Monitoring of Wear in Elasto-Hydrodynamic Lubricated Contacts

Running-in and Failure Propagation

Stephan Schnabel
MONITORING OF WEAR IN ELASTO-
HYDRODYNAMIC LUBRICATED CONTACTS

— RUNNING-IN AND FAILURE PROPAGATION —

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Cover figure: disassembled
Angular contact ball bearing

Title page figure: Surface topography of rolling element bearings
contaminated with iron oxide

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– RUNNING-IN AND FAILURE PROPAGATION –

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Preface

The work presented in this licentiate thesis was carried out at Luleå University of Technology at the Division of Machine Elements and is part of the project TriboAct. I would like to thank the Swedish Foundation for Strategic Research (ProViking) for financial support of this project. I would also like to thank the industrial partners to this project, Bosch-Rexroth Sweden, LKAB and SKF CMC Luleå for the financial support, shared knowledge and interesting discussions. Further I would like to express my gratitude to my supervisors Associate-Professor Pär Marklund and Professor. Roland Larsson for the guidance throughout this work and for the discussions which inspired me in my research. I would also like to thank Pär especially for the support during the first months in a new culture and an unknown system.

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Tore Serrander, Martin Lund and Jan Granström were a tremendous help during the experimental work. Further acknowledgment belongs to Ichiro Minami for patiently explaining for me the chemical background of the experiments. Thanks also to all my colleges for a fantastic and inspiring work environment. It is a pleasure to work with all of you and to be a part of the division of machine elements.

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of them, but I want to express that I appreciate their support and the still good connection even though we are separated by more than 2500 km. Finally I would like to thank my family, which supported my step to move to another country and to start a doctoral thesis. I would like to thank them for their help, their advices and their love. I am thankful for the support I have received over the years especially from my parents, my brother and my grandfather.

"Dank'schön"
Abstract

Elasto-hydrodynamic lubricated (EHL) contacts can be found in various machine elements or systems, like rolling element bearings, cam followers or gear transmissions. The service life of these elements and systems are depending to some extent on the performance of EHL contacts. Today most tribological contacts are lubricated with the same type of lubricant throughout the entire service life. However operating conditions can change over the components service life and the contacts will therefore require different lubricant properties. In order to expend the service life of the component, the lubrication of the tribological contacts has to be optimized based on the current operating conditions. A future vision is to develop machine elements which can adopt to the actual operating condition, so called triboactive systems. A first step of necessary research in order to develop such systems is presented in this work. In order to enable operation dependent lubrication the mechanism of monitoring techniques and their interaction with different operating conditions have to be investigated. In this work the effect of surface topography, slide to roll ratio and additives on the running-in and the monitoring by contact impedance were studied. Characteristic dependences between the surface parameter $R_q$ and the contact capacitance and between the surface parameter $R_z$ and the contact resistance were found. Further tests with iron oxide ($Fe_3O_4$) contaminated bearings, monitored by vibration and acoustic emission were carried out. Premature failure due to iron oxide contamination is the most common problem for rolling element bearings operating in mining environment. Thereby the effect of iron oxide contamination on the vibration and acoustic emission monitoring for two different types of greases were investigated. It was found that a simple RMS analysis of the vibration and acoustic emission signals enables the detection of improvements of contaminated contacts by lubrication. Both vibration and acoustic emission from the investigated bearings were reduced by adding extreme pressure additives (EP) to the contaminated contacts. Monitoring of the lubrication condition is necessary to generate information
about the current performance of the tribological contact. However, in order to improve the performance of tribological contacts by changes of the lubricant or additives, the effect of such additives on the lubrication condition and the performance of the tribological contact need to be studied more in detail. The presented running-in tests in this work showed that EP additives are only favourable in the very first stage of running-in. The advantage of EP additives for running-in increases with increased surface roughness and increased slide to roll ratio.

Another advantage of EP additives was observed during the tests with iron oxide contaminants. The use of EP additives reduced the acoustic emissions of the tribological contact by 70% and reduced the increase of surface roughness of the raceways by as much as 60%. Furthermore the tests indicate a lower wear rate for contaminated EHL contacts lubricated with greases containing EP additives in comparison to plain grease without EP additives, in case of iron oxide contaminated EHL contacts.
Contents

Preface i
Abstract iii

I Comprehensive Summary 1

1 From passive to active Tribology 3

2 EHL contacts 7
  2.1 Monitoring of EHL contacts 9
  2.1.1 Contact resistance and capacitance 9
  2.1.2 Vibration 10
  2.1.3 Acoustic emission 11
  2.2 Lubrication of EHL contacts 13
  2.2.1 Lubricants for EHL contacts 13
  2.2.2 Extreme pressure (EP) additives in EHL contacts 13
  2.3 Wear in EHL contacts 14

3 Test setup 17
  3.1 Test methods 17
  3.1.1 Running-in tests 18
  3.1.2 Contaminated bearing tests 19
  3.2 Test equipment 20
  3.2.1 Ball on disc test rig 20
  3.2.2 Rolling element bearing test rig 22

4 Results and discussion 23
  4.1 Monitoring 23
4.1.1 Contact impedance .................................. 23
4.1.2 Vibration and acoustic emission .................... 25
4.2 Wear and lubrication effects .......................... 28
  4.2.1 Running-in ........................................ 28
  4.2.2 Three body abrasive wear ........................ 30

5 Conclusions .............................................. 33

6 Future work .............................................. 35

II Appended Papers ........................................ 37

A Running-in of an EHL contact .......................... 39
  A.1 Introduction ......................................... 42
  A.2 Experimental ........................................ 43
    A.2.1 Rig setup ........................................ 43
    A.2.2 Specimens ....................................... 46
    A.2.3 Experimental scheme .......................... 47
  A.3 Results and discussion ................................ 48
    A.3.1 Dependency of running-in and surface topography .. 48
    A.3.2 Contact impedance behavior during running in .... 51
    A.3.3 Correlation between contact impedance and surface to-
          pography ......................................... 53
  A.4 Conclusions .......................................... 54
  A.5 Acknowledgments ...................................... 54

B Contaminated rolling element bearings ................. 59
  B.1 Introduction ......................................... 62
  B.2 Experimental setup ................................... 63
    B.2.1 Rolling bearing test rig ........................ 63
    B.2.2 Test setup ....................................... 64
  B.3 Results .............................................. 66
    B.3.1 Test duration of 24 hours ....................... 67
    B.3.2 Test duration of 168 hours ..................... 69
  B.4 Discussion .......................................... 72
  B.5 Conclusions .......................................... 74
  B.6 Acknowledgments ...................................... 74

Bibliography ............................................. 76
Appended Papers

"Monitoring of running-in of an EHL contact using contact impedance"
To be submitted
The ability of contact impedance as a monitoring tool for running-in was investigated. Experimental tests were carried out in an in-house built ball on disc test rig with two different lubricants. The experimental work was carried out by Stephan Schnabel, who also wrote the paper. The setup of the experiments was designed by Stephan Schnabel and Pär Marklund, while Ichiro Minami designed the chemical composition of the lubricants. Both Pär Marklund and Roland Larsson were involved in the discussions of the results.

"Study of the short term effects of Fe₃O₄ particles in rolling element bearings"
Accepted for publication of Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology
In this work the effect of iron oxide contaminants on the performance of rolling element bearings was investigated. Further the influence of extreme pressure additives on contaminated contacts was studied. The experimental work and scheme was set up by Stephan Schnabel, who as well wrote the article. Both Pär Marklund and Roland Larsson were involved in the discussions regarding the results. Pär Marklund took furthermore part in the development of the experimental scheme.
Part I

Comprehensive Summary
Chapter 1

From passive to active
Tribology

Tribology is a quite young research subject even though the core problems of
tribology (friction, wear and lubrication) have challenged humans far before
the research subject was established. The origin of tribology as a research
subject started with both the book "The friction and lubrication of solids" [1]
and the definition of the word tribology in the Jost report 1966 [2]. Tribol-
ogy is defined as the science of interacting surface in relative motion to each
other [2]. A more industry orientated definition of tribology formulated Czi-
chos and Habig [3]:

"Tribology is an interdisciplinary research subject on optimization
of mechanical technologies through minimization of energy losses
caused by friction and wear" [3]

Tribology is a broad research area and has impact on many different products
and systems. All kind of bearings, gears, motors, skies, tires and even mascaras
are related to tribology and its topics friction, wear and lubrication. However
the work of this licentiate thesis is narrowed down to a specific kind of tribo-
logical contact. Elasto- hydrodynamic lubricated (EHL) contacts are the focus
of the presented investigations in this work. EHL contacts can be found in
rolling element bearings, hydraulic motors and gear boxes. The invention of
rolling element bearings is more than 500 years old, as shown by Leonardo da
Vincis sketch in Figure 1.1 and ever since, rolling element bearings and EHL
contacts have been optimized.

This work is part of a project called "TriboAct". The aim of this project is to
contribute to the development of triboactive or tribotronic systems [5], which
CHAPTER 1. FROM PASSIVE TO ACTIVE TRIBOLOGY

Figure 1.1: Sketch of a rolling element trust bearing designed by Leonardo da Vinci [4]

are able to adapt the tribological interface to the actual operating conditions. Active tribological contacts would be revolutionary. It would not only change the definition of tribology by Czichos and Habig, but it would also transform tribological systems like rolling element bearings, gears or cam followers from passive to active machine elements. However, triboactive contacts are currently just an idea. In order to realize such contacts a huge amount of research needs to be carried out and several questions need to be answered, e.g.:

- How is the propagation of certain failures in a specific tribological contact?
- Which effect has lubrication on failure propagation?
- How can tribological properties be changed in an existing contact?
- How can different failure types be monitored through online condition monitoring?
However, this perspective of triboactive systems is too general for a single PhD-project. Therefore the presented work is narrowed down to more specific problems. The type of tribological contact was limited to EHL contacts and different monitoring tools (vibration, acoustic emission and contact impedance) were chosen. Based on these limitations more specific research questions were formulated:

- What is the typical failure propagation for point defects and three body abrasive wear in EHL contacts?
- How do extreme pressure additives influence elasto- hydrodynamic lubricated contacts?
- How do extreme pressure additives influence failure propagation and wear?
- Can vibration and acoustic emission monitoring be used to distinguish between different kind of point defects?
- Can vibration and acoustic emission monitoring be used to distinguish between different lubrication regimes?

These research questions can be seen as the project objectives and are partly addressed in this licentiate thesis.
Chapter 2

Elasto- hydrodynamic lubricated contacts

An elasto- hydrodynamic lubricated (EHL) contact is a tribological interface in which both elastic deformation of the surfaces in contact and load carrying by the lubricant occurs. In order to deform the surfaces elastically high pressures are required. For EHL contacts with metallic surfaces the pressures can reach up to 2.5 GPa or in certain cases even 3 GPa [6]. The surfaces will deform, a continuous film will be built up and the pressure will be distributed according to the schematic shown in Figure 2.1. The surface relaxation and the caused pressure spike at the outlet of the contact, is typical for EHL contacts and was first observed by Petrushevich [7]. Once the film has built up and the surfaces are fully separated from each other, friction is minimized and only caused by

![Figure 2.1: Schematic of the deformation of an EHL contact](image)
the viscous-shear of the lubricant. The described conditions is one of three lubrication regimes and is called full-film- or EHL-regime, in which the complete load is carried by the lubricant film. However, it is only valid as long as rolling speed and viscosity of the lubricant are high enough for a given load. If this is not the case, the lubricant film thickness will not be thick enough to separate the highest asperities of the surfaces and the contact will enter the mixed lubrication regime. The asperity collisions will further contribute to a friction increase. As the film gets thinner, asperity collisions and contact friction will increase and asperity contact will carry more load successively. If all load is carried by the asperities of the surfaces, the contact will enter the boundary regime. A common way to illustrate this regimes and the friction behavior is the Strubeck-curve (Figure 2.2). R.Strubeck investigated 1902 the behavior of journal and rolling bearings and introduced these kind of diagrams [8].

Figure 2.2: Schematic illustration of a Strubeck-curve and the different lubrication regimes

Another method to distinguish between the different lubrication regimes is the value. is calculated according to equation (2.1), where represents the
2.1. **MONITORING OF EHL CONTACTS**

minimum film thickness and $R_q$ the root mean square of the surface roughness.

$$\Lambda = \frac{h_{\text{min}}}{\sqrt{R_q^2 + R_q^2}} \quad (2.1)$$

For $\Lambda$ values smaller 1 the contact is considered to work in the boundary regime. Values between 1 and 3 are equivalent to the mixed regime and if $\Lambda$ exceeds 3, a full film regime is assumed [9].

### 2.1 Monitoring of EHL contacts

Monitoring of EHL contacts is essential for both scientific research and condition based maintenance of e.g. rolling element bearing, gears and hydraulic motors. Vibration sensors are the most common monitoring tool [6], but also contact capacitance/resistance, acoustic emission, temperature, particle and "Total-Acid-Number" (TAN) measurements are used to evaluate the condition of EHL-contacts.

#### 2.1.1 Contact resistance and capacitance

Monitoring of the electrical properties of the tribological contacts is a tool which is almost only used in laboratory experiments. Contact resistance and capacitance measurements make use of the changes of electrical properties in the tribological contact. Property changes can be caused by either changes of the electrical properties of the lubricant or changes of the film thickness. Equations (2.2) and (2.3) illustrate the mathematical relation between the local surface separation $h(x)$ and the contact resistance $R_{\text{contact}}$ or contact capacitance $C_{\text{contact}}$ over the contact length $l_{\text{contact}}$.

$$R_{\text{contact}} = \rho \int_{x=0}^{x=l_{\text{contact}}} \frac{h(x)}{A_{\text{contact}}(x)} \, dx \quad (2.2)$$

$$C_{\text{contact}} = \varepsilon_0 \varepsilon_r \int_{x=0}^{x=l_{\text{contact}}} \frac{A_{\text{gap}}(x)}{h(x)} \, dx \quad (2.3)$$

where $\rho$ [Ωm] is the specific resistance, $\varepsilon_0$ [$\frac{F}{m}$] is the vacuum permittivity and $\varepsilon_r$ [-] is the relative static permittivity of the lubricant. A change of the film thickness would result in changes of the local mean film thickness $h(x)$, the
real area of contact \( A_{\text{contact}} \) and a change of the free area \( A_{\text{gap}} \), which is
dependent on \( A_{\text{contact}} \) according to the following equation:

\[
A_{\text{gap}} = A_{\text{nominal}} - A_{\text{contact}}
\]  

(2.4)

where \( A_{\text{nominal}} \) is the nominal area of contact. However, not only film thickness
will influence the contact resistance \( R_{\text{contact}} \) and the contact capacitance \( C_{\text{contact}} \). Danforth [10] observed the pressure dependency of \( \varepsilon_r \). Further did Jablonka et al. [11, 12] show that equation (2.3) is not valid for the cases of cavitation and tribofilm formation. Thereby the group derived more general forms for both cases. Temperature has as well an influence on the measurement due to the temperature dependency of \( \varepsilon_r \) and \( \rho \) [10].

Knowing these mathematical relations and the electrical properties of the lubricant, contact resistance/capacitance measurement can be used to measure film thickness [12, 13]. Further use of the method is the detection of tribofilm formation [14] and surface separation (“lift off”) [15].

2.1.2 Vibration

Vibration measurement with its different analyzing techniques is probably the most common tool for condition monitoring of EHL contacts [16]. Vibrations are excited resonances caused by severe enough asperity collisions [17]. However, even a well functioning EHL contact emits vibrations. Sunnersjö [18] has investigated the vibrations of rolling element bearings with geometrical imperfections. He observed that both the bearing design and the production inaccuracy are causing vibrations. Therefore vibration analysis of EHL contacts is challenging and several methods were introduced. The most common methods are:

- Crest factor and root mean square (RMS) analysis
- Kurtosis spectral analysis
- Shock pulse method
- Defect frequency analysis

Crest factor and RMS analysis methods are the most simple methods, where the both these factors are calculated for either the frequency or time domain and are observed over time. The signal to noise ratio is essential for the quality of the results and therefore literature reports only limited success for this methods [19].
Kurtosis spectral analysis is a statistical approach to characterize the vibrations of a tribological contact. Dyer and Stewart [20] defined that the vibration emission of a healthy bearing has a kurtosis of 3 and that any value above 3 will indicate a failure. However, after a fully developed failure the kurtosis might get back to a value of a healthy bearing [19].

Shock pulse method is based on a patent of Eivind Schoel in 1966. Thereby the measured vibration will be filtered with a bandpass filter (typically between 35 kHz to 40 kHz). Within this frequency range common vibration noise reaches a minimum [21]. The amplitudes of this filtered vibration signal are compared to threshold values and once the threshold value is exceeded a failure is indicated. One of the biggest drawbacks of this method is thereby to set the threshold values for a certain application.

Defect frequency analysis is another analysis method which is especially used for rolling element bearings. A failure in a tribological system is rolled over periodically. This periodicity is used to calculate frequencies at which the failure appears in the tribological contact. The calculated frequencies are called "defect frequencies". Analysis of the defect frequencies and their harmonics in the frequency domain will lead to failure detection. Further is the location of the failure detectable to some extent [17]. However, noise [22] and the slide to roll ratio of the EHL contact [23] can either mask or shift the signals. In order to minimize these effects different processing techniques like high frequency resonance technique (HFRT) [16], adaptive noise canceling (ANC) [24] and the wavelet transform method [25] are used.

2.1.3 Acoustic emission

Sometimes experienced maintenance staff can diagnose the condition of a machine by the emitted sound. The method does not seem very professional. However, Scanlan [26] investigated already 1965 the relation between machine noise and vibration and proofed their dependency. In combination with the knowledge about vibration and EHL contacts this method is almost scientifically corroborated. Acoustic emission is a technology which makes use of these sound phenomenons. However, research showed the most effective frequency range (0.1 MHz to 1.5 MHz) [27–29] is far above the range sensible for human ears.

Acoustic emissions (AE) for condition monitoring were defined in the eighties by Mathews [30] as:

"...transient elastic waves generated from a rapid release of strain energy caused by a deformation or damage within or on the sur-
Even though he defined acoustic emissions already 1983, it took more than one decade until acoustic emissions in EHL contacts were studied in more detail. This delay is probably caused by the huge amount of data generated by AE monitoring [31].

The advantage of AE monitoring is the possibility to detect failure both on the surface [32] and beneath [33]. Figure 2.3 shows the broader operating field of acoustic emission in comparison to vibration monitoring. Especially the ability to detect subsurface defects of tribological contacts which still operate in the full film regime is superior to vibration monitoring. Furthermore showed

Yoshioka and Fujiwara [34] that AE monitoring enables an earlier detection in comparison to vibration monitoring. However, beside the previous mentioned data storage problem of acoustic emission monitoring exist also the problem of signal to noise ratio. Acoustic emission need to be measured as close to the contact as possible, in order to be able to separate signal and noise [28].
2.2 Lubrication of EHL contacts

In general EHL contacts are lubricated with oils or greases (mineral, synthetic or biodegradable). In rolling element bearings is grease the major type of lubricant [35]. Further include modern lubricants additives to optimize the performance. The most common additives for lubricants used in EHL contacts are viscosity modifiers, antioxidants, defoamers, corrosion inhibitors, detergents and dispersants.

2.2.1 Lubricants for EHL contacts

The main task of the lubricant in an EHL contact is to separate the surfaces like shown in the schematic in Figure 2.1. Operating in the full film regime leads to a minimization of friction and eliminates abrasive and adhesive wear. The degree of surface separation depend on the viscosity of the lubricant, the load onto the tribological contact and the rotational speed. Both load and speed are usually determined by the application, which leaves viscosity as the design parameter for EHL contacts.

Viscosity depends on temperature, pressure and shear rate. All these measures can reach extreme values in an EHL contact. The temperature can reach up to several hundred degrees on a micro scale [36], with pressure that can reach several GPa and the shear rate has a huge variation within short time spans. Therefore the viscosity prediction of the lubricant in the tribological contact is challenging. Temperature dependency is usually minimized by use of viscosity modifiers. These additives increase lubricant viscosity at high temperatures more than they do at low temperatures and result in a lubricant which therefore will be less temperature dependent [37]. Beneficial is the pressure dependency. The pressure-viscosity behavior of oils enable lubricants to carry the loads occurring in EHL contacts and is responsible for the surface separation. If the oil is an ideal Newtonian fluid, the viscosity of the oil would be shear rate independent. However, at high pressures and high slide to roll ratios oil does not behave like a Newtonian fluid. Especially at high pressures the viscosity dependency with shear rate is not fully understood [38].

Further tasks for the lubricant in EHL contacts are corrosion prevention, sealing of the contact and heat transfer.

2.2.2 Extreme pressure (EP) additives in EHL contacts

Extreme pressure additives are surface active and consist of a polar head and a hydrocarbon tail [37]. The main task is to prevent scuffing, severe wear and
seizure, by formation of tribofilms with low shear strength. In order to form these films an activation energy is needed. An energy dependency of the reactivity of EP additives has been reported several times. Björling et al. [39] observed this dependency in a ball on disc test rig, while Staub et al. [40] observed the same dependency in a reciprocating pin on plate rig. This activation energy can be either provided by external heat or by asperity collisions [41]. In the boundary lubrication regime the probability for asperity collisions is highest. Therefore EP additives are most active for tribological contacts operating in the boundary lubrication regime [37]. On the other hand, EP additives can be harmful for EHL contacts, while operating in the full-film regime. Several researchers have reported a reduction of fatigue life of rolling element bearings lubricated with EP additives [42, 43]. This reduction can be as high as a factor 5 [44].

2.3 Wear in EHL contacts

Wear is a form of surface degradation and is the main limitation of service life [41]. Different types of wear are defined based on the mechanisms of the surface degradation:

- Surface fatigue
- Abrasive wear (two- and three-body wear)
- Adhesive wear
- Corrosive wear
- Erosive wear

Surface fatigue and abrasive wear are the most common failure sources in EHL-contacts. Surface fatigue is common for contacts operating in the full film regime, while abrasive wear is the most likely failure source for contacts in mixed and boundary lubrication.

Also well maintained and designed rolling element bearings eventually fail due to surface fatigue. The probability of failure is well studied [45] and can be calculated according to the following equation:

\[ L_{10\%} = a_1 \times 10^9 \left( \frac{C}{P} \right)^n \]  \hspace{1cm} (2.5)
where $L_{naa}$ is the bearing lifetime in million revolutions, $a_1$ is the reliability coefficient, $a_{SKF}$ is a manufacturer coefficient, $C$ is the dynamic load capacity, $P$ is the equivalent load and $p$ is the principle exponent for ball and roller bearings respectively.

Abrasive wear can be either two- or three-body wear. Two-body wear is caused by asperity collision in the mixed and the boundary lubrication regime and is called running-in for the period immediately after the initial start of an application. Running-in is the only type of wear which is considered to be positive and can have a favorable effect on fatigue life [46]. Three-body abrasive wear is mainly caused by contamination, but also by wear particles of other mechanisms. Dwyer-Joyce et al. [47, 48] studied both the particle movement and the effect of the particle size. Dwyer-Joyce concluded only a certain particle size can enter the contact and causes the failures, dependent on the conditions (Film thickness, rotational speed and lubricant). The typical shape of failures caused by three-body abrasive wear is an arrow (Figure 2.4). A initial dent caused a flake followed by a tale of smaller dent [49].

![Figure 2.4: SEM picture of a point defect caused by particle contamination](image-url)
Chapter 3

Test setup

All tests presented in this thesis were carried out in a laboratory environment. The investigations were based on two different kinds of test rigs. With respect to the investigated system, elasto-hydrodynamic lubricated contacts, a component test rig, Ball on disc rig ("BonD-rig") and a full scale test rig, Rolling bearing test rig ("RBT-rig"), were used. Both rigs were built in-house and optimized for the different investigations. The detailed explanation of the rigs can be found in the appended papers (BonD-rig: Section A.2.1 and RBT-rig: Section B.2.1).

In general, rolling element bearing tests are usually based on statistical methods. The most common method is probably the calculation of statistical life time of rolling element bearings based on the Weibull slope. However, the test presented in this thesis are not designed to determine the lifetime of the bearings. The tests are designed to investigate the performance (Vibrations, acoustic emissions, friction, lubrication regime and surface topography) of the tribological contacts or systems with respect to the test conditions (Slide to roll ratio, type of lubrication and contamination). Therefore the tests are based on the "paired comparison" method. Thereby the tests are designed to compare the performance of pairs were only one parameter was changed at the time.

3.1 Test methods

As mentioned earlier, both tests are based on the paired comparison design. However, the investigated performance parameter and the chosen test conditions are different.


3.1.1 Running-in tests

During running-in tests, contact impedance, friction and surface topography were investigated, while the type of lubricant, the slide to roll ratio (SRR) and the initial surface roughness were changed successively. The setup scheme used during the running-in investigation is shown in Table 3.1. The entrainment speed, contact pressure (1.7 GPa), lubrication rate (55 ml/h) and test duration (6 h) were constant during the tests. These contact condition were chosen to simulate a slow rotating rolling element bearing contact or cam follower contact. The balls used in the investigation had the tolerance G20 and a hardness of 62 HRC which is slightly harder than the discs (57 to 58 HRC). The hardness is equivalent to the hardness of rolling element of rolling bearings. The difference in hardness between balls and discs, was chosen to have mainly wear on the discs instead of the balls.

The initial disc roughness was reached through different surface finishes. The rough disc (Rq=1275 nm) and the medium disc (Rq=556 nm) were grounded.

Table 3.1: Test schedule

<table>
<thead>
<tr>
<th>test no.</th>
<th>disc type</th>
<th>SRR(%)</th>
<th>lubricant</th>
<th>speed [m/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Disc rough</td>
<td>1</td>
<td>non-EP oil</td>
<td>0.1</td>
</tr>
<tr>
<td>2</td>
<td>5</td>
<td>non-EP oil</td>
<td>0.1</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>11</td>
<td>non-EP oil</td>
<td>0.1</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>1</td>
<td>EP oil</td>
<td>0.1</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>5</td>
<td>EP oil</td>
<td>0.1</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>11</td>
<td>EP oil</td>
<td>0.1</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>Disc medium</td>
<td>1</td>
<td>non-EP oil</td>
<td>0.1</td>
</tr>
<tr>
<td>8</td>
<td>5</td>
<td>non-EP oil</td>
<td>0.1</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>11</td>
<td>non-EP oil</td>
<td>0.1</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>1</td>
<td>EP oil</td>
<td>0.1</td>
<td></td>
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<td>11</td>
<td>5</td>
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<td>0.1</td>
<td></td>
</tr>
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<td>13</td>
<td>Disc smooth</td>
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<td>non-EP oil</td>
<td>0.1</td>
</tr>
<tr>
<td>14</td>
<td>5</td>
<td>non-EP oil</td>
<td>0.1</td>
<td></td>
</tr>
<tr>
<td>15</td>
<td>11</td>
<td>non-EP oil</td>
<td>0.1</td>
<td></td>
</tr>
<tr>
<td>16</td>
<td>1</td>
<td>EP oil</td>
<td>0.1</td>
<td></td>
</tr>
<tr>
<td>17</td>
<td>5</td>
<td>EP oil</td>
<td>0.1</td>
<td></td>
</tr>
<tr>
<td>18</td>
<td>11</td>
<td>EP oil</td>
<td>0.1</td>
<td></td>
</tr>
</tbody>
</table>
3.1. TEST METHODS

and the smooth disc (Rq=56 nm) was polished. These difference in initial surface roughness secured an investigation in different lubrication regimes. While rough (Λ=0.14) and medium (Λ=0.32) discs operated in the boundary regime, the smooth disc (Λ=2.21) was operating in the mixed lubrication regime. Two different lubricants were used in the investigation. Both lubricants consist of paraffin oil with a high purity (99.7%). While one of the lubricants was completely additive free (non-EP oil), the other lubricant (EP oil) contained $5 \text{ mmol} / \text{kg}$ of dibenzyl-disulfide, which is an extreme pressure additive. The crystalline additive powder and the paraffin oil were blended for 24 hours before the tests.

3.1.2 Contaminated bearing tests

The investigated performance parameters for the contaminated bearing tests were vibration emissions, acoustic emissions, friction and surface topography of the bearing raceways. The test conditions were set up based on the scheme in Table 3.2. Thereby lubrication type, test duration and amount of contamination were changed, while the other test conditions like axial load (2 kN), rotational speed (180 rpm), type of contamination, amount of lubricant (3 grams of grease) and type of rolling element bearing (SKF 7204BEP) were constant throughout the entire investigation.

Table 3.2: Test setup

<table>
<thead>
<tr>
<th>test-no.</th>
<th>lubricant type</th>
<th>contamination (wt)</th>
<th>number of samples</th>
<th>test duration [h]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>SL12-602</td>
<td>0%</td>
<td>2</td>
<td>24</td>
</tr>
<tr>
<td>2</td>
<td></td>
<td></td>
<td>1</td>
<td>168</td>
</tr>
<tr>
<td>3</td>
<td>SL12-603</td>
<td>0%</td>
<td>2</td>
<td>24</td>
</tr>
<tr>
<td>4</td>
<td></td>
<td></td>
<td>1</td>
<td>168</td>
</tr>
<tr>
<td>5</td>
<td>SL12-602</td>
<td>9.1%</td>
<td>4</td>
<td>24</td>
</tr>
<tr>
<td>6</td>
<td></td>
<td></td>
<td>2</td>
<td>168</td>
</tr>
<tr>
<td>7</td>
<td>SL12-603</td>
<td>9.1%</td>
<td>4</td>
<td>24</td>
</tr>
<tr>
<td>8</td>
<td></td>
<td></td>
<td>2</td>
<td>168</td>
</tr>
</tbody>
</table>

The contaminants were iron oxide particles (Fe$_3$O$_4$) collected in the actual working environment in the production line of LKAB in Kiruna, Sweden. The particle size distribution of the contaminants is shown in section B.2.2 Figure B.3. Another investigated test condition was the type of lubricant. Two
different lubricants, SL 12-602 and SL 12-603 were used. Both are greases with Lithium thickener and mineral oil including antioxidants. The only difference is an additional additive. While SL 12-603 contains sulfur based extreme pressure (EP) additives, the grease SL 12-602 is an EP-additive free grease.

3.2 Test equipment

Component and full scale test are terms used within research and development areas. While a rolling element bearing test is a full scale test for a bearing manufacturer, it is a component test for a car manufacturer. In the following section the rolling element bearing test is called a "full scale test" and the ball on disc test is the "component test".

Both types of tests have advantages and drawbacks, which are important to be aware of before choosing a test type. The advantage of the full scale test is that it matches the reality. But due to the probabilistic nature of components like rolling element bearings, the test results of different samples may differ. Even a small system like a rolling element bearing has parameters which are not fully controllable and which can interact with each other. The angular contact ball bearing used in this work has 11 rolling elements, which leads to 22 contacts in the system. In addition to the rolling contacts, each rolling element has a sliding contact at the ball-cage interface. Further depends the slide-to-roll ratio on the tribological interfaces and is therefore not constant. The temperature and the contact pressure are additional parameters which vary and which are not fully controllable. This high amount of degrees of freedom can lead to a poor repeatability.

The component test on the other hand allows a better control of all these parameters and while reducing the complexity of the system to one single contact. But being able to control these parameters means as well, that the right test conditions need to be chosen. The test conditions are essential for the interpretation of the test results and for how the results can be applied to the actual application.

3.2.1 Ball on disc test rig

In order to reduce the complexity of the system to one single tribological contact, the ball on disc (BonD) rig was used for the investigation regarding running-in and its monitoring with contact impedance. Especially the monitoring of the contact was simplified, by using the BonD-rig. Instead of monitoring the average of two bearings with 22 tribological contacts each in the RBT-rig,
the fundamentals of contact impedance of a single tribological contact were studied using the BonD-rig. Further the ball on disc rig allowed to prepare discs with different surface roughness and thereby enabled the investigation of different lubrications regimes and its impact on contact impedance measurements.

The contact impedance was measured by applying an alternating voltage on the disc and the ball, which are electrically isolated from each other (Figure 3.1). This voltage across the tribological contact leads to an alternating current response depending on the electrical properties of the contact interface. The current and the phase shift to the initial voltage were measured. Measuring both current and phase shift enables to calculate the resistance and the capacitance component of the tribological interface.

![Figure 3.1: Schematic of BonD-rig](image)

The lubrication rate to the contact is controlled by injecting a defined amount of lubricant onto the ball (Figure 3.1). After the lubrication nozzle the rotational motion of the ball drags the oil into the contact. This lubrication system enables the lubrication with fresh oil during the test with fully flooded condi-
tions. The BonD-rig is explained more in detail in section A.2.1.

3.2.2 Rolling element bearing test rig

The rolling element bearing test rig (RBT-rig) was used for the investigation of contaminated bearings. The main reason to choose the RBT-rig was to have test conditions as close to reality as possible. Rolling element bearing test rigs have also been used in several earlier investigations [32, 50–54], which makes the test more comparable to previous investigations. The problem to control test parameters and their influence on the results, were considered to be small in comparison to the effect of the particle distribution in the grease.

The RBT-rig is constructed to load the bearings axially by a spring. This allows the bearing carrier (violet part in Figure 3.2) to move freely, hence enables friction measurements from the bearings. At the bearing carrier are vibration and acoustic emission sensor mounted.

![Diagram of RBT-rig](image)

Figure 3.2: Schematic of the free moving bearing carrier of the RBT-rig

The use of two test bearings averages the results even more and minimizes the influence of both the probabilistic nature of rolling element bearings and the contamination distribution in the grease. The RBT-rig is explained more in detail in section B.2.1.
Chapter 4

Results and discussion

In this chapter the results of the thesis work are presented. The results are divided into two sections; condition monitoring and wear and lubrication effects.

4.1 Monitoring

As the reliability of tribology-based products like rolling element bearings, gears and hydraulic motors are becoming more and more important, monitoring of the tribological interfaces becomes an essential part of the mechanical system. Today’s monitoring tools allow to collect huge amount of data from the tribological interface, but the interpretation of this data is still difficult. One part of this thesis work was to learn more about the monitoring techniques based on vibration, acoustic emission and contact impedance. The aim was to study the monitoring signals with respect to wear and lubrication conditions in order to contribute to the knowledge of condition monitoring and its interpretation.

4.1.1 Contact impedance

The run-in process of discs in a ball on disc rig was investigated using contact impedance (resistance and capacitance) as a monitoring tool. During running-in an EHL contact experience all three lubrication regimes (boundary, mixed and full film). These regimes are detectable either by contact resistance or by contact capacitance. Figure 4.1 and Figure 4.2 are illustrating the average surface roughness versus the average of the last 10,000 measurement points (20 seconds) of the contact
capacitance and contact resistance respectively. Figure 4.1 shows that the capacitance increases as the surface parameters $R_q$ and $R_z$ increase. This trend agrees with the work of Lord and Larsson [55]. Contact capacitance is mainly used to determine the film thickness of EHL contacts and was used by several researchers [11–13]. However, the obtained results show that contact capacitance is not only able to determine the film thickness, but also to detect the transition of boundary and mixed lubrication regime. The film thickness factor $\Lambda=1$ defines the transition of boundary to mixed lubrication [56]. For the test conditions used in this investigation, this transition point will correspond to a surface roughness $R_q$ of 180 nm. This point of 180 nm correspond as well to the point were the capacitance values reach the lower level, see Figure 4.1a.

Figure 4.1: Capacitance measures versus the surface roughness parameters $R_q$ and $R_z$

The contact resistance on the other hand show an abrupt increase when reach-
ing the mixed lubrication regime. However, contact resistance is known for an asymptotic increase when reaching full-film regime [15, 57] and not for reaching the mixed lubrication regime. The increase already in the mixed lubrication regime could be caused by either tribofilm formation or micro-EHL. Further the asymptotic curve is more clear and continuous for the surface parameter Rz in comparison to the surface parameter Rq. This indicates that contact resistance is rather dependent on the highest asperities of a surface than on the average surface roughness.

Chou and Lin [14] used contact resistance to measure the tribofilm formation of EP additives. The obtained results in this investigation on the other hand indicate that the EP film did not have any influence on the contact resistance measurement.

### 4.1.2 Vibration and acoustic emission

The experiments with iron oxide contaminated rolling bearings were monitored by vibration and acoustic emission. Figure 4.3 show the average values for vibration and acoustic emission during the experiments with contaminated
rolling element bearings. While vibration was measured during both 24 hour tests and 168 hour tests, the acoustic emission was only measured during the 168 hour tests. The results in Figure 4.3 show a significant difference for

![Graph](image1)

(a) Non-EP grease

![Graph](image2)

(b) Non-EP grease with particle contamination

Figure 4.4: Observation of acoustic emission signal for the different test setups with a test duration of 168 hours. The Figures show the trend of the double envelope signal of test bearings.

![Graph](image3)

(a) Non-EP grease with particle contamination

![Graph](image4)

(b) EP grease with particle contamination

Figure 4.5: Observation of acoustic emission signal for the different test setups with a test duration of 168 hours. The Figures show the trend of the double envelope signal of test bearings.

both the addition of contamination and the type of lubricant. The increase of acoustic emission by adding contamination is as high as 175% (Figure 4.3 and Figure 4.4) and is conform with results obtained by Miettinen and Andersson [32]. In their investigation they reached up to 500% increase in acoustic emission due to contamination. The vibration signal also shows the same behavior as the acoustic emission. The increase of vibration by adding contamination is as distinct and the increase is of the same magnitude as the acoustic
emission signal.

With both monitoring techniques (vibration and acoustic emission) it was possible to distinguish between the different types of lubricants. The acoustic emission measurements shows the clearest difference. However, the vibration measurement was as well able to detect a significant difference between EP grease and non-EP grease for contaminated rolling element bearings.

Figure 4.4 shows the acoustic emission signal over the test duration. In comparison to the mean values of Figure 4.3, the difference due to contamination is even more distinct. This could be caused by the probability of particles entering the EHL contact, which was studied by Dwyer-Joyce et al. [47].

The difference of acoustic emissions caused by the type of lubricant follows the difference in contamination. This is even more clear when looking at the signals over time (Figure 4.5). In comparison to the mean values of Figure 4.3 the difference in amplitudes is three times higher in Figure 4.5.
4.2 Wear and lubrication effects

Both wear during running-in and three body abrasive wear were studied during the investigations in this thesis. Surface topography measurements carried out with a 3D-profiler called WYKO NT100 were used to investigate the worn surfaces.

4.2.1 Running-in

As mentioned running-in is contrary to other types of wear positive for the EHL contact. However the mechanism and the monitoring of running-in is not fully understood even though several researchers [46, 57, 58] report about the importance of running-in for fatigue life of the EHL contact.

![Figure 4.6: Change of the surface parameter Rq during running-in for different disc types and lubricants](image)

The surface profiles of the used discs of the running-in experiments (section 3.1.1) were measured both before and after running-in. Parts of the results are presented in Figure 4.6. The diagram shows the change of the surface parameter Rq during running-in of the discs using a slide to roll ratio of 5%. While the surfaces got smoother or remained at the same surface roughness for both rough and medium disc, the smooth disc became clearly rougher. Both the
behavior of the rough and the medium discs are expected results for running-in [41]. However, the increase of surface roughness for the smooth disc was unexpected, but was still observed previously by Chou and Lin [14]. A closer look on the wear track (Figure 4.7) shows the wear track consists of two areas. Area number 1 includes the edges of the wear track, which caused the increase of surface roughness shown in Figure 4.6. Apart from that a slight decrease of surface roughness is observed for the inner wear track (no.-2 Figure 4.7).

Figure 4.7: SEM images of a wear track on a smooth disc runned-in with non-EP oil and a slide to roll ratio of 5%

A dependency between the slide to roll ratio (SRR) and the surface roughness after running-in was also observed (Figure 4.8). As the SRR increased the surface roughness of the wear tracks was getting smoother. For the rough and the medium discs a smoothening effect of the extreme pressure additives was noticed also. However the effect was not observed for the smooth discs. In case of the smooth disc the ineffectiveness might be caused by the high activation energy that the EP additives requires. EP additives need a relatively high activation energy in comparison to other additives [37]. This energy is gen-
erated by the contact pressure and the asperity collisions in the contact. The probability of this collisions decrease with an increase of the film thickness factor $\Lambda$ [41], which is surface roughness dependent. In general the results show that smoothening of the surfaces increase for harsher test conditions, i.e. high initial roughness and high SRR.

![Figure 4.8: Change of the surface parameter Rq during running-in for different slide to roll ratios](image)

### 4.2.2 Three body abrasive wear

Three body abrasive wear is caused by either contaminants in the systems or wear particles. In order to address the contamination problem of rolling element bearings in mining industries, iron oxide particles $\text{Fe}_3\text{O}_4$ were used as contaminants in this investigation. Figure 4.9 shows the results for the test with a duration of 24 hours and 168 hours respectively. Both the uncontaminated samples showed an increased surface roughness, $R_q$, for both test durations. However, the change is small and not significant in contradistinction to the increase of the contaminated samples, which in general show an increase of the raceway surface roughness. For samples lubricated with EP grease an increase of surface roughness was observed, but in comparison to the non-EP
4.2. WEAR AND LUBRICATION EFFECTS

Figure 4.9: Surface roughness of the raceways after the test duration of 24 hours and 168 hours respectively

grease lubricated samples the increase was only half as large. This results are in line with the monitoring measurements presented earlier in Figure 4.3 and Figure 4.5.

The advantage of EP additives indicated by these results could be caused by several mechanisms. A possible explanation would be the theory of sacrificing layers. Thereby the EP-additive reacts with the surface of the raceways and forms a layer which can be worn in a more controlled way. However, surface roughness profiles of samples with a test duration of 24 hours are not supporting the theory of sacrificing layers. Figure 4.10-a show grinding marks for the new and unused sample. These grinding marks remain after the test for both uncontaminated samples (Figure 4.10-b,c). Even for the contaminated bearing lubricated with EP grease (Figure 4.10-e) the grinding marks are visible. However, the contaminated sample lubricated with non-EP grease did not show any grinding marks. This could indicate a higher wear rate for the non-EP grease which is not consistent with the theory of sacrificing layers.

Another explanation could be a reaction of the EP-additive with the contaminants. Thereby the iron oxide contaminants could transform into iron sulfide and iron sulfate, like the chemical reactions (4.1) and (4.2) show. Both
CHAPTER 4. RESULTS AND DISCUSSION

Figure 4.10: Images of the surface measurements with interferometric spectroscopy after a test duration of 24 hours. a) unused bearing, b) bearing with non-EP grease, c) bearing with EP grease, d) contaminated bearing with non-EP grease, e) contaminated bearing with EP grease

molecules (FeS and FeSO₄) are used as solid lubricants in journal bearings. Yuan-Dong et al. [59] have shown the good anti-wear properties of these molecules, which might be beneficial for the used test setup as well. Further Martin et al. [60] proofed a reduction of abrasive wear by iron oxide particles caused by reaction between hematite (Fe₂O₃) with ZDDP.

\[
2 \text{Fe}_3\text{O}_4 + 4 \text{S} = \text{R} + \text{O}_2 \rightarrow 2 \text{FeSO}_4 + 2 \text{FeS} + 2 \text{FeO} + 4 \text{R}
\]

\[
\text{Fe}_3\text{O}_4 + 3 \text{S} = \text{R} + 2 \text{O}_2 \rightarrow 2 \text{FeSO}_4 + \text{FeS} + 3 \text{R}
\]

The theory of additive-contaminant reaction is in line with all obtained results of the investigation. It explains the reduction of vibration and acoustic emission signal as well as the results for the surface measurement. However, proofs are still missing that this is what actually occurs in the contact.
Chapter 5

Conclusions

The work included in this thesis contains investigations both regarding lubrication/additive effects on wear and the condition monitoring of these mechanisms. In general both investigations lead to the following conclusions:

- Contact impedance is a powerful monitoring tool to distinguish between different lubrication regimes at a laboratory scale

- Both vibration and acoustic emission monitoring are able to detect lubrication effects on wear. However, if sensing close to the tribological contact is possible, acoustic emission monitoring is providing a more distinctly detection of the lubrication effects on wear in comparison to vibration.

- Sulfur based extreme pressure additives can be favorable in the early stage of running-in

- Sulfur based extreme pressure additives are beneficial for rolling element bearings exposed to iron oxide contamination

However the laboratory environment in which the investigations were executed, need to be taken into account. Both the results for condition monitoring and study of lubrication effects are easier to obtain in a controlled environment than in actual applications in industry.
Chapter 6

Future work

The overall aim of the project, the investigation of "triboactive" systems, is ambitious and more research regarding condition monitoring, failure propagation and lubrication effects need to be done, in order to create "triboactive" systems on a laboratory scale. However, the work shows as well the potential of "triboactive" systems and can be seen as a base for the future work towards active tribology. To reach the main goal, future work should focus on:

- The experiments in the investigation regarding contaminated rolling element bearings should be further analyzed with respect to the hypothesis of a chemical reaction of contaminants and additives. The energy required for the chemical reactions should be determined and compared to the conditions in the tribological interface. Further improvement would be to test contaminants, which cannot react with the EP additive as an additional reference sample.

- Lubrication effects on other failures than contamination should be studied. Possible failure modes of interest are pitting, flaking, indentations or corrosion. To be able to study these effects, test methods to simulate these failures need to be developed.

- In order to control tribological contacts actively, condition monitoring methods need to distinguish between different failure modes. Therefore the signal behavior of acoustic emission, contact impedance and vibration should be studied with respect to the failure mode and failure propagation.
Part II

Appended Papers
Monitoring of running-in of an EHL contact using contact impedance
Monitoring of running-in of an EHL contact using contact impedance

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Abstract Running-in is an important process for EHL contacts, which affect both service life and operating performance. However the possibilities of monitoring running-in are still poor. Therefore the properties of contact impedance as a monitoring tool were studied by using an in-house made ball on disc apparatus. The contact impedance was monitored during run-in experiments with different initial surface roughness’s of the discs, different slide to roll ratios (SRR) and with pure or additive containing paraffin. In order to investigate the relation between surface roughness parameters and both contact resistance and contact capacitance. While the contact resistance seems to be affected by the parameter Rz, the contact capacitance seems more dependent on Rq. In addition the experiments showed that surface active additives do not necessarily need to influence the contact impedance.

Keywords: contact resistance, contact capacitance, running-in, ball on disc, condition monitoring, EHL
A.1 Introduction

The amount of rolling element bearings in industrial production lines is continuously increasing, thereby the productivity is becoming more and more dependent on the service life of rolling element bearings. Cann [61] estimated the amount of rolling bearings in use to be 18 billion back in 1998. Nowadays the figure can be considered to be even higher. Knowing that a single breakdown can cause a standstill in a production line, this figure indicates the importance of service life of rolling element bearings for productivity.

Service life of rolling element bearings depends on several factors like environment, lubrication, speed and loading. But already the running-in conditions can significantly influence the lifetime of a tribological interface. The importance of running-in is well known in the field of tribology, since Abbott and Firestone [58] already 1933 published their pioneer work. According to Berthe et al. [46] a defined running in process can be a method to optimize fatigue life of rolling element bearings. Based on Abbotts and Firestones work of surface characterization, Crook [57] investigated running in of both sliding and rolling contacts by using pin on disc and twin disc apparatus. His work shows a clear relationship between running in condition, pitting failure and wear. Furthermore Crook was a pioneer in monitoring running in, as he introduced resistance measurement as a method to detect the separation of the discs during run-in.

Even though Abbott, Firestone and Crook early addressed the topic of running in, investigations regarding running in of rolling contacts are rare and investigations regarding monitoring of running in are even more rare. Some researchers continued the work of Crook and investigated running in and its monitoring. Chou and Lin [14] used a roller on disc apparatus to investigate the dependency of tribofilm formation and additive concentration during run-in. Lugt et al. [15] looked at the lift off behavior of EHL contacts due to running in conditions. The correlation between running in and surface finish was published by Lord and Larsson [55]. All these investigations were based on contact resistance as a monitoring tool to detect either the formation of tribofilms or the formation of a lubrication film. However the electrical capacitance of the tribological contacts was not taken into account in these investigations [14,15,55,57]. capacitance measurements of tribological contacts are used to measure the lubricant film thickness, rather than to use it as a monitoring tool for running-in. Especially in automotive tribology capacitance is a common measure to evaluate film thickness of piston-ring cylinder liner assemblies. Hamilton and Moore investigated this method already in 1974 [13].
Jablonka et al. [11] used contact capacitance instead to measure the exact film thickness of an elasto- hydrodynamic contact in a ball on disc apparatus. Further they did show that film thickness measurements by capacitance are influenced by the polarity of the lubricant [12].

When it comes to monitoring of run-in processes, electrical resistance measurement is mainly used to detect the end of running in [15, 57]. Full surface separation is thereby used as the end criteria defined by Jacobson [62]. He defined that the run-in state is reached as soon as the surfaces are separated by a continuous lubricant film. But as mentioned contact resistance is as well used to measure tribofilm formation [14]. Therefore this investigation was carried out in order to study the impact of surface active additives on contact impedance. The behavior of contact impedance in the boundary and mixed lubrication regime was another focus of this investigation. Especially the behavior of contact capacitance in these regimes is not well known in comparison to the full film regime. Further the investigation gave an idea of the suitability of contact impedance measurement as a monitoring tool for the run-in process of EHL contacts and gave a better understanding of the correlation between running-in, contact impedance and surface topography.

A.2 Experimental

The investigation was carried out in a laboratory environment using a ball on disc type of rig. Further all experiments were carried out in room temperature.

A.2.1 Rig setup

For the investigation an in-house made ball on disc rig (BonD-rig) was used, were both disc and ball can rotate independently. Both disc and ball are driven by electric step motors (Figure A.1-no.1), which are connected to the disc and to the ball respectively by a belt drive. These motors allow the use of the rig for entrainment speeds between $0.001 \text{ m/s}$ to $3.0 \text{ m/s}$. The upper limit is limited due to the maximum speed of the step motors itself, while the limit of $0.001 \text{ m/s}$ exist because the step motors cannot secure a linear motion beneath this speed. As mentioned the speed of the disc and the ball are independently controllable and therefore allows tests at any slide-to-roll ratio (SRR).

The discs used in the BonD-rig have a diameter of 104 mm and can consist of any material like plastic, glass or metal due to the universal sample holder (Figure A.1-no.2) of the rig. While the position of the ball is fixed, the position of
the disc is adjustable. The whole sample holder of the disc is horizontally movable in one axis by metric fine pitch thread adjustment (Figure A.1-no.3). The adjusting screw allows to chose manually the position of the ball and disc contact. Different contact radius can be chosen and thereby discs can be used for several tests. The position of the disc is shown by a digital display (Figure A.1-no.4). It enables to adjust the position of the disc as accurate as 0.1 mm.

As mentioned the sample holder of the ball has a fixed position (Figure A.1-no.5) and is made for balls between 22.5 and 27 mm in diameter. The balls are mounted in a conical sample holder and clamped by a screw. For the use of the rig the ball size is not important, but it influences to some extend the borders for the entrainment speed and determines the contact pressure. The standard ball used has a diameter of 1" (25.4 mm). Using this ball size a maximum Hertzian contact pressure of 1.8 GPa can be achieved. The balls are loaded by a pneumatic cylinder (Figure A.1-no.6), which applies a force onto the sample holder in order to press the ball against the disc. It allows to operate between 0.75 GPa and 1.8 GPa using the standard ball size of 1". The lower limit exist because a stable control of the pneumatic cylinder cannot be secured beneath a contact pressure of 0.75 GPa.

The loading of the ball in combination with the entrainment speed and the lubrication situation causes a horizontal friction force at the disc. The sample holder for the disc is connected to the foundation by a flat spring (Figure A.1-no.7). This leads to a small horizontal movement of the sample holder due to
the friction force generated in the contact. The small movement is an indirect measure of the friction force and measured by an inductive sensor (Figure A.1-no.8). In combination with the pressure measurement of the pneumatic cylinder the friction coefficient can be calculated.

Additional to the friction force and friction coefficient, the BonD-rigg is able to measure film formation by electrical resistance and capacitance measurement. Disc and ball are electrically connected as shown in Figure A.2. The ball is electrically isolated from the rig by using ceramic ball bearings for the ball sample holder. A carbon brush (Figure A.1-no.9) connects the shaft of the metallic ball sample holder to the electric circuit. To avoid carbon particles in the tribological contact the disc was connected by a mercury contact (Figure A.1-no.10). The electric circuit consists basically of a system impedance, precision resistors of the Wheatstone bridge and the impedance of the tribological contact. While the system impedance and the precision resistors are constant, the impedance of the tribological contact can vary over time by for-
mation of lubrication films or additive films [55]. The electric circuit is supplied by a 100mV AC-signal of a frequency of 100MHz. The low voltage is chosen to minimize the influence of the impedance measurement on the tribological contact [63].

The 100mV measuring voltage applied to the circuit will lead to a alternating current response. A micro controller (AD8302ARUZ) detects the changes of the contact impedance by a Wheatstone bridge. Both the signal of the Wheatstone bridge and the supply source are used by the signal processing unit of the micro controller. The processed data are two analog continuous current signals, which represent the contact current response of the tribological contact and the phase shift in comparison to the applied voltage. This analog continuous signals are connected to the computer and digitalized with a sample rate of 500 Hz. The same sample rate is used to calculate the contact impedance and to store the calculated values.

The changes of R (resistance) and X (imaginary resistance) over time are proportional to the changes in the tribological contact (fluid film thickness or tribofilm formation).

\[
Z = |Z|e^{-j\varphi} = \frac{u}{i} e^{-j(\varphi_u - \varphi_i)} = R + jX
\]  

(A.1)

where \( Z \) is the impedance of the contact, \( u \) is the measurement voltage, \( i \) is the current through the contact, while \( \varphi_i \) in this equation represents the phase shift angle. Knowing the measured voltage, the current response and the phase shift of these signals, the impedance of the system can be calculated according to equation (A.1). The real part of the impedance is equal to the contact resistance and the imaginary part (\( jX \)) enables to calculate the contact capacitance by using the following equation:

\[
C = -\frac{1}{\omega X} = -\frac{1}{2\pi f X}
\]  

(A.2)

where \( C \) is the contact capacitance, \( \omega \) is the angular frequency of the measured signal and \( f \) is the frequency of the measured signal.

The BonD-rig is continuous lubricated by a constant flow rate. The lubricant drops through a nozzle (Figure A.1-no.11 and Figure A.2) onto the ball and will be dragged into the contact by the rotational speed. It enables continuously lubrication with fresh oil in order to secure a fully flooded contact.

A.2.2 Specimens

For disc specimens the steel type OVAKO825B was used and all disc specimens had a hardness between 57 HRC and 58 HRC with a failure range of
A.2. EXPERIMENTAL

±0.7 HRC. Further three different surface finishes were used, to achieve the large difference in surface roughness shown in Table A.1. The ball specimens were standard G20 balls from SKF with a hardness of 62±1.0 HRC, i.e. harder than the disc samples.

Table A.1: Test specimens

<table>
<thead>
<tr>
<th>Spec.</th>
<th>Surface type</th>
<th>$R_q$ [nm]</th>
<th>$R_z$ [nm]</th>
<th>Hardness HRC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ball G20 polished</td>
<td>59</td>
<td>980</td>
<td>62</td>
<td></td>
</tr>
<tr>
<td>Disc, rough ground</td>
<td>1275</td>
<td>15500</td>
<td>57</td>
<td></td>
</tr>
<tr>
<td>Disc, smooth polished</td>
<td>56</td>
<td>1100</td>
<td>58</td>
<td></td>
</tr>
<tr>
<td>Disc, medium ground</td>
<td>556</td>
<td>4610</td>
<td>57</td>
<td></td>
</tr>
</tbody>
</table>

For the investigation two different lubricants were used. One of the lubricants was pure paraffinic oil with a purity of 99.7% (non-EP oil). Further known properties of the paraffinic oil are the viscosity (255 mPa s) and the pressure coefficient $\alpha$ (32.2 GPa$^{-1}$). The other lubricant was the same paraffinic oil, but containing an extreme pressure (EP) additive (EP oil), which was sulfur based (Dibenzyl disulfide $C_{14}H_{14}S_2$). This paraffinic oil was blended with the crystalline EP powder for 24 hours in order to dissolve the EP additive totally. The blending process was done in a glass beaker with magnetic steering and the amount of EP additives in the EP oil was $5 \text{ mmol kg}^{-1}$.

A.2.3 Experimental scheme

Before the tests, the discs and balls were cleaned with acetone and mounted in the BonD-rig. The test settings for the rig are shown in the test schedule (Table B.1). All tests were performed with an entrainment speed of 0.1 m/s and a lubrication rate of 55 ml/h. These test settings in combination with the difference of initial surface roughness (Table A.1) lead to different film thickness parameter ($\Lambda$) for the discs. While the rough ($\Lambda = 0.14$) and medium ($\Lambda = 0.32$) discs are operating in the boundary regime, the smooth disc ($\Lambda = 2.21$) is operating in the mixed lubrication regime towards the full film regime. The general test duration was 6 hours. In order to avoid EP contaminations settings with the pure paraffinic oil (non-EP oil) were tested first on different tracks on the disk, followed by the tests with the EP oil. After testing all six combinations for a disk type, the disc was dismounted, cleaned and analyzed. The analysis consisted of 3D surface profiling of the wear tracks with a WYKO NT1100.
Table A.2: Test schedule

<table>
<thead>
<tr>
<th>test no.</th>
<th>disc type</th>
<th>SRR [%]</th>
<th>lubricant</th>
<th>speed [m/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Disc rough</td>
<td>1</td>
<td>non-EP oil</td>
<td>0.1</td>
</tr>
<tr>
<td>2</td>
<td>Disc rough</td>
<td>5</td>
<td>non-EP oil</td>
<td>0.1</td>
</tr>
<tr>
<td>3</td>
<td>Disc rough</td>
<td>11</td>
<td>non-EP oil</td>
<td>0.1</td>
</tr>
<tr>
<td>4</td>
<td>Disc rough</td>
<td>1</td>
<td>EP oil</td>
<td>0.1</td>
</tr>
<tr>
<td>5</td>
<td>Disc rough</td>
<td>5</td>
<td>EP oil</td>
<td>0.1</td>
</tr>
<tr>
<td>6</td>
<td>Disc rough</td>
<td>11</td>
<td>EP oil</td>
<td>0.1</td>
</tr>
<tr>
<td>7</td>
<td>Disc medium</td>
<td>1</td>
<td>non-EP oil</td>
<td>0.1</td>
</tr>
<tr>
<td>8</td>
<td>Disc medium</td>
<td>5</td>
<td>non-EP oil</td>
<td>0.1</td>
</tr>
<tr>
<td>9</td>
<td>Disc medium</td>
<td>11</td>
<td>non-EP oil</td>
<td>0.1</td>
</tr>
<tr>
<td>10</td>
<td>Disc smooth</td>
<td>1</td>
<td>EP oil</td>
<td>0.1</td>
</tr>
<tr>
<td>11</td>
<td>Disc smooth</td>
<td>5</td>
<td>EP oil</td>
<td>0.1</td>
</tr>
<tr>
<td>12</td>
<td>Disc smooth</td>
<td>11</td>
<td>EP oil</td>
<td>0.1</td>
</tr>
<tr>
<td>13</td>
<td>Disc smooth</td>
<td>1</td>
<td>non-EP oil</td>
<td>0.1</td>
</tr>
<tr>
<td>14</td>
<td>Disc smooth</td>
<td>5</td>
<td>non-EP oil</td>
<td>0.1</td>
</tr>
<tr>
<td>15</td>
<td>Disc smooth</td>
<td>11</td>
<td>non-EP oil</td>
<td>0.1</td>
</tr>
<tr>
<td>16</td>
<td>Disc smooth</td>
<td>1</td>
<td>EP oil</td>
<td>0.1</td>
</tr>
<tr>
<td>17</td>
<td>Disc smooth</td>
<td>5</td>
<td>EP oil</td>
<td>0.1</td>
</tr>
<tr>
<td>18</td>
<td>Disc smooth</td>
<td>11</td>
<td>EP oil</td>
<td>0.1</td>
</tr>
</tbody>
</table>

A.3 Results and discussion

In the following section the results will be presented. Thereby the results are divided into observations on the surface topography analysis, the contact impedance and the correlation of surface roughness and contact impedance.

A.3.1 Dependency of running-in and surface topography

In Figure A.3 and Figure A.4 the change of surface roughness during the running-in test are shown. A positive value indicates a surface smoothening, while a negative one indicates an increase of the surface roughness. Comparing the surface measurements before and after the running-in of 6 hours with different test setups, they do not show an obvious pattern (Figure A.3 and Figure A.4). For the rough discs all surface parameters (Rq and Rz) are decreasing. However, the parameters Rq and Rz are increasing for the smooth discs, while for the medium discs both increase and decrease were observed.
A.3. RESULTS AND DISCUSSION

But by analyzing the different disc setups separately the different patterns could still be explained. The rough discs are continuously operating in the boundary lubrication regime. It is one reason for the huge changes in surface roughness (in absolute values) in comparison to the other test setups. Further the operation in the boundary regime does explain the advantage of EP-additives during the tests with rough discs. Even though Dibenzyl-disulfide is a surface active EP-additive it needs some activation energy in order to react with the metallic surface. The energy in the contact is considered to be highest in the boundary lubrication regime (either due to flash temperature or mechanical disruption of surface bonds [64]) and increases even further with the increase of SRR [65]. Therefore the biggest difference in surface topography between the pure paraffinic oil and the additive containing oil is observed for the rough disc and the highest slide to roll ratio (SRR 11%).

The smooth disc on the other hand is considered to operate in the mixed lubrication regime, close to the border between the mixed and the elasto-hydrodynamic lubrication (EHL) regime. Both Figure A.3 and Figure A.4 show an increase in surface roughness, even though a decrease after running-
Figure A.4: Change of surface roughness parameter $R_z$ of the discs dependent on slide-to-roll ratio and initial surface roughness

in was expected. A closer look into the wear track (Figure A.5) shows that the major contribution of the increase in surface roughness is caused by the outer regions of the wear track.

Table A.3: Surface parameters regarding wear track zone (Figure A.5) of test no.14 (Table B.1)

<table>
<thead>
<tr>
<th>parameter</th>
<th>initial</th>
<th>zone 1</th>
<th>zone 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>$R_q$ [nm]</td>
<td>56</td>
<td>117</td>
<td>56</td>
</tr>
<tr>
<td>$R_z$ [nm]</td>
<td>1100</td>
<td>2590</td>
<td>1010</td>
</tr>
</tbody>
</table>

The measurement area marked with no.1 in Figure A.5 was used to measure the average surface roughness parameter shown in Figure A.3 and A.4. However, the inner wear track (no.2 Figure A.5) is much smoother. This difference in surface roughness is illustrated in Table A.3 for the test conditions no.14 (Table B.1) and shows that the inner wear track still gets smoother even if the
A.3. RESULTS AND DISCUSSION

Figure A.5: SEM images of a wear track on a smooth disc generated under test condition no. 14 of Table B.1. While the whole wear track shows an increase in surface roughness due to the edges (no. 1), is the center of the wear track still smooth (no. 2).

The surface roughness of the whole wear track is increasing. The surface roughness parameter results of the medium disc does not show any clear pattern within the test duration of 6 hours. While the surface roughness parameters of test samples with lower SRR tend to increase, the higher ones tend to decreasing.

A.3.2 Contact impedance behavior during running in

The online measurement of friction, contact capacitance and contact resistance of test no.1 to no.6 of the test setup (Table B.1) are shown in Figure A.6. The contact resistance is decreasing during the whole test duration. Starting with a sharp decrease in the beginning and flattening out over time. The high initial value and the sharp decrease in the beginning of the running in tests could be
caused by oxide layers, which are worn down during the first 5 to 15 minutes of the tests. This would as well explain the friction behavior (Figure A.6 a and b) during this early stage of running in, due to the reason that oxide layers are easier to shear. The general decrease of the contact resistance during the whole test duration is probably caused by the increase of contact area. During running in the summits of the asperities in contact were worn down and thereby the real area of contact will increase, which leads to a higher conductivity. The contact capacitance did not show a significant behavior over time or with change of SRR, which is not surprising due to fact the tests are operating in the boundary lubrication regime ($\Lambda = 0.14$). In this regime metal to metal contact is dominating and therefore the contact capacitance is not the dominant component of contact impedance.

The initial behavior of friction and resistance are identical for the rough (Figure A.6 a,b,c,f) and the medium (Figure A.7 a,b,c,f) discs. But after an initial increase, both friction and capacitance are decreasing, while the contact resistance is continuing to increase. The increase of contact resistance and the decrease of contact capacitance indicate a surface separation. However, film thickness parameter and friction value indicate the contact is still operating in the boundary lubrication towards the mixed regime. One explanation could be the formation of electrically insulating tribofilms, another one might be micro EHL. Figures A.6 to A.8 consist of almost 11 million measurement points and are smoothened with a moving average including 5,000 points (10 seconds). Therefore the scattering of the resistance does indicate that the contact is separated for a short period of time (milliseconds). Individual values can be as high as $100 \text{ M} \Omega$. The short time period does support the theory of micro EHL. However the fact that this phenomenon increase with SRR does support the theory of tribofilm formation.

The smooth disc is as mentioned considered to operate in the mixed lubrication regime at the boarder towards the elasto-hydrodynamic lubrication regime ($\Lambda = 2.21$). Analyzing the measured friction coefficients (Figure A.8 a,b), this assumption is reasonable. For both lubricant types the capacitance stabilizes after an initial decrease. Once again the decrease of the contact capacitance indicates an increase of surface separation during running-in. However, the contact resistance indicates only partial surface separation. Especially the low values of the contact resistance for the test with 1% slip (blue curve marked with circles in Figure A.8 e,f) indicates metal to metal contact. The behavior can be caused by the same mechanisms suggested for the medium discs. However the operation towards the full film regime, leaves as well the possibility of fluid film formations. The observations show that the capacitance is more
influenced by the mean separation of the surface, rather than the actual surface separation in the asperities.

The contact resistance on the other hand seems to detect the actual surface separation. Therefore the resistance shows the scattering effect, while the capacitance signal remains smooth. Further the tests (Figure A.6, A.7 and A.8) show that the used surface active additive (dibenzyl-disulfide) does not influence contact resistance or contact capacitance measurement. The results in Figure A.3 and Figure A.4 indicated a benefit of the EP-additive regarding surface roughness for the rough discs, but the formation of a EP-film was not detected by the resistance or by the capacitance measurement (Figure A.6). Hence a Tribofilm formation of the surface active additives is not necessarily detectable by either contact resistance or capacitance measurement. Another indicator is that both capacitance and resistance measurements fit for the same curves in a plot versus the surface roughness (Figure A.9 and A.10 regardless the additivation of the used lubricant.

A.3.3 Correlation between contact impedance and surface topography

In order to study the correlation between contact impedance and surface topography the average value of the last 10,000 measurements (20 seconds) of contact capacitance (Figure A.9) and contact resistance (Figure A.10) were plotted against the measured surface roughnesses Rq and Rz for each test setup respectively. The results for the contact capacitance shown in Figure A.9 decreases with an decrease of surface roughness. Considering the operating conditions Λ=1 would be equal to a surface roughness of Rq=180nm. Therefore the main decrease of contact capacitance occurs while the contact operates in the boundary regime. The contact resistance on the other hand does not show significant changes in the boundary lubrication regime, but in the mixed lubrication regime. However, the reason (tribofilm, micro-EHL or fluid film) of this increase is hard to identify. Further show the results a clear relationship between contact resistance and the surface parameter Rz (Figure A.10b), were a clear asymptotic correlation is visible in contradiction to the parameter Rq (Figure A.10a). The contact resistance seems to be more depended on Rz, which is reasonable by having in mind that the slightest metal to metal contact between the highest asperities of the surfaces in contact has a major impact on the measured value. The more clear correlation between contact capacitance and Rq can be explained by the strong dependency of the capacitance signal to mean surface separation of the contact.
A.4 Conclusions

The main aim of the work was to give an idea of the suitability of contact impedance measurement as a monitoring tool for run-in processes in boundary and mixed lubrication regime. The investigation proved the suitability to some extent, but further research is required, in order to get a detailed understanding of the detection mechanisms of contact impedance. The experiments showed as well, that surface active additives does not necessarily need to have an effect on contact impedance measurements. But it is still necessary to study the effect of additives and tribofilm formation on contact impedance measurements more in detail in order to use it for monitoring run-in. Further showed the investigation that both contact resistance and contact capacitance follow characteristic curves in a plot versus surface roughness parameters and that the contact resistance seems to be mainly affected by the parameter Rz, while the contact capacitance is more dependent on Rq.

A.5 Acknowledgments

The authors would like to thank the Swedish research council (SSF) for sponsorship of the research. Further acknowledgment belongs to SKF CMC, LKAB and Bosch-Rexroth Sweden for their sponsorship and the contribution of their know-how.
Figure A.6: Online measurement results of the rough disc during the run-in with a test duration of 6 hours
Figure A.7: Online measurement results of the medium disc during the run-in with a test duration of 6 hours
Figure A.8: Online measurement results of the smooth disc during the run-in with a test duration of 6 hours
Figure A.9: capacitance measures versus the surface roughness parameters Rq and Rz

Figure A.10: Resistance measures versus the surface roughness parameters Rq and Rz
Paper B

Study of the short term effect of Fe$_3$O$_4$ particles in rolling element bearings
Study of the short term effect of Fe₃O₄ particles in rolling element bearings
Observation of vibration, friction and change of surface topography of contaminated angular contact ball bearings

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Luleå University of Technology, Division of Machine Elements, SE-97187 Luleå, Sweden

Abstract The harsh environment rolling element bearings are exposed to in iron mining industries is replicated in a laboratory scale in this work. Bearings (SKF 7204BEP) were tested both with and without magnetite oxide (Fe₃O₄) contamination. In order to study the interaction between contaminants and extreme pressure (EP) additives the rolling element bearings were lubricated with two different greases. A grease without EP and a grease containing sulfur based EP additives were used. Further the effect of the contamination-additive interaction on rolling bearing performance and monitoring signals (vibration and acoustic emission) was investigated. The obtained results indicate an advantage of EP additive in case of the investigated operating conditions. More in detail the use of EP additives decreased wear, surface roughness, vibration and acoustic emission for both test durations of 24 and 168 hours. The decrease of the acoustic emissions and the surface roughness parameter Rq in case of the tests with a duration of 168h as high as 70 % and 60 % respectively using EP additives in comparison with the plain grease were observed. The major cause for this reduction seems to be the interaction between contaminants and EP additives.

Keywords: Rolling element bearing, condition monitoring, contaminants, extreme pressure (EP) additives
B.1 Introduction

Rolling element bearings located in the production lines of mining industries are mainly affected by the contamination of iron ore particles. This contamination leads to a significant shortening of the bearing life and decrease thereby the productivity due to standstills. Even the use of advanced sealing and lubrication systems can not prevent the significant decrease of the life of rolling bearings working in the mining environment.

Third body abrasive wear due to contamination is a common failure source of rolling element bearings [47]. As a result, many researchers have investigated the effect of contamination on rolling element bearing performance, bearing life time and the detection of debris. In these investigations different types of contaminants were used. The most common used contaminant is silica [32, 50–53], but diamonds [47] and iron oxides [32, 51] were also used.

While Sayles and Ioannides [66, 67] as well as Gabelli et al. [68] investigated the influence of debris on fatigue life of rolling element bearings and its statistical calculation, did other authors focus on the performance of the EHL contact itself. The wear behavior of EHL contacts and the particle movement were studied by Dwyer-Joyce et al. [47]. The same research team characterized the probability of particles entering the contact with the result that wear increases with increased particles size due to the higher probability that particle enter the EHL contact [48]. A more industrial focused summary on rolling bearing performance with contaminated lubricants was carried out by Wennehorst and Poll [49].

Vibration and acoustic emission are common tools to monitor contaminated bearings. Both Maru et al. [53] and Tandon et al. [51] have investigated the effect of particle size and concentration on rolling element bearings, using either oil and vibration [53] or grease and acoustic emission [51] for their test setups. The wear behavior of contaminated bearings were studied by Kahlman et. al. [52] and Akagaki et. al. [54] using both vibration and surface imaging techniques. Another investigation using vibration as a monitoring tool was carried out by Hariharan and Srinivasan [50]. They showed the interaction of the root mean square (RMS) of the vibration signal and the rotational speed of the bearing at different concentrations of contamination. But the type of particles has also an influence on the signals of the monitoring tools, like Miettinen and Andersson, [32] have shown. In their work they presented the activity of the acoustic emission signal for steel, iron, Fe₃O₅ and quartz particles.

Even though iron oxides were used in previous studies [32, 51], they focused
more on the detection of the contaminants rather than the influence on the rolling element bearing. Further, do the previous investigations not take the effect of additives into account. Some studies [32, 69] use different base oils in order to examine the effect of base oil properties on a contaminated contact. However knowledge about the effect and mechanisms of additives on contaminated systems is missing. Therefore this work was carried out to study the short term effect of contamination on surface topography of raceways, vibration and acoustic emission signals with respect to the type of lubricant.

B.2 Experimental setup

The test samples were prepared in a laboratory environment and the all test setups were executed at room temperature.

B.2.1 Rolling bearing test rig

The investigation was carried out in an in-house built rolling bearing test rig (Figures B.1 and B.2). The principle of the rig is based on a free moving bearing carrier (pos. 4 in Figure B.1) to be able to measure the sum of the friction torque from both test bearings (pos. 2 in Figure B.1). The tested bearings are loaded with an axial load by a spring shown in Figure B.2-a. Thereby is the maximum adjustable axial load 2.5 kN.

1: support bearing
2: test bearing
3: rack
4: free moving bearing carrier

Figure B.1: Schematics of the Rolling bearing test rig used for this investigation.
The test rig is capable to run with a rotational speed between 100 rpm and 2500 rpm, due to a variable frequency drive. The measured variables were in case of the 24 hour tests friction torque and vibrations of the test bearings. For the tests with a duration of one week the rig was additionally equipped with acoustic emission sensors. The friction torque is measured by a force sensor connected to a torque arm (Figure B.2-b). A commercial vibration system called WindCon is used to investigate vibrations from the tested bearings [70]. The accelerometer (SKF-607M69) was placed in the center of the test bearing carrier in order to measure vibrations from both test bearings (Figure B.2). In order to get as close to the contacts as possible, the acoustic emission sensors (CMSS786M) were mounted above each of the tested angular contact ball bearings. The measured frequency ranges are 0-10 kHz for the vibration and 0.1-0.5 MHz for the acoustic emission. Both signals where measured with a sample storage rate of 0.625Hz. however, the sample rate of the measurement was 2.56 kHz for the vibration signals and 1.28 MHz for the acoustic emission signals. The hardware of the sensors create an envelope signal of the measured acoustic emission, due to the amount of data, which is otherwise not processable by the WindCon system.

During this investigation mainly the trend analysis function of the WindCon system was used. The trend value (B.1) in this case is the quadratic mean value across the frequency spectrum for one measured sample.

\[
T = \frac{1}{n} \sqrt{n \sum_{i=0}^{f_E} a_i^2}, \quad \text{with} \quad n = \frac{f_E}{\Delta f} 
\]

where \(T\) is the trend value, \(f_E\) is the end value of the measured frequency range, \(\Delta f\) is the discretization of the frequency range and \(a_i\) is the amplitude at a certain frequency. The discretization of the system for the presented results is 1 Hz. The Trend values were calculated for both the vibration, the acoustic emission and their envelope signals respectively. The presented trend value diagrams are based on the envelope of the vibration signal and the double envelope of the acoustic emission signal. The 40 trend values during a period of one minute are compared with each other and the highest value is stored. These stored values within the test duration represent the trend of vibration and acoustic emission level respectively.

### B.2.2 Test setup

Eight different test setups were used in the investigation (Table B.1), whereby four test setups (test-no 1 to 4) acted as reference samples without contamina-
B.2. EXPERIMENTAL SETUP

Figure B.2: Image of the free moving bearing carrier. a) top view, b) side view.

Table B.1: Test setup

<table>
<thead>
<tr>
<th>test-no.</th>
<th>lubricant type</th>
<th>contamination (wt)</th>
<th>number of samples</th>
<th>test duration [h]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>SL12-602</td>
<td>0%</td>
<td>2</td>
<td>24</td>
</tr>
<tr>
<td>2</td>
<td></td>
<td></td>
<td>1</td>
<td>168</td>
</tr>
<tr>
<td>3</td>
<td>SL12-603</td>
<td>0%</td>
<td>2</td>
<td>24</td>
</tr>
<tr>
<td>4</td>
<td></td>
<td></td>
<td>1</td>
<td>168</td>
</tr>
<tr>
<td>5</td>
<td>SL12-602</td>
<td>9.1%</td>
<td>4</td>
<td>24</td>
</tr>
<tr>
<td>6</td>
<td></td>
<td></td>
<td>2</td>
<td>168</td>
</tr>
<tr>
<td>7</td>
<td>SL12-603</td>
<td>9.1%</td>
<td>4</td>
<td>24</td>
</tr>
<tr>
<td>8</td>
<td></td>
<td></td>
<td>2</td>
<td>168</td>
</tr>
</tbody>
</table>

Tests and for each sample one pair of bearings was tested. Before the test the bearings were cleaned by an ultrasonic cleaning in heptane as a solvent and then lubricated with 3 g of grease. The reference samples were lubricated with uncontaminated grease and the samples of test 5 to 8 were lubricated with a mixture of 3 g grease and 0.3 g magnetite particles (Fe₃O₄). The size distribution of the contaminant is shown in Figure B.3. The contaminated grease was mixed in a beaker before it was applied to the bearings.
The greases SL12-602 and SL12-603 were used as lubricants. SL12-602 (non-EP grease) is a grease with a lithium thickener and a mineral base oil. Antioxidants are the only additives in the grease. SL12-603 (EP grease) is based on the grease SL12-602, but contains in addition a sulfur based extreme pressure additive added to the base oil of the grease.

The prepared bearing samples were tested during 24 hours and 168 hours respectively. For both cases the bearings were tested at a rotational speed of 180rpm and by applying an axial load of 2 kN, which results in a lifetime \( L_{10h} \) of 147,000 hours. During all tests the friction torque was measured in order to calculate the coefficient of friction and the vibration trend was monitored. In addition to the vibration trend, the trend of the acoustic emission signal was monitored for tests with a duration of 168 h. After the test durations of 24 h and 168 h the tested bearings were disassembled and cleaned by ultrasonic cleaning in a solvent (heptane) and the surface topography of the raceways was measured. For this measurements a 3D optical interferometry spectroscope (WYKO NT1100) was used.

### B.3 Results

All results showed good repeatability and the results of the different test duration are in line with each other. Patterns for surface measurement and trends in both vibration and acoustic emission measurements are consistent throughout the whole investigation.
B.3. RESULTS

B.3.1 Test duration of 24 hours

From a coefficient of friction (COF) point of view, the tests did not indicate any advantage of EP additives. Figure B.4 shows the coefficient of friction over the test duration of 24 hours. Addition of contaminants as well as EP-additives lead to an increase of COF in comparison to the reference sample.

![Graph showing COF over time for different samples]

Figure B.4: Average coefficient of friction over the test duration of 24 hours for the used test setups.

The measurements of the surface topography (Figure B.5) on the other hand showed a clear difference between EP and non-EP grease. Analyzing the surface features of the images of the raceways (Figure B.5) indicates similarities between four of the pictures. The image marked with a) is a new bearing, while the images b) and c) show tested bearing raceways of the non-EP grease and EP grease references without contamination. In all these samples the grinding grooves of the manufacturing process are still visible. The grinding grooves still remain in the sample with EP grease and a 10% particle contamination (Figure B.5-e). However the grinding grooves are not visible for the contaminated sample without EP-additives, as shown in Figure B.5-d.

Further enable the images (Figure B.5) of the surface topography a analysis of the surface roughness. For all uncontaminated tests (Figure B.5-a,b,c) the sur-
Figure B.5: Images of the surface measurements with interferometric spectroscopy after a test duration of 24 hours. a) unused bearing, b) bearing with non-EP grease, c) bearing with EP grease, d) contaminated bearing with non-EP grease, e) contaminated bearing with EP grease

face roughness parameter Rq is within a range of 100 nm to 130 nm. But for both contaminated tests (Figure B.5-d,e) the surface roughness has increased significantly. However the increase for tests using EP grease (Figure B.5-e) is

Figure B.6: Observation of vibration signal for the different test setups. Yellow: trend of an envelope signal with a range of 10 Hz-1 kHz. Blue: trend of an envelope signal with a range of 10 Hz-100 Hz. Observe the difference in scale!
much lower than for the non-EP grease tests (Figure B.5-d). While the surface roughness parameter Rq of non-EP grease samples increase to about 500 nm, the EP grease samples had a end value of about 300 nm. Further are pitting dents only visible for images of the contaminated non-EP grease samples (Figure B.5-d).

The vibration signals shown in Figure B.6 are the trend (B.1) of envelope signals with different frequency ranges. Beside test setup 5 (non-EP grease with contamination) all tests show a constant vibration trend signal. The vibration trends of the tests with non-EP grease and contamination are increasing during the 24 hours of testing. The end value increases to as much as three times the initial value, while for all other test-setups (test-no. 1,3,7 of Table B.1) the end value does not differ significantly in comparison to the initial one.

### B.3.2 Test duration of 168 hours

The friction behavior (Figure B.7) follows the exact same pattern as the 24 hour tests (Figure B.4), where both addition of EP additives and contamination increases the coefficient of friction. The values for COF are decreasing in all cases during a running in period, followed by a period were the COF stabilizes on a certain level. Further does the COF, as expected, fluctuate more by adding contaminants. Another observation is that the systems without EP additives and with contamination (Figure B.7-C) have a prolonged running in period in comparison to the other test setups.

As expected, the surface roughness did not change significantly for the uncontaminated test setups. In comparison to the unused reference sample, with a surface roughness of Rq=100 nm (Figure B.8-a), the surface roughness parameter Rq of the uncontaminated non-EP grease sample (Rq=150 nm, Figure B.8-b) and the EP-grease sample (Rq=115 nm, Figure B.8-c) increased only slightly. For the contaminated cases the elevated wear rate seen in the 24 hour tests maintained through the 168 hour tests and resulted in even rougher surfaces as seen after the 24 hour tests. Not just the surface roughness increase during the 168 hours tests, but also the deviation of the measured surface parameters increased for the contaminated samples. The surface roughness parameter Rq varied from 800 nm to 1000 nm in case of the non-EP grease (Figure B.8-d). While Rq for EP-grease samples were measured in between 300 nm and 500 nm (Figure B.8-e).

Comparing the vibration signals from the 24 hour tests and the 168 hour tests (Figures B.6 and B.9) some differences are observed, but the overall pattern seems to be maintained throughout both tests. As for the 24 hour tests, the 168
Figure B.7: Average coefficient of friction over the test duration of 168 hours for the used test setups.

Figure B.8: Images of the surface measurements with interferometric spectroscopy after a test duration of 168 hours. a) unused bearing, b) bearing with non-EP grease, c) bearing with EP grease, d) contaminated bearing with non-EP grease, e) contaminated bearing with EP grease
B.3. RESULTS

(a) Non-EP grease
(b) EP grease
(c) Non-EP grease with particle contamination
(d) EP grease with particle contamination

Figure B.9: Observation of vibration signal for the different test setups with an test duration of 168 hours. Red: trend of an envelope signal with a range of 10 Hz-100 Hz. Observe the difference in scale!

Hour tests have low constant vibration levels for uncontaminated samples (Figure B.9-a,b) and accelerated vibration levels for the samples contaminated with iron oxide (Figure B.9-c,d). Thereby is in general the vibration level lower for samples with EP grease (Figure B.9-d) in comparison to the samples with non-EP grease (Figure B.9-c). However the increasing trend seen from the previous 24 hour tests (Figure B.6-c) was not observed for the 168 hour tests (Figure B.9-c).

The detection of acoustic emission was only used for the samples with an test duration of 168 hours. But the results of the acoustic emission (Figure B.10) substantiate both the previous mentioned vibration responses of the tests and the observed differences between uncontaminated and contaminated as well as the differences between the two types of greases. Especially the difference of the acoustic emission between non-EP grease (Figure B.10-c) and EP grease is even more distinct in compare to the vibration responses. The differences in trend values of the double envelope of the acoustic emission signal of the test setups no. 6 and no. 8 (Table B.1) were as high as 70%.

Ploting the mean values of vibration and acoustic emission signals allows a di-
Figure B.10: Observation of acoustic emission signal for the different test setups with a test duration of 168 hours. Circles: trend of the double envelope signal of test bearing A. Squares: trend of the double envelope signal of test bearing B. Observe the difference in scale!

rect comparison of the different test setups (Figure B.11). For all measures the reference samples for non-EP grease and EP grease do not show a significant difference. Further are all reference measures several factors lower than the contaminated samples. The clearest difference was observed for the acoustic emission signals of the contaminated bearings used during the 168 hour tests. Also the average of the vibration signals of the contaminated tests for both the 24 and the 168 hours show a significant difference. However the difference is not as extreme for the vibration signals in comparison to the acoustic emission signals.

B.4 Discussion

All eight setups with contaminated rolling element bearings simulate the harsh environment in the iron mining industry and show a difference in vibration trend and surface topography already after the relatively short test duration of respectively 24 hours and 168 hours. The observed advantage of EP ad-
additives could have several reasons. The most obvious would be a reaction of the additive with the bearing surfaces in order to form sacrificing layers. The theory would explain the differences in both acoustic emission and vibration responses of different test samples as well as the difference in surface roughness. But this would not explain the results of the surface topography images shown in Figure B.5. The missing grinding grooves of samples with non-EP grease and contaminations (Figure B.5-d) compared to the other test samples could be a indication of higher wear rate, which is contradictory to the theory of sacrificing layers. This difference in wear rate between test setup no. 5 and 7 (Table B.1) could be a result of a possible chemical reactions (B.2) and (B.3) of the sulfur based EP-additive with the contaminants itself.

\[
2 \text{Fe}_3\text{O}_4 + 4 \text{S} = \text{R} + \text{O}_2 \rightarrow 2 \text{FeSO}_4 + 2 \text{FeS} + 2 \text{FeO} + 4 \text{R} \quad (B.2)
\]
\[
\text{Fe}_3\text{O}_4 + 3 \text{S} = \text{R} + 2 \text{O}_2 \rightarrow 2 \text{FeSO}_4 + \text{FeS} + 3 \text{R} \quad (B.3)
\]

Martin et al. [60] have shown that abrasive wear of iron oxide particles can be
reduced by a chemical reaction with additives. In Martin et al. [60] the reaction between hematite (Fe₂O₃) and ZDDP was investigated. Another study was carried out by Yuan-Dong et al. [59]. In this study they showed the favorable effect of FeS and FeSO₄ particles on the wear behavior of grease lubricated steel on steel interfaces. Both studies support the theory of a reduced wear rate due to a chemical reaction of parts of the contaminants with the EP-additives. Further the theory of a reaction between contaminants and EP additives is in line with the measured vibration and acoustic emission. Another result of the investigation is that vibration and acoustic emission measurements are consistent with the surface topography measurements, whereas the measurement of the friction coefficients is not related to the measured surface roughness topographies. At least not within the short test durations of 24 hours and 168 hours.

B.5 Conclusions

The investigation replicated the severe operating conditions for rolling element bearings in production lines of mining industries. It was shown that the use of EP additives could be favorable for those bearings working in such environments. In particular for the used operating conditions during the experiments EP additives were favorable. Contaminated test samples with EP additives showed decreased surface roughness, wear rate, vibration and acoustic emission in compare to the contaminated samples without EP additives. However in order to explain the mechanism for these improvements further analysis has to be done, especially chemical analysis of raceways and grease samples.

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Bibliography


