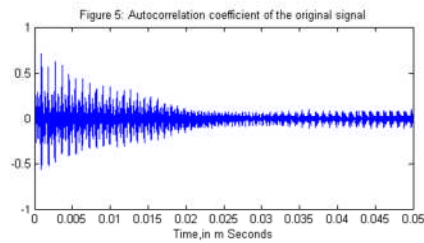
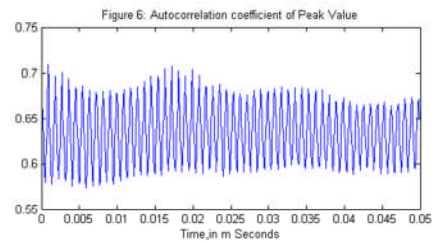
D.2 PeakVue Results

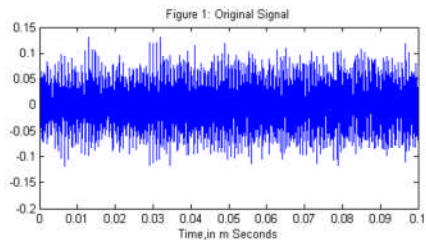
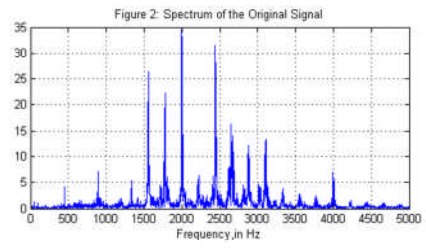
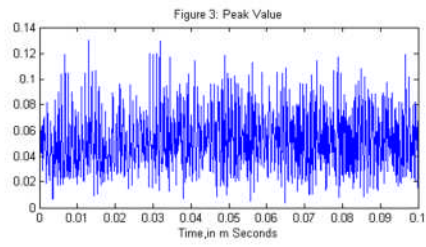
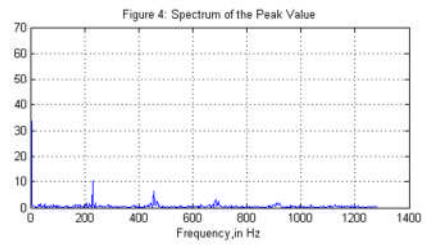
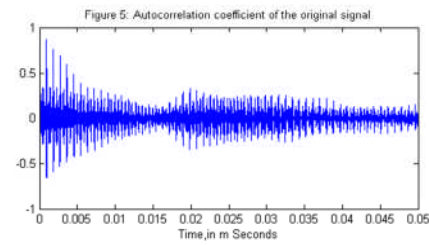
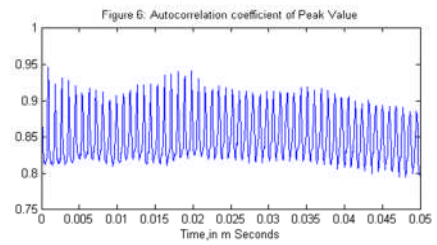
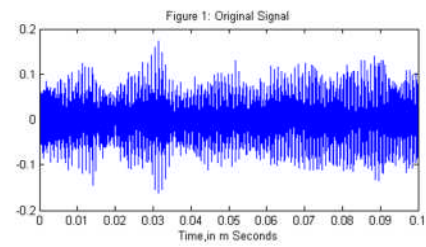
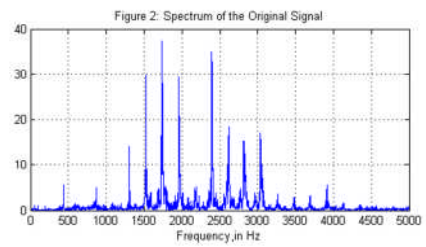
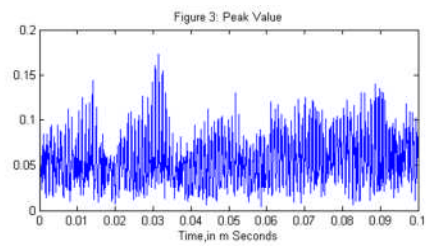
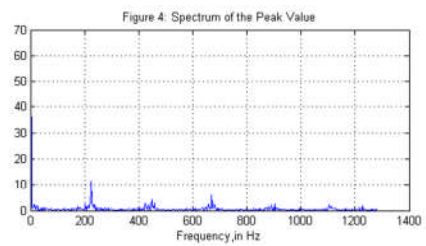
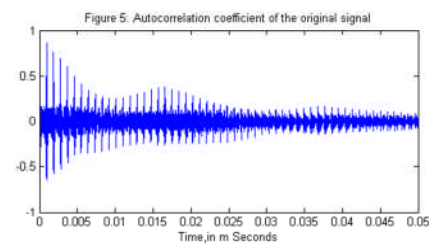
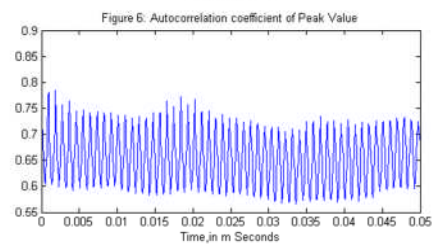
Different waveforms, PeakVue analysis:

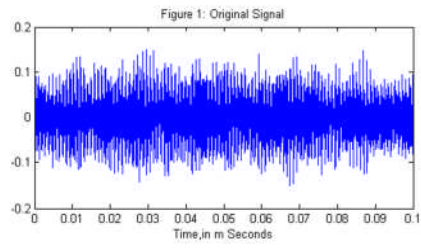
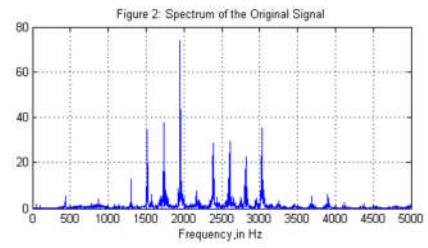
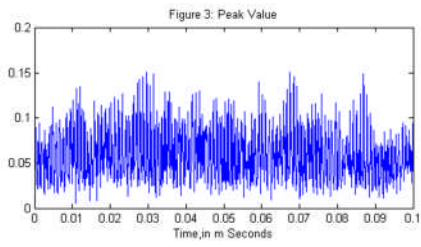
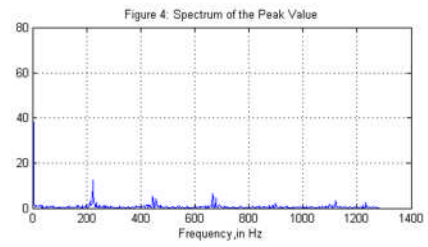
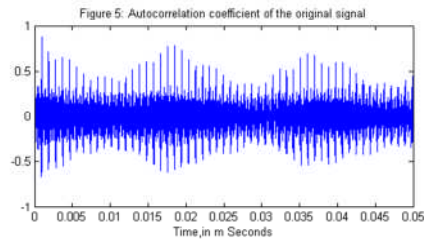
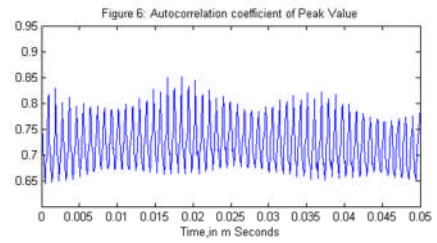
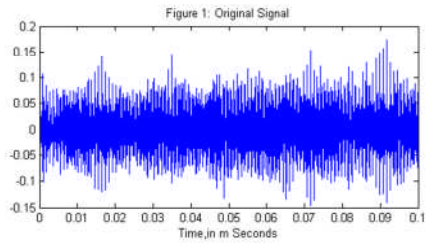
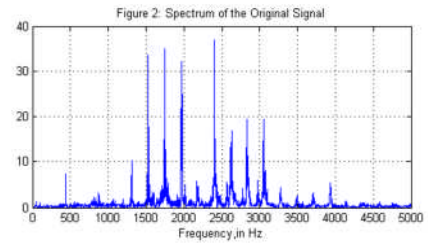
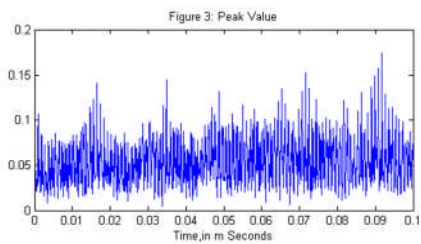
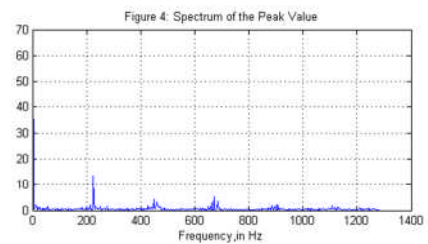
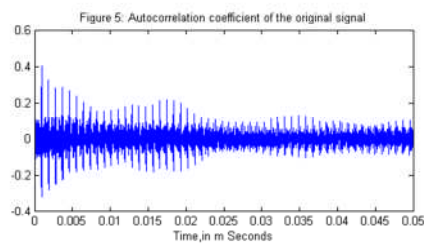
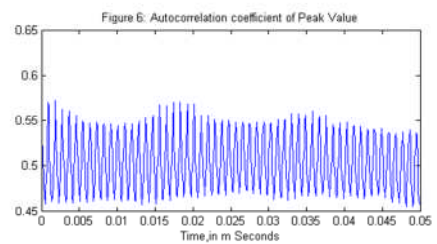






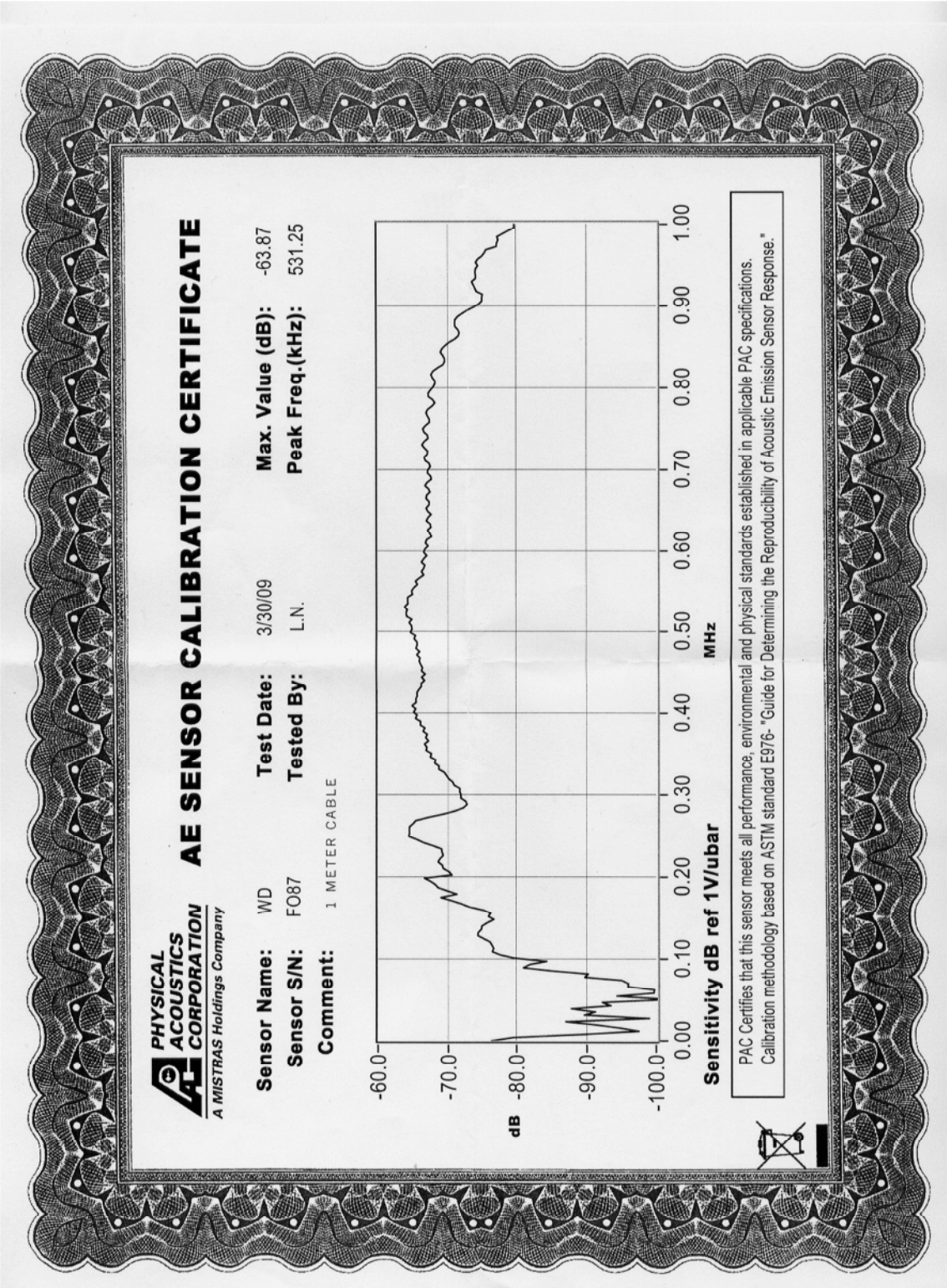






Appendix E

E.1 Calibration Certificate for WD type sensor



Appendix F

Vibration spectrum and sound for assessment and testing [108]

