Juha Miettinen

Condition Monitoring of Grease Lubricated Rolling Bearings by Acoustic Emission Measurements

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Thesis for the degree of Doctor of Technology to be presented with due permission for public examination and criticism in Festia Small Auditorium 1, at Tampere University of Technology, on the 6th of November 2000, at 12 o'clock noon.

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ABSTRACT

The behaviour of grease lubrication of rolling bearings is less known than the behaviour of oil lubrication in the same type of tribological systems, although greases are the most common lubricants for rolling bearings. Problems connected with the lubrication of grease lubricated rolling bearings arise from starvation of the contact zone and contaminants in the lubricant.

This thesis presents a new tool for predictive condition monitoring of grease lubricated rolling bearings. The tool consists of acoustic emission measurement technology, of guidelines for interpreting the AE measurement results and of an empirical model for prediction of the acoustic emission pulse count rate. Significant features of grease lubrication in rolling bearings are the starvation and the time dependence of the lubrication situation and the behaviour of the lubrication situation that does not follow the classical EHL theory for oil lubricated rolling bearings. The six papers presented in this thesis deal with mechanisms of grease lubrication and verification of the behaviour of grease lubrication in a real rolling bearing application. The conclusions regarding the running situation of the bearing are based on the measurement of the acoustic emission generated by the running of the bearing. Paper A gives an overview of the vibration signature monitoring of rolling bearings. A special case, an extremely slowly rotating rolling bearing, is presented in Paper C. Papers B and D deal with monitoring of the contaminants in the lubricating grease. The influence of the grease and running parameters on the lubrication situation of a rolling bearing has been the goal in Papers E and F.
The present study was carried out during the years 1996 to 2000 in the Institute of Machine Design at Tampere University of Technology. The research was financed by the National Technology Agency (Tekes) in Finland, Finnish Maintenance Society, VTT Manufacturing Technology, SKF Engineering & Research Centre B.V. in The Netherlands, the Finnish companies Mobil Oy oy ab, Rautaruukki Steel, VR Ltd. and Acutest Oy. The study was connected with the international research programme COST 516 GRIT. I wish to thank Professor Kenneth Holmberg, Pertti Leinonen, Esa Kantola Miika Kojonen, Pekka Saranto, Matti Pyylami and Pentti Pataniitty. I owe special gratitude to Dr. Victoria Wikström from SKF and Peter Andersson from VTT for improving the Papers comprising the study.

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Finally, warm thanks to my wife Hilkka, my son Joni and my parents for their support during the years I spent working on my thesis.

Tampere, November 2000
Juha Miettinen
This thesis comprises a survey and the following appended papers:


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<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Dislocation slip area</td>
</tr>
<tr>
<td>AE</td>
<td>Acoustic emission</td>
</tr>
<tr>
<td>$A_{es}$</td>
<td>AE severity key-figure</td>
</tr>
<tr>
<td>$A_s$</td>
<td>Actual area of contact in frictional contact</td>
</tr>
<tr>
<td>$A_{tot}$</td>
<td>Total contact area in frictional contact</td>
</tr>
<tr>
<td>$\dot{A}(t)$</td>
<td>Derivative of the dislocation area function</td>
</tr>
<tr>
<td>a</td>
<td>Final radius of a dislocation</td>
</tr>
<tr>
<td>$a_c$</td>
<td>Half of the crack length</td>
</tr>
<tr>
<td>$a_{cmin}$</td>
<td>Half width of the minimum detectable crack</td>
</tr>
<tr>
<td>$\hat{a}_{dc}$</td>
<td>Rms. value of the distance of dislocation motion</td>
</tr>
<tr>
<td>$\bar{a}_{dc}$</td>
<td>Average value of the distance of dislocation motion</td>
</tr>
<tr>
<td>$a_f$</td>
<td>Proportionality factor</td>
</tr>
<tr>
<td>$a_m$</td>
<td>Exponent whose value depends on the material</td>
</tr>
<tr>
<td>$a_1 ... a_{10}$</td>
<td>Coefficients in the empirical model</td>
</tr>
<tr>
<td>B</td>
<td>Thickness of the plastic zone</td>
</tr>
<tr>
<td>$B_{40}$</td>
<td>Base oil bleeding rate at temperature of 40 °C</td>
</tr>
<tr>
<td>b</td>
<td>Multiplier whose value depends on the type of the transducer and on the settings of the measurement device</td>
</tr>
<tr>
<td>b</td>
<td>Burger’s vector</td>
</tr>
<tr>
<td>$b_1, b_2, b_3$</td>
<td>Magnitudes of the components of Burger’s vector in the directions $x_1, x_2$ and $x_3$</td>
</tr>
<tr>
<td>C</td>
<td>Basic dynamic load rating</td>
</tr>
<tr>
<td>CF</td>
<td>Signal Crest Factor value</td>
</tr>
<tr>
<td>c</td>
<td>Phase velocity of a stress wave</td>
</tr>
<tr>
<td>$c_e$</td>
<td>Group velocity of a wavefront</td>
</tr>
<tr>
<td>$c_L$</td>
<td>Phase velocity of a longitudinal plane wave</td>
</tr>
<tr>
<td>$c_R$</td>
<td>Phase velocity of Rayleigh wave</td>
</tr>
<tr>
<td>$c_T$</td>
<td>Phase velocity of a transverse plane wave</td>
</tr>
<tr>
<td>$c_{w1}, c_{w2}$</td>
<td>Coefficients in the mother wavelet function</td>
</tr>
<tr>
<td>D</td>
<td>Proportionality constant</td>
</tr>
<tr>
<td>d</td>
<td>Unit vector which defines the direction of the plane wave motion</td>
</tr>
</tbody>
</table>
\( d_c \) Peak amplitude of the surface displacement caused by the longitudinal stress wave

\( d_j \) Minimum displacement of the surface that can be detected with the measurement sensor

\( d_3 \) Minimum displacement of the surface in the direction of surface normal that can be detected with the measurement sensor

\( E \) Modulus of elasticity

\( \dot{E}_c \) AE energy rate generated by a crack formation

\( \dot{E}_d \) AE energy rate caused by the motion of a large number of dislocations

\( \dot{E}_f \) AE energy rate generated from a sliding frictional contact

\( \dot{E}_{fr} \) Total AE energy rate in sliding frictional contact

EHL Elastohydrodynamic lubrication

\( F \) Force

\( F_f \) Frictional force

\( F_n \) Normal force

\( f \) Frequency of the wave motion

\( f_{\text{max}} \) Maximum sampling rate

\( G \) Dimensionless material parameter

\( G_s \) Modulus of rigidity

\( G_{ij}(r,r',t) \) Components of the dynamic elastic Green’s tensor representing displacement in the \( x_i \) direction at \( r \) as a function of time \( t \) due to a unit strength force impulse applied at \( r' \) and with \( t = 0 \) in the \( x_j \) direction

\( G_{ij,k}(r,r',t) \) Solution of the wave equation for a unit force derivative impulse

\( g(t) \) Mother wavelet function

\( H_{\text{min}} \) Dimensionless minimum film thickness in fully flooded oil lubricated elliptical contact

\( h \) Half plate thickness

\( h_c \) Depth of the crack from the surface of the material

\( h_d \) Depth of the dislocation from the surface of the material

\( h_{\text{min}} \) Minimum film thickness

\( h_p \) Thickness of the piezoelectric element of the AE sensor
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>$K_I$</td>
<td>Stress intensity factor in crack opening mode</td>
</tr>
<tr>
<td>$k$</td>
<td>Ellipticity parameter</td>
</tr>
<tr>
<td>$k_w$</td>
<td>Wavenumber of the wave motion</td>
</tr>
<tr>
<td>$k'_w$</td>
<td>Wavenumber of the reflected wave motion</td>
</tr>
<tr>
<td>$m$</td>
<td>Constant determined by the material system and the detection equipment</td>
</tr>
<tr>
<td>$N$</td>
<td>Number of samples</td>
</tr>
<tr>
<td>$\dot{N}_p$</td>
<td>Pulse count rate</td>
</tr>
<tr>
<td>$\sum N_p$</td>
<td>Number of cumulative AE counts</td>
</tr>
<tr>
<td>NLGI</td>
<td>National Lubricating Grease Institute</td>
</tr>
<tr>
<td>$n$</td>
<td>Herschel-Bulkley rheological index</td>
</tr>
<tr>
<td>$n_w$</td>
<td>Integer multiple of wave length</td>
</tr>
<tr>
<td>$\mathbf{n}_1, \mathbf{n}_2, \mathbf{n}_3$</td>
<td>Unit vectors of the components of Burger’s vector</td>
</tr>
<tr>
<td>$P$</td>
<td>Equivalent dynamic bearing load</td>
</tr>
<tr>
<td>$p$</td>
<td>Pressure in the lubricating film</td>
</tr>
<tr>
<td>$\mathbf{p}$</td>
<td>Unit vector which defines the direction of the plane wave motion</td>
</tr>
<tr>
<td>$q$</td>
<td>Wear volume in dry sliding contact</td>
</tr>
<tr>
<td>$R_r$</td>
<td>Reflected wavefront</td>
</tr>
<tr>
<td>$R_x$</td>
<td>Equivalent radius in rolling direction</td>
</tr>
<tr>
<td>$r_d$</td>
<td>Radial distance from a crack tip</td>
</tr>
<tr>
<td>$r_y$</td>
<td>Cylinder radius of the plastic zone in Irwin model</td>
</tr>
<tr>
<td>$r_u$</td>
<td>Radius of the uniform strain zone</td>
</tr>
<tr>
<td>$r$</td>
<td>Place of surface displacement</td>
</tr>
<tr>
<td>$r_{ep}$</td>
<td>Radius of the plastic region in Dugdale model</td>
</tr>
<tr>
<td>$r'$</td>
<td>Place of disturbance inside the material</td>
</tr>
<tr>
<td>$S'_k$</td>
<td>Partial derivative of the vector normal to the surface of the structure</td>
</tr>
<tr>
<td>$s$</td>
<td>Sliding distance</td>
</tr>
<tr>
<td>$T$</td>
<td>Temperature</td>
</tr>
<tr>
<td>$T_d$</td>
<td>Lifetime of dislocation growth</td>
</tr>
<tr>
<td>$t$</td>
<td>Time</td>
</tr>
<tr>
<td>$t_c$</td>
<td>Crack formation time</td>
</tr>
<tr>
<td>$t'$</td>
<td>The point of time of the occurrence of the disturbance</td>
</tr>
<tr>
<td>$U$</td>
<td>Dimensionless speed parameter</td>
</tr>
<tr>
<td>$u$</td>
<td>Surface displacement</td>
</tr>
</tbody>
</table>
\( u_i(r,t) \)  Displacement of the surface in direction \( x_i \)

\( u \)  Displacement vector of a plane wave

\( V_c \)  Volume of the crack

\( V_p \)  Strained material volume inside the material

\( \Delta V_p \)  Increase in the volume \( V_p \)

\( v \)  Dislocation velocity

\( v_c \)  Crack formation velocity

\( \hat{v}_{dc} \)  Rms. value of dislocation velocity

\( \bar{v}_{dc} \)  Average value of dislocation velocity

\( v_s \)  Sliding velocity

\( W \)  Dimensionless load parameter

\( W_t \)  Frictional work in sliding contact

\( \bar{y} \)  Signal mean – value

\( y_i \)  Individual signal value

\( y_{\text{max}}, y_{\text{min}} \)  Signal maximum and minimum values

\( Z_1 \)  Dimensionless viscosity-pressure index

\( \alpha_3 \)  Signal Skewness value

\( \alpha_4 \)  Signal Kurtosis value

\( \delta_c \)  Half of the crack opening

\( \delta_{j1}, \delta_{j2}, \delta_{j3} \)  Kronecker’s delta functions (1 for \( j = k \) and 0 otherwise) of directions \( x_1, x_2 \) and \( x_3 \)

\( \varepsilon \)  Linear strain

\( \varepsilon_{dc} \)  Strain in the dislocation area

\( \dot{\varepsilon}_{dc} \)  Strain rate in the dislocation area

\( \varepsilon_y \)  Yield strain

\( \varepsilon_u \)  Uniform strain

\( \kappa \)  \( v_u / v_1 \)

\( \kappa(\tau_d) \)  Coefficient that depends on the linearity of the sensitivity of the sensor

\( \Lambda \)  Wavelength
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \Lambda_r )</td>
<td>Wavelength of the Rayleigh wave</td>
</tr>
<tr>
<td>( \Lambda' )</td>
<td>Wavelength of a reflected wave</td>
</tr>
<tr>
<td>( \lambda )</td>
<td>Lamé’s elastic constant</td>
</tr>
<tr>
<td>( \lambda_s )</td>
<td>Wavelength of a standing wave</td>
</tr>
<tr>
<td>( \dot{\gamma} )</td>
<td>Shear rate of the lubricant</td>
</tr>
<tr>
<td>( \eta )</td>
<td>Dynamic viscosity of lubricant</td>
</tr>
<tr>
<td>( \eta_p )</td>
<td>Plastic viscosity of a grease</td>
</tr>
<tr>
<td>( \eta_0 )</td>
<td>Dynamic viscosity of lubricant at ( p = 0 )</td>
</tr>
<tr>
<td>( \mu )</td>
<td>Coefficient of friction</td>
</tr>
<tr>
<td>( \mu_L )</td>
<td>Lamé’s elastic constant</td>
</tr>
<tr>
<td>( \mu_2 )</td>
<td>Signal variance</td>
</tr>
<tr>
<td>( \nu )</td>
<td>Poisson’s ratio of a material</td>
</tr>
<tr>
<td>( \nu_a )</td>
<td>Actual viscosity</td>
</tr>
<tr>
<td>( \nu_1 )</td>
<td>Required viscosity for adequate EHL lubrication</td>
</tr>
<tr>
<td>( \Theta_0 )</td>
<td>Reflection angle of a stress wave</td>
</tr>
<tr>
<td>( \rho )</td>
<td>Density</td>
</tr>
<tr>
<td>( \rho_{me} )</td>
<td>Mobile dislocation density</td>
</tr>
<tr>
<td>( \sigma )</td>
<td>Stress</td>
</tr>
<tr>
<td>( \sigma_d )</td>
<td>Standard deviation</td>
</tr>
<tr>
<td>( \sigma_y )</td>
<td>Yield stress</td>
</tr>
<tr>
<td>( \Delta \sigma_{jk} )</td>
<td>Volume stress change associated with the source of the stress wave</td>
</tr>
<tr>
<td>( \tau )</td>
<td>Shear stress of lubricant</td>
</tr>
<tr>
<td>( \tau_d )</td>
<td>Duration time of the AE source</td>
</tr>
<tr>
<td>( \tau_s )</td>
<td>Shear strength of the interfacial layer in frictional contact</td>
</tr>
<tr>
<td>( \tau_0 )</td>
<td>Yield shear stress of the lubricant</td>
</tr>
<tr>
<td>( \Delta \tau_{jk} )</td>
<td>Volume surface traction change associated with the source stress wave</td>
</tr>
<tr>
<td>( \omega )</td>
<td>Angular velocity of a stress wave</td>
</tr>
</tbody>
</table>
1. INTRODUCTION

The maintenance strategies of machines can be divided into corrective, *i.e.* Run-to-Failure Maintenance, and predictive maintenance, which comprises Scheduled Maintenance and Condition-based Maintenance [43]. Which one of these is employed in a firm depends on the policy and practice of the firm as well as on the duties fixed by laws in some branches, for example in nuclear power stations. The Run-to-Failure Maintenance strategy causes minimum maintenance costs – up to the first failure. When the Run-to-Failure principle is employed, the user takes a great risk. This principle is no longer acceptable in large complex machines, like paper or steel milling machines, where the losses, due to an unexpected shut down of production, can be appreciable. Machine manufacturers often recommend the Scheduled Maintenance principle. This method is a traditional maintenance principle and is widely accepted in the process industry. One speaks about “revision” per year or every two years for example in the case of power plants. The Scheduled Maintenance principle may lead to excessive maintenance costs caused by an unnecessarily short time between overhauls and where machines are maintained whether overhaul is needed or not.

Condition-based Maintenance is the modern way to fulfil the needs of productivity and operational safety today. Applying Condition-based Maintenance lead-time to the maintenance action prior to the predicted failure can be attained using suitable measurement methods. The maintenance action can be carried out “Just-in-Time”.

In the area of rotating machinery Condition-based maintenance is executed prevalently by vibration signature monitoring methods. Development work on methods of analysing vibration signals has been active during recent years. This development has been made possible by using the powerful and cost-effective computers which can be used in field environment. Some of the new methods are Joint-Time Frequency Analysis (JTFA) and Wavelet Analysis (WA), which can be applied in the analysis of transient signals and the applications of neural networks for classifying measurement data. New signal analysis softwares give the possibility to build up “personalised analysers” that contain the needed signal analysis features.
Bearings are in a central position in the monitoring of the condition of rotating machinery. Measurement is usually carried out at the points at which a shaft is supported with bearings and hence the vibration generated by the bearing is included in the vibration signal whether the signal is analysed or not. The goal in the development of vibration measurement methods for rolling bearings has usually been to develop techniques for detection of bearing faults in their earliest stage. The characteristic feature of most of these methods is that they are designed to detect cyclic bearing faults. This means that the fault must already exist in the bearing or the bearing is already faulted. Otherwise the method does not give a signal concerning some impending fault.

Considering the situation before a cyclic fault has originated, the main problems are concentrated on the lubrication situation of the bearing. Figure 1 gives the classification and distribution of the reasons for which rolling element bearings did not reach their calculated lifetime [42]. Fifty per cent consists of lubrication problems: poor lubrication and contamination. This problem will also exist in the future unless development work is done.

![Figure 1. Classification and distribution of the causes why rolling element bearings did not reach their calculated lifetime [42].](image)

If the bearing is correctly dimensioned to correspond to the running situation and correctly mounted, problems in lubrication can be considered as the earliest warning of the approach of problems in the condition of the bearing. The problems in lubrication appear as non-cyclic phenomena during the running of the bearing. The results of a research project carried out in 1993 [24] concerning condition monitoring methods of centrifugal pumps in the Finnish process industry showed that the monitoring of the lubrication situation of rolling bearings is usually carried out by key-figure monitoring, see App. 2, Chapter 7, using for example the shock pulse measurement method. The
problem with the methods for monitoring the lubrication situation of rolling bearings is that the methods do not differentiate lubrication mechanisms of the bearings whether it is an oil or grease lubricated bearing.

In this study a new on-line monitoring tool for predictive condition monitoring of grease lubrication of rolling bearings is presented. This tool consists of acoustic emission (AE) measurement technology which is used in monitoring of the lubrication situation of a grease lubricated rolling bearing and of the guidelines for interpreting the AE measurement results. In the present study fundamentals of the behaviour of grease lubrication in rolling bearings has been investigated. The influence of fundamental properties of the grease and the influence of running parameters of the bearing on the AE measurement result have been verified in a real rolling bearing application. Furthermore, an empirical model for prediction of the acoustic emission of a grease lubricated rolling bearing lubricated with clean grease has been formulated. The characteristic features of the suggested measurement technology are: continuous acoustic emission pulse count measurement in a frequency band over 100 kHz, the use of reference AE measurement, measurement or knowledge of the running parameters of the bearing during AE measurement and knowledge of the properties of the lubrication grease.

The primary hypothesis of this investigation is:

“The acoustic emission signal measured from a grease lubricated rolling bearing during its running indicates risks in the lubrication of the bearing. By reducing the level of the acoustic emission the risk of premature failure of the bearing can be reduced.”

1.1 The basis and the goals of the investigation

This investigation is restricted to a consideration of one lubrication area, the grease lubrication of rolling bearings. The reasons why the grease lubrication of rolling bearings has been chosen as the main object of this investigation are:

1. *Greases are the most common lubricants for rolling bearings.* About 90 % of all rolling bearings are lubricated with grease [44,47].
2. *Most of the present methods for measuring and analysing the vibration of rolling bearings are not specified especially for grease lubricated rolling bearings.* The lubrication mechanisms of grease lubricated rolling bearings are less known than the mechanisms in oil lubricated rolling bearings. Grease is a non-newtonian lubricant which has complex behaviour in rolling bearings. The grease lubrication mechanisms do not follow the traditional elastohydrodynamic lubrication (EHL) mechanisms for oil lubricated rolling bearings.

3. *The contaminants of the grease cannot be filtered away like in oil lubricated bearing applications.* The contaminants accumulate in the vicinity of the contact zone, making the lubrication situation worse. On-line particle monitoring of the lubricating grease is very difficult to carry out due to difficulties in obtaining representative grease samples and due to lack of suitable methods of analysis of the grease samples.

The following factors summarise why research work must be carried out to develop condition monitoring methods for grease lubrication of rolling bearings:

− problems related to poor lubrication and contaminants in the lubricant,
− meagre knowledge about the behaviour of grease lubrication of rolling bearings,
− the great relative proportion of grease lubrication in rolling bearings and
− the need to meet the productivity and safety needs of machinery today.

The goals of the present investigation are:

1. *To study and apply acoustic emission measurement technology for predictive condition monitoring of grease lubricated rolling bearings and to test it in laboratory and field environments.*

2. *To determine guidelines for interpreting the AE measurement results by determining the influence of fundamental properties of the grease and by determining the influence of running parameters of the bearing on the lubrication situation in a real rolling bearing application.*

3. *To determine the correlation between contaminants in the lubricating grease and the acoustic emission of a grease lubricated rolling bearing.*
1.2 The structure of the thesis

The thesis consists of a survey and six appended papers. Chapter two deals with the fault modes and with the vibration monitoring methods of rolling bearings and is connected to Paper A.

In Chapter three the phenomenon of acoustic emission and the propagation of stress waves in material are discussed. Furthermore, the chapter deals with the basic features of the measurement techniques for acoustic emission. This chapter is connected mainly to Papers A, B, C and F. Acoustic emission measurement and signal processing techniques are presented in Papers A and C. Furthermore, in Paper C a special case, an extremely slowly rotating rolling bearing, is presented. Paper B deals with applying the AE measurement method in field environment and the use of statistical values in the analysis of the AE signal and Paper F deals with the mounting of the AE sensor.

Chapter four deals with the basic differences between the classical EHL theory for oil lubricated rolling bearings and the grease lubrication of rolling bearings from the viewpoint of lubrication film formation. The chapter deals also with grease parameters which influence the formation of the grease film and the influence of contaminants in the lubricant on the operation of a rolling bearing. At the end of the chapter an interpretation of the AE measurement results is given. This chapter is connected mainly to Papers D, E and F. Paper D deals with the influence of contaminants on starvation of the lubrication situation and the correlation between different types of contaminants in the lubrication grease and acoustic emission generated by the contaminants. Paper E focuses on the grease lubrication mechanisms and the influence of important properties of the grease on the formation of the lubricating grease film. Paper F examines the influence of bearing running parameters on acoustic emission.

The following original contributions were developed in the course of this work:

1. Application of the measurement of acoustic emission for predictive condition monitoring of grease lubricated rolling bearings and verification of the suitability of the method in field environment.
2. Verification of the influence of the fundamental properties, thickener concentration, base oil bleeding rate, base oil viscosity and consistency of the grease on the lubrication situation of the bearing in real rolling bearing application.

3. Verification that the influence of the base oil viscosity of the grease on the lubrication situation in real rolling bearing application does not follow the classical EHL theory for oil lubricated rolling bearings.

4. Formulation of an empirical model for prediction of the acoustic emission pulse count rate of a rolling bearing lubricated with clean grease.

5. Applications of the statistical and time signal methods of analysis of acoustic emission signal in monitoring of the lubrication situation of rolling bearings lubricated with contaminated grease and guidelines for interpretation of the AE measurement results of grease lubricated rolling bearings in different malfunction situations.

1.3 Scientific method

According to Niiniluoto [35], science can be characterised as a systematic totality of the knowledge related to nature, people and society or as a purposeful and systematic pursuit of this kind of knowledge. The systematic totality of the knowledge describes the results of scientific investigation and the systematic pursuit of the knowledge describes the process of scientific investigation. The OECD handbook divides research and development practice into fundamental research, applied research and development work.

A question or problem related to a research goal is always the basis of a research process [35]. According to Haaparanta and Niiniluoto [19], the construction of a scientific argumentation is such that its different stages together give the answer to the original problem, which often is a question that contains the word “why”. In the area of natural science this kind of explanation model is the Hypothetical-Deductive explanation model. Popper [38] calls the Hypothetical-Deductive explanation model also the Deductive Testing of Theories. In the process from a new idea or hypothesis conclusions are drawn by means of logical deduction, which means testing the conclusions among themselves and with other theories and accepting and rejecting the conclusions. Popper [39] describes the growth of knowledge with a process: problem –
theoretical trial solution – criticism of the solution – new problem. In experimental sciences and studies inductive explanation methods are widely used [38]. In an inductive explanation method universal statements are inferred from singular ones or from numerous statements using induction. According to Popper [38] this kind of scientific explanation model leads to a so-called “problem of induction”, which means the question: Are inductive inferences justified and under what conditions?

Because the area of grease lubrication of rolling bearings is not known very well, there is not much knowledge available based on experimental results of the causalities between different factors concerning grease and running situation, especially from real bearing installations. For that reason the main emphasis of the scientific method of this study has been on the Hypothetical-Deductive explanation model in finding out the causalities in the behaviour of the grease lubrication of rolling bearings. As support for the explanations, the measurement results of grease lubrication obtained with EHL ball-on-disc and disc-on-disc devices, which have not been carried out in the present investigation, have been used. The conclusions have been compared with the classical elastohydrodynamic theory for oil lubricated rolling bearings. Because there is a great number of parameters and combinations of them which influence the behaviour of grease lubrication in rolling bearings, the investigation does not pursue to present universal generalisations using induction. The measurement method used in the investigation, i.e. acoustic emission method, can be classified as a holistic measurement method in the case of rolling bearings. This is because it is not only the contact zone that generates acoustic emission, but acoustic emission can be generated by contacts between all the different components of the bearing. The complex behaviour of grease lubrication in rolling bearings and the features of the measurement methods used have led to the scientific view in this investigation being also slightly holistic.

2. VIBRATION SIGNATURE MONITORING OF ROLLING BEARINGS

In this chapter the classification of rolling bearing fault modes in terms of the type of vibration they generate during the running of the bearing is presented. The chapter comprises the classification of the vibration measurement methods and short
descriptions of some low frequency measurement methods in addition to the methods that are discussed in Paper A.

2.1 Multi-Parameter Monitoring

Vibration measurements occupy a central position in on-line condition monitoring of rolling bearings. Other on-line monitoring methods for rolling bearings are for example temperature measurement with contacting sensors or thermographic photographing. The measurement of airborne vibration, sound, has been applied in on-line fault monitoring of rolling bearings in the low frequency band as well as in the ultrasonic frequency band. Commonly used off-line monitoring methods are the wear particle analysis methods, e.g. ferrography and the particle counting method, as well as visual inspection of damaged bearings.

None of the condition monitoring methods for rolling bearings is suitable for all the fault modes or running situations of the bearings. The best result is achieved by simultaneous use of various monitoring methods, which is called Multi-Parameter Monitoring [42]. The methods, which are used simultaneously, should be chosen so that they are suitable for the running situation of the bearing and that they indicate different types of fault modes. The rotational speed of the bearing and the fault mode being monitored can be considered criteria in choosing adequate vibration signature monitoring methods for rolling bearings.

2.2 Fault modes of rolling bearings from the aspect of vibration monitoring

The fault modes which cause vibration during the running of the bearing can be roughly divided into cyclic and non-cyclic fault modes.

Cyclic faults

A fault which causes vibrations with a regular period during running of the bearing is classified as a cyclic bearing fault. Cyclic faults in rolling bearings typically appear in the raceways, in rolling elements or in rolling element cages. The cycle time of the vibration period depends on the geometry, on the rotational speed and on the direction
of the bearing load. The equations for calculation of the cycle time or the corresponding frequency, which is called the bearing fault frequency, are commonly used [5] and nowadays integrated in bearing data banks in commercial computer softwares for rolling bearing monitoring.

Non-cyclic faults

Faults which cause random vibration or non-periodic vibration bursts during running of the bearing are classified as non-cyclic faults. A non-cyclic vibration in rolling bearings is generated typically by severe wear or early stage faults in the raceways or rolling elements, contaminants in the lubricant, inadequate lubrication and sliding of the bearing components [42].

2.3 Monitoring methods

Rao [42] classifies the vibration monitoring methods of rolling bearings, depending on the vibration frequency, into low frequency methods (frequency range from 0 to 20 kHz), middle frequency methods (from 20 kHz to 100 kHz) and high frequency methods (over 100 kHz).

Low frequency vibration is mechanical vibration of the construction or a part of the construction and is typically caused by mechanical unbalance or misalignment of machine components. Depending on the machine construction, structural resonances can occur also in the middle frequency area. The natural frequency of a mounted bearing assembly can occur in the low frequency as well as in the medium frequency region. Correspondingly, medium and high frequency vibration is mainly caused by the changes inside the material [42]. In this thesis the measurement methods are divided into low and high frequency methods, keeping the limiting frequency in the area of 20 kHz.

2.3.1 Low frequency methods

The monitoring of cyclic faults is complicated by the low intensity, the burst-like character and high frequency of the vibration. The enveloping method, described in
Paper A, is one of the most common low frequency methods used today in monitoring the cyclic faults of rolling bearings. There exist some modifications of the enveloping method, which have been described by Berry [4] and Randall [41]. Berry uses the name High Frequency Enveloping (HFE) method, because the highest frequency in the method can be as high as 40 kHz.

The most common low frequency methods for monitoring rolling bearing vibration are presented in Paper A. In the following, some methods which are less used in field environment are described. Kim, Lim and Cheoung [27] have presented the use of the Moving Window method that is presented schematically in Figure 2. In the method, a time window is moved in steps over the time signal measured from the faulted bearing. A spectrum, by using scan windowing techniques [40], is calculated from every window. By studying the periodicity of the spectra of the stepped windows, the fault frequencies of the bearing can be defined. Kim, Lim and Cheoung state that in the Moving Window method the signal to noise ratio is better than for example in the enveloping method.

Figure 2. Schematic description of the Moving Window method presented by Kim, Lim and Cheoung [27].

Mechefske and Mathew [32,33] have presented the Parametric Spectra method for monitoring slowly rotating rolling bearings. In the method, a signal which contains a couple of fault cycles of the bearing is measured. This signal is modified into a continuous signal using the regression analysis method and the spectrum is calculated from the manipulated signal. The authors have stated that the result is an estimate of the
supposed to exist in the signal that is under analysis. The Wavelet Transform compares the similarity of the mother wavelet and the real measured signal. Li and Ma [28] have applied the WT in monitoring local defects in a rolling bearing. As mother wavelet they have used the function

\[ g(t) = \exp(-c_{w1} t) \sin(c_{w2} t) \] (1)

which describes the sinusoidal, exponentially damped signal. The coefficients \( c_{w1} \) and \( c_{w2} \) are chosen so that the mother wavelet resembles as closely as possible the excitation impulse of the bearing fault. The advantage of the Wavelet Transform, according to the authors, is the better resolution in time and in the frequency domain compared with the traditional FFT-based methods.

2.3.2 High frequency methods

The high frequency measurement methods in the condition monitoring of rolling bearings are based on the measurement of stress waves. Depending on the fault mode and the generation mechanism of stress waves, high frequency vibrations can occur in a wide frequency range. The high frequency measurement methods included in this thesis are presented in Paper A and the basis of the acoustic emission measurement technology is presented in more detail in Chapter three of the survey part and in the appended papers of this thesis.
3. ACoustic Emission

In this chapter the phenomenon of acoustic emission and the basic features of the technique for its measurement are presented. The chapter includes also the most important features of the measurement installation used in the present investigation. The description of the AE measurement installation and the technical data of the measurement devices are given in Appendix 1.

3.1 The phenomenon of acoustic emission

The first documented observation of the phenomenon which later came to be called acoustic emission was made by an Arabian alchemist, Jabir ibn Hayyan, who is known also under the name Geber, in the eighth century A.D. In his work "Summa Perfectionis Magisterii", which was published in Latin in the year 1545, he wrote that Jupiter (tin) gives off a "harsh sound" or "crashing noise". He also wrote that Mars (iron) "sounds much" during forging. This "sound" of iron probably comes from the formation of martensite during cooling [52,55]. The first result obtained with measurement devices of the observation of acoustic emission was published by the German scientists Friedrich Förster and Erich Scheil in the year 1936 [16]. They measured the noise following from the formation of martensite in steel which contained 29 % nickel.

The investigations of acoustic emission carried out by Joseph Kaiser at the Technische Hochschule München in Germany are considered the genesis of current acoustic emission technology. Kaiser published his Doctor-Ingenieur dissertation from his investigations in the year 1950 [26]. His greatest discovery in the research work was the irreversibility phenomenon of acoustic emission. This phenomenon is nowadays known as the Kaiser effect and is one of the basic phenomena of acoustic emission. The Kaiser effect is described in Paper C, Chapter 1. Kaiser presented also the division of the acoustic emission phenomenon into two parts: continuous emission and burst emission. He concluded in his studies that acoustic emission is generated by friction mechanisms between grains in polycrystalline materials and by deformation of intergranular cracks [26,52].
3.2 Source mechanisms

Acoustic emission means the stress waves inside a material which are generated by rapid release of local instabilities. The reference [8] divides the instabilities or the source mechanisms of acoustic emission into four groups:

- dislocation movements,
- phase transformations,
- friction mechanisms,
- crack formation and extension.

Acoustic emission can be generated by many other mechanisms than those mentioned, which can make it difficult to monitor the phenomenon under consideration, as described in Paper B. Scruby [48] states that in practice there are only a few basic point sources, that each of these generates a characteristic pattern of ultrasound and that more complicated sources can be built up by combining the different source types. Depending on the source mechanism, acoustic emission can appear as continuous emission (typically friction mechanisms) and burst emission (typically crack formation). The released energy appears as elastic stress waves inside the material. The elastic waves propagate as spherical wavefronts to the surrounding material. While propagating in the material, the wavefront diffuses when it travels through nonhomogeneous areas and finally strikes the surface of the material and is reflected back from it. In consequence of the internal friction in the material the wave motion is transformed mainly to heat and dies away.

The time dependence of the source is an important factor in determining the detectability of the AE source. A slow growing of a crack produces a weak signal, while rapid crack growth of the same size produces a sharp and strong signal, Figure 3. The weak signal can be swamped by background noise [48].

This investigation concentrates on the acoustic emission of rolling bearings lubricated with clean and contaminated greases. Thus it can be assumed that the source mechanisms of the acoustic emission are the friction mechanisms and material
deformation mechanisms like dislocation, crack nucleation and crack extension mechanisms in the contaminants and in the surface lawyer of the material.

Figure 3. (a) Rapid crack extension produces a sharp and strong stress wave burst, while (b) slow extension of a crack produces a wide and weak stress wave [48].

3.3 Propagation of stress waves

3.3.1 Waves in infinite medium

In infinite, homogenous, isotropic and linearly elastic material, only two kinds of stress waves can exist: longitudinal stress waves, or dilatational waves, and transversal waves, or distortional waves. The displacement vector $u$ of a plane wave which is propagating with phase velocity $c$ is defined by equation [1]

$$u = f(x \cdot p - ct)d$$

(2)

where $x$ is a position vector and $d$ and $p$ are unit vectors defining the direction of the motion. The displacement vector $u$ is governed by the differential equation of motion [1]

$$\mu_L \nabla^2 u + (\lambda + \mu_L)\nabla \nabla \cdot u = \rho \ddot{u}$$

(3)

where $\rho$ is the density of the material and $\mu_L$ and $\lambda$ are Lamé’s elastic constants, which are defined as
\[
\mu_L = \frac{E}{2(1-\nu)} \quad ; \quad \lambda = \frac{E\nu}{(1+\nu)(1-2\nu)}
\] (4)

where \( E \) is the modulus of elasticity and \( \nu \) the Poisson’s ratio of the material.

The solution of equation (3) gives two different kinds of propagation models for the plane waves: the longitudinal plane wave, Figure 4a, which travels with velocity \( c_L \) and the transversal wave, Figure 4b, which travels with velocity \( c_T \), see App. 2. The phase velocities of these waves depend on the density and Lamé’s elastic constants of the material but they are independent of the wavelength of the motion.

Figure 4. a) Propagation of longitudinal plane wave, or dilatational wave, and b) transversal wave, or distortional wave, inside material [3].

### 3.3.2 Waves in half-space

In material bounded by half-space the stress waves appear in the surface of the material as surface waves or Rayleigh waves, Figure 5. A specific feature of Rayleigh waves is that they decay exponentially in the layers under the surface of the material.

Figure 5. Surface wave or Rayleigh wave in the material. The wavelength of the Rayleigh wave is marked with \( \lambda_r \) [1].
Using the exponential model for the displacement $u$, the Rayleigh phase velocity $c_R$ can be obtained in the case of a plane wave from the differential equation of motion (3) and presented with equation [1]

$$(2 - \frac{c_R^2}{c_L^2})^2 - 4(1 - \frac{c_R^2}{c_L^2})^{1/2}(1 - \frac{c_R^2}{c_T^2})^{1/2} = 0$$

(5)

Since the displacement components $u_1$, *i.e.* the displacement component in the direction of $x_1$, and $u_2$, *i.e.* the displacement component in the direction of $x_2$, are $90^\circ$ out of phase, the trajectories of the particles in the surface of the material are ellipses, see Figure 5. At a free surface the normal component is about 1.5 times the tangential displacement [1].

In Table 1 values of density and longitudinal, transversal and Rayleigh wave phase velocities are given. The velocity of the Rayleigh wave in the material is slightly lower than the velocity of the transversal wave. The phase velocity of the Rayleigh wave as well as the phase velocities of the longitudinal and transversal waves are independent of the wavelength and therefore also independent of frequency.

<table>
<thead>
<tr>
<th>Material</th>
<th>$\rho$ ($\text{kg/m}^3$)</th>
<th>$c_L$ (m/s)</th>
<th>$c_T$ (m/s)</th>
<th>$c_R$ (m/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>steel</td>
<td>7800</td>
<td>5900</td>
<td>3200</td>
<td>3000</td>
</tr>
<tr>
<td>copper</td>
<td>8900</td>
<td>4600</td>
<td>2300</td>
<td>2100</td>
</tr>
<tr>
<td>aluminium</td>
<td>2700</td>
<td>6300</td>
<td>3100</td>
<td>2900</td>
</tr>
</tbody>
</table>

Lamb waves are the fourth important mode of the stress waves which appear in material. A Lamb wave is a plane wave which is born in a material body with two boundaries where the material has a thickness of a couple of wavelengths of the stress wave [3]. The wave builds up from two surface waves which are in the same or in opposite phase, Figure 6. Lamb waves differ from longitudinal, transversal and Rayleigh waves in that the phase velocity of Lamb waves depends on its wavelength and frequency.
3.3.3 AE waves in practical measurement

Real constructions are seldom so regular and "clean" as is assumed in theoretical examinations of stress wave propagation. Materials can contain non-homogeneties, material dimensions are finite and the way from the source point to the measuring sensor can contain boundaries which the travelling stress wave has to pass. The movement of the surface of the material, surface waves, which is measured in the acoustic emission method consists mainly of waves which are "like" Lamb waves. The complication of real constructions in the case of wave propagation causes dispersion of the wavefronts. Dispersion of the wavefront means that the phase velocity varies with frequency and is called velocity dispersion [37].

Figure 7 illustrates the dispersion phenomenon in a plane. The line $PBE$ describes a wavefront of wavelength $\Lambda$ travelling in the direction of $x_1$ and line $ADEF$ describes another wavefront ($R_r$) of wavelength $\Lambda'$ which has been reflected from each boundary. If the wavefront $PBE$ interferes with the reflecting wavefront which has travelled the additional path $BDE$, it is required that
\[ \frac{1}{\Lambda'} \times \text{length } BDE = n_w \]  

(6)

where \( n_w \) is an integer number (integer multiple of wavelengths). From Figure 7 we get

\[ \text{length } BDE = \frac{2h}{\cos\theta_0} + \frac{2h \cos 2\theta_0}{\cos\theta_0} = 4h \cos\theta_0 \]  

(7)

The phase velocity of the wave motion in the direction \( x_1 \) is \( c \) and the phase velocity of the reflected wave motion (\( R_r \), Figure 7) is \( c_T \). The ratio of the phase velocities can be expressed

\[ \frac{BC}{PQ} = \frac{c_T}{c} = \sin\theta_0 \]  

(8)

The angular velocity \( \omega \) of both wavefronts is the same and we can write

\[ \omega = k_w c = k_w' c_r \]  

(9)

and

\[ k_w = \frac{2\pi}{\Lambda} = k_w' \sin\theta_0 = \frac{2\pi \sin\theta_0}{\Lambda'} \implies \Lambda = \frac{\Lambda'}{\sin\theta_0} \]  

(10)

where \( k_w \) and \( k_w' \) are the wavenumbers of the wave motion in the direction of \( x_1 \) and in the direction of the wave motion of \( R_r \), respectively.

Substituting equation (7) into (6) and taking into account equation (10) we get

\[ k_w \cot\theta_0 = \frac{n_w \pi}{2h} \]  

(11)

Using the function \( \cot\theta_0 \) and taking into account the equation (11) we can write
\[
\cot \theta_0 = \left(\frac{1}{\sin^2 \theta_0} - 1\right)^{\frac{1}{2}} = \left[\left(\frac{c}{c_T}\right)^2 - 1\right]^{\frac{1}{2}}
\]  
(12)

Finally, the equation (11) with the help of the equation (12) can be written [1]

\[
\left(\frac{c}{c_T}\right) = 1 + \left(\frac{n_w \pi}{2k_w h}\right)^2
\]  
(13)

where \( k_w \) is the wavenumber \( (k_w = 2\pi / \Lambda = 2\pi f / c) \). The equation (13) shows that the phase velocity \( c \) depends on the wavenumber \( k_w \) or the frequency \( f \) of the wave motion, except when \( n_w \) has the value of 0.

When a stress wave burst with a narrow frequency band propagates in the material, it spreads due to dispersion. The dispersed wavefront forms an envelope which propagates with a speed \( c_e \). Velocity \( c_e \) is called the group velocity and is defined by equation [1]

\[
c_e = c + k_w \frac{dc}{dk_w}
\]  
(14)

The group velocity of the surface waves is observed due to the dispersion of the wavefront in the measurements of the acoustic emission. In Figure 8a the velocity dispersion curves of the first four Lamb wave modes in a steel plate are presented.

Figure 8. (a) Group velocities \( (c_e) \) of the first four Lamb wave modes in steel plate. Horizontal unit \( (\text{mm x MHz}) \) is defined as following: \( (\text{material thickness}) \times (\text{detected frequency}) \), (b) \( S_0 \) and \( S_1 \) are symmetric and \( a_0 \) and \( a_1 \) are asymmetric modes [37].
In Figure 8 the horizontal axle has a unit of (mm x MHz), which is defined as follows: (material thickness) x (detected frequency). From Figure 8a it can be observed that if the detected frequency is constant, the group velocity approaches Rayleigh wave velocity when the material thickness increases.

3.4 Acoustic emission models

The displacement of the surface due to internal (volume stress) and surface (traction) sources can be written as follows [56]

\[
 u_i(r,t) = \int d\mathbf{r}' \int G_{ij,k}(r,\mathbf{r}', t - t') \Delta \sigma_{jk}(\mathbf{r}', t') dt' - \int dS' \int G_{ji,k}(r,\mathbf{r}', t - t') \Delta \tau_{jk}(\mathbf{r}', t') dt' \tag{15}
\]

where \( u_i(r,t) \) is the displacement of the surface, see Figure 9, and \( G_{ij}(r,\mathbf{r}', t) \) are the components of the dynamic elastic Green’s tensor representing displacement in the \( x_i \) direction at \( r \) as a function of time \( t \) due to a unit strength force impulse applied at \( \mathbf{r}' \) and at time \( t = 0 \) in the direction \( x_j \) [56]. Green’s tensor is the solution to the wave equation for the unit force impulse source. \( G_{ij,k}(r,\mathbf{r}', t) \) is the corresponding wave equation solution for a unit force derivative impulse, \( \Delta \sigma_{jk} \) and \( \Delta \tau_{jk} \) are the volume stress and surface traction changes associated with the source, \( S' \) is a vector normal to the surface of the structure and \( S'_{k'} \) is its partial derivative. In order to detect the acoustic emission, the displacement of the surface has to be greater than the sensitivity of the AE sensor.

![Figure 9. Schematic diagram of the acoustic emission process [56].](image_url)
3.4.1 Dislocation model

The criterion for the smallest detected surface displacement, the detectability criterion, of the surface epicenter where the stress waves arrive can be written [56]

\[
2\pi h_d d_j \leq \kappa(\tau_d) \left[ \frac{2c_T^2 b_1 n_3}{c_L^3} \delta_{j1} + \frac{(b_1 n_3 + b_2 n_1) \delta_{j2} + (b_2 n_3 + b_3 n_2) \delta_{j3}}{c_T} \right] \max_t \dot{A}(t) \tag{16}
\]

where \( d_j \) is the minimum displacement of the surface that can be detected with the measurement sensor, \( h_d \) is the depth of the dislocation from the surface in the direction of surface normal, see Figure 10, \( b_1, b_2 \) and \( b_3 \) are the magnitudes of the Burger’s vector, \( n_1, n_2 \) and \( n_3 \) are the unit vectors of the Burger’s vector, \( \delta_{j1}, \delta_{j2}, \) and \( \delta_{j3} \) are the Kronecker’s delta functions (1 for \( j = k \) and 0 otherwise) and \( \dot{A}(t) \) is the derivative of the dislocation area function, e.g. the rate of the change in the area. \( \kappa(\tau_d) \) is a coefficient that depends on the linearity of the sensitivity of the AE sensor and \( \tau_d \) is the duration of the source, in this case the duration of the dislocation.

![Figure 10. Schematic diagram showing the dislocation at an angle of 45° to the normal of the surface [51].](image)

The first term in brackets in equation (16) corresponds to the motion of the surface at the epicenter in the direction of the surface normal associated with the arrival of a longitudinal wave. The second term in brackets corresponds to tangential motion at the epicenter associated with the arrival of a shear wave. The sensor used in the measurements of the AE is insensitive in the tangential direction. Therefore the first term in brackets is dominant and the equation (16) simplifies to the form
where \( d_3 \) is the minimum surface displacement in the direction of surface normal. Assuming that the dislocation grows from zero to its final radius \( a \) with constant velocity \( v \) we can simply obtain a rough estimate for the minimum detectable dislocation radius, when one dislocation is born. With a constant dislocation velocity the lifetime \( T_d \) of the dislocation is

\[
T_d = \frac{a}{v} \tag{18}
\]

and we can write

\[
A = \pi a^2 = \pi T_d^2 v^2 \quad \Rightarrow \quad \max_t \dot{A}(t) = \dot{A}(T_d) = 2\pi T_d^2 v^2 = 2\pi a v \tag{19}
\]

Denoting \( \kappa(\tau_d) = 1 \) signifies a constant sensitivity of the sensor in the frequency band from 0 to \( 1/\tau_d \), which is not true with real sensors. The detectability criterion can now be written in the form [50,51]

\[
|d_3| = d_3 \leq \frac{c_f^2 b_3}{2h_d c_L^3} 2\pi a v = \frac{c_f^2 b_3 a v}{h_d c_L^3} \tag{20}
\]

A typical sensitivity value for a resonant-type piezoelectric sensor is \( 10^{-14} \) m in the direction of the measurement axle (the axle perpendicular to the surface of the material), so we can put \( d_3 = 10^{-14} \) m. For example with following example values [50]: steel \( c_L = 5960 \) m/s, \( c_f = 3200 \) m/s, \( b_3 = 3 \times 10^{-10} \) m, \( v = 200 \) m/s and \( h = 0.05 \) m (the distance from the bearing outer raceway to the mounting place of the transducer) a rough estimate of the minimum detectable dislocation radius, when one dislocation is born, is

\[
a = 172 \mu m \tag{21}
\]
According to the model, dislocations of smaller size can be detected if they either grow at greater velocity or many dislocations are situated close to each other expanding together over the same time interval so that their combined radii exceed the minimum detectable dislocation radius [50].

Long and Huazi [31] have presented that the AE energy rate $\dot{E}_d$ caused by the motion of a large number of dislocations can be expressed as

$$\dot{E}_d \propto \rho_{\text{me}} (\hat{v}_{dc} \hat{a}_{dc})^2$$  \hspace{1cm} (22)

where $\rho_{\text{me}}$ is the mobile dislocation density and $\hat{v}_{dc}$ and $\hat{a}_{dc}$ are the rms values of dislocation velocity and the distance of the motion of the dislocation. By expressing the strain rate $\dot{\varepsilon}_{dc}$

$$\dot{\varepsilon}_{dc} = b \rho_{\text{me}} \overline{v}_{dc}$$  \hspace{1cm} (23)

where $\overline{v}_{dc}$ is the average dislocation velocity at strain $\varepsilon_{dc}$ and assuming that

$$\hat{v}_{dc} = \overline{v}_{dc} \quad \text{and} \quad \hat{a}_{dc} = \overline{a}_{dc}$$  \hspace{1cm} (24)

where $\overline{a}_{dc}$ is the average distance of the motion of the dislocation, the energy rate of acoustic emission can be written as proportional to the strain rate, the average dislocation velocity and the average distance of the motion of dislocations as follows [31]

$$\dot{E}_d \propto \dot{\varepsilon}_{dc} \overline{v}_{dc} \overline{a}_{dc}$$  \hspace{1cm} (25)

3.4.2 Crack formation model

In the case of crack formation in the material that is under stress, the edges of the crack are moving into the position of lower stress state. When the stress in the area of the crack sinks, energy is released, generating acoustic emission. Scruby [49] classifies the
acoustic emission generated by crack formation into primary events and secondary events. Figure 11 describes the mechanisms connected with both of these events.

![Schematic illustration of the AE generation mechanisms of a crack](image)

Figure 11. Schematic illustration of the AE generation mechanisms of a crack [49].

Scruby [50] has presented that the peak amplitude $d_c$ of the surface displacement (perpendicular to the surface) caused by the longitudinal stress wave in the case of crack formation with crack opening mode (mode I, see App. 2) is

$$d_c = \frac{V_c}{\pi c t_c h_c}$$  \hspace{1cm} (26)

where $t_c$ is the crack formation time, $V_c$ is the volume of the crack and $h_c$ is the depth of the crack from the surface of the material. Scruby [50] has supposed that the crack volume has ellipsoidal form and that in this case, according to the linear elastic model (Irwin model, see App. 2.), the volume can be expressed as

$$V_c = \frac{4\pi a_c^2 \delta_c}{3}$$  \hspace{1cm} (27)

where $a_c$ is half of the crack length (first and second radius of the ellipsoid) and $\delta_c$ is half of the crack opening (the third radius of the ellipsoid). Figure 12 illustrates the volume $V_c$ of the crack. In this case the ellipsoid surface in plane $x$-$y$ becomes circular.
By substituting $\delta_c$ [50]

$$\delta_c = \frac{2(1-v^2)\sigma}{E} a_c$$  \hspace{1cm} (28)$$

into the equation (26) and assuming the constant crack formation velocity as $v_c = a_c / t_c$, the equation for the minimum detectable crack size can be written in the form [50]

$$a_{c,\text{min}} = \sqrt{\frac{3c_1 Eh_c d_3}{8(1-v^2)v_c \sigma}}$$  \hspace{1cm} (29)$$

For example, with the same values as in the case of the dislocation and in addition with $E = 210*10^9$ N/m$^2$, $h_c = 0.05$ m, $v = 0.3$, $v_c = 500$ m/s and $\sigma = 600*10^6$ N/m$^2$ (stress of $600*10^6$ N/m$^2$ is a typical value of the yield stress for a case hardening steel), a rough estimate of the minimum detectable crack size is

$$a_{c,\text{min}} = 1 \mu\text{m}$$  \hspace{1cm} (30)$$

The AE energy rate generated by a crack formation $\dot{E}_c$ is proportional to the square of the surface displacement amplitude and is [31]

$$\dot{E}_c \propto \sigma^2 \frac{a_c^6}{l_c^2}$$  \hspace{1cm} (31)$$
Bassim [2] has presented the plastic deformation model and the linear elastic fracture model for the AE pulse count rate generated by crack formation. In the plastic deformation model the AE pulse count rate is associated with plastic deformation of the material ahead of the tip of the crack.

**Plastic deformation fracture model**

In the plastic deformation model the following assumptions are made:

- A metal or alloy gives the highest AE pulse count rate when loaded to the yield strain.
- The shape of the plastic zone is determined according to the linear elastic fracture mechanics concept, the Irwin model (see App. 2.), as a cylinder of radius $r_y$ (radius of the yield strain zone) and it can be expressed in plain stress conditions [2,23] as

$$r_y = \frac{1}{2\pi} (\frac{K_I}{\sigma_y})^2$$  \hspace{1cm} (32)

where $K_I$ is the stress intensity factor in crack opening mode and $\sigma_y$ is the yield stress.
- The strains at the crack tip vary as $r_d^{-1/2}$, where $r_d$ is the radial distance from the crack tip.
- The AE pulse count rate $\dot{N}_p$ is proportional to the increase $\Delta V_p$ for the volume $V_p$ of the material that is strained between the yield strain $\varepsilon_y$ and the uniform strain $\varepsilon_u$ (the uniform strain corresponding to a stress lower than 0.4 $\sigma_y$ [23]), or

$$\dot{N}_p \propto \Delta V_p$$ \hspace{1cm} (33)

The increase in the volume is

$$\Delta V_p = \pi (r_y^2 - r_u^2)B$$ \hspace{1cm} (34)
where \( r_y \) and \( r_u \) are the radii of zones according to the yield strain and uniform strain and \( B \) is the thickness of the plastic zone. With the equation (32), by substituting \( \sigma = \varepsilon E \), the equation (34) can be written in the form [2]

\[
\Delta V_p = \frac{B}{4\pi} \left[ \frac{\varepsilon^4_u - \varepsilon^4_y}{E \varepsilon_u \varepsilon_y} \right] K_i^4 \quad \text{or} \quad \Delta V_p \propto K_i^4
\]  

Finally, it can be seen that the AE pulse count rate generated by the crack formation in the case of the plastic deformation model is proportional to the stress intensity factor according to the equation

\[
\dot{N}_p \propto K_i^4
\]  

The linear elastic fracture model

In the linear elastic fracture model the AE pulse count rate is proportional to the area of the elastic-plastic boundary ahead of the crack tip, or [2]

\[
\dot{N}_p = D r_{ep}
\]  

where \( D \) is a proportionality constant and \( r_{ep} \) is the radius of the plastic area that can be expressed with the Dugdale (see App. 2.) model [2,23] as

\[
r_{ep} = \frac{a_c}{2} \left[ \sec \left( \frac{\pi \sigma}{2 \sigma_y} \right) - 1 \right]
\]  

Finally, the AE pulse count rate generated by the crack formation in the case of the plastic deformation model can be expressed as

\[
\dot{N}_p = \frac{a_c}{2} D \left[ \sec \left( \frac{\pi \sigma}{2 \sigma_y} \right) - 1 \right]
\]
3.4.3 Friction model

Jiaa and Dornfeld [25] have determined with pin-on-disk experiments in dry sliding contact the following empirical relation for the AE energy rate $\dot{E}_f$ generated from a sliding frictional contact

$$\dot{E}_f \propto (\tau_s A_s v_s)^{\frac{m}{2}}$$  \hspace{1cm} (40)

where $\tau_s$ is the shear strength of the interfacial layer, $A_s$ is the actual area of a contact, $v_s$ is the sliding velocity and $m$ is a constant determined by the material system and the detection equipment and can be assumed to be 1.0. Concerning the equation (40), Mullins et al. [34] point out that with increasing sliding velocity, or with stronger interfacial layers, the average frequency and amplitude of AE events increase. These increases produce a non-linear increase in the energy detected by the AE monitoring equipment. A complex contact situation is a sum of incremental small contacts and the total AE energy $\dot{E}_{\text{fr}}$ can be expressed as [34]

$$\dot{E}_{\text{fr}} = a_f (F_f v_s)^{\frac{m}{2}} dA_{\text{tot}} = a_f (F_f v_s)^{\frac{m}{2}} dA_{\text{tot}}$$  \hspace{1cm} (41)

where $a_f$ is a proportionality factor, $A_{\text{tot}}$ is the total contact area and $F_f$ is the frictional force.

Lingard and NG [30] have investigated the relationship between the AE signal and wear rate of a specimen in dry sliding contact with a two-disc wear machine (Amsler type A135). They have stated that the AE output is more likely to relate to frictional forces than to wear parameters. The AE output here is understood as AE pulse count rate or total AE pulse count value. From experiments they observe that in unlubricated friction the cumulative AE count $\sum N_p$ is related to frictional work $W_f$ as

$$\sum N_p = b(W_f)^{\alpha_m}$$ \hspace{1cm} (42)
where the multiplier $b$ depends on the type of the transducer and on the settings of the measurement device. The value of the exponent $a_m$ is dependent on the material and its value is predominantly in the range 1.2 to 1.5. Lingard and NG derived the equation (42) in the form

$$\dot{N}_p = a_m b(W_r)^{a_m-1} \frac{dW_r}{dt} = a_m b(\mu F_n)^a v_s s^{(a_m-1)}$$

(43)

where $\mu$ is the coefficient of friction, $F_n$ is the normal force, $v_s$ is the sliding velocity and $s$ is the sliding distance. By assuming the wear volume per unit sliding distance and the sliding velocity to be constant, Lingard and NG found the relationship between the AE pulse count rate and the wear volume $q$ in dry sliding contact

$$\dot{N}_p \propto q^{(a_m-1)}$$

(44)

Lingard and NG state that the relationship in the equation (44) is strange because it means that the AE pulse count rate should be related to the material already removed. They explained the equation in the case of a two-disc wear machine, stating that $\dot{N}_p$ is related to the wear scar length of the specimen, because the width in the specimen, the disc, is constant. One other explanation for the increase in the AE pulse count rate during the wear of the specimen can be that the particles, as a result of the wear volume of the specimen, accumulate in the sliding race and increase the AE pulse count rate when they enter the contact zone. This kind of observation is made by Jiaa and Dornfeld [25] with pin-on-disc tests, although they have measured the AE rms value but not the AE pulse count rate.

3.5 Frequency of the AE vibration

All measurement systems have some kind of detection threshold. Only those acoustic emission signals whose amplitudes exceed the detection threshold of the measurement system can be detected. The detection threshold is determined on the sensitivity of the measurement sensor and on the properties of the measurement unit.
In examination of the smallest dislocation and crack size, in previous chapters, which can be detected it was assumed that the measurement sensor has linear output in the frequency band where acoustic emission exist ($\kappa(\tau_d) = 1$, see page 22). The minimum detected dislocation radius 172 $\mu$m corresponds to frequency of 1.2 MHz with the example values presented in page 22. Correspondingly the minimum detected crack size, see page 25, corresponds to the frequency of 500 MHz. In here it is assumed the AE signal as pure sine wave signal. Although, the situation described here is idealised it gives some view of the sensitivity of the AE measurement method to detect faults and supports its use in predictive condition monitoring of rolling bearings.

Because of the complicated source mechanisms and dispersion of the stress waves while propagating inside the material the acoustic emission detected from the surface of the material has a wide frequency band width. The norm for AE is that no two signals have exactly the same frequency content [3]. Generally, the spectrum of an AE burst will contain numerous peaks at different frequencies, see Figure 18. A short impulsive source will generate an emission with a broad frequency content. The spectrum of a delta function, a large pulse with zero width, will contain all frequencies equally. After propagation from source point to the detection sensor the resulting signal is far from the idealised sine wave signal [3].

### 3.5.1 Choice of the frequency band width

In order to obtain some understanding of the sensitivity of the AE measurement system used in the present study a test with a new, unlubricated bearing (washed clean from slushing oil with white spirit) and with an oil lubricated bearing was carried out. The results (Figure 13) of this test showed strong AE activity with the unlubricated bearing while with the oil lubrication hardly any acoustic emission could be detected.
Figure 13. Acoustic emission of a deep groove ball bearing. a) Bearing lubricated with oil (SAE 80W/90) and b) unlubricated bearing. Results are from different measurements but with the same measurement device settings. Rotational speed was 450 rpm.

In this investigation the sensors were piezoelectric, resonance-type sensors, model PAC R15. In a piezoelectric AE sensor the thickness and the diameter of the piezoelectric element are the main characteristic factors which define the properties of the sensor [15]. The thickness defines the frequencies at which the sensor has its highest output. At the point of highest output there arise standing waves with wavelengths of \( \lambda_1 = 2h_p, \lambda_2 = 2h_p/3, \lambda_3 = 2h_p/5 \) and so on (\( h_p \) is the thickness of the piezoelectric element) inside the piezoelectric element and the element attains its natural frequency or the resonant frequency. The resonance frequencies of the sensors which were used in the measurements were 150 kHz, 240 kHz and 350 kHz. The frequency responses vary slightly between individual sensors. A typical frequency response graph of the AE sensors which were used in the measurements is shown in Figure 14.

Figure 14. Frequency response graph of the AE sensors (type PAC R15) used in the measurement of this investigation.
The AE signal was high-pass filtered to erase the low frequency disturbances from the measurement. Environmental conditions can disturb the AE measurement strongly. These arise from the numerous mechanisms that can be sources of acoustic emission and from the non-directionality of the propagation of the AE waves. The frequency bands that some disturbances can generate are shown in Figure 15 [7].

![Figure 15](image)

Figure 15. Frequency bands caused by disturbances from environment. BW is measurement band width. (A) available signal, (B) transducer response from true signal, (C) flow noise interference, (D) mechanical noise interference, (E) electromagnetic interference [7].

In the case of the small test rig used in this investigation, the influence of the mechanical disturbances was reduced by rubber and plastic insulators, described in Paper E, Chapter 3. In the case of the large test rig, the acoustic emission generated by the support bearings is always present in the measurement results, but the support bearings and the type of grease for lubricating the support bearings were the same in all the measurements. The electric motors were isolated from the shaft in both test rigs with rubber insulators or by using a belt-drive system. Williams [57] points out that the electrical or electromagnetic interferences are not easy to eliminate. This was observed in the field measurement also in this study, see Paper B.

Low-pass filtering of the AE signal was needed to adjust the signal adequately for the data acquisition unit. The maximum sampling rate of the AE time signal measurement unit was 500 kHz, so the 350 kHz area was in principle too high a frequency for the sampling unit. In Figure 16 the AE time signals measured without filtering and with filters of central frequency 150, 240 and 350 kHz and a band width of 10 kHz are
shown. The results show that the background noise without filtering is rather high and that the signal induced from a bearing fault becomes very weak at higher frequencies. The measurement was carried out with a large bearing test rig, a faulted bearing and clean grease. The description of the bearing fault and the grease type are given in Paper C, Chapter 4.1.

Figure 16. AE time signal from faulted bearing. a) Without filtering and with 10 kHz wide band-pass filter of central frequencies b) 150, c) 240 and d) 350 kHz. The results are from the same measurement.

FFT analysis showed that the dominating frequency that raises the background noise level of the unfiltered signal, Figure 16a, is 8 kHz. When the background noise is filtered with a 30 kHz high-pass filter, the spectrum of the filtered background noise signal is rather flat in the frequency band from 50 kHz to 180 kHz, so the frequency content of the background noise is very wide, Figure 17. The weakening of the signal in frequencies in the area of 200 kHz is probably due to the frequency response of the sensor.
Figure 17. Spectrum of the high-pass filtered background noise signal. The high-pass filtering frequency is 30 kHz. The unfiltered signal is shown in Figure 16a.

One emission burst and its frequency content in the case of the unfiltered signal is shown in Figure 18. In the spectrum, frequencies 15, 43, 59, 66, 85, 125, 137, 156 and 183 kHz can be detected. The frequency of 43 kHz which is dominant in the spectrum is in the frequency area of shock pulse vibration and is generated by the impulse when the rolling element passes the bearing fault.

Conclusions of the examination of the high frequencies which arise in the running of a damaged and undamaged rolling bearing are that the frequency spectrum of an undamaged bearing is wide, in the range of 50 – 180 kHz, and the fault in the bearing causes vibrations in the band under 50 kHz. The choice of the frequency for the measurement is a compromise. Lower frequencies have to be filtered away so that they do not disturb the measurement. At higher frequencies the energy of the vibration becomes low and very high frequency makes the processing of the measurement data troublesome. Based on the factors presented here and also in the appended Papers, the acoustic emission measurements were carried out in the present study using the
frequency band from 100 kHz to 240 kHz and filtering the signal with a narrow band-pass filter. The amplifier used in all measurements of the study was a low-noise type amplifier with 60 dB amplification.

The size range of the thickness of the material where the transducers were mounted was 50 mm. In this case, according to Figure 8a, with a measurement frequency of 150 kHz the group velocities of 3000 m/s, 2900 m/s, 2700 m/s and 2300 m/s can be detected. These wave velocities are near the velocity of a Rayleigh surface wave. This kind of situation should be propitious for measurement of the AE activity because the Rayleigh waves are perpendicular to the surface of the material and the sensor is most sensitive in that direction.

3.6 Measurement and signal analysis techniques

The measurement and signal analysis techniques of the acoustic emission have been described in Papers B, C and F. In this investigation the most used signal analysis method was the pulse count method. The reason for the use of this method was the long measurement times, which were derived from the time-dependence of the lubrication situation in a grease lubricated rolling bearing, as discussed in Paper E. The characteristic feature in the measurement of acoustic emission is that it is nondirectional. The sensor must be mounted on a clean, smooth surface via couplant grease, but the direction of the sensor is not important. The use of wave guides, as in the measurements in this investigation, makes the fixing of the sensor tight. The wave guide also eliminates base strain or "base bending" of the sensor, which is discussed in Paper F.

The diameter of the piezoelectric element can be the reason for which the response of the sensor for defined surface waves is low. In the theoretical case where the pure sine wave form of a Rayleigh wave is acting on the surface of the sensor so that one or more full wavelengths match the diameter of the piezoelectric element, the output of the element is zero. This is the so-called aperture effect and the sensor cannot detect this kind of surface wave. Figure 19 illustrates the situation of zero output of the sensor and the situation with a sensor of smaller diameter.
3.6.1 Calibration

The basic construction of a piezoelectric sensor is simple and very reliable. The drift of the sensor is insignificant. However, errors in measurements can be caused for example by bad measurement cables, cable connectors or incorrect mounting of the sensor. In this investigation the absolute values of the AE activity was not under examination, so the calibration of the sensor was not such an important element. The condition of the measurement devices was confirmed by testing the measurement chain before and after measurements. The test method was the simple "Pencil Lead Break" method according to the ASTM standard number E 976-84 [54]. In the method a pencil lead is broken against the surface where the sensor is mounted. The breaking generates an AE burst signal for testing the measurement chain. In the method the lead is supported by a special guide ring during the breaking, see Figure 20.

Higo and Inaba [22] have shown a modification of the ASTM method in which the shape of the teflon guide ring is slightly different. The lead used in the "Pencil Lead Break" method in this investigation was 0.7 mm Pentel HB lead. The point against which the
lead was broken in the small bearing test rig was the front surface of the bearing housing opposite the side of the AE sensor, see Figure 2 in Paper E. In the large bearing test rig the breaking point was on the surface of the bearing housing, see Figure 6 in Paper C. The breaking of the lead induces a burst-type AE signal. The AE signal generated by breaking the lead was analysed using the pulse count method and in these tests the gain and threshold settings were always the same. One typical result of the measurement chain test using the "Pencil Lead Break" method is shown in Figure 21.

![Figure 21](image)

Figure 21. Result of the measurement chain test using the "Pencil Lead Break" method. The result contains six pencil breakings and the respective number of pulses per each emission burst.

4. GREASE LUBRICATION IN ROLLING BEARINGS

This chapter deals with the basic differences between the classical EHL theory for oil lubricated rolling bearings and the grease lubrication of rolling bearings from the viewpoint of lubrication film formation. The chapter deals also with parameters which influence the formation of the grease film thickness and the influence of contaminants in the lubricant on the operation of a rolling bearing. At the end of this chapter an interpretation of the AE measurement results is discussed and the guidelines for interpreting the AE measurement results and an empirical model for prediction of the AE pulse count rate when clean grease is used as lubricant are presented.
4.1 Classical EHL theory and grease lubrication

The Reynolds equation governs the pressure distribution in fluid film lubrication. The terms of the Reynolds equation are the Poiseuille term, which describes the lubricant flow rates due to the pressure gradients in the film, the Couette term, which describes the flow rates due to surface velocities, the Squeeze term, which describes the flow rates due to squeezing motion and the Local expansion term, which describes the flow rates due to local expansion, see App. 2.

On the basis of the Reynolds equation and taking into account the dynamic viscosity \( \eta \) (cP) according to the Roelands model, defined by the dimensioned equation \[20\]

\[
\log \eta + 1.200 = (\log \eta_0 + 1.200)(1 + \frac{p}{2000})^{Z_1}
\] (45)

where \( \eta_0 \) is the dynamic viscosity (cP) at \( p = 0 \), \( p \) is the pressure in the lubricating film (bar) and \( Z_1 \) is the viscosity-pressure index (dimensionless), as well as the elastic deformation of the material in the contact zone, the dimensionless minimum film thickness \( (H_{\min}) \) in a fully flooded oil lubricated situation in elliptical contact, like in a deep groove ball bearing, can be derived to the form of Hamrock and Dowson \[20\]

\[
H_{\min} = \frac{h_{\min}}{R_x} = 3.63U^{0.68}G^{0.49}W^{-0.073}(1 - e^{-0.68k})
\] (46)

where \( U \) is the dimensionless speed parameter, \( G \) is the dimensionless material parameter, \( W \) is the dimensionless load parameter, \( k \) is the ellipticity parameter, \( h_{\min} \) is the minimum film thickness and \( R_x \) is the equivalent radius in rolling direction. The film thickness calculated with the equation (46) may be considered the maximum value of film thickness. In real bearing applications the lubrication situation is often starved to some degree, which reduces the film thickness.

The assumptions in the equation (46) regarding the lubricant and lubrication situation are the newtonian behaviour of the lubricant, defined as [21]
\( \tau = \eta \dot{\gamma} \) \hspace{1cm} (47)

where \( \tau \) is the shear stress and \( \dot{\gamma} \) the shear rate of the lubricant, and the fully flooded lubrication situation. In the case of grease lubrication, the assumptions regarding the newtonian behaviour and the fully flooded situation are not realised. If the equation (46) is used for calculating the film thickness in the case of grease lubrication, based on the base oil viscosity of the grease the obtained value can be considered a very rough estimate of the film thickness.

Unlike the shear stress in lubricating oils, which is linearly proportional to the shear rate, the behaviour of the shear stress in lubricating greases is non-linear and is often described with the Herschel-Bulkley model \([21,58,18]\)

\[ \tau = \tau_0 + \eta_p (\dot{\gamma})^n \] \hspace{1cm} (48)

where \( \tau_0 \) is the yield shear stress, \( \eta_p \) is the plastic viscosity and \( n \) is the rheological index or Herschel-Bulkley shear stress index. The non-linearity of the shear stress appears particularly with low shear rate values. Palacois and Palacois \([36]\) have argued that with high shear rates the greases will behave like their base oils, like shown in Figure 22.

![Figure 22. Shear stress (\( \tau \)) of grease and oil as a function of shear rate (\( \dot{\gamma} \)).](image)

The fully flooded film thickness in grease lubrication is generally greater than that of the oil when oil is the base oil of the grease, as Cann and Lubrecht \([11]\) have shown in measurements involving point contact with an EHL ball-on-plate device. Cann and Lubrecht explain that the greater film thickness is due to the presence of the thickener in the inlet, contributing to a higher effective viscosity.
In practical grease lubricated bearing applications the fully flooded situation may appear only in certain running conditions, like in the starting situation if the bearing is full of grease or just after re-greasing of the bearing. The fully flooded situation may last only a few minutes from the start, as described in Paper E. In the AE measurements in this study the fully flooded lubrication situation, or a situation “like” that, was best observed when grease was added to the bearing when the bearing was running. A measurement result from that kind of situation is seen in Paper F, Figure 5, the point "Re-greasing". If the bearing is completely full of grease and starts to rotate directly with high rotational speed, the running of the bearing is shaky, which induces a high AE level. This may be the explanation for the situation which was observed in the measurement in Paper F, Chapter 4.4.

After greasing of the bearing the fully flooded running time depends also on the type of the bearing, but after a running time of some minutes the lubrication situation in a grease lubricated rolling bearing becomes starved. Cann [9] has stated that starvation with grease lubrication is analogous to the fluid film case, except for the complication of the time-dependent rheology of the grease. It can be concluded that the characteristic features of grease lubrication in rolling bearings are starvation of the rolling contact, the non-newtonian behaviour of the grease and the time dependence of the running situation.

4.2 Parameters in formation of the grease film thickness

The grease lubrication mechanisms of rolling bearings are discussed in Paper E of this thesis. The important grease parameters in the formation of the lubricating film, according to present knowledge, are the base oil viscosity, the thickener structure and concentration, the bleeding rate of the base oil and the consistency of the grease [9]. The influence of the base oil viscosity was the opposite of that given by the traditional EHL theory for oil lubricated rolling bearings. The AE measurement results indicated a decreasing starvation of the lubrication situation with decreasing base oil viscosity.

The other parameters which influence the formation of the film thickness are the type and design of the bearing and the running parameters. The influence of the vibration of the bearing, which can be considered ad one running parameter, on the grease
lubrication situation may be complex. Vibration can cause the bulk grease to move inside the bearing, reducing the degree of starvation of the lubrication situation. On the other hand the vibration may increase the acoustic emission level of the bearing, which means more contacts between surfaces. A measurement result presented in Figure 23 describes this phenomenon in a deep groove ball bearing. At the beginning of the measurement in Figure 23 the vibration velocity overall level was 1.4 mm/s. With external excitation the vibration level was raised to 10 mm/s. The increased vibration of the bearing generated a higher AE pulse count rate. When the vibration was lowered to the level 1.8 mm/s, the AE level first dropped for a time lasting a couple of minutes to a very low level and then rose back to the “normal” level. The reason for the very low AE level just after the end of the vibration overall level of 10 mm/s is the movement of the bulk grease that degreases the starvation of the lubrication situation.

![Figure 23](image)

Figure 23. Influence of the vibration of the bearing on the lubrication situation of a deep groove ball bearing. The vibration velocity overall values (1.4 mm/s, 10 mm/s and 1.8 mm/s) are measured in axial direction from the test bearing housing, see Figure 2 in Paper E.

In the case of vibration, the viscous damping properties of the lubricating grease may have some effect on the AE level of the bearing. The influence of the grease damping properties on the lubrication situation of the rolling bearing was not a primary goal of this investigation. However, in order to gain some understanding of the differences in the damping properties of greases, a comparison between two very different greases was carried out. The greases which were compared were the greases G1 and G6, whose compositions are explained in Paper E, Table 1. These greases were chosen for the comparison because the AE levels of the bearing lubricated with these greases were very different, see Figures 5 and 6 in Paper A.
The comparison of the viscous damping properties of the greases was carried out by measuring the attenuation of the amplitude of the angular velocity of a free torsional vibration system. The torsional vibration of the system was damped by the viscous friction of the grease. The viscous damper was composed of a steel rod with a diameter of 10 mm pushed 80 mm into a large grease sample, see Figure 24. The temperature in the measurements was 25 °C.

Figure 24. Schematic illustration of the installation for measuring the attenuation of the amplitude of the free torsional vibration system damped with the viscous friction of the grease.

The measurement results are shown in Figure 25. The damping of the grease G1 is clearly stronger than that of the grease G6. Correspondingly, the AE level of the grease G6 was in the measurements about 25 times higher than that of the grease G1. It is possible that the different damping properties of these greases may have some influence on the different AE levels of the greases. The influence of the vibration properties, viscous damping and nominal vibration properties on the lubrication of rolling bearings could be one interesting research area in grease lubrication in the future.
Figure 25. Attenuation of the amplitude of the angular velocity of a free torsional vibration system damped with the viscous friction of a grease. a) Grease G1 and b) Grease G6. The compositions of the greases are explained in Table 1 of Paper E.

4.3 Contaminants in the lubricating grease

In Figure 26 the effect of a contaminant in the lubricant on the pressure distribution in EHL line contact is described. The contaminants cause high pressure peaks which travel through the contact zone. These pressure peaks cause additional fatigue stress in the rolling elements and the raceways, which reduces the lifetime of the bearing.

Figure 26. Influence of a contaminant in the lubricant on the pressure distribution in EHL line contact where \( u_1 \) and \( u_2 \) are the velocities of the surfaces.

Figure 27 presents the result of an endurance test carried out by Sayles and MacPherson [45]. They have investigated the influence of filtration of the oil on the fatigue life of the bearings. The small change in the fatigue life with filter ratings from 1 to 3 \( \mu \)m is due to the limit of the filter effectiveness.
Dwyer-Joyce [13] have suggested three mechanisms for the deformation of the debris particles when they enter the contact zone in oil lubricated rolling contact: ductile particles deform plastically and contact surfaces elastically or plastically, Figure 28 (a), ceramic materials fracture and fragments contact surfaces plastically, Figures 28 (b) and (c), small ceramic particles remain undamaged and contact the surfaces plastically, Figure 28 (d).

Figure 28. Schematic illustration of the entry of different kinds of debris into the contact zone in rolling contact [13].

Figure 29 illustrates the behaviour of the debris in oil lubricated rolling contact based on the studies with an EHL ball-on-disc device [13]. In grease lubricated rolling contact the flow of the particles may not be as strong as in the oil lubricated situation, but particles
on the central line may behave the same way in grease lubricated contact as in oil lubricated contact. Furthermore, in grease lubrication particles do not float away from the contact zone with lubricant but accumulate in both sides of the raceway. The accumulation of the particles may impede the bleeding of the base oil from the thickener into the contact zone and the starvation of the lubrication situation will increase. Because the particles stay in the vicinity of the contact zone, they will be ground smaller and smaller, which will raise the AE level. This can explain the AE measurement result obtained in field measurement. In the latter, after two months of running the acoustic emission RMS (Root Mean Square) value rose from a value of 0.060 V to the value of 0.092 V and the shape of the time signal was very smooth, Figure 9 in Paper B. This indicates that the particles probably were ground into very small pieces.

Figure 29. Schematic illustration of the flow of the debris around oil lubricated rolling contact [13].

Dwyer-Joyce points out that the entry of the debris into the contact zone in real bearing application depends a lot for example on the bearing type. When the width of the contact zone becomes wider, a greater number of the particles become entrained into the contact zone and a smaller number is swept to the sides.

The effect of cleaning the bearing of debris and re-greasing it with new clean grease on the AE of a deep groove ball bearing was investigated in Paper D, Chapter 4.3. The cleaning and re-greasing decreased the AE pulse count rate to a half of the level that it was with grease containing debris. However, the pulse count rate after cleaning and re-greasing was about 30 times higher than that of the bearing before running with contaminated grease. This higher AE pulse count rate is a result of the rough surface
roughness of the raceways of the bearing caused by the running of the bearing with contaminated grease. SEM pictures from the outer raceway of the test bearing as new, after running-in and after running with contaminated grease are shown in Figure 30. The debris in the test was quartz and the contaminant concentration was 0.2 weight percentage. In the new bearing, Figure 30a, only machining grooves can be observed, while the surface after running with contaminated grease is strongly damaged, Figure 30c.

Figure 30. SEM picture from the outer raceway of the test bearing, (a) new bearing, (b) bearing after running-in and (c) bearing after running with contaminated grease.

5. INTERPRETATION OF AE MEASUREMENT RESULTS

When the measurement results are compared with the results achieved with EHL interferometer devices using point contact [9,10], some differences appear. The explanation for these differences is found in the different operation situations between the EHL apparatus and a real rolling bearing. There is a great difference in the time available for lubricant replenishment in the contact zone. In EHL ball-on-disc test devices there is only one contact per revolution, but in a real bearing the rolling elements follow each other with a space of only some millimetres. In addition, in typical rolling bearing applications the rolling speeds are higher than in the EHL devices. The AE measurement indicated an increasing AE level, which means increasing starvation of the lubrication situation with increasing rotational speed of the bearing, as shown also in the EHL measurement results presented by Cann [9] for a starved grease lubricated situation. The interpretation of the AE measurement result in that case is not so unambiguous. In the normal range of use of rotating machinery the amplitude of
mechanical vibration rises with increasing rotational speed up to some nominal frequency. Furthermore, with higher rotational speed there occur more contacts per time unit between bearing components that increase the acoustic emission. The influence of the rotational speed on the AE of the bearing is presented in Paper F, Chapter 4.2. Thus, the higher AE level with higher rotational speed may not be considered as indicator of a lower film thickness or increasing starvation of the lubrication situation. The influence of the rotational speed on the film thickness in a fully flooded situation is not easy to verify with a real bearing installation because the fully flooded situation is not the normal lubrication situation in a grease lubricated rolling bearing. Cann [9] has measured increasing film thickness with increasing rolling speed in EHL apparatus in point contact in the case of a fully flooded grease lubrication situation.

In addition to the factors, the influence of which has been verified in this investigation, there are other factors which influence also the lubrication situation and the acoustic emission of grease lubricated rolling bearings. In grease these factors are e.g. solid particles of the grease thickener, insoluble ingredients from the manufacturing of the thickener, solid additive particles and liquid additives.

5.1 Empirical model for prediction of the AE pulse count rate

The dependence of the different grease properties on each other makes it difficult to determine the influence of one separate property on the acoustic emission of the bearing. Furthermore, most of the properties depend on the running temperature and other running parameters. An empirical model was formulated for predicting the AE pulse count rate in a grease lubricated rolling bearing. With this model, a key-figure called AE severity ($A_{es}$), is calculated. The $A_{es}$ indicates the direction of the change on the AE pulse count rate of the bearing in a stable running situation or with constant bearing load and rotational speed when the lubricating grease is clean. In practical applications the AE severity value can be used for predicting the AE level of the bearing if for example the lubrication grease is changed. In the model one grease is defined as “base grease” and the $A_{es}$ is calculated in relation to this “base grease”. In this case the grease” was the grease G1 (NLGI grade 2 lithium soap grease with mineral base oil with a viscosity of 150 mm²/s at a temperature of 40 °C), see Paper E.
The AE severity model is formulated based on the measurement results presented in Figs. 3 - 6 in Paper E, and verified with the measurement results presented in Fig. 6 in Paper F. As a result, it was found that the model gave the direction of the change of the AE pulse count rate when verifying the model with the measurement results of Paper F. The model indicated also the different AE pulse count rates between the greases Grease 1 and Grease 2, whose measurement results are presented in Figs. 5 and 6 in Paper A.

5.1.1 Construction of the model

Based on the measurement results and on the literature survey of clean greases of this study, the model was formulated with the following principles:

- When the thickener concentration increases, the starvation of the lubrication situation increases. The parameter in the model is "thickener concentration".
- When the bleeding rate increases, the starvation of the lubrication situation decreases. The parameter in the model is “base oil bleeding rate”.
- When the viscosity of the base oil is very high, the starvation of the lubrication situation increases and when the viscosity is very low, the film thickness does not carry the bearing load. The parameter in the model is "kappa (κ) value".
- When the consistency of the grease is low, the grease does not stay in the vicinity of the contact zone, which increases the starvation of the lubrication situation. The consistency of the grease is defined with the penetration of the grease. The parameter in the model is “penetration”.

*The influence of running temperature and scaling of the grease properties*

The bleeding rate increases with increasing temperature along a quadratic function. In addition to the temperature, the consistency of the grease influences the base oil bleeding rate of the grease [46]. The influence of the temperature on the bleeding rate was modelled with a quadratic function

\[
\text{Bleeding rate} = \frac{1}{2} B_{r,40} \left( \frac{T}{40} \right)^2 + a_0
\]  

(49)
where $B_{40}$ is the bleeding rate at the standardised measurement temperature of 40°C and $a_9$ is a factor for scaling the bleeding rate to correspond to the measured bleeding rate value at the temperature of 40°C. $T$ is temperature in degrees °C.

The influence of the temperature on the base oil viscosity was modelled with the Vogel formula [53]

$$\eta = a_6 e^{\frac{a_7}{T-a_8}}$$  \hspace{1cm} (50)

where $\eta$ is the dynamic viscosity and $a_6$, $a_7$ and $a_8$ are constants which were determined from the base oil viscosity measurement results of the greases G1 and G6. The base oil viscosity model of the grease G1 was used also with the greases G2 – G4 and G8 – G10 and the base oil viscosity model of the grease G6 was used also with the grease G7, scaling in all cases the model to correspond to the value of the viscosity at the temperature of 40°C. $T$ is the absolute temperature.

The influence of the viscosity of the base oil on the AE pulse count rate of the bearing was modelled with the kappa value according to the measurement results shown in Fig. 7 in Paper E, scaled so that the AE pulse count rate with the kappa value of 1.7 corresponded to the relative AE pulse count rate value of 1. The model is:

$$\text{Relative AE pulse count rate} = \frac{2}{(\text{Kappa})^{0.33}} + 0.2 \times (\text{Kappa})^{0.935} - 1$$  \hspace{1cm} (51)

The influence of the temperature on the penetration of the grease was modelled based on the measurement results of the penetration at different temperatures. The model was done by linear line fitting to the measurement results. The penetration model of the grease G1 – G4 and G8 – G10 is

$$\text{Penetration at temperature } T = 0.83T + a_{10}$$  \hspace{1cm} (52)

and of the greases G6 and G7
Penetration at temperature $T = 0.93T + a_{10}$ \hfill (53)

where the constant $a_{10}$ is for scaling the penetration to correspond to the penetration at the standardised measurement temperature of 25 °C separately with each grease.

*Scaling of the model*

The model was scaled so that the thickener concentration, base oil bleeding rate and penetration were scaled with the values of the test grease G1, defined at standardised measurement temperatures, bleeding rate at the temperature of 40°C and penetration at the temperature of 25°C, see Table 1 in Paper E. So the grease G1 was a “base grease”.

The $A_{es}$ model for estimating the AE pulse count rate was formulated as follows:

$$
A_{es} = a_1 \times \left( \frac{\text{Concentration}}{8} \right)^{a_2} \times \left( \frac{\text{Bleeding rate}}{4.3} \right)^{a_3} \times \left( \frac{2}{(\text{Kappa})^{0.33} + 0.2 \times (\text{Kappa})^{0.935} - 1} \right)^{a_4} \times \left( \frac{\text{Penetration}}{265} \right)^{a_5}
$$ \hfill (54)

where $A_{es}$ is the key-figure of acoustic emission severity, $a_1 - a_5$ are coefficients that are defined with the AE pulse count rate measurement results of the five test greases G1, G2, G4, G6 and G7. These measurement results are presented in Paper E and are tabulated in Table 2 in the column “AE pulse count rate (pulses/s)” of this chapter. The units of factors of the equation (54) are shown in Table 2.

In determining the coefficients $a_1 - a_5$ the AE pulse count rates are scaled with the value of the grease G1 so that the AE pulse count rate of grease G1 is 1 and correspondingly $A_{es}$ is 1. Then the coefficients $a_1 - a_5$ are resolved approximately with the five non-linear equations with Mathcad software. The following values for the coefficients were obtained: $a_1 = 5$, $a_2 = 0.14$, $a_3 = 1$, $a_4 = 1.4$ and $a_5 = 4.5$. Table 2 contains the data for calculating the values of the coefficients. The base oil bleeding rate of the greases G8, G9 and G10 were missing and the $A_{es}$ values are calculated with the bleeding rate value of 1. In these cases the model indicated the lowest AE rate for the grease G10, which is in agreement with the measurement results.
Table 2. The values for calculating the coefficients $a_1 - a_5$ for the model. Values marked with $^1$ are only for calculating the $A_{es}$ values, not real values. Values marked with $^2$ are reduced from the measurement results to correspond to the $C/P$ value of 30, see Chapter 4.5 Summary of experiments, in Paper E.

<table>
<thead>
<tr>
<th>Used in formulation of the model</th>
<th>Grease No.</th>
<th>Thickener concentration (%)</th>
<th>Oil bleeding rate (%)</th>
<th>Kappa value</th>
<th>Penetration (mm/10)</th>
<th>Bearing temperature (C)</th>
<th>Rotational speed (rpm)</th>
<th>Bearing load (C/P)</th>
<th>AE pulse count rate (pulses/s)</th>
<th>Relative AE pulse count rate</th>
<th>$A_{es}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>x G1</td>
<td>8</td>
<td>3.5</td>
<td>10.6</td>
<td>265</td>
<td>32</td>
<td>800</td>
<td>30</td>
<td>2300</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>x G2</td>
<td>5</td>
<td>11.5</td>
<td>10.6</td>
<td>292</td>
<td>32</td>
<td>800</td>
<td>30</td>
<td>1000</td>
<td>0.43</td>
<td>0.43</td>
<td></td>
</tr>
<tr>
<td>G3</td>
<td>2.5</td>
<td>33.8</td>
<td>10.6</td>
<td>378</td>
<td>32</td>
<td>800</td>
<td>30</td>
<td>1500</td>
<td>0.65</td>
<td>0.42</td>
<td></td>
</tr>
<tr>
<td>x G4</td>
<td>5</td>
<td>24.9</td>
<td>1.6</td>
<td>354</td>
<td>30</td>
<td>800</td>
<td>30</td>
<td>500</td>
<td>0.22</td>
<td>0.21</td>
<td></td>
</tr>
<tr>
<td>x G6</td>
<td>14</td>
<td>3.5</td>
<td>14.7</td>
<td>393</td>
<td>46</td>
<td>820</td>
<td>30</td>
<td>7000</td>
<td>3.04</td>
<td>3.00</td>
<td>2.04</td>
</tr>
<tr>
<td>G7</td>
<td>15</td>
<td>2.8</td>
<td>3.0</td>
<td>244</td>
<td>46</td>
<td>820</td>
<td>30</td>
<td>1000</td>
<td>0.43</td>
<td>0.42</td>
<td></td>
</tr>
<tr>
<td>G8</td>
<td>7</td>
<td>$^1$</td>
<td>10.2</td>
<td>326</td>
<td>36</td>
<td>800</td>
<td>30</td>
<td>6000</td>
<td>2.61</td>
<td>8.40</td>
<td></td>
</tr>
<tr>
<td>G9</td>
<td>14</td>
<td>$^1$</td>
<td>10.2</td>
<td>284</td>
<td>36</td>
<td>800</td>
<td>30</td>
<td>7000</td>
<td>3.04</td>
<td>5.07</td>
<td></td>
</tr>
<tr>
<td>G10</td>
<td>7</td>
<td>$^1$</td>
<td>10.2</td>
<td>244</td>
<td>36</td>
<td>800</td>
<td>30</td>
<td>500</td>
<td>0.22</td>
<td>2.37</td>
<td></td>
</tr>
</tbody>
</table>

5.1.2 Testing of the model

The model was tested with the measurement results of Paper F. The measurements of Paper F were carried out by raising the temperature of a bearing lubricated with the grease G1. The bearing was of the same type as in the present study, the rotational speed was 800 1/min and $C/P$ was 10. The measurement results, the AE pulse count rate with regard to the bearing temperature, are shown in Table 3 in the column “AE pulse count rate (pulses/s)” and graphically in Figure 31 a.

The values of the grease parameters and the corresponding $A_{es}$ values at different temperatures are defined with the $A_{es}$ model. The results are shown in Table 3 in the column “AE pulse count rate (pulses/s)” and the $A_{es}$ value with regard to temperature is shown graphically in Fig. 31 b. The relative acoustic emission and $A_{es}$ values are scaled in the position of the minimum AE rate (200 pulses/s).
Table 3. The AE pulse count rate, the relative AE pulse count rates, the $A_{es}$ key-figure values and the values of the grease parameters of a deep groove ball bearing with regard to the bearing temperature. The grease parameters and $A_{es}$ are calculated with the model.

<table>
<thead>
<tr>
<th>Thickener concentration (%)</th>
<th>Oil bleeding rate (%)</th>
<th>Kappa value</th>
<th>Penetration (1/10 mm)</th>
<th>Bearing temperature (°C)</th>
<th>Rotational speed (rpm)</th>
<th>Bearing load (C/P)</th>
<th>AE pulse count rate (pulses/s)</th>
<th>Relative AE pulse count rate</th>
<th>$A_{es}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>8</td>
<td>2.9</td>
<td>16.5</td>
<td>265</td>
<td>25</td>
<td>800</td>
<td>10</td>
<td>25000</td>
<td>125</td>
<td>1.86</td>
</tr>
<tr>
<td>3.3</td>
<td>12.0</td>
<td>269</td>
<td>30</td>
<td></td>
<td></td>
<td></td>
<td>12000</td>
<td>60</td>
<td>1.20</td>
</tr>
<tr>
<td>3.7</td>
<td>8.9</td>
<td>273</td>
<td>35</td>
<td></td>
<td></td>
<td></td>
<td>3000</td>
<td>15</td>
<td>0.81</td>
</tr>
<tr>
<td>4.3</td>
<td>6.6</td>
<td>277</td>
<td>40</td>
<td></td>
<td></td>
<td></td>
<td>2500</td>
<td>12.5</td>
<td>0.58</td>
</tr>
<tr>
<td>4.6</td>
<td>5.6</td>
<td>280</td>
<td>43</td>
<td></td>
<td></td>
<td></td>
<td>2000</td>
<td>10.0</td>
<td>0.49</td>
</tr>
<tr>
<td>5.5</td>
<td>3.8</td>
<td>286</td>
<td>50</td>
<td></td>
<td></td>
<td></td>
<td>1000</td>
<td>5.0</td>
<td>0.37</td>
</tr>
<tr>
<td>6.9</td>
<td>2.3</td>
<td>294</td>
<td>60</td>
<td></td>
<td></td>
<td></td>
<td>300</td>
<td>1.5</td>
<td>0.32</td>
</tr>
<tr>
<td>8.7</td>
<td>1.4</td>
<td>302</td>
<td>70</td>
<td></td>
<td></td>
<td></td>
<td>200</td>
<td>1.0</td>
<td>0.34</td>
</tr>
<tr>
<td>10.7</td>
<td>0.9</td>
<td>311</td>
<td>80</td>
<td></td>
<td></td>
<td></td>
<td>300</td>
<td>1.5</td>
<td>0.39</td>
</tr>
<tr>
<td>13.0</td>
<td>0.6</td>
<td>319</td>
<td>90</td>
<td></td>
<td></td>
<td></td>
<td>1000</td>
<td>5.0</td>
<td>0.47</td>
</tr>
<tr>
<td>14.2</td>
<td>0.5</td>
<td>323</td>
<td>95</td>
<td></td>
<td></td>
<td></td>
<td>3000</td>
<td>15.0</td>
<td>0.51</td>
</tr>
<tr>
<td>15.5</td>
<td>0.4</td>
<td>327</td>
<td>100</td>
<td></td>
<td></td>
<td></td>
<td>5000</td>
<td>25.0</td>
<td>0.56</td>
</tr>
</tbody>
</table>

Fig. 31. (a) The relative AE pulse count rate and (b) the $A_{es}$ key-figure values of a deep groove ball bearing with regard to the bearing temperature. The number values are tabulated in Table 3. Bearing rotational speed was 800 rpm and $C/P$ value was 10.

The results show that the model has the same kind of shape as the AE rate of the bearing with regard to the temperature of the bearing. However, there is a great difference between the relative AE rate values and $A_{es}$ values. The minimum of the...
relative $A_{es}$ value is very near the point of the minimum AE rate. In this case with the $A_{es}$ key-figure the direction of the change of the AE rate can be predicted.

In the Figure 32 the influence of the base oil bleeding rate, the base oil viscosity (kappa value) and the consistency (penetration) on the $A_{es}$ are presented separately. Examination of the model allows the following conclusions to be drawn regarding the influence of the basic properties of the grease on the lubrication situation based on the $A_{es}$:

- The factor of greatest influence when indicating the lubrication situation is the base oil viscosity of the grease. It indicates increasing starvation at low temperatures and insufficient film thickness at higher temperatures.
- Increasing bleeding rate decreases starvation of the lubrication situation.
- Low consistency strongly starves the lubrication situation and eliminates the influence of increasing bleeding rate.
- The influence of only the thickener concentration by itself on the lubrication situation is low, but it must be pointed out that it is typically connected with the consistency of the grease.

![Graphs showing the influence of grease properties on the $A_{es}$ key-figure value](image)

Fig. 32. Influence of grease properties on the $A_{es}$ key-figure value in the case of a deep groove ball bearing with regard to the bearing temperature. (a) The base oil bleeding rate, (b) the base oil viscosity (kappa value), (c) the consistency (penetration) of the grease. The $A_{es}$ of the whole model is presented in Figure 31 b.
5.2 AE interpretation guidelines

The guidelines for interpreting the AE measurement results are summarised in Table 4. The table contains the factors whose influence on the acoustic emission in a grease lubricated rolling bearing have been investigated in this thesis. The grades of the influence of the factors must be considered separately in individual factor groups.
Table 4. Guidelines for interpreting the AE measurement results of grease lubricated rolling bearings. A = AE pulse count rate measurement, B = AE signal rms value measurement, C = AE time signal measurement. Grade-of-influence values are valid in individual factor groups.

<table>
<thead>
<tr>
<th>Factor</th>
<th>Grade-of-influence</th>
<th>Manner of influence</th>
<th>Suitable signal analysis method</th>
<th>Figure of a sample signal or figure about the influence</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>High AE rate</td>
<td>Low AE rate</td>
<td>Cyclic or unstable AE rate</td>
</tr>
<tr>
<td>Grease parameters</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Base oil viscosity</td>
<td>3</td>
<td>High viscosity</td>
<td>Low viscosity</td>
<td>A, B</td>
</tr>
<tr>
<td>Thickener concentration</td>
<td>1</td>
<td>High concentration</td>
<td>Low concentration</td>
<td>A, B</td>
</tr>
<tr>
<td>Consistency</td>
<td>2</td>
<td>Low consistency</td>
<td>High consistency</td>
<td>A, B</td>
</tr>
<tr>
<td>Base oil bleeding rate</td>
<td>2</td>
<td>High bleeding rate</td>
<td>Low bleeding rate</td>
<td>A, B</td>
</tr>
<tr>
<td>Viscous damping</td>
<td>1</td>
<td>Low damping</td>
<td>High damping</td>
<td>A, B</td>
</tr>
<tr>
<td>Contaminants</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Particle concentration</td>
<td>3</td>
<td>High concentration</td>
<td>Low concentration</td>
<td>A, C</td>
</tr>
<tr>
<td>Particle size</td>
<td>2</td>
<td>Small-size particles</td>
<td>Large-size particles</td>
<td>A, B</td>
</tr>
<tr>
<td>Particle hardness</td>
<td>1</td>
<td>Hard particles</td>
<td>Tough particles</td>
<td>C, D</td>
</tr>
<tr>
<td>Running parameters</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Rotational speed</td>
<td>2</td>
<td>High rotational speed</td>
<td>Low rotational speed</td>
<td>A, B</td>
</tr>
<tr>
<td>Bearing load</td>
<td>1</td>
<td>High load</td>
<td>Low load</td>
<td>A, B</td>
</tr>
<tr>
<td>Running temperature</td>
<td>3</td>
<td>Low temperature</td>
<td>High temperature</td>
<td>A, B</td>
</tr>
<tr>
<td>Re-greasing</td>
<td>3</td>
<td>Continuous</td>
<td></td>
<td>Bearing vibration</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>A, B</td>
</tr>
<tr>
<td>Bearing fault</td>
<td></td>
<td></td>
<td></td>
<td>Long stable running</td>
</tr>
<tr>
<td>Rubbing</td>
<td>3</td>
<td>Bearing component</td>
<td></td>
<td>A, C</td>
</tr>
<tr>
<td>Fault in bearing component</td>
<td>1</td>
<td>Cyclic fault</td>
<td></td>
<td>A, C</td>
</tr>
<tr>
<td>Measurement disturbances</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Grounding</td>
<td>3</td>
<td>Poor grounding</td>
<td></td>
<td>A, B</td>
</tr>
<tr>
<td>Measurement cabling</td>
<td>3</td>
<td>Bad cabling, long cables</td>
<td></td>
<td>A, B</td>
</tr>
<tr>
<td>Contact of sensor</td>
<td>1</td>
<td>Lack of contact glue</td>
<td>Surface strain</td>
<td>A, B</td>
</tr>
<tr>
<td>Environment disturbances</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Bearing seals and mounting</td>
<td>3</td>
<td>Contacting seals</td>
<td>Loose mounting</td>
<td>A, B</td>
</tr>
<tr>
<td>Electrical disturbances</td>
<td>2</td>
<td>Starting of motor</td>
<td></td>
<td>A, B</td>
</tr>
<tr>
<td>Hydraulic and air pressure systems</td>
<td>2</td>
<td>Flow and leakage</td>
<td></td>
<td>A, B</td>
</tr>
</tbody>
</table>
6. CONCLUSIONS

This thesis comprises a survey and six papers dealing with monitoring of the grease lubrication of rolling bearings and concentrates on the use of the acoustic emission (AE) measurement method in monitoring the lubrication situation of the bearing. Paper A gives an overview of the vibration measurement methods for rolling bearings, covering both the low and high frequency vibration measurement methods. Paper B deals with applying the AE measurement method in field environment. A special case, an extremely slowly rotating rolling bearing, is presented in Paper C. The influence of contaminants on the lubrication situation and the correlation between contaminants in the grease and acoustic emission are investigated in Paper D. Paper E focuses on the grease lubrication mechanisms and the influence of grease properties on the lubrication situation of a bearing. Finally, Paper F examines the influence of bearing running parameters on acoustic emission. The conclusions and discussions of separate experiments and analyses of the results are presented in different papers. Here the main results are collected together in order to specify what has been achieved in the thesis.

The goals of the present investigation were:

1. To study and apply acoustic emission measurement technology for predictive condition monitoring of grease lubricated rolling bearings and to test it in laboratory and field environments.
2. To determine guidelines for interpreting AE measurement results by determining the influence of fundamental properties of the grease and by determining the influence of running parameters of the bearing on the lubrication situation in a real rolling bearing application.
3. To determine the correlation between contaminants in the lubricating grease and the acoustic emission of a grease lubricated rolling bearing.

Achievement of the goals:

The thesis contains an explanation of the technology for measurement of the acoustic emission of grease lubricated rolling bearings and this technology has been tested in the laboratory and field environments. The influence of typical disturbances on AE
measurement that appear in field environment has been clarified and tested. The influence of the fundamental grease parameters and the running parameters of the bearing have been verified and an empirical model for predicting the AE pulse count rate for a deep groove ball bearing lubricated with clean grease has been formulated. The guidelines for interpreting the AE measurement results are based on a large number of measurements during a time period of two years. However, because of the complex behaviour and the lack of theoretical models of the behaviour of grease lubrication in rolling bearings, the interpretation guidelines are given in a holistic way. The original contributions of this thesis are presented in Chapter 1.2.

In this thesis a new on-line monitoring tool for predictive condition monitoring of grease lubrication of rolling bearings is presented. The tool consists of acoustic emission measurement technology, of the guidelines for interpreting the AE measurement results and of an empirical model for predicting the acoustic emission pulse count rate. The characteristic features of the suggested method are: continuous AE pulse count measurement in the frequency area over 100 kHz, the use of reference AE measurement, measurement or knowledge of the running parameters of the bearing during AE measurement and knowledge of the properties of the lubrication grease. The AE time signal and the analysis of the time signal using statistical methods give additional information, especially on contaminants in the lubricating grease.

The application area of the condition monitoring tool presented here is in predictive condition monitoring of grease lubricated rolling bearings. Taking into account the investments costs, comprising the AE measurement equipment and measurement devices for running parameters and the knowledge, typical suitable applications may be found in the process industry, where maintenance actions are timed, for example in the case of steel milling machines and paper machines. One practical application could be to use the AE measurement of the lubrication situation for adaptive control of the grease supply to a rolling bearing in order to optimise the costs of grease consumption.

6.1 Recommendations for future work

The complex influence of vibration on the grease lubrication of a bearing was mentioned in Chapter 4.2 of the survey part of this thesis as one investigation area for
future work. This work 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 bearing types that have been used in this investigation are the deep groove ball
bearing and the two row spherical roller bearing. The bearing load has been purely
radial. In this kind of applications no spin motion of the rolling elements is present.
Investigation area could be influence of the spin motion, which is present for example in
an angular contact ball bearing, of the rolling elements on the lubrication situation. The
spin motion of the rolling elements is mentioned as one of the continuous replenishment
mechanisms of the grease in the contact zone in Paper E, Chapter 2 \[E4,E5,E6\].

This study indicated that the running parameters of the bearing influence strongly the
AE measurement result. Furthermore, the running situations influence the behaviour of
the lubrication grease, for example as a result of the running temperature. The
interpretation of the AE measurement results may often be difficult and may take a lot
of time, especially in field environment, where external disturbances may influence
measurement. One investigation area could be the use of the neural network method to
interpret the AE measurement results by classifying the measurement data from the
bearing and from the environment and taking into account the properties of the grease.
The first goal in that kind of investigation could be to filter the disturbances away from
the measured signal and the second goal could be the classification of condition of the
grease lubrication situation of the bearing.
REFERENCES


APPENDIX 1. Measurement installation and measurement devices

PC 1  Computer, collecting data from data acquisition cards
PC 2  Computer, collecting extra information from measurements

WBK 512 and WBK 14  12-bit Data acquisition units
Fluke hydra data acq. unit  Data acquisition unit
PeakVue  CSI PeakVue signal processing device
D-94  Mitsol D-94 signal processing device for signal derivation
AETTC  4-channel acoustic emission pulse count measurement unit
          with analogic AE signal output
Microlog  SKF Microlog CMVA 10 vibration measurement device
Acceleration sensors  PCB 353B34, SKF CM55786M, METRA KD-35 and
          B&K 4378
AE sensors  PAC R15
Load sensor  Interface and HBM load sensors
RPM sensor  B&K optical pulse sensor
Temperature sensors  K-type temperature elements
SPM BAS-10  Measurement device using Shock Pulse Method
SPM T-2000  Measurement device using Shock Pulse Method
SEE pen  Pen indicating overall value of the SKF SEE-method
APPENDIX 2. Definitions

1. Base oil and thickener of grease
Lubricating greases contain 5 - 30 per cent of thickener, 70 - 95 per cent of base oil and additives like antioxidants, rust inhibitors, EP-additives and anti-wear additives. The base oil is bound to the thickener as described in Figure A2-1. The most common thickener types are metal soaps, like lithium, sodium and calcium, and metal complex soaps, which are mixtures of different metal soaps. Other thickener types are inorganic compounds, e.g. silicagel, and organic compounds, e.g. polyurea. The base oils can be classified as mineral oils, e.g. paraffinic and napthenic oils, and synthetic oils, e.g. polyalohaolefins and polyglycols.

![Figure A2-1. Schematic illustration of the composition of a grease.](image)

2. Base oil bleeding rate
Base oil bleeding rate describes the capability of the base oil to separate from the thickener of the grease. The bleeding rate is determined as the percentage of the oil separated from the grease in 168 hours at a temperature of 40°C. The test method is in accordance with the standard DIN 51817 [6]. The test measurement device is presented in Figure A2-2. The separated oil will trickle through the sieve into the beaker.

![Figure A2-2. Device for measuring the bleeding rate of a grease according to the standard DIN 51817 [6].](image)
3. Consistency of a grease
Consistency means the stiffness of a grease. The degree of stiffness mainly depends on the type and concentration of the thickener. Greases are classified into consistency classes according to the NLGI (National Lubricating Grease Institute) classification. The consistency is determined with a penetrometer measurement device: How deep a standardised cone will sink due to gravity force into the grease during a time of five seconds at a temperature of 25 °C determines the NLGI class of the grease. The principle of measurement of the penetration is shown in Figure A2-3.

![Penetration](image)

Figure A2-3. Principle of the measurement of penetration [47].

4. Ball-on-disc device and disc-on-disc device
A ball-on-disc device is used for measuring lubricant film thickness. The device consists of a hard rotating glass disk and a steel ball that is in loaded rolling contact on the disc. The measurement of film thickness is based on the light interference technique. In Figure A2-4 one type of ball-on-disc device is shown.

![Ball-on-disc device](image)

Figure A2-4. Ball-on-disc device for measuring lubricant film thickness [12].

A disc-on-disc device consists of two discs which are in loaded radial contact against each other. The measurement of film thickness is based on the capacitance measurement technique.
5. Plastic viscosity
The viscosity of the bulk grease.

6. Burger’s vector
Burger’s vector defines the direction and magnitude of the slide between the lattice planes in a dislocation.

Figure A2-5. Burger’s vector $b$. [29].

7. Key-figure monitoring
Monitoring of a measured or calculated value which describes the properties of a phenomenon, for example the vibration overall value and the Kurtosis value of a signal.

8. Terms of Reynolds equation
The Reynolds equation can be expressed in the direction of motion ($x$ direction) when the flow in the direction $y$, i.e. the side-leakage term ($\frac{\partial}{\partial y}$), is ignored in the form [18]

$$
\frac{\partial}{\partial x} \left( \frac{\rho h^3}{12 \eta} \frac{\partial p}{\partial x} \right) = \frac{\partial}{\partial x} \left[ \rho h (u_a + u_b) \right] + \rho \left( \frac{w_a - w_b - u_a}{\partial h}{\partial x} \right) + h \frac{\partial p}{\partial t} \tag{55}
$$

where

$$
\frac{\partial}{\partial x} \left[ \frac{\rho h (u_a + u_b)}{2} \right] \tag{56}
$$

is the Poiseuille term, which describes the lubricant flow rates due to the pressure gradients in the film,
\[
\frac{\partial}{\partial x} \left( \frac{\rho h^3}{12\eta} \frac{\partial p}{\partial x} \right)
\]

is the Couette term, which describes the flow rates due to surface velocities,

\[
\rho \left( w_a - w_b - u_a \frac{\partial h}{\partial x} \right)
\]

is the Squeeze term, which describes the flow rates due to squeezing motion and

\[
\frac{\partial p}{\partial t}
\]

is the Local expansion term, which describes the flow rates due to local expansion.

Nomenclature in the Reynolds equation:

- \( h \): fluid film thickness
- \( p \): pressure in the film
- \( u_a \): velocity of surface \( a \)
- \( u_b \): velocity of surface \( b \)
- \( t \): time
- \( \rho \): density of the fluid

9. Irwin model

In the Irwin model the plastic region is presumed cylindrical in form. The radius of the plastic region can be calculated in plain stress conditions with equation [21]

\[
r_y = \frac{1}{2\pi} \left( \frac{K_f}{\sigma_y} \right)^2
\]

The crack is assumed to have a length of \( \Delta a_c = r_y \), deeper than it is in reality, Figure A2-6.

Figure A2-6. Crack in Irwin model [21].
10. Dugdale model

In the Dugdale model a compression pressure distribution on the sides of the crack is presumed, so that the crack tip just closes, see Figure A2-7. This causes a drop in stress intensity factor. The radius of the plastic region can be calculated with equation [21]

\[
 r_{ep} = \frac{a_c}{2} \left[ \sec\left(\frac{\pi \sigma}{2 \sigma_y}\right) - 1 \right] \tag{61}
\]

Figure A2-7. Crack in Dugdale model [21].

11. \( K_I \) is stress intensity factor in crack opening mode

In the crack opening mode (Mode \( I \)) the crack is presumed to be opened by a force symmetric in the \( x-y \) plane, see Figure A2-8.

Figure A2-8. The crack opening mode (Mode \( I \)) [21].

12. Phase velocity of longitudinal and transversal stress wave

Phase velocities of longitudinal (\( c_L \)) and transversal (\( c_T \)) stress wave in elastic solid material are defined as follows [1]:

\[
 c_L = \sqrt{\frac{E}{\rho}} \tag{62}
\]

\[
 c_T = \sqrt{\frac{G_s}{\rho}} \tag{63}
\]
APPENDED PAPERS