An Investigation into Scuffing and Subsurface Fatigue in a Lubricated Rolling/Sliding Contact
MATTI SAVOLAINEN

An Investigation into Scuffing and Subsurface Fatigue in a Lubricated Rolling/Sliding Contact

ACADEMIC DISSERTATION
To be presented, with the permission of the Faculty Council of the Faculty of Engineering and Natural Sciences of Tampere University, for public discussion in the Auditorium K1702 of the Konetalo Building, Korkeakoulunkatu 6, Tampere, on the 28th of June 2019, at 12 o’clock.
ACADEMIC DISSERTATION
Tampere University, Faculty of Engineering and Natural Sciences
Finland

Responsible supervisor
Professor Arto Lehtovaara
Tampere University
and Custos
Finland

Pre-examiners
Professor Ulf Olofsson
Royal Institute of Technology
Sweden
Professor Peder Klit
Technical University of Denmark
Denmark

Opponents
Professor Ulf Olofsson
Royal Institute of Technology
Sweden
Associate Professor Niels Pedersen
Technical University of Denmark
Denmark

The originality of this thesis has been checked using the Turnitin OriginalityCheck service.

Copyright ©2019 author

Cover design: Roihu Inc.

ISBN 978-952-03-1079-0 (print)
ISBN 978-952-03-1080-6 (pdf)
ISSN 2489-9860 (print)
ISSN 2490-0028 (pdf)

PunaMusta Oy – Yliopistopaino
Tampere 2019
The work presented in this thesis has been done at the Tampere University of Technology in the group of Tribology and Machine Elements, which is led by Professor Arto Lehtovaara. The group belongs to the laboratory of Materials Science. Before finalizing the thesis, the Universities at Tampere joined and the new organization became Tampere University. Professor Arto Lehtovaara guided the work and I would like to thank him for the valuable advices, support and fruitful discussions during this period.

I would also like to thank Lic. Tech. Olli Nuutila for the guidance on measurements and instrumentation, MSc Jouni Ketola for his contribution to the design and assembly of the upgrades to the twin-disc test device, PhD Suvi Santa-Aho for the support with the ultrasonic inspections and MSc Gabor Szanti, MSc Tami Komssi and MSc Jesse Rontu from ATA Gears for their co-operation. Additionally, I am grateful to the staff and colleagues working in various organizations within Tampere University and especially the members of the research group of Tribology and Machine Elements for their determinations on generating a supportive environment for enhancing ones skills on the studied topics.

A large part of the work was carried out under the auspices of the ArTEco (Arctic Thruster Ecosystem) project, which was a part of the Maritime Technologies II Research Programme (MARTEC II) under the European Commission ERA-Net scheme. I would like to gratefully acknowledge the financial support of Business Finland (formerly Tekes), and other partners in the project. Additionally I wish to thank TTY foundation, and the foundation of Yrjö and Senja Koivunen for the financial support towards the activities.

Finally yet importantly, I would like to express my thanks to my wife, the children and my parents for their efforts and patience over the years.

Tampere, 9th of April, 2019

Matti Savolainen
ABSTRACT

Unexpected failures of gears still occur although their durability has been well studied and comprehensive dimensioning guidance has long been available. One of the reasons for these failures is thought to originate from occasional overload cycles, which the gear sets may experience under extreme operating conditions.

Scuffing is a failure mode which occurs under heavy contact pressure and intensive sliding between the interacting surfaces in elevated temperatures. Scuffing failures result in damaged surfaces and a consequent decrease in load carrying capacity of the gear. This may also lead to other failure modes and ultimately to the breakdown of the machine component. Although this phenomenon has been widely studied, there are still some uncertainties, that are reflected in the dimensioning and use of gear sets. Another failure mode typically occurring in gears can be described as subsurface initiated fatigue. In contrast to traditional fatigue failure, in this mode, the crack initiation originates beneath the hardened layer of the case-hardened surface of a gear tooth.

Firstly the objective of this thesis is to shed light on the effect of occasional overloads on the initiation of scuffing. The second objective is to establish a procedure for investigating subsurface fatigue under rolling/sliding loading with high contact pressure; a typical condition in a gear contact. The experimental part consists of several tests with a pre-existing, but upgraded twin-disc test device with case-hardened specimens, where the loading in the tests mimics the various contact conditions between the tooth flanks in real gears. In addition to the analysis of the experimental results, the stress and strain states and the consequent formation of fatigue damage in the discs were calculated utilizing an elastoplastic finite element model of the test assembly.

The scuffing test results demonstrate the importance of running-in the interacting surfaces as overloading during the running-in process increased the risk of scuffing. Conversely, the introduction of overload cycles in a controlled manner and having periods of nominal load cycles between each series of overload cycles enhanced the
components resistance to scuffing. The tests also revealed that small deviations in the contact path allowed the use of increased normal load levels without the occurrence of scuffing failures.

The fatigue tests confirmed the suitability of the twin-disc test device for studying subsurface fatigue in case-hardened components. The related numerical analysis, including the influences of increased material hardness and residual stress, revealed critical depths beneath the hardened layer that coincided with the ones found in the experimental tests. Furthermore, a shear strain-based damage calculation method (Fatemi-Socie) seemed to be more sensitive to the effect of residual stresses, while inaccuracies in components due to manufacturing tolerances only have a minor effect on the formation of damage in the subsurface region.
Odottamattomia hammaspyörävaihteiden vaurioita esiintyy edelleen, vaikka niiden kestävyyttä on tutkittu laajasti, ja kattavia mitoitusohjeita on ollut saatavilla jo pitkän aikaa. Yhtenä syynä tällaisiin vaurioitumisiin on ehdotettu olevan satunnaiset ylikuormitusyklit, joita vaihteet saattavat kokea käytön aikana ankarissa olosuhteissa. Tahmautuminen (scuffing) on vauriomuoto, joka esiintyy vuorovaikutuksessa olevien pintojen välillä voimakkaassa kosketuspaineessa ja intensiivisen liukumisen vallitessa kohonneissa lämpötilaosuhteissa. Tahmautuminen voidaan havaita vauroituneina pintoina ja siten pienentyneenä kuormankantokapasiteettina, joka voi johtaa muihin vaurioitumoihin sekä lopulta komponentin hajoamiseen. Vaikka tästä ilmiöltä on tutkittu laajalti, on edelleen epävarmuustekijöitä, jotka heijastuvat vaihteistojen mitoitukseen ja käyttöön. Toinen vauriomuoto, joka tyypillisesti esiintyy hammaspyörissä, voidaan kuvata käsitteellä ”pinnan alta ydintyvä väsyminen”. Toisin kuin perinteisessä väsymisvaurioissa, tässä muodossa särön ydintyminen tapahtuu hiiletyskarkaistun hampaan pinnan karkaisukerroksen alapuolella.

Tahmautumistulosten mukaan vuorovaikutuksessa olevien pintojen sisäänajo osoittautui merkitykselliseksi; ylikuormitussyklit pintojen sisäänjautumisen aikana lisäsi tahmautumisvaurion ilmenemisriskiä, kun taas ylikuormitussyklien hallittu vastaanottaminen sekä nimelliskuormitusjaksojen sisällyttäminen ylikuormitussyklisarjojen välille lisäsi vastustuskykyä tahmautumisvaurion ilmenemistä vastaan. Lisäksi testit osoittivat, että pienet poikkeamat kosketusreitistä mahdollistivat suhteellisen korkean kuormitustason käytön ilman tahmautumisvauriota.

Väsytskokeet osoittivat kiekko-kiekkotestilaitteen soveltuvuuden hiiletyskarkastujen komponenttien pinnan alta ydintyvän väsymisen tutkimiseen. Tähän liittyvä numeerinen analyysi, sisältäen lisääntyneen materiaalikovuuden ja jäännösjännitysten vaikutukset, paljasti karkaisukerroksen alla olevat kriittiset syvyyt, jotka olivat yhteneviä kokeellisesti löydettyjen kanssa. Lisäksi leikkausvenymään (liukumaan) perustuva vaurion laskentamenetelmä (Fatemi-Socie) näytti olevan herkempi jäännösjännitysten vaikutukseissa, kun taas valmistuksen toleransseista johtuvilla eroilla komponentissa on vain vähäinen vaikutus vaurioiden muodostumiseen pinnan alla olevalla alueella.
# CONTENTS

Preface .......................................................................................................................................... i

Abstract ....................................................................................................................................... iii

Tiivistelmä ...................................................................................................................................... v

Abbreviations ............................................................................................................................... ix

Original publications ................................................................................................................... xiii

Author’s contribution .................................................................................................................. xv

## 1 Introduction

1.1 Bevel gear sets and testing gears ....................................................................................... 18
1.2 Scuffing ............................................................................................................................. 21
1.3 Subsurface fatigue ............................................................................................................. 23
1.4 The core of the thesis ......................................................................................................... 24
  1.4.1 Objectives and scope ................................................................................................. 25
  1.4.2 Outline and scientific contribution .......................................................................... 26

## 2 Methods

2.1 Twin-disc test device .......................................................................................................... 29
2.2 Test specimen and lubricant ............................................................................................. 32
2.3 Evaluation of the coefficient of friction ............................................................................ 33
2.4 Test procedures .................................................................................................................. 35
  2.4.1 Scuffing tests ............................................................................................................. 35
  2.4.2 Fatigue tests ............................................................................................................. 39
2.5 Finite element analysis ...................................................................................................... 40
  2.5.1 Material model ......................................................................................................... 42
  2.5.2 Residual stresses .................................................................................................... 43
  2.5.3 Calculation of damage ............................................................................................. 45
    2.5.3.1 Evaluation of shear stress/strain amplitude and normal stress .................. 47

## 3 Results and discussion

3.1 Scuffing failure .................................................................................................................... 49
  3.1.1 Findings related to scuffing initiation ................................................................. 53
3.1.2 Evolution of the friction coefficient................................................55
3.1.3 Evolution of bulk temperature.........................................................56
3.1.4 Analysis of the contact surfaces and material hardness ...............58

3.2 Subsurface fatigue failure....................................................................................61
3.2.1 Finite element analysis results ..........................................................62
3.2.1.1 Stress and strain results ...................................................65
3.2.1.2 Plastic strain results .........................................................68
3.2.1.3 Damage.............................................................................69
3.2.1.4 The effect of residual stresses.........................................69
3.2.1.5 The effect of the surface curvature..................................71
3.2.1.6 The effect of the case-hardening depth.........................72

4 Conclusions ..................................................................................................................73

References ....................................................................................................................77

Publications..................................................................................................................83
ABBREVIATIONS

a
b
DF
DFS
E
FN
k_F
k_FS
n
p
p_{max}
R_a
R_z
R_1, R_2, R_{disk}
r_1, r_2
T_c
T_b
T_{fmax}
T_{dc}
T_m
T_r
T_{ra}
W_{GSE}

Semi-major axis of an ellipse  
Semi-minor axis of an ellipse  
Findley damage  
Fatemi-Socie damage  
Young’s modulus  
Normal load  
Findley material constant  
Fatemi-Socie material constant  
Ramberg-Osgood material constant  
Pressure  
Hertzian maximum contact pressure  
Arithmetic mean height of surface roughness  
Average separation of five highest peaks and lowest valleys within the sampling lengths of surface roughness  
Test disc radius  
Radius of test disc curvature  
Surface temperature  
Bulk temperature  
Flash temperature  
Torque from disc contact  
Measured torque  
Torque from seals and bearings  
Approximated torque from seals and bearings  
Maximum generalized strain energy
x, y, z  Coordinate axis
x', y', z'  Rotated coordinate axis
z_{ss}  Depth of maximum shear stress

\alpha  Ramberg-Osgood material constant, stress tensor rotation angle
\beta, \gamma  Stress tensor rotation angle
\gamma_{12}, \gamma_{13}  Shear strain components
\Delta \gamma_{\text{max}}  Maximum shear strain range
\Delta \gamma_{e}  Elastic shear strain range
\Delta \gamma_{p}  Plastic shear strain range
\Delta \varepsilon_{\text{ne}}  Elastic normal strain range
\Delta \varepsilon_{\text{np}}  Plastic normal strain range
\Delta \tau  Shear stress range
\Delta \tau_{F}  Findley shear stress range
\varepsilon  Strain
\mu, \mu_{c}  Coefficient of friction
\sigma  Stress
\sigma_{0}  Yield strength
\sigma_{F}  Maximum normal stress in Findley damage criterion
\sigma_{n,\text{max}}  Maximum normal stress in Fatemi-Socie damage criterion
\tau_{12}, \tau_{13}  Shear stress component
\tau_{\text{max}}  Maximum shear stress
\omega_{1}, \omega_{2}  Angular velocity

C3D8R  Element type in Abaqus
CHD  Case hardening depth
DHD  Deep hole drilling
EHL, EHD  Elastohydrodynamic lubrication
<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>EP</td>
<td>Extreme pressure</td>
</tr>
<tr>
<td>FE</td>
<td>Finite element</td>
</tr>
<tr>
<td>FEA</td>
<td>Finite element analysis</td>
</tr>
<tr>
<td>FD</td>
<td>Fast rotating disc</td>
</tr>
<tr>
<td>GSE</td>
<td>Generalized strain energy</td>
</tr>
<tr>
<td>HL</td>
<td>Hydrodynamic lubrication</td>
</tr>
<tr>
<td>LE11, LE12</td>
<td>Logarithmic strain components</td>
</tr>
<tr>
<td>HRC</td>
<td>Hardness Rockwell</td>
</tr>
<tr>
<td>HV0.2</td>
<td>Hardness Vickers with load of 0.2 kgf</td>
</tr>
<tr>
<td>HV3</td>
<td>Hardness Vickers with load of 3 kgf</td>
</tr>
<tr>
<td>MCC</td>
<td>Minimum circumscribed circle</td>
</tr>
<tr>
<td>PEEQ</td>
<td>Equivalent plastic strain</td>
</tr>
<tr>
<td>RCF</td>
<td>Rolling contact fatigue</td>
</tr>
<tr>
<td>SD</td>
<td>Slow rotating disc</td>
</tr>
<tr>
<td>S11, S12</td>
<td>Stress components</td>
</tr>
<tr>
<td>TIFF</td>
<td>Tooth interior fatigue fracture</td>
</tr>
<tr>
<td>XD</td>
<td>X-ray diffraction</td>
</tr>
</tbody>
</table>
This thesis is based on the following publications:


Unpublished manuscript

IV M. Savolainen, A. Lehtovaara, *An investigation into subsurface fatigue in a rolling/sliding contact*. (Submitted)
AUTHOR’S CONTRIBUTION

By agreement of the authors of the publications:

Authors’ roles: Matti Savolainen was the main author in all the publications. His role comprised planning of the activities, preparation of testing and execution of them, building of the calculation models and carrying out solving of them, analysis of the results and writing of the publications. Prof. Lehtovaara was the second author in all the publications and he supervised the work, participated in planning and contributed in writing.

Publication I: As stated in the authors’ roles. Additionally, Savolainen designed updates to the twin-disc test device. Prof. Lehtovaara participated in designing and MSc. Jouni Ketola took part in designing, made the drawings and contributed in assembly of the updates to the apparatus.

Publication II: As stated in the authors’ roles. Additionally, Savolainen designed updates to the twin-disc test device. Prof. Lehtovaara participated in designing and MSc. Jouni Ketola took part in designing, made the drawings and contributed in assembly of the updates to the apparatus.

Publication III: As stated in the authors’ roles. Additionally, MSc Taru Karhula carried out the X-ray based residual stress measurements and the hardness measurements. Veqter Ltd. executed the residual stress measurements using the Deep-Hole-Drilling technique.

Publication IV: As stated in the authors’ roles.
1 INTRODUCTION

Tribology is the scientific study of interacting surfaces in relative motion and comprises topics such as friction, wear and lubrication. The term ‘tribology’ was defined by Jost in 1966 and is derived from the Greek root of the word ‘tribos’, ‘rubbing’. However, the concepts of friction, wear and lubrication have long been recognised. There are several examples of how early civilizations developed bearings and used different lubricants such as water or animal fat to reduce friction in wheel bearings or under dragged pieces of stone [1].

It was during the renaissance, when engineer-artist Leonardo da Vinci (1452-1519) first suggested a scientific approach to friction. Da Vinci deduced the two classic laws of friction; the friction force is directly proportional to the normal load and is independent of the apparent contact area. However, Da Vinci didn’t publish his findings at the time and these laws were rediscovered and formally proposed by Guillaume Amontons in the 17th century. In the 18th century Charles Augustin de Coulomb suggested the third basic law to cover kinetic friction; kinetic friction is independent of the sliding velocity. He also distinguished between static and kinetic friction [1].

Hydrodynamic lubrication (HL) was not identified until much later after the experimental studies of Beauchamp Tower (1884) and the theoretical analysis of Osborne Reynolds (1886) along with the work of N.P. Petroff (1883). Reynolds provided the first analytical proof that a viscous liquid can separate sliding surfaces by hydrodynamic pressure, which leads to low friction and theoretically to zero wear [1]. In addition, Heinrich Hertz published papers in the 1880’s, which provided the basis for the field of contact mechanics. His analytical formulae, based on the theory of elasticity, are still widely used in engineering for analysing contacts under high loading [2].
Classic HL theory assumes the bodies in contact to be rigid. However, pioneering studies in the field of elastohydrodynamic lubrication (EHL, EHD) overcoming this deficiency, were performed by Martin (1916) [3]. Later Grubin (1949) [4] proposed that the combination of hydrodynamics, elastic deformation of the metal surfaces and the increase in the viscosity of oil under extreme pressure contribute to the lubrication action. Later, Dowson and Higginson (1959) continued the work by computing numerical solutions for the EHL line contact [5].

Wilhelm Albert was probably the first person to record observations of metal fatigue. In the 1830’s he studied the failures of hoist chains in an iron mine and built a machine which repeatedly loaded a chain. From his experiments he deduced that metal fatigue was dependent on the load and the number of load cycles. In addition, in the 1840’s William Rankine recognized the effect of stress concentrations and in the 1870’s August Wöhler proposed that the cyclic stress range is more damaging than peak stress. Wöhler also introduced the concept of the endurance limit [6].

Since Wöhler’s time there have been massive developments in the field of fatigue research, of which only a few are mentioned here. For example in 1903 Ewing and Humfrey observed the formation of slip bands on the surface of a rotating-bending specimen; probably the first metallurgical description of the fatigue process. Later, in 1934, Polanyi proposed the dislocation theory [7], which was further developed into fatigue by Orowan in 1939 [8]. In the early 1950’s Coffin and Manson introduced the relation between fatigue crack-growth and plastic strain in the tip of crack [9, 10] and in 1973 Brown and Miller concluded that fatigue life under multiaxial conditions is determined in the critical plane which receives the most damage due to tension and shear loads [11].

1.1 Bevel gear sets and testing gears

In bevel gears, the axes of the two shafts intersect. The angle between the shafts is often 90 degrees, although other angles are used. One of the most common uses for bevel gears is in a vehicle differential gear. Another typical application for bevel gears can be found in ships as a part of a maneuvering thruster. An example of a ship’s bow thruster is shown in Fig. 1. In this system a motor in the ship drives the propeller via bevelled gears.
Bevel gears can be categorized into four basic types according to the shape of the teeth: straight, Zerol\(^1\), spiral and skew tooth. In addition, the method of manufacture such as the face milling (face hobbing and tapered hobbing) produces slightly different tooth geometries [12].

![Figure 1. A bow thruster with a bevel gear [13].](image)

Experimental research into gears has traditionally focused on testing spur gears by utilizing the FZG test device [14]. Several studies have been carried out, which have focused on comparing different lubricants, the effects of extreme thermal conditions, the viscosity of the lubricant and the role of the surface roughness on gear failures such as pitting or micropitting [15-18]. Most recently, research into bevel gear-based devices have concentrated on improving the gear design through geometric parameters and the development of finite element-based methods for analysing of gear teeth in contact [19-21].

The above approaches take into account the precise geometry, surface roughness and surface treatment of the gear teeth. However, in gears, the contact condition varies greatly along the line of action. During operation, the combined radius of curvature, contact pressure distribution and the sliding/rolling

---

\(^1\)Zerol is a registered trademark of Gleason Works, Rochester, New York.
velocities on the tooth surface are constantly changing. Therefore, it is challenging to analyze instantaneous contact locations in detail and to apply complicated load histories to these systems. Nevertheless, it is widely accepted that the elliptic trace of the pressure distribution between the teeth of a bevel gear can be mimicked with an precise arrangement of the twin-disc test device. This justifies the use of a twin-disc test system, as it is possible to generate steady-state conditions in the contact and implement accurate dynamic control of the loads. However, it should be noted that the use of the achieved results for the dimensioning of gears requires additional attention of many factors such as the formation of the lubricant film, the coefficient of friction and the topography of the surfaces in the contact [22].

Fig. 2 illustrates the fundamentals of the twin-disc test arrangement. The contact pressure is generated by pushing the discs together with the normal force $F_N$. The elliptic pressure trace between the discs can be reproduced by adjusting the radius values $r_1$ and $r_2$. The rolling/sliding condition in the contact and the variation of the relation between rolling and sliding is handled by adjusting the rotation speeds of the discs ($\omega_1, \omega_2$).

![Figure 2. The principle of the twin-disc test device.](image-url)
1.2 Scuffing

Wear between interacting surfaces after a short period of operation under intense surface pressure and high sliding velocity is described as scuffing. According to Ref. [12] scuffing can be categorized as follows:

**Cold scuffing:** this failure mechanism is more filing or abrading away of material and usually occurs in slow-speed gears at a relatively low temperature, where there isn’t any or only a partial elastohydrodynamic (EHD) oil film present (boundary or mixed lubrication regimes).

**Hot scuffing:** this failure mechanism is more of welding and tearing away material, which usually occurs in high-speed applications with high sliding velocity in elevated temperature condition.

This study focuses on hot scuffing (hereafter referred to as ‘scuffing’). These scuffing failures can be attributed to the high temperatures, which occur in the EHD contacts due to frictional heating. The temperatures may be so high that the lubricant decomposes resulting in an insufficient thickness of the lubricant film. This in turn increases the probability of asperity collisions, which simultaneously increases local pressure and temperature values within the EHD contact and thus raises the probability of the initiation of scuffing. The resulting damage can usually be observed as a rough or matte texture on the flank of the gear tooth. Although the system may recover from a mild localized scuffing, severe scuffing over a large area often causes increased noise and vibration levels, which may lead to other gear failure modes [2, 23].

One significant special characteristic of scuffing is that it can be initiated by a single destructive load cycle. In contrast, most other failures are usually initiated and progressed after a relatively large number of load cycles. Furthermore, running-in period of a new gear set, when the surface asperities on a tooth have not yet been smoothed, is also a high-risk period for the initiation of scuffing [24].

The risk of scuffing failure varies according to certain properties of the gear set such as the gear material, the lubricant, the surface roughness of the tooth flanks, the
sliding velocities and the loading. Probably the most commonly used method to estimate the scuffing risk, also utilized by standardization organization [26], is the flash temperature approach, which was originally developed by Blok (1937) [26] and further developed by Jaeger (1942) [27] and later by Archard (1959) [28]. The basis of the flash temperature method is to calculate the rise in the tooth surface temperature at a given point along the line of action over the teeth interaction. The maximum surface temperature \( T_c \) is considered to be the sum of the bulk temperature \( T_b \) of the interacting solids before entering the contact and the flash temperature \( T_{fmax} \) occurring in the contact.

\[
T_c = T_b + T_{fmax}
\]  

In order to obtain the risk of scuffing, the calculated temperature is compared with the corresponding tested value of the specific lubricant and the material pair. This load carrying capacity can be found by conducting scuffing tests according to ISO 14635-1 [29]. Alternatively, it can be determined by field investigations.

Regarding the estimation of heat generation in contacts, Fernandes et al. [30] recently proposed an approach to predict both the bulk and the flash temperatures based on thermal finite element analysis using the capabilities of a validated gear power-loss-model. Although they concentrated on the load carrying capacity of polymer gears, which are often operated in dry conditions, their model also enables the prediction of the bulk temperature for oil jet and dip lubrication conditions. In addition, Li et al. [31] investigated the flash temperature rise of a spur gear contact under tribo-dynamic condition, in which the interaction of the gear dynamics and the gears’ tribological behaviour is included in the analysis. By comparing the tribo-dynamic and quasi-static conditions, they concluded that gear dynamics play an important role and that not only the line-of-action direction but also the off-line-of-action direction vibratory motion influences the surface flash temperature.
1.3 Subsurface fatigue

Fatigue failure is usually initiated from or close to the surface of the component. In surface hardened components, such as gear wheels, the failure may also initiate beneath the hardened layer where the strength of the material is significantly lower than at the surface. In addition, although the compressive residual stresses generated during the surface treatment process do not have any effect in this region, balancing tension residual stresses may occur. These residual stresses can have a detrimental effect on the component’s fatigue life by accelerating crack growth. Consequently, Tooth Interior Fatigue Fracture (TIFT), which describes a mode of gear fatigue failure, has also been suggested [32]. In this failure mode, the cracks are initiated inside the tooth under specific load conditions.

Fatigue testing of gear tooth flanks has long been conducted utilizing the FZG test device and by noting the generation of pitting or micropitting marks on the contact surfaces [33-36]. The strength of the material, in turn, has generally been investigated with bending or push-pull tests on specimens machined out of the particular material and handled with a specific hardening treatment [37, 38]. In addition, the effects of the geometry, the surface hardening and the surface roughness of the tooth have been taken into account in the results of bending tests on a single gear tooth [39–41]. However, studying the effects of the combination of dynamically adjustable rolling/sliding loading with high contact pressure on the surface and crack initiation beneath the hardened layer at the same time is more challenging.

Fatigue beneath the surface under rolling/sliding loading is complex. The stress history is usually non-proportional, meaning that the stress components do not rise and fall at the same time, and the state of the stress is multiaxial. It is also commonly accepted that cracks mainly nucleate and grow in certain planes rather than being randomly oriented. The direction of these planes is defined by the material characteristics and the state of the loading. This concept has been indicated in several studies [42-44], which support and substantiate the use of a critical plane approach. Several critical plane approaches have been proposed over the years for observing the maximum shear or maximum principal planes [45-52]. They can usually be
categorised into stress-based, strain-based, stress-strain-based or energy-based models.

Various numerical analyses of rolling contact fatigue (RCF) have focused on the wheel-rail contact of trains [53-55]. A study by Ringsberg showed that a strain-life approach with elastic-plastic FE analysis is an effective combination for predicting crack initiation in RCF [56]. On the other hand, MackAldener successfully utilized a stress-based criterion for analysing the fatigue life of gears in the event of tooth interior fatigue fracture [32]. Furthermore, Miyashita et al. studied the rolling contact fatigue of a sintered alloy in a two-cylinder contact. According to the results of their experimental and finite element analysis, the depth of the subsurface cracks coincided with the depth of the maximum shear stress range [57].

1.4 The core of the thesis

The motivation for this research originates from the unending demand to improve the performance of machines. In most cases this means that the components in a machine must be able to withstand higher loads without any substantial increase in their physical size and/or mass. The best approach to tackling this issue is to increase the understanding of how failure modes are initiated and how they lead to a failure of the component. Any knowledge gained can then be used in dimensioning the components and modifying the way in which the machines are used.

Bevel gears are commonly dimensioned to directly account for several failure modes such as pitting, scuffing and tooth root bending fatigue. Nevertheless, even though the behavior of these gear sets have been studied for years and dimensioning guidance is widely available, component failures still occur. The root cause of these failures is not always clear and therefore, it has been proposed that the gears experience loads, which are not sufficiently accounted for by the existing dimensioning rules. Since scuffing is a failure mode, which may occur after only a couple of destructive load cycles, more information about the failure initiation under such load situations is required.
A failure mode of subsurface initiated fatigue leading to fracture usually causes the whole gearing system to stop working. The challenge for research into this failure mode is to find a means to investigate the mode under controlled conditions in a laboratory environment with high contact pressure and damaging rolling/sliding loading. Although the obvious option is to utilize real gears for the research this often restricts the possibilities of applying widely varying dynamic loads to the system.

Modelling and calculating subsurface fatigue is an intricate procedure. The gear teeth are often surface hardened, which means that the material properties in the hardened region change and the occurring residual stresses may also have an effect on the results. In addition, it is important to take the non-proportional and multiaxial stress or strain state beneath the surface into account in the fatigue analysis in order to improve the accuracy of the fatigue life prediction for a particular gear set.

1.4.1 Objectives and scope

Much of this work has been done under the auspices of a project, whose goal was to improve the durability of thruster technology in ships. The bevel gear sets, in these systems, are part of a chain delivering power from the engine or electric motor to the propeller. These gear sets may be exposed to occasional overload cycles because of the ship’s propeller impacting on solid objects such as ice floes, when operating in polar regions. Such impacts can lead to scuffing failure, or at least have an effect on the initiation of subsurface fatigue failure. Therefore, the research questions of this thesis, formulated by the author in co-operation with the supervisor, are:

1. What is the effect of occasional overload cycles on scuffing initiation?
2. How can the twin-disc test arrangement be effectively exploited in research into subsurface-initiated fatigue?

It is commonly accepted that the twin-disc test apparatus can be used to mimic the pressure trace between bevel gear teeth by generating an elliptic pressure distribution between the discs. This research first focuses on scuffing tests carried out with a pre-existing twin-disc test device. However, before these experiments were done, the device was updated in order to extend the magnitude of loading in the tests and to
enable the use of dynamically varying loads. Several tests were carried out with case-hardened discs made out of 18CrNiMo7-6 (EN 10084). These tests included variations in the loading in both the radial direction of the discs, which has a direct effect on the contact pressure, and in the axial direction of the discs, which influences the lubrication condition between the discs. The achieved results were analyzed in order to gain an understanding of the initiation of scuffing and the behavior of the coefficient of friction under occasional overload cycles in a sliding/rolling contact under high contact pressure.

The second part of the research consists of fatigue testing of case-hardened discs with the twin-disc test device by concentrating on subsurface initiated cracking and analysis of the test conditions using the finite element method. A procedure for finding the cracks beneath the surface was established and a finite element model of the test arrangement was constructed, which enabled the study of stress/strain states in the subsurface domain. Furthermore, using the discs as an example, the role of the residual stresses and tolerances related to manufacturing, were evaluated. Finally, the fatigue in case-hardened components is discussed, which opens up the possibilities for further research and implementation of the outcomes for dimensioning gear sets and for modifying operating procedures.

1.4.2 Outline and scientific contribution

The main part of the thesis is composed of four chapters: 1. Introduction, 2. Methods, 3. Results and Discussion and 4. Conclusions. This first chapter began with a literature review of the relevant research into scuffing and fatigue. The chapter went on to describe the motivation for the research, its objectives and scope and an outline of the scientific contribution of the work.

The second chapter describes the methods, materials and equipment used in the experiments. This includes an explanation of the twin-disc test system and the instrumentation used with the test apparatus. It also describes the test specimens, the test procedures and the test matrix. The chapter concludes by clarifying the calculation approaches and the models utilized to analyze the test cases.
The main results of the tests are presented in the third chapter. This chapter is divided into two sections. The first section deals with the scuffing failure data. The second section processes the subsurface fatigue test data, the results of the related finite element analysis and the formation of fatigue damage in the discs. The chapter also discusses the effects of the tolerances related to the manufacture of case-hardened components and evaluates the effect that the residual stresses have on subsurface fatigue of the discs.

The final chapter summarizes and presents the main conclusions of the work.

The main scientific contribution of the thesis can be summarized as follows:

a) An increased understanding on the relation of overload cycles to scuffing initiation under high contact pressure. The overload cycles are applied in both the normal direction to the contact and as additional axial displacement loading, which causes deviations in the nominal rolling/sliding direction.

b) An improved understanding of the behavior of the coefficient of friction and bulk temperature in a rolling/sliding contact under high surface pressure and overload cycles, and the effect of the load conditions on the material at and near to the component surface.

c) The development of an approach to studying subsurface-initiated fatigue in case-hardened components under rolling/sliding loading and high surface pressure.

d) Enhancement of knowledge in the effect of manufacturing tolerances on subsurface fatigue in case-hardened components under contact loading and the role of residual stresses in the formation of fatigue damage.
2 METHODS

The first part of the work focuses on experimentation with a twin-disc test device aiming to produce scuffing failure at the surface or crack initiation in the subsurface region. The second part concentrates on modelling the fatigue test arrangement using the finite element method.

2.1 Twin-disc test device

The twin-disc test system used in the experiments was originally developed in-house [58]. As a preliminary task for this research work, the device was modified to be able to implement dynamic loading to the contact in the radial direction of the disc, and also to be able to vary the axial position of one of the discs in order to deviate the contact path on the opposite disc. Both of these loads are generated using hydraulic cylinders, which are controlled via fast regel valves. The update required the renewal of the nominal force cylinder, the installation of a new cylinder for the axial movement, a new hydraulic valve, a new controller for the hydraulics and reprogramming. In addition, a new hydraulic unit was purchased and the mechanics of the twin-disc assembly were modified so that they are suitable for the axial displacement loading. The twin-disc test device and the load frame of the system are shown in Figs. 3 and 4.
Figure 3. The twin-disc test device.

A separate hydraulic unit was set to provide lubricant for the rolling bearings inside the load frame and to the disc contact. The lubricant feed was placed to flow from the inlet side to the contact according to the rotational direction of the discs. The lubricant temperature before entering the contact is automatically controlled by a microcontroller, while the flow is adjusted by setting the speed of the hydraulic pump in this system.

A new system for collecting the measurement data was built as well. A force transducer, mounted on the back of the cylinder providing the normal force, measures the load $F_N$ applied to the contact. The axial displacement of one of the discs is measured by sensing the location of the free end of the shaft of the through-rod-type cylinder that provides this movement. This cylinder is connected to the bearing housing of the fast rotating shaft. This housing and the cylinder for the normal force are coupled to the frame structure with joints that allow the axial movement.
Figure 4. The load frame of the twin-disc test device.

The rotation speeds and directions of the shafts were adjusted with frequency converters controlling the electric motors. The motors were mounted on the shafts with flexible couplings to allow small deviations between the joined axles during operation. The measurement system collected data about the rotation speeds of both shafts. The torque in the slow rotating shaft was also measured and this measurement was used to calculate the friction coefficient between the discs. Moreover, the bulk temperature of the disc on the fast rotating shaft was recorded using a telemetry system. A thermocouple sensed the temperature at approximately 3.5 mm beneath the surface in the middle of the contact of the discs. In addition, the control system of the device was built in the way that enabled unmanned operation. Some performance values of the device are presented below:

- Rotation speed: Max 6000 RPM
- Normal load: Max 50 kN
- Lubricant temperature: 25…120 °C
- Lubricant flow rate: 0.5…20 l/min
2.2 Test specimen and lubricant

The discs were made of 18CrNiMo7-6 (EN 10084) and case-hardened to a depth of approximately 1.1 mm with a surface hardness of 59-61 HRC. The width of the ground contact area is 10 mm and the outer diameter is 70 mm. The elliptical contact trace was generated by grinding the discs on the fast-rotating shafts to a radius of 100 mm in the axial direction \( r_1 \) in Fig. 2, while the discs on the other shaft were manufactured to be flat (infinite radius \( r_2 \) in Fig. 2). The grinding was done with a special device for the purpose [58] resulting in a surface condition comparable to the surface of a real gear tooth with perpendicular grooves into the direction of the sliding and rolling. The grinding removed roughly 0.25 mm from the disc surface and a typical finished contact surface is shown in Fig. 5.

![Contact surface of a test specimen with perpendicular grinding marks.](image)

All the discs were ground after being assembled on a shaft. This was done in order to minimise the risk of eccentricity error between the contact surface and the mounting areas of the support bearings. The requirement for the surface roughness in terms of the average surface finish \( (R_a) \) ranged from 0.28 to 0.32 \( \mu \text{m} \) and the average distance of the highest peak from the lowest valley \( (R_z) \) had to be under 3 \( \mu \text{m} \).

A commercial mineral oil-based lubricant that is commonly used in marine propulsion equipment was applied to the tests. The oil is equipped with an EP-
additive system and is recommended for heavy-duty industrial gears. The oil specifications are shown in Table 1.

Table 1. Specification of the lubricant used in the tests.

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Kin. viscosity @40°C [mm²/s]</td>
<td>150</td>
</tr>
<tr>
<td>Kin. viscosity @100°C [mm²/s]</td>
<td>15</td>
</tr>
<tr>
<td>Density @15°C [kg/m³]</td>
<td>897</td>
</tr>
<tr>
<td>VG class [-]</td>
<td>150</td>
</tr>
<tr>
<td>Flash point (COC) [°C]</td>
<td>240</td>
</tr>
<tr>
<td>Pour point [°C]</td>
<td>-24</td>
</tr>
</tbody>
</table>

2.3 Evaluation of the coefficient of friction

The measured torque signal $T_m$ on the slow-rotating shaft, can be divided into the torque $T_r$, which originates from the shaft seals and the bearings, and the torque $T_{dc}$, which is derived from the contact between the discs.

$$T_{dc} = T_m \pm T_r$$  \hspace{1cm} (2)

The value of $T_r$ depends on the load conditions, meaning that when a decelerating torque is recorded on the slow-rotating shaft the sign of the $T_r$ is negative and in the opposite situation, it is positive. Typical values for the torque $T_r$ were found by measuring the torque on the shaft as a function of the normal force in pure rolling state, while the rotation speed of both the shafts was set at 175.8 RPM. The working temperature during these tests was 60 °C and the test was repeated with three different disc pairs. The resulting data is shown in Fig. 6.
A load-dependent equation fitted into the results, can be formulated as follows:

\[ T_{ra} = 0.0801 \cdot \ln(F_N) - 0.0265 \]  \tag{3} 

where the units are [N] for the \( F_N \) and [Nm] for the resulting torque \( T_{ra} \). Due to the variation of the torque values in the range of 5 to 12 kN in the normal force as shown in Fig. 6, a sensitivity check was performed utilising the following equation to describe the friction between the discs:

\[ \mu_c = \frac{T_m + T_r}{R_{disc} F_N} \]  \tag{4} 

Firstly, the torque \( T_m \) was calculated by setting the radius of the disc \( R_{disc} \) to 0.035 m and the friction coefficient \( \mu_c \) to 0.03 in the disc contact. The \( T_r \) was approximated with Eq. (3). Secondly, the torque \( T_r \) was first set to 0.4 Nm and then to 1.0 Nm to evaluate the ensuing deviation in the friction coefficient. Fig. 7 illustrates the resulting error range in the coefficient of friction as a function of the normal load. The error band for the friction values was considered insignificant as it varied between roughly 0.003 and 0.001. The values provided by Eq. (3) were utilized in the data analysis in all the test runs.
2.4 Test procedures

All the tests began with a preliminary step, where the temperatures of both of the shafts and the lubricant were stabilized to 60 °C. This is within the range for industrial gears in power transmission, which typically vary from 40 °C to 100 °C [59]. During this step, both of the shafts were rotated at a speed of 175.8 RPM without any loading, but with lubricant circulation. The lubricant flow to the inlet of the contact was set to 6 l/min, which was found to deliver an adequate lubrication, but enabled a clear increase in the bulk temperature during the rise of a load step according to preliminary testing.

2.4.1 Scuffing tests

The rationale for planning the scuffing tests was to produce load situations, which may take place in a bevel gear set working as part of the propulsion system of a ship travelling in harsh conditions. This could mean for example passing through icy seas, where the gear set experiences high loading for a long time just to maintain the ship’s movement. Under such conditions, the propeller may also be hitting ice floes, which in turn, can lead to additional loading (overload cycles) in the tooth contact.
Furthermore, the rough environment can cause deformation in the ship’s power line generating displacement in the alignment of any two gear wheels in contact. This can disturb the lubrication film between the gear teeth causing a detrimental increase in the number of asperity contacts between the interacting gear teeth.

Two types of scuffing tests were carried out. The first one was based on varying the normal force and the second concentrated on observing the effect of applying axial movement to the fast-rotating disc. During the scuffing tests, the speed of rotation of the fast-rotating shaft was 2039.3 RPM, while the slow-rotating one revolved at a speed of 175.8 RPM. As a result, with the disc diameter of 70 mm, the sliding speed at the surface was 6.8 m/s and the slide-to-roll ratio 1.68. This can be considered as a relatively high value increasing the probability of a scuffing failure to occur. The scuffing experiments can be categorized into four groups:

A. Reference test
B. Addition of a series of rapid sudden overload cycles with high magnitude
C. Addition of a series of sudden overload cycles with increasing magnitude
D. Addition of axial movement cycles
   a. Starting simultaneously with normal force increase
   b. Starting in the middle of the execution of a normal force step (stabilized normal load condition)

In all the test groups, a reference test (A) with stepwise increasing normal force $F_N$ was used as the starting point. The outline of this test is shown in Fig. 8.
Figure 8. Ten steps of increasing loading in the scuffing tests.

The steps in Fig. 8 illustrate the maximum Hertzian pressure $p_{\text{max}}$ in the contact of the discs as follows:

$$p_{\text{max}} = \frac{3F_N}{2\pi ab}$$

where $b$ and $a$ are the semi-minor and semi-major axes of the contact ellipse respectively. The load rises from base level to step 1 and from step 1 to step 2 were slightly greater than the other steps because controlling the normal force at low load levels with the current test setup for the device is less accurate. The step increases from 2 to 6 were about 230 MPa, which represented the running-in of newly-ground gear teeth surfaces in real gear sets. The subsequent load increases after the settling period were 150 MPa per step. Each step was set at 1000 s, which allowed the temperature and the torque signals to stabilise after each load rise. In the reference test, the increases in the normal load continued until scuffing occurred.

In test type B, the goal was to apply overload cycles rapidly after an increase of the normal load meaning during the settling process. The overload cycles were added in a series of $25 + 25$ cycles. This means that the first series were at the definite overload level and the following series at the nominal level. This alternation continued without any increases in the normal load. Fig. 9 (a) illustrates a series of overload cycles from level 7 to level 9.
In test type C the aim was at controlled progressive increase of the magnitude of the overload cycles. In contrast to type B, the overload cycles were introduced to the system at the chosen level after stabilisation. The first overload cycles were introduced in magnitudes that reached the following load level and were then steadily increased to be one level greater than the previous ones. The overload period always contained a series of 25 + 25 cycles in the same way as with the type B tests. Each series of overload cycles was followed by a 1000 s period at the starting load level. Fig. 9 (b) illustrates the load scheme for the type C tests.

In test type D, the increase of the normal force followed the practice for the reference tests. In addition, when a certain load level was reached a series of continuous axial displacement cycles were applied. The difference between test types Da and Db is the moment of the start of these cycles. In the first one, the cycles were added at the same time as the normal force increase and in the latter, they were applied in the middle of the load step, which means that the stabilization process to that particular normal load level has finished. The displacement load moved the fast-rotating disc and the oscillation was ±1 mm on both sides around the middle point of the disc contact. This amplitude corresponds to about 50 % of the length of the semi-major axis of the contact ellipse at load level 7, where the maximum Hertzian pressure is about 2.1 GPa. The frequency of the oscillation was about 12 Hz and the profile represented a saw-tooth pattern as illustrated in Fig. 10. This track is roughly 0.3 % longer than the path of a straight-line and therefore caused insignificant
change in the sliding speed. It is also worth of noting that the track is not exactly the same on the discs in the circumferential direction between consecutive rotations.

![Disc circumference [mm]](image)

**Figure 10.** When axial movement was applied, the middle point of the contact trace follows the dotted line on the slow-rotating shaft with an inclination of 4.3 degrees.

All the tests were continued until either 3 GPa in maximum Hertzian pressure was reached or scuffing occurred.

### 2.4.2 Fatigue tests

In contrast to the scuffing tests, the aim of the subsurface fatigue experiments was to collect the load cycles on the discs, which may lead to crack initiation under the surface layer. A further target was to establish a procedure for analyzing the discs in order to reveal cracks in this region.

The surface sliding speed was relatively low, which is also what occurs during operation midway up a bevel gear tooth, where the rolling load dominates and contact load is the highest [13]. For all the fatigue tests, the speed of rotation was 2039.3 RPM for the fast-rotating shaft and 1447.4 RPM for the slow-rotating one. The resulting sliding velocity was thus approximately 2.2 m/s, the mean velocity for rolling 6.4 m/s and the slide-to-roll ratio 0.34.

The normal force was ramped up for about 60 s at the start of each test. After reaching the planned load level, it was kept at the same value until it reached the
The cycles during the ramp-up were disregarded when counting the number of cycles.

After the tests, all the discs were removed from the shafts with a hydraulic press and inspected with ultrasonic equipment in order to find possible locations of subsurface cracks. Then each disc was cut into four segments at crack-free locations allowing the cross section of a disc to be studied in the plane that pierces the mid-point of the elliptic pressure trace at the contact surface. Fig. 11 illustrates the inspection surface of a typical quarter piece of a disc.

![Quarter section of a disc.](image)

Figure 11. Quarter section of a disc.

Only two out of every four of the quarter pieces were inspected, since cutting removed a fair amount of material from the two other pieces. The surface of the two remaining quarter-pieces were ground and polished until they were smooth enough for any cracks to be seen with an optical microscope. During this procedure the disc pieces were cooled with a continuously circulating water-based coolant in order to minimize heat generation. Before the optical inspections, the surfaces were etched with a 4% solution of nitric acid and ethyl alcohol (Nital).

### 2.5 Finite element analysis

Case-hardening has a significant effect on the material properties near the surface of the treated component. Therefore, the change in hardness and the residual stresses were taken into account when building the FE-model of the twin-disc assembly. The
modelling was done using Abaqus R2017x. The model and a close-up of the cross-section of the contact region are shown in Fig. 12.

Figure 12. The FE-model of the shafts in the twin-disc test device on the left; a cross section of the contact region on the right.

The FE-model is about 2.9 MDOF in size and was made with first-order hexahedron elements handled with the reduced-integration method (C3D8R). This model consists of four parts: Two discs, which have been shrink-fitted onto two shafts using an overclosure (negative clearance) of 0.015 mm between the surfaces in contact. The roughness of the surfaces was not modelled. The mesh near the contact was attached with tied-contact to the adjacent elements as shown in Fig. 12. The elements near the contact were modelled as 0.1 mm-thick layers up to a depth of 3 mm. The size of the elements in the other directions was set to about 0.2 mm.

There are two bearings supporting each of the shafts. Nodes were positioned at the longitudinal axes of the shafts in the middle of each bearing at locations, where the bearings are assembled in reality. The surface nodes on the shafts at these four locations, each covering the area of a bearing width, were connected to the corresponding middle node using rigid beam elements. Boundary conditions and loads were applied to these nodes: All the rotary degrees of freedom were free for both of the shafts to allow torque loading and a realistic simulation of the behaviour of the bearings. The fast shaft was also allowed to travel in the radial direction to be able to engage contact with the slow shaft when the normal load was active.

A “finite sliding” type of contact was used between the discs with a coefficient of friction of 0.024 that was derived from earlier tests. The coefficient of friction in the shrink-fit between the shafts and the discs was set at 0.12 and the “small sliding” contact-type was applied.
The very thin lubricant film between the discs was not specifically modelled, since it’s effect on the stresses and strains in the subsurface region was considered to be insignificant. However, the performance of the model was checked by comparing the maximum Hertzian contact pressure values, which had been found analytically with the corresponding data from an elastic FE-model of the disc assembly. The pressure data between 1.5 to 2.5 GPa were the maximum of 4% lower with the finite element approach in comparison to the analytical results and thus it was concluded that the model was adequately accurate.

2.5.1 Material model

The elasticity (Young’s) modulus, the Poisson’s ratio and the thermal expansion of the material, were set at 210 GPa, 0.3 and 1.2·10⁻⁵ K⁻¹, respectively. The elastoplastic behaviour was modelled using the connection between stress and strain according to Ramberg-Osgood [60]:

\[
\varepsilon = \frac{\sigma}{E} + \alpha \frac{\sigma}{E} \left( \frac{\sigma}{\sigma_0} \right)^n
\]

where \( \varepsilon \) is strain, \( \sigma \) is stress, \( E \) is Young’s modulus, \( \sigma_0 \) is the yield strength of the material and \( \alpha \) and \( n \) are material parameters. The \( n \) can be calculated by fitting with the tensile data and \( \alpha \) can be evaluated from the yield strength by assuming a 0.2% yield offset. The yield and the tensile strengths were solved separately for each element layer through the hardened region and for the bulk material according to Pavlina et al [61] as a function of hardness. Fig. 13 illustrates the resulting curves and the material hardness as fitted through the measurement points by including the effects of removing the material through grinding.
Because of the case-hardening, compressive residual stresses occur near the surface because of volumetric expansion in this region. In the FEA, the residual stresses were modelled by introducing a temperature load to the layer of elements near the surface. Thus compressive residual stresses were generated due to thermal expansion. Consequently, balancing tension residual stresses occurred deeper in the material.

The applied temperature field is based on two separate residual stress measurements. The first one used the x-ray diffraction (XD) method, while the other used the deep hole drilling (DHD) approach. These two measurements were taken from different unground discs in the middle of the contact surface perpendicular to the surface. A characteristic of the XD method is that the results may be less accurate the deeper in the material the measurement is taken. This is due to possible stress relaxation and curvature changes in the measured surface, caused by the removal of material by the electrochemical etching before each measurement cycle. In contrast, the DHD measurements taken from near the surface of the component tend to be less accurate than the measurements from deeper in the material [62]. Therefore, the two curves were combined in order to construct the residual stress field for the FE-model. Fig. 14 illustrates the measured residual stress data and the calculated curve occurring in

---

**Figure 13.** Yield and tensile strengths along with hardness.

2.5.2 Residual stresses

Because of the case-hardening, compressive residual stresses occur near the surface because of volumetric expansion in this region. In the FEA, the residual stresses were modelled by introducing a temperature load to the layer of elements near the surface. Thus compressive residual stresses were generated due to thermal expansion. Consequently, balancing tension residual stresses occurred deeper in the material.

The applied temperature field is based on two separate residual stress measurements. The first one used the x-ray diffraction (XD) method, while the other used the deep hole drilling (DHD) approach. These two measurements were taken from different unground discs in the middle of the contact surface perpendicular to the surface. A characteristic of the XD method is that the results may be less accurate the deeper in the material the measurement is taken. This is due to possible stress relaxation and curvature changes in the measured surface, caused by the removal of material by the electrochemical etching before each measurement cycle. In contrast, the DHD measurements taken from near the surface of the component tend to be less accurate than the measurements from deeper in the material [62]. Therefore, the two curves were combined in order to construct the residual stress field for the FE-model. Fig. 14 illustrates the measured residual stress data and the calculated curve occurring in
the finite element model. The figure also shows the hardness according to curve fitting through the measured points.

![Graph showing residual stress and hardness data](image.png)

**Figure 14.** Calculated (FEA) and measured (XD and DHD) residual stress data with the fitted hardness curve.

The measured residual stress curves in Fig. 14 were taken in the disc’s circumferential direction. The difference of the residual stress curves in the axial direction, which were also measured, was negligible. The shear residual stresses were zero. Due to the absence of any shear residual stresses, the residual stress curve in the FE-model was constructed by combining the principal stress results from a preliminary solving of the model. The maximum principal stress data (positive values) on the tension side were merged through the heat affected region with the minimum principal stress data (negative values) on the compressive side using nodal stress values in Abaqus (UNIQUE NODAL).

The effect of grinding of the discs were taken into account in both the FEA-based residual stress curve and the hardness curve. This can be seen in Fig. 14 as a shift of 0.25 mm in the starting point of these curves, which corresponds to the thickness of the layer that is typically removed from the disc’s surface during grinding. The grinding induced residual stresses were considered to be minimal.
2.5.3 Calculation of damage

Generally, component failure due to fatigue can be divided into three phases; the crack initiation, the crack growth and the final rupture. Typically, cracks initiate due to dislocation movement, which generates slip bands and consequent extrusions and intrusions in the material. These are often considered as crack initiation locations particularly in materials with ductile-like behaviour. In brittle-behaving materials, the slip may arise at discontinuities, such as at the matrix-inclusion interface, however, cracks can also nucleate along slip systems because of shearing.

It is commonly acknowledged that most of the cracks nucleate and grow in certain planes, controlled by the state of the loading and the characteristics of the material [42-44]. The direction of the growth is believed to follow either the maximum shear stress or strain or maximum principal stress planes. However, even if the crack growth is governed by shearing, tensile normal stress in the shear plane acts as reducing crack closure and thus decreases component fatigue life as noted also in Refs. [63, 64].

The fatigue life of a component is often defined by comparing damage due to loading with the material strength. In this thesis, the focus is on investigating fatigue damage formation in the discs through the heat-affected region. Two presumptions were made for the calculation of the damage, the first one being that there is no substantial plastic deformation and the shear stresses are significant in the subsurface region near the contact. Three damage calculation methods combining shearing and normal stress were compared. Findley [45] originally proposed the first of the approaches:

\[ D_F = \left( \frac{\Delta \tau_F}{2} + k_F \sigma_F \right)_{\text{max}} \]  

(7)

where \( \Delta \tau_F \) is the shear stress range, \( \sigma_F \) is the maximum of the normal stress in the shear stress plane at any time during the load cycle and \( k_F \) is a factor accounting for the material’s sensitivity to normal stress. Fatemi and Socie [52] proposed another damage calculation model:

\[ D_{FS} = \frac{\Delta \gamma_{\text{max}}}{2} \left( 1 + k_{FS} \frac{\sigma_{n,\text{max}}}{\sigma_0} \right) \]  

(8)
where $\Delta \gamma_{\text{max}}/2$ is the maximum shear strain amplitude in any plane and $\sigma_{n,\text{max}}$ is the maximum normal stress acting on the maximum shear strain plane over the load cycle. The $\sigma_0$ is the yield strength of the material and $k_F$ is a constant linking the shear strain and the normal stress terms.

Ince et al. suggested a fatigue damage parameter, which is based on generalized strain energy (GSE) [65]. This $W_{\text{GSE}}$ parameter consists of the normal and the shear strain energy terms in a plane where the maximum generalized strain energy occurs:

$$W_{\text{GSE}} = \left( \tau_{\text{max}} \frac{\Delta \gamma^e}{2} + \frac{\Delta \tau \Delta \gamma^p}{2} + \sigma_{n,\text{max}} \frac{\Delta \epsilon_n^e}{2} + \frac{\Delta \sigma_n \Delta \epsilon_n^p}{2} \right)_{\text{max}}$$

(9)

where $\tau_{\text{max}}$ is the maximum shear stress, $\Delta \gamma^e/2$ is the elastic shear strain amplitude, $\Delta \tau/2$ is the shear stress amplitude and $\Delta \gamma^p/2$ is the plastic shear strain amplitude. The $\sigma_{n,\text{max}}$ is the maximum normal stress acting on the maximum shear strain plane over the load cycle, $\Delta \epsilon_n^e/2$ is the elastic normal strain amplitude, $\Delta \sigma_n/2$ is the normal stress amplitude and $\Delta \epsilon_n^p/2$ is the plastic shear strain amplitude.

In the search for the critical plane of maximum damage, the stress tensors needed to be rotated at each examined location. This was done around two axes ($x$ and $y$ or 1 and 2) from 0 to 180 degrees. An angle increment of 5 degrees was applied for both axes as a smaller increment of 2 degrees, requiring more calculation time, was tested, but yielded no significant change in the results. Fig. 15 illustrates the rotation of a stress tensor.

Figure 15. Principle of the rotation of a stress tensor.
2.5.3.1 Evaluation of shear stress/strain amplitude and normal stress

Although there are a number of approaches available to characterize the amplitude of the shear stress [66], probably the most common is the Minimum Circumscribed Circle (MCC) method. This method draws the smallest possible circle around a path, which is described by the shear stress (strain) components $\tau_{12}$ and $\tau_{13}$ ($\gamma_{12}$ and $\gamma_{13}$) during a load cycle in the critical plane in the $\tau_{12}-\tau_{13}$ ($\gamma_{12}-\gamma_{13}$) system. The shear stress (strain) amplitude $\Delta \tau_{F}/2$ ($\Delta \gamma_{\text{max}}/2$) is then the radius of this circle. The advantage of this method is that it takes into account the non-proportionality of the loading.

In this thesis, the normal stress/strain values were taken from the shear plane at any time during the load cycle. The factors $k_F$ and the $k_{FS}$ in Equations 7 and 8, emphasizing the effect of the normal stress, can be defined from test data for the particular material. However, due to the absence of such information a series of calculations was performed varying the magnitude of these factors.
3 RESULTS AND DISCUSSION

The first part of this chapter (publications I and II) deals with the result of the scuffing tests and concentrates on the evolution of the friction coefficient in the contact, the bulk temperature and the comparisons of the occurrences of scuffing initiation under varying load conditions. In the second part, the focus is on the subsurface fatigue experimentation process and analyses of the stress/strain states and fatigue damage beneath the surface (publications III and IV).

3.1 Scuffing failure

The roughness of the interacting surfaces has an effect on the formation of friction, which in turn affects the generation of heat in the contact. Generally, the rougher the surface the higher the possibility that the asperity contacts will increase the coefficient of friction and the consequent creation of heat. An additional parameter affecting the probability of asperity contacts is the thickness of the lubricant film in the contact. This is dependent at least on the viscosity of the lubricant, which is depending on the pressure and temperature of the fluid in the contact. In order to minimize uncertainty in the results these parameters were closely controlled.

The average surface roughness of the discs was evaluated by measuring three locations around the disc. For all the scuffing tests, the disc’s average surface roughness ($R_a$) ranged from 0.27 to 0.34 $\mu$m and the average separation of the five lowest valleys and the five highest peaks ($R_z$) was from 1.49 to 3.68 $\mu$m. In addition, the radii of the curvatures of the fast-rotating discs varied from 95.0 to 102.9 mm. The lubricant temperatures for all the tests were taken at the inlet of the contact, and were all within the range of 60 ± 1.5 °C.

The measured signals were averaged using moving average approach with a changing length that is dependent on the observed load step. This was done in order to remove any additional variation because of geometrical imprecision caused by deformation of the components, manufacturing tolerances or electrical distortion from the
measurement equipment. The fluctuation in the normal force around the average value was from ±1.5...±5 %, which corresponds to ±0.5...±1.7 % in the maximum Hertzian contact pressure at the load level before a scuffing failure. The upper limit of ±1.7 % converts to ±40 MPa indicating that a load step of 150 MPa was sufficient to provide a clear difference between the steps. The oscillation of the torque signal was ±6...±8 %, so the upper limit of ±8 % leads to ±0.002 variation around the mean of the calculated coefficient of friction. This also suggests that the tests are of adequate accuracy. The scuffing results are presented in Table 2.

In Table 2 the test parameters for the groups A, B and C are roughly the same, although there were slight differences. For instance A3 used a step length of 1500 s and the speeds of the shafts were reversed, meaning that the fast-rotating shaft was run at the slow shaft’s speed, and vice versa. Nevertheless, the experimental results were similar to A1 and A2. Furthermore, the starting level of the overload cycles for C1 was one level higher than it was for C2 or C3. Despite of that the scuffing level resulted to be the same as it was for C2.

The torque arising from the shaft seals and bearings was removed from the measured curves by utilizing Eq. 3 for the evaluation of the coefficient of friction in the

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>A1</td>
<td>N/A</td>
<td>10</td>
<td>9.67</td>
<td>178</td>
<td>2.59</td>
<td>1.73</td>
</tr>
<tr>
<td>A2</td>
<td>N/A</td>
<td>10</td>
<td>9.45</td>
<td>177</td>
<td>2.57</td>
<td>1.71</td>
</tr>
<tr>
<td>A3</td>
<td>N/A</td>
<td>10</td>
<td>9.73</td>
<td>175</td>
<td>2.60</td>
<td>1.73</td>
</tr>
<tr>
<td>B1</td>
<td>7</td>
<td>9</td>
<td>7.88</td>
<td>136</td>
<td>2.42</td>
<td>1.61</td>
</tr>
<tr>
<td>B2</td>
<td>7</td>
<td>9</td>
<td>7.89</td>
<td>133</td>
<td>2.42</td>
<td>1.61</td>
</tr>
<tr>
<td>B3</td>
<td>7</td>
<td>9</td>
<td>7.73</td>
<td>127</td>
<td>2.41</td>
<td>1.61</td>
</tr>
<tr>
<td>C1</td>
<td>8</td>
<td>13</td>
<td>15.0</td>
<td>-</td>
<td>3.00</td>
<td>2.00</td>
</tr>
<tr>
<td>C2</td>
<td>7</td>
<td>13</td>
<td>15.5</td>
<td>169</td>
<td>3.04</td>
<td>2.03</td>
</tr>
<tr>
<td>C3</td>
<td>7</td>
<td>11</td>
<td>11.6</td>
<td>159</td>
<td>2.76</td>
<td>1.84</td>
</tr>
<tr>
<td>Da1</td>
<td>7</td>
<td>N/A</td>
<td>15.4</td>
<td>189*</td>
<td>3.03</td>
<td>2.02</td>
</tr>
<tr>
<td>Da2</td>
<td>7</td>
<td>N/A</td>
<td>15.2</td>
<td>199*</td>
<td>3.02</td>
<td>2.01</td>
</tr>
<tr>
<td>Da3</td>
<td>7</td>
<td>7</td>
<td>5.21</td>
<td>140</td>
<td>2.12</td>
<td>1.41</td>
</tr>
<tr>
<td>Db1</td>
<td>7</td>
<td>12</td>
<td>12.2</td>
<td>216*</td>
<td>2.80</td>
<td>1.87</td>
</tr>
</tbody>
</table>

Table 2. Bulk temperature and normal force values with maximum Hertzian pressure and friction coefficients in the contact just before the initiation of scuffing or at the end of the test. The ‘*’ indicates the highest recorded bulk temperature value before disconnection of the thermocouple from the disc.
contact. Typical torque and coefficient of friction curves are shown in Figs. 16 and 17. In these figures, there are visible inaccuracies between 0 and roughly 1000 s originating from the torque generated by the bearings and seals. The reason for this is that the torque and the frictional torque induced from the contact of the discs are at the same level leading to a partly labile behavior at low load levels.
Figure 16. Calculated maximum Hertzian contact pressure, torque and coefficient of friction arising from the disc contact for cases A1 (a), B1 (b) and C2 (c). The initiation of scuffing is visible as a rapid increase in the torque and friction curves at the end of the test.
3.1.1 Findings related to scuffing initiation

In general, Table 2 shows that the initiation of scuffing in a twin-disc test device requires high contact pressure and bulk temperature. The results also show relatively good repeatability for each test type. In the analysis of the results the series A data is used as a reference due to its simple load type of stepwise increasing normal force.

A comparison of the A and B test series reveals that the initiation of scuffing occurred at a one-step lower load level and at about a 30 % lower bulk temperature for the B type tests. However, the coefficient of friction was roughly 30 % greater in series B at the moment of initiation of the scuffing failure. That may have had an increasing effect on the generation of heat in the surface resulting in the initiation of scuffing with the overloading. This occurrence of the failure was just after a normal load increase, during which the asperities may have been undergoing smoothing through plastic deformation because of rise of the contact pressure. The smoothing in turn is most clearly seen as a reduction in the torque curve shortly after an increase.
in the normal load especially between 4000 to 7000 s for the A1 test. The duration of the stabilization period seems to be about 200 to 400 s.

The results for the series A and C tests show that an up to 18 % higher load in terms of maximum Hertzian pressure (about 2.45 GPa versus 2.90 GPa) could be applied without scuffing failure. An interesting difference in the normal force required to initiate scuffing can be seen from the series B and C tests. If C3 is disregarded, nearly 94 % higher force is used in the C than in the B series tests. Compared to series A, this means an increase of roughly 58 %. Hence, the surface condition and temperature due to the load history seems to have a substantial impact on the evolution of a scuffing failure under overloading. Pressure-wise, this can be interpreted to indicate that after a proper running-in and by using cooling cycles between each sequence of overload cycles, overloading of roughly 450 to 750 MPa in the maximum Hertzian pressure can be tolerated. Interestingly, the results showed that overload cycles of 300 MPa during the stabilization period leads to failure under high loading, but increase of about 150 MPa in a stabilized state is not detrimental. In regard to the settling of the surfaces, Castro et. al. [67] have suggested an approach to calculating scuffing safety of gears by dividing the analysis into local and global parts according to the contact pressure and the state of the running-in.

The test discs in series Da1 and Da2, with the axial displacement load, withstood about 60 % greater normal force without scuffing failure than the discs in series A. The maximum Hertzian contact pressure was approximately 3.0 GPa and thus the difference to series A was roughly 17 %. However, the test results are not unambiguous. In Db1 scuffing failure occurred at a much lower level than the other tests there. The gain for that was only 12 % in the normal force, which corresponds to about an 8 % increase in maximum Hertzian contact pressure when compared to the series A tests. Additionally, there was an instant scuffing failure (Da3) immediately after the introduction of the axial displacement load.

The introduction of the axial displacement loading seems to generate a self-healing scuffing initiation. This appears as a sharp peak in the torque curve at 6000 s as shown in Fig. 17 (a). An analogous peak is visible in Fig. 17 (b) for Db1 at about 6500 s, which is the starting point of the axial movement in that test. This is probably caused by an immediate reduction in the lubricant film thickness, while less oil is
dragged into the contact, which in turn is a result of disturbed lubricant flow due to the start of the displacement loading. In addition, the condition of the surface roughness differs in this new sliding/rolling path in comparison to the previous straight contact track because of the effect that running-in has had on this original path. However, the similarities of tests Da1 and Db1 suggest that an increase in the normal load is not a key factor for the initiation of scuffing when the axial displacement movement load is added.

According to the results, after the occurrence of self-healing scuffing, the rises in the torque signal are approximately on an equivalent level for A1 and, for instance, Da1. However, eventually higher normal load values could be applied without scuffing initiation for the test with axial displacement loading. This indicates that the contact condition stabilizes rapidly after the start of the axial movement load and the performance of the EHL between the discs seems to improve due to the displacement cycles.

The small differences between the surface roughness values of the tested discs did not correlate with the initiation of scuffing. However, the results for the C and D series tests emphasise the demanding character of scuffing. Even a small variation probably in the surface condition or the lubrication under specific loading can initiate scuffing.

3.1.2 Evolution of the friction coefficient

According to Figs. 16 and 17, the coefficient of friction increases in line with the load until it reaches the maximum at between 3000 and 4000 s, which corresponds to roughly 1.5 to 1.75 GPa in maximum Hertzian contact pressure. Thereafter it begins to decrease. There are at least two possible scenarios, or a combination of them, which can explain this behaviour. Firstly, before this turning point between 3000 and 4000 s, the lubrication condition is mainly in the mixed lubrication regime, while the coefficient of friction is heavily dependent on the asperity contacts. However, due to several normal load increases, the aggressive asperities are flattened and the condition evolves further towards the elasto-hydrodynamic regime with decreased level of friction. In this working state, the specification of friction is dominated by the shear stresses in the oil, which in turn depends on the pressure
and temperature of the lubricant. Characteristic for this condition is that the friction coefficient tends to slightly drop along with the increasing load and temperature caused by reduction of the lubricant’s limiting shear stress [68]. In the second scenario, the role of the systems of extreme pressure or friction improver additives in the lubricant change their level of activity, which has an influence on the lubrication condition.

A temporary rise in the coefficient of friction after a rise in the normal load indicates an increase in the probability of asperity contacts between surfaces. The friction value stabilizes after a settling in period of roughly 200 to 400 s to a lower level caused by the levelling of the asperities. Additionally, there are visible sharp peaks in the coefficient of friction curves in Fig. 17 (at 6000 s and at 6500 s) at the moment of starting the axial displacement cycles as has also been discussed above. Nevertheless, the curves show that a drop in the coefficient of friction will follow after this self-healing scuffing as shown in Fig. 17 (b). The fundamental reason for this was not studied in this thesis.

3.1.3 Evolution of bulk temperature

Fig. 18 illustrates the climbing of the typical bulk temperature curves of the fast-rotating shaft to a higher level after stepwise increases in the normal load. In roughly 300 seconds after a normal load step-increase, the local maximum of the bulk temperature is reached and the frictional heat generation in relation to cooling stabilizes. In addition, a minor reduction in the temperature occurred after the stabilization process indicating a drop in the friction coefficient, which is also visible as a decrease in the signal for torque. Furthermore, the initiation of scuffing can be noticed as a rapidly rising peak at the end of each test.

The temperature curves of tests A2 and C2 in Fig. 18 (a) characterise the effect of the overload cycles between 7000 and 8000 s. The temperature rise caused by the overload-type loading is about half of the rise that occurs under full-time loading at that level. Furthermore, after the load scheme returns to the base level, the bulk temperature falls to a lower level than it was before the period of overloading occurred. This indicates a reduction in the corresponding friction coefficient, probably due to the continued flattening of the surface asperities. The trend of ever
falling temperature as the test progresses and high normal load values for the series C tests suggests that by controlling the heat generation and running-in of the surfaces, high overload cycles can be applied.

![Bulk Temperatures at Fast Shaft](image_url)

**Figure 18.** Typical bulk temperature curves where the scuffing initiation is visible as a peak at the end of the test period.

At the moment of applying the axial displacement loading (6000 s), Fig. 18 (b) illustrates a clear separation between the curves of temperature for tests A2 and Da1. Similarly, a drop in the temperature is visible in Db1 at 6500 s, which is the moment of introduction of the axial displacement load in that case. This behaviour may be caused by the evolution of the friction coefficient and consequent reduction in the generation of heat, as discussed earlier. Alternatively, the saw-tooth like path may have spread the heat energy to a broader area on the surface, which leads to the transfer of less heat through the measurement location. However, it seems that the drop in the temperature only occurs at the instant when the axial displacement movement is introduced, since there was no clear decrease in the temperatures for the subsequent load levels.

The higher general level of the bulk temperature for Db1 in comparison to the other tests in Fig. 18 (b) is probably caused by the difference in the location of the measurement point so that it is marginally closer to the location of heat generation and thus shows higher temperature values. Another explanation may be differences in the surface topographies of the discs leading to minor variations in the coefficient of friction and then in the bulk temperature. As the results also prove, the bulk
temperature cannot be utilized as an accurate parameter for predicting the risk of scuffing; however, it indicates stabilization between heat generation and cooling after a change in the load condition. It is also worth mentioning that a thermocouple disconnection caused the falling temperature signal at about 12000 s for Da1 in Fig. 18 (b) from the disc.

3.1.4 Analysis of the contact surfaces and material hardness

An optical profilometer was used to analyse the contact surfaces of the discs. Figs. 19 and 20 illustrate typical images of the discs after testing with and without experiencing a scuffing failure. Fig. 21 shows surface profiles measured over the discs along the measurement lines, which are visible below in Figs. 19 and 20.

Figure 19. Typical contact surfaces after the occurrence of scuffing. The curved surface of the fast-rotating disc is on the left and the right is the slower-rotating disc with flat grinding. In the middle are horizontal lines, which illustrate the paths for measuring the surface profiles.

Tearing of the material after scuffing caused severe damage to the contact surfaces. There are grooves visible in the circumferential direction on both discs and the dark surface colour indicates high temperature during operation. In Fig. 19 clearly noticeable pits can be observed at the edges of the contact areas, where larger particles of material have been torn off.
Figure 20. Typical contact surfaces after testing without the occurrence of scuffing. The curved surface of the fast-rotating disc is shown on the left and the flat surface curvature of the slower-rotating disc is on the right. In the middle are horizontal lines, which illustrate the paths for measuring the surface profiles.

The non-scuffed contact surfaces clearly differ from the scuffed ones. On the slower-rotating disc in Fig. 20 the darker area of the surface is significantly wider indicating that the spread of the heat energy has been over a broader area on the surface. Signs of plastic deformation are visible in the central area of the slow-rotating disc. These deformations produce inclined ridges on the contact surface. The magnitude of the inclination is about 11 degrees and these ridges were probably caused by the axial displacement loading even though the inclination slightly differs from the path in Fig. 10.

Since no large-scale plastic deformation or scuffing failure has occurred, grinding marks are still visible on the fast-rotating disc in Fig. 20. However, minor plastic deformation can be seen on the surface starting from a grinding groove and continuing in the direction of sliding. These may be due to a discontinuity in the topography of the surface, which has deformed significantly under heavy loading.

Figure 21. Surface profiles of the tested discs.
The profile curves in Fig. 21 express that a notable quantity of material has detached from the surfaces of the scuffed discs. On the other hand, there is also significant plastic deformation on both of the non-scuffed discs. The magnitude of the deformation is the highest in the centre of the contact, where the contact pressure has been the greatest. The shape of the deformation probably has an effect on the lubrication condition between the discs, since in the tests with the axial displacement loading, the locations with the large deformation on both of the discs are not directly against each other all the time. Therefore, the lubricant film thickness inside the contact probably differs significantly from the condition of keeping the discs directly against one another.

Fig. 22 illustrates the material hardness (HV0.2) for the test cases A2 and Da1 on a cross section in the middle of the contact area. The results for the untested disc were taken from an unground but case-hardened test disc. The starting location for that disc was moved to a depth of 0.25 mm in order to be comparable with the ground and tested discs.

![Graph](image)

**Figure 22.** Hardness profiles of untested and tested discs.

In general, the hardness of the used discs is about 15 to 20% lower than the untested one. The reduction in the hardness is greatest near the surface, where the generation of heat has also been the highest during testing. The softening is probably caused by tempering of the material due to the high prevailing temperature in the contact and is also applicable to Da1 although scuffing was not initiated in that case. The decrease of the material hardness can be regarded as a reduction in the load carrying capacity of the component.
3.2 Subsurface fatigue failure

The surface roughness in the fatigue tests did not play such an important role, because the focus was on crack initiation in the subsurface region. In addition, sliding between the discs was set at a clearly lower level thus inducing insignificant stresses at the surface. The measured surface roughness $R_a$ varied between 0.27 and 0.34 μm and $R_z$ from 1.16 to 2.29 μm between the discs. The radius of curvature of the fast discs ranged from 101.0 to 102.9 mm and the lubricant inlet temperature remained within $60\pm1.5$ °C during the tests. The test results are presented in Table 3. Fig. 23 illustrates typical subsurface cracks along with fatigue data as a function of maximum Hertzian contact pressure.

Table 3. Results of the subsurface fatigue tests at different load levels with the data about which of the discs the cracks were found in. Additionally, the parameters of the elliptical contact of two elastic bodies are presented. These values were evaluated in accordance with Ref. [2]. The effect of residual stresses and stresses due to shrink-fit of the discs as well as sliding were not considered.

<table>
<thead>
<tr>
<th>Max Hertzian pressure [GPa]</th>
<th>Contact ellipse axes</th>
<th>Max shear stress and depth</th>
<th>Load cycles [millions]</th>
<th>Cracks found</th>
<th>Crack location depth</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a$ [mm]</td>
<td>$b$ [mm]</td>
<td>$\tau_{\text{max}}$ [GPa]</td>
<td>$z_{\text{ss}}$ [mm]</td>
<td>FD SD</td>
<td>FD SD [mm]</td>
</tr>
<tr>
<td>A 1.8</td>
<td>1.60</td>
<td>0.51</td>
<td>0.58</td>
<td>0.36</td>
<td>7.0 5.0 No No -</td>
</tr>
<tr>
<td>B 1.8</td>
<td>1.60</td>
<td>0.51</td>
<td>0.58</td>
<td>0.36</td>
<td>7.0 5.0 No No -</td>
</tr>
<tr>
<td>C 2.0</td>
<td>1.80</td>
<td>0.58</td>
<td>0.65</td>
<td>0.41</td>
<td>5.1 3.6 No No -</td>
</tr>
<tr>
<td>D 2.2</td>
<td>2.00</td>
<td>0.64</td>
<td>0.73</td>
<td>0.45</td>
<td>7.0 5.0 No No -</td>
</tr>
<tr>
<td>E 2.2</td>
<td>2.00</td>
<td>0.64</td>
<td>0.73</td>
<td>0.45</td>
<td>72.5 51.5 Yes No 1.2, 1.3</td>
</tr>
<tr>
<td>F 2.2</td>
<td>2.00</td>
<td>0.64</td>
<td>0.73</td>
<td>0.45</td>
<td>23.0 16.3 No Yes 1.0</td>
</tr>
<tr>
<td>G 2.5</td>
<td>2.30</td>
<td>0.74</td>
<td>0.84</td>
<td>0.52</td>
<td>21.1 15.0 Yes No 0.8, 1.4</td>
</tr>
</tbody>
</table>

A large number of load cycles and relatively high contact pressure (16 million cycles and above 2 GPa) were needed to produce subsurface cracks according to Table 3. However, even after 72.5 million load cycles no propagation of the cracks to the surface was observed. Along with the images in Fig. 23 this suggests that the crack growth happened in the region beneath the hardened layer where the material strength is reduced in comparison to the surface. In addition, the cracks were significantly deeper in the material than the location of the maximum shear stress.
according to the analytical evaluation ($z_{ss}$ in Table 3). However, the strategy of cutting and grinding the discs may have removed smaller cracks from the examined surfaces and also the preparation process may not have revealed cracks that are below the processed surface.

![Figure 23. Fatigue results and subsurface cracks for tests E (at the depth of 1.2 mm), F (at the depth of 1.0 mm) and G (at the depth of 0.8 mm). The contact surface of each disc is horizontal and located above the crack.](image)

### 3.2.1 Finite element analysis results

The solving of all the finite element models included four static calculation steps. First, the shrink-fit was solved and secondly the residual stresses near the surface of the discs were evaluated. Then the introduction of the normal load was calculated and the effect of sliding was applied to the discs as forced rotations of the shafts. The calculations were performed in increments, which lead to a quasi-static simulation of sliding and rolling in the last calculation step.
Four different normal load stages were examined. These loads resulted in maximum Hertzian contact pressures of 1.8, 2.0, 2.2 and 2.5 GPa between the discs. Additionally, at the load level of 2.2 GPa, a series of models were solved. These calculations aimed to shed light on the effects of tolerances in the manufacturing on the formation of damage in the subsurface region by varying the case hardening depth (CHD) and the radius of curvature in the axial direction of the fast-rotating disc. In addition, the role of the residual stresses was examined. Table 4 presents the calculated variants.

Table 4. Calculated cases and parameter variation. "FD" stands for fast-rotating disc and "SD" to the slow-rotating disc.

<table>
<thead>
<tr>
<th>#</th>
<th>Normal force [kN]</th>
<th>Max Hertzian pressure [GPa]</th>
<th>Radius of curvature (FD) [mm]</th>
<th>Radius of curvature (SD) [-]</th>
<th>CHD (FD) [mm]</th>
<th>CHD (SD) [mm]</th>
<th>Residual stresses</th>
</tr>
</thead>
<tbody>
<tr>
<td>SFA1</td>
<td>3.0</td>
<td>1.8</td>
<td>100</td>
<td>Flat</td>
<td>0.85</td>
<td>0.85</td>
<td>Yes</td>
</tr>
<tr>
<td>SFA2</td>
<td>4.3</td>
<td>2.0</td>
<td>100</td>
<td>Flat</td>
<td>0.85</td>
<td>0.85</td>
<td>Yes</td>
</tr>
<tr>
<td>SFA3</td>
<td>5.9</td>
<td>2.2</td>
<td>100</td>
<td>Flat</td>
<td>0.85</td>
<td>0.85</td>
<td>Yes</td>
</tr>
<tr>
<td>SFA4</td>
<td>9.0</td>
<td>2.5</td>
<td>100</td>
<td>Flat</td>
<td>0.85</td>
<td>0.85</td>
<td>Yes</td>
</tr>
<tr>
<td>SFB1</td>
<td>5.9</td>
<td>2.2</td>
<td>100</td>
<td>Flat</td>
<td>0.85</td>
<td>0.85</td>
<td>No</td>
</tr>
<tr>
<td>SFC1</td>
<td>5.9</td>
<td>2.2</td>
<td>90</td>
<td>Flat</td>
<td>0.85</td>
<td>0.85</td>
<td>Yes</td>
</tr>
<tr>
<td>SFC2</td>
<td>5.9</td>
<td>2.2</td>
<td>110</td>
<td>Flat</td>
<td>0.85</td>
<td>0.85</td>
<td>Yes</td>
</tr>
<tr>
<td>SFD1</td>
<td>5.9</td>
<td>2.2</td>
<td>100</td>
<td>Flat</td>
<td>0.95</td>
<td>0.75</td>
<td>Yes</td>
</tr>
</tbody>
</table>

For the alteration of the case hardening depth in case SFD1, the applied residual stress field and the elastoplastic behavior of the material needed to be modified. The hardness and the resulting yield and tensile strength curves along with the residual stress curves for case hardening depths of 0.75 and 0.95 are show in Fig. 24. The figure also includes a comparison of the original curve with the CHD of 0.85 mm.
Figure 24. Hardness (a), yield and tensile strengths (b) and residual stress (c) curves for the calculated cases.
3.2.1.1 Stress and strain results

The stresses and strains were studied in a cross section of the model that coincided with the maximum pressure value in the middle of the elliptic contact trace on the disc surface. Typical contours are shown in Fig. 25. The upper plots illustrate the stress and logarithmic strain near the contact in the x-direction. The lower contours show the stress and logarithmic strain concentrations due to shearing in the xy-plane. The logarithmic strain data was used instead of engineering strain because such a parameter gives a more realistic value of the final strain when deformation takes place in a series of increments [69].

The stress and strain data from the cross-sections are shown in Fig. 26. The stress results were converted from the integration points of the elements to nodes in Abaqus (UNIQUE NODAL). The upper graphs in Fig. 26 show the stress and strain history during a load cycle at the depth of 0.5 mm with a radial load resulting in 2.2 GPa maximum Hertzian pressure in the contact. The greatest amplitude values were found for the components S11 and S12 as well as strain components LE11 and LE12. The middle graphs present the maximum and minimum values for these components as a function of load and depth captured during one load cycle. These curves specify that the maximum range for the shear components is between the depths of about 0.3 mm and 0.6 mm. For the S11 stress component the maximum amplitude occurs at the surface. In contrast, the maximum amplitude of the strain component LE11 can be seen at a depth of 0.3 mm. This is probably caused by the combination of working and residual stresses, whose effect is visible in this coordinate direction (x). This is also reflected as a significant change in the curves for LE11 between 1.0 mm and 1.5 mm, which can also be seen in Fig. 14 as an amendment in the residual stress curve. The corresponding curves for the slow-rotating disc (SD) did not differ significantly from the curves of the fast-rotating disc (FD) as is illustrated in the contour plots in Fig. 25 showing the similarity of the stress fields on both of the discs.
Figure 25. Stress components $S_{11}$ (a) and $S_{12}$ (b) and logarithmic strains $LE_{11}$ (c) and $LE_{12}$ (d) at the cross section of the contact on the normal load level generating the maximum of 2.2 GPa in pressure at the contact (SFA3).
Figure 26. The two upper graphs (a, b) show the stress and strain histories at the depth of 0.5 mm during one load cycle with a load generating 2.2 GPa maximum Hertzian pressure at the contact. The graphs in the middle (c, d) present the maximum and the minimum values for components S11 and LE11 during one load cycle, while the two bottom graphs (e, f) illustrate these values for components S12 and LE12. “FD” stands for fast-rotating disc.
3.2.1.2 Plastic strain results

Plastic deformation occurred only for the loading scheme that generated 2.5 GPa contact pressure. Fig. 27 illustrates the maximum of the equivalent plastic strain (PEEQ) as a function of its depth in the material. The results showed a concentration at a depth of 0.4 mm to 0.8 mm, which coincides with the occurrence of the maximum range of the stress and strain components S12 and LE12.

![Image of graph illustrating plastic strain results](image.png)

**Figure 27.** The maximum equivalent plastic strain (PEEQ) with the loading that produces 2.5 GPa maximum Hertzian contact pressure on the disc surface and the fitted hardness values through the case-hardened region. “FD” stands for fast-rotating disc and “SD” for the slow-rotating disc.

Another region of large plastic deformation appears between the depths of 1.1 mm to 1.7 mm. At these depths, the strength of the material is substantially lower than at the surface although the stresses and strains are also clearly lower. Additionally, the residual stresses in this area are mostly on the tension side, which can be detrimental for fatigue life. The test outcome in Table 2 also suggested that this area is critical from the crack initiation point of view even at minor load levels than the loads which generate the contact pressure of 2.5 GPa. In addition, since the PEEQ curves in Fig. 27 do not differ significantly from each other the difference between the curvatures of the contacting surfaces has a negligible effect on the plastic deformation in the subsurface zone.
3.2.1.3 Damage

The shear stress or strain amplitudes were evaluated with the MCC approach for the calculation of damage over a load cycle. The comparison of the methods presented in section 2.5.3, was carried out utilizing the FE-results of the case SFA3. Due to the lack of material data for the calculation of the factors $k_F$ and $k_{FS}$, a series of analyses was conducted by varying these parameters. A range, covering typical values for steels [70-72], was tested: 0.1, 0.3 and 0.8 for the $k_F$ and 0.5, 1.0 and 2.0 for the $k_{FS}$. Fig. 28 illustrates the resulting damage curves with all the methods.

![Damage curves](image)

**Figure 28.** The damage as a function of depth with all the utilized methods and by varying the factors $k_F$ and $k_{FS}$.

The curves in Fig. 28 show that the maximum damage with all the methods occurred at the same depth where the maximum of shear stress or strain ranges appear in Fig. 26. The factors $k_F$ and $k_{FS}$, which emphasise the effect of normal stress, did not have a substantial effect on the results. This is because the tension normal stress values, which are believed to accelerate crack growth, were low in the observed region. However, a clearly visible deviation can be seen between the depths from 1.3 mm to 1.8 mm, especially between the $D_{FS}$ curves. In this region, there are also tension residual stresses present as illustrated in Fig. 14.

3.2.1.4 The effect of residual stresses

A challenge in analysing the fatigue life of case-hardened components is the consideration of the residual stresses in the material. In addition, changes in these stresses may occur because of the way the component is used. Hence, two cases were compared; in the first one (SFA3), the residual stresses are comprised in the model
and in the second (SFB1), these stresses are omitted. The resulting maximum and minimum stress and strain values during the load cycle and the consequent damage are presented in Fig. 29.

**Figure 29.** Graphs in the upper row illustrate the maximum and minimum values of the stress and strains components S12, LE12 and S11 over the load cycle. The lower row presents the damage curves according to equations presented in chapter 2.5.3, while the parameters \( k_F \) and \( k_F S \) were set at 0.3 and 1.0.

All the damage evaluation methods were able to distinguish a difference in the damage values regardless of whether or not the residual stresses were included in the analysis. In general, the differences were small and with the shear strain based approaches \( (D_{FS}, W_{GSE}) \) this is probably caused by a minor deviation between the strain curves of LE12 as shown in Fig. 29 (b). However, Fig. 29 (e) illustrates a step-up in the damage (SFA3) starting at a depth of about 1.3 mm. This probably originates from the tension residual stresses at this depth, whose effect is also visible as a minor deviation between the curves in Fig. 29 (c) starting from at roughly the same depth.

The above indicates that the residual stresses in the subsurface region increase the damage. In addition, the damage calculation method according to Fatemi and Socie...
appears to be lightly more sensitive to changes in the stress and strain data than the others, and hence it was utilized in further analyses with the \( k_{FS} \) set at 1.0.

3.2.1.5 The effect of the surface curvature

The curvature of the disc has an effect on the maximum Hertzian contact pressure at the surface, which is also reflected in the stresses and strains deeper in the material. The pressure can be calculated with Eq. 5. The resulting values for cases SFA3, SFC1 and SFC2 are 2.53 GPa, 2.60 GPa and 2.47 GPa respectively, and the FEA based values didn’t differ significantly either. A decrease of roughly 11 % in the radius down to 90 mm increased the maximum Hertzian pressure by 2.8 %. Correspondingly, an increase of 10 % up to 110 mm resulted in a decrease of 5 % in the pressure. The effect of the variation of the curvature in the strain, stress and damage values is illustrated in Fig. 30 for the fast-rotating disc.

Figure 30. The strain and stress ranges for components LE12 and S11 during the load cycle. On the right is the corresponding damage curves \( D_{FS} \), while the factor \( k_{FS} \) was set at 1.0.

The differences between the curves in Fig. 30 can be seen mainly at those locations where the stress or strain values are high. Accordingly, these occur relatively near to the surface. However, deeper in the material the variation between the curves becomes insignificant. This suggests that a fairly large variation in the curvature of the interacting surfaces can be allowed without having any considerable effect on the strains or stresses starting below a depth of about 1.0 mm.
3.2.1.6 The effect of the case-hardening depth

After case-hardening the contact surfaces need to be ground to achieve sufficient surface curvature and roughness. The thickness of the removed material may vary within manufacturing tolerances leading to variations in the case-hardening depths of the components. This, in turn, can have an influence on the fatigue strength in the subsurface region. The effect of small changes in the grinding depth on the strain and stress components and the resulting damage is shown in Fig. 31.

![Graphs showing strain, stress, and damage](image)

**Figure 31.** The graphs on the left and in the middle show the range of the strain and stress components $LE_{12}$ and $S_{11}$ during the load cycle. The graph on the right shows the damage $D_{FS}$ when factor $k_{FS}$ was set at 1.0. “FD” means the fast-rotating disc and “SD” the slow-rotating one.

The results in Fig. 31 indicate that small differences in the CHD do not have a clear effect on the strain, stress or damage states in the subsurface region. Any minor variation between the curves probably originated from small deviances in the residual stresses between the cases. The effect of changes in the case-hardening depths may be more visible in the material strength data.
4 CONCLUSIONS

This thesis first concentrated on responding to the first research question, namely widening the knowledge of scuffing initiation under occasional overloading. The focus in the response to the second research question was on establishing a procedure to study subsurface fatigue in a contact under rolling/sliding loading and high contact pressure. Experiments were conducted using a twin-disc test device, where the elliptical pressure distribution between the discs is considered to be comparable to the pressure trace between the gear teeth in a bevel gear set. The work started by upgrading an existing twin-disc test device so that it was possible to apply dynamically changing loading to the contact in the radial and in the axial directions in relation to the discs’ orientation. This was followed with several tests looking for scuffing or subsurface-initiated fatigue failure in the discs and a subsequent post-processing approach aimed at revealing possible cracks beneath the surface of the discs. The following part of the thesis concentrated on analyzing the results. The evolution of the coefficient of friction between the discs and the occurring bulk temperature in one of the discs were studied and any scuffing events were compared. Furthermore, the twin-disc test arrangement was modelled with the finite element method in order to be able to examine stresses and strains in the subsurface region. The calculation of the fatigue life of case-hardened components was discussed and the effect of manufacturing tolerances and residual stresses on the formation of fatigue damage was evaluated using the test system as an example.

According to the results of the scuffing tests, the running-in of the interacting surfaces is clearly important. Overload cycles on top of the nominal normal load during the running-in period substantially increased the scuffing risk. However, if the overload cycles were introduced in a controlled manner so that they gradually increased in magnitude in each series of overloads, and there were periods of nominal load cycles between the series of overload cycles, the durability against scuffing was enhanced. The reason for this is probably due to the decrease of the coefficient of friction between the discs caused by flattening of asperities during the previous overloading period. The consequent decrease in the production of frictional heat, along with the systematic cooling periods after the overload phases, made it
possible to reach higher normal load levels without scuffing than by directly stepwise increasing the normal load.

The scuffing tests with the axial displacement loading showed that a stabilized condition of small deviations in the contact path enabled the use of higher normal load without the occurrence of a scuffing failure in comparison to a load condition where the same rolling/sliding path is maintained throughout the test. The start of the reciprocating axial displacement movement seemed to briefly increase the risk of scuffing by instantly reducing the thickness of the lubricant film. This was also seen as an increase in the coefficient of friction although after stabilization of the lubrication condition between the discs, the coefficient of friction decreased. However, the high temperature and extensive surface pressure even without the scuffing initiation resulted in plastic deformation of the contact surfaces and changes in the material hardness in this region.

The experiments on scuffing revealed its complexity as a gear failure mode. Although some means to increase the load without the occurrence of scuffing were shown (at least up to 3 GPa in maximum Hertzian contact pressure at slide-to-roll ratio of 1.68), the results were not definitive. In some experiments, scuffing occurred unexpectedly in comparison to earlier tests. This leaves room for speculation about the effects of other parameters such as the role of lubricant additives and the details of surface roughness profiles indicating the need for a more thorough investigation of the lubricant and the surfaces of the discs. In addition, a higher number of tests with varying load parameters could increase the statistical significance of the results.

The fatigue tests proved that a twin-disc apparatus is suitable for studying subsurface fatigue in case-hardened components. Crack initiation occurred below the hardened layer, but no propagation to the surface was observed. According to the tests data, a relatively high contact load (2.2 GPa in maximum Hertzian contact pressure) and a significant number of load cycles (16 million cycles) was required to generate cracks. However, there may be an uncertainty with the detection process: The destructive method of crack inspection, which was used, may have removed some of the smaller cracks in particular from the specimen. Additionally, an elastoplastic FE-model of the test discs for evaluating the case-hardening effects of the residual stresses and increased material hardness, performed well at finding critical depths beneath the
hardened layer. The experimentally found depths coincided with these FE-based ones.

The fatigue life of a component is often calculated by comparing damage due to loading to the fatigue strength of the component. The formation of damage was studied utilizing stress/strain–based multiaxial damage criteria, which combined shearing and normal stress. All the examined criteria showed that the maximum damage was at the depth where the shear stress or strain amplitudes were at their highest. Nevertheless, the shear strain-based method (Fatemi-Socie) seemed to be more sensitive to the occurrence of the residual stresses even though the tension normal stress in the subsurface area was relatively low. The analysis also showed that small manufacturing-induced deviations in the curvature of the contacting surfaces or in the case-hardening depths only have a small or insignificant effect on the fatigue damage in the subsurface region.

Implementing these results on real case-hardened components requires further work. For instance, the loads used in the scuffing tests were high in comparison to the loading usually experienced by real gear sets. Correspondingly the effects of the differences in the size, the material, the surface topography, the lubrication condition and the engagement of the contact need to be investigated further as do the fatigue strength of the material and the subsequent calculation of fatigue life of the components.
REFERENCES


[31] Li S, Anisetti A. On the flash temperature of gear contacts under the tribo-dynamic condition. Tribol Int 2016;97;6-13.


[61] Pavlina EJ, Van Tyne CJ. Correlation of yield strength and tensile strength with hardness for steels. J of materials engineering and performance 2008;17(6);888-93.


[71] Everitt CM, Alfredsson B. Contact fatigue initiation and tensile surface stresses at a point asperity which passes an elastohydrodynamic contact. Trib Int 2018;123;234-55.

PUBLICATIONS
An experimental approach for investigating scuffing initiation due to overload cycles with a twin-disc test device

M Savolainen, A Lehtovaara

Tribology International, 109 (2017), 311-318
doi:10.1016/j.triboint.2017.01.005

Publication reprinted with the permission of the copyright holders.
An experimental approach for investigating scuffing initiation due to overload cycles with a twin-disc test device

Matti Savolainen, Arto Lehtovaara

Group of Tribology and Machine Elements Department of Materials Science, Tampere University of Technology, PO BOX 589, 33101 Tampere, Finland

ARTICLE INFO

Keywords:
Scuffing
Twin-disc
Overload
Friction

ABSTRACT

This paper presents an investigation into the effect of unexpected overloading on scuffing initiation with a twin-disc test device. An existing twin-disc test device was modified to be suitable for applying overloading dynamically to the disc contact. Three test series including three pairs of discs in each series were subjected to combined normal force and rolling/sliding loading. First a base level for scuffing initiation was defined by increasing the normal load stepwise until failure, while keeping the rolling and sliding between the discs constant. In the subsequent test series, the overload cycles were applied at a significantly lower level in two different patterns leading to scuffing at both an earlier and later stage than was observed for the base level.

1. Introduction

Bevel gears in ship shafting lines often experience occasional overload cycles caused by propeller impacts on hard obstacles e.g. ice floes. The gear teeth have long been dimensioned against dynamic loads using load influence factors. However, occasional overload cycles are usually not noticed as a separate factor and therefore their role is neither visible nor accounted for in gear dimensioning.

The exact initiating mechanism of scuffing failure is not fully understood, although there appears to be a close relationship between a scuffing failure and lubricant film breakdown as noted in e.g. Ref. [1]. High surface pressure and sliding velocity with consequent frictional heating may affect the thickness of the lubrication film, whose decreasing viscosity leads to damaging asperity contact.

In addition, shear stresses play a key role in determining the film’s thickness in an EHL contact between the rolling surfaces and the lubricant. However, a lubricant can only withstand a limited shear stress [2,3]. If this limit is exceeded and if the maximum allowable shear stress in the oil is more dependent on the oil’s temperature than viscosity, a thinner oil film may result because less oil is dragged into the contact area [4].

Another factor to consider is that a transient load rise instantaneously reduces the lubricant film thickness at the same time increasing the friction coefficient [5], which increases the potential for asperity contacts. It has also been reported that a temperature rise in the lubricant film profile occurs under heavy loading, due to the compression and the viscous friction [6,7] increasing the risk for a scuffing failure.

The risk of scuffing is also known to be high during the initial settling of contacting surfaces often termed ‘running-in’, which is defined as the series of processes of wear rate and friction stabilisation for lubricated contacts [8]. In fact, the nature of scuffing is different than other gear failure modes in that it may be initiated after only a short period of severe load cycles in harsh conditions. Hence, rarely occurring occasional overload cycles may shorten the gear’s life time significantly through a scuffing failure.

The flash temperature method is a widely-used approach to calculate the risk of scuffing. It was originally proposed by Blok [9] in 1937 and further enhanced by Jaeger [10] and Archard [11]. The basis of the theory is in the calculation of the maximum contact temperature $T_c$, which is an indication of the risk for a scuffing failure. This temperature is considered to be the sum of the bulk temperature $T_b$ and the flash temperature $T_{r_{\max}}$ generated in the contact as follows:

$$T_c = T_b + T_{r_{\max}}$$

One of the challenges with the above theory is the evaluation of the bulk temperature. In many cases this can be found by measurement, but an analytical approach based on loading is presented in Ref. [12]. Also recently Xue et al. proposed a method for predicting scuffing failure in spur gear pairs by means of a transient thermal solver [7]. This includes a temperature rise formula based on transient heat flux and a transient thermal elastohydrodynamic lubrication (TEHL) model, which takes into account the dynamic loads.

The experimental approach for finding the scuffing load-carrying capacity of a lubricant-material pair is traditionally defined by conducting FZG test procedures described in the standards [13]. Additionally, several scuffing studies have been performed with both

http://dx.doi.org/10.1016/j.triboint.2017.01.005
Received 11 November 2016; Received in revised form 4 January 2017; Accepted 6 January 2017
Available online 06 January 2017
0301-679X/ © 2017 Elsevier Ltd. All rights reserved.
the FZG device [1] and with bevel-gear-based test devices [14,15]. The focus has been on varying the used lubricant, the influence of the oil temperature and the applied loading, amount of the oil supplied and the effect of a coating on the contact surface. Despite the stability of the results delivered, the possibilities for investigating a wider range of parameters is somewhat limited; often requiring a new design for the gear wheels. In addition, several scuffing studies have been performed by varying the above mentioned parameters, but observing the behaviour of a piston ring and liner contact [16–18]. However, an obvious difference exists between the conditions in comparison to a gear teeth contact.

A new approach to the scuffing test has recently been presented [19]. This utilises a ball-on-disc test device with contra-rotating surfaces to limit the EHD films. This arrangement enables the determination of the boundary lubrication conditions over a wide range of sliding speeds.

With the twin-disc test device, the elliptical pressure distribution, which is seen between the gear teeth in e.g. spiral bevel gears, can be reproduced [20,21]. This is done through the precise definition of the discs’ geometries. Furthermore, the structure of the device enables the actual loading of the contact to be highly adjustable. Over the years, the twin-disc test device has been used to study the behaviour of friction, pitting formation, wear prediction and the influence of surface roughness [22–25]. The effect of different surface conditions on scuffing initiation has been also investigated [26]. However, there has not been much research focusing on the effect of variable loading on scuffing failures, which is the main topic of this paper. The paper presents the first result of an investigation of the relationship between overload cycles and the initiation of scuffing using a twin-disc test device.

2. Test equipment

The tests are conducted with an existing in-house-built twin-disc test device as shown in Fig. 1. The original version of the device is described in detail in Ref. [27]. However, the device was updated in order to be able to apply dynamic loading on the contact. The hydraulic cylinder and its control valve, which place a variable normal force \( F_N \) on the discs, were renewed. In addition, the valve control system hardware has been updated and reprogrammed. The load frame with the test discs are illustrated in Fig. 2. The applied normal load is measured with a force transducer situated behind the normal force cylinder and its signal is used as an input for the valve controller.

The adjustable rotation speed, direction and slide-to-roll ratio are controlled with frequency converters, which drive the electric motors. These are joined to the shafts via couplings allowing minor misalignment between the axles.

A separate hydraulic unit provides lubricant for the disc contact injected from the inlet side. The same unit also lubricates the support rolling bearings inside the test device. Stepless adjustment of the lubricant flow is provided by means of a frequency-converter-driven pump motor and manual valves. The lubricant inlet temperature is automatically controlled by a separate microcontroller unit.

The test device is highly adjustable and automated enabling the possibility for unmanned operation. Some of the performance values are listed below:

- Rotation speed: Max 6000 RPM
- Normal load: Max 50 kN
- Lubricant temperature: 25...120 °C
- Lubricant flow rate: 0.5...20 l/min

2.1 Measurements

The measurement system collects data on the above-mentioned normal force, lubricant flow and inlet temperature, as well as the shaft rotation speeds and the torque on the slow shaft. In addition, the test disc bulk temperature on the fast shaft was recorded using a telemetry system and a thermocouple located approximately 3.5 mm underneath the surface at the disc contact.

The torque signal \( T_m \), measured by the torque transducer mounted on the slow shaft, can be separated into the torque \( T_T \) originating from the bearings and the shaft seals, and the torque \( T_R \), which originates from the disc contact. These two values have to be separated in order to analyse the contact behaviour more precisely. The actual torque at the disc contact can be calculated with the following equation:

\[
T_c = T_m \pm T_R
\]  

(2)

The \( T_R \) value is negative, when measuring the braking torque on the slower shaft and positive in the opposite situation.

Typical values for the torque \( T_T \) were found by measuring the shaft torque as a function of the normal force. This was done by rotating the shafts with the same speeds 175.8 RPM, resulting in a pure rolling condition. The operating temperature was 60 °C and three different disc pairs were used. A load-dependent equation was fitted to the measured values shown in Fig. 3:

\[
T_T = 0.0801 \cdot \ln(F_N) - 0.0265
\]  

(3)
The results, as illustrated in Fig. 3, show clear variation between an interesting normal force range of 5–12 kN. Hence, a sensitivity check was performed. This was done in two stages utilising the following equation:

\[ \mu_t = \frac{T_m + T_r}{r_{disc}} \]

Firstly, the torque \( T_m \) was calculated with a friction coefficient \( \mu_c \) of 0.03 in disc contact, a disc radius \( r_{disc} \) of 0.035 m and by estimating the \( T_r \) with Eq. (3). Then the variation in the friction value was evaluated by setting the torque \( T_r \) first to 0.4 Nm and then to 1.0 Nm. The used friction value of 0.03 represents a typical value according to the experiments and the used \( T_r \) values were found in Fig. 3. The results in Fig. 4 show a corresponding error range in the friction coefficient. The height of the friction error band is found to be insignificant, being roughly from 0.003 to 0.001 depending on the normal force.

The values found with Eq. (3) were used for all the test runs since the bearings, seals and their lubrication do not change.

3. Test specimen and lubricant properties

The discs are 70 mm in diameter and the width of the contacting surfaces is 10 mm. In an axial direction the radius is set to 100 mm on the fast rotating disc while the other is flat. This allows an elliptical contact trace to be formed. The discs are turned from a round bar of 18CrNiMo7-6 (EN 10084) and case hardened to a surface hardness of 59–61 HRC.

A special grinding device was designed for manufacturing the desired radius and a surface topography of a real gear-flank’s surface. The basic principle of this device is to rotate the disc to be ground against a rotating grinding stone at a specified radius in order to produce perpendicular grinding marks on the specimen’s surface, as shown in Fig. 5 and explained in detail in Ref. [27]. The discs were assembled on shafts before the final grinding in order to minimise any possible eccentricities of components arising from the manufacturing tolerances. The average surface finish (Ra) of both discs has to be between 0.28 and 0.32 μm and the average distance of the highest peak from the lowest valley (Rz) should be below 3 μm.

The lubricant is a mineral-based commercial gear oil with an EP-additive system for heavy duty industrial gears, which is commonly used in marine thruster systems. The oil’s properties are shown in Table 1.

4. Experimentation procedure and test cases

The experiments commenced with a pre-run, during which the shafts were rotated at the same low speed for about an hour with lubricant circulation and without normal load. During this time the lubricant and the temperature of both shafts were stabilized to a constant level.

For the actual test runs the shaft rotation speeds were adjusted in order to achieve a high surface sliding speed, which is known to increase the risk of a scuffing damage. The fast shaft rotated at 2039.3 RPM and the slow one at 175.8 RPM. With a specimen diameter of 70 mm this leads to a surface sliding speed of 6.8 m/s and a slide-to-roll ratio of 1.68. The relatively high sliding speed was determined by preliminary tests delivering scuffing failure at an achievable force level for a reasonable test duration.

The lubricant flow has an obvious effect on the temperature of the contact through heat energy transfer. However, its influence in real

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Kin. viscosity @40 °C</td>
<td>[mm²/s] 150</td>
</tr>
<tr>
<td>Kin. viscosity @100 °C</td>
<td>[mm²/s] 15</td>
</tr>
<tr>
<td>Density @15 °C</td>
<td>[kg/m³] 897</td>
</tr>
<tr>
<td>VG class</td>
<td>[-] 150</td>
</tr>
<tr>
<td>Flash point (COC)</td>
<td>[°C] 240</td>
</tr>
<tr>
<td>Pour point</td>
<td>[°C] 24</td>
</tr>
</tbody>
</table>
applications is often relatively challenging to predict. During the testing
the oil fow to the contact inlet was 6 l/min, which according to the
preliminary tests is an adequate oil supply providing clear bulk
temperature increase after load rise. The lubricant inlet temperature
was set to 60 °C, which can be considered as a typical value or slightly
below average for the common temperature range of 40–100 °C used
for industrial gears for power transmission applications [28]. These
parameters were kept constant during the tests.

All the tests follow a common stepwise increasing normal load
structure as shown in Fig. 6. The loading is increased after 1000 s at
that particular load level. During this period aggressive asperities on
contact surfaces are flattened and the specimen’s temperature is
simultaneously increased and stabilised at a higher level.

The step magnitudes in Fig. 6 represents the Hertzian maximum
contact pressure \( p_{\text{max}} \) for elliptical contact between the discs:

\[
p_{\text{max}} = \frac{3F_{\text{c}}}{2\pi a b}
\]

where a and b are the semimajor axes of the contact ellipse. The
increases from zero to step 1 and from step 1–2 are somewhat larger
than the other steps due to the current test setup’s lower accuracy in
force control with relatively low normal forces. The step increases from
2 to 6 are about 230 MPa per step and they are considered as running-
in steps covering a running-in up to full loading of a newly-ground gear
teeth in real-life applications. These steps allow the specimen’s initial
settling of the surfaces. During this period the temperature and the
torque signals are stabilised. The scufing failure and overload cycles
are not applied until level 6 and upwards, which proceed in steps in
150 MPa.

The test matrix contains three different types of test runs:

A. Reference test
B. Series of rapid sudden overload cycles with high magnitude
C. Series of sudden overload cycles with increasing magnitude

For the reference test A, the stepwise load increase is continued
until scufing occurs. The purpose of this is to set the nominal level for
failure initiation.

In series B the objective is to apply overload cycles instantly after
the step increase i.e. during the stabilisation process. Fig. 7 shows an
example of a series of overload cycles starting from level 7. The
magnitude is set to reach level 9.

Series C is more aimed at the running-in process. The overloading
begins at a clearly lower level than the scufing level of the reference
tests, but in contrast to the series B tests, the overload cycles are
introduced to the system at steadily increasing rates. After stabilisation
at the chosen level the first overload cycles are applied in magnitudes
reaching to the next level. This 1000 s period is followed with another
1000 s period at the starting level, which in turn is followed with
overload cycles one level higher than the previous ones. This procedure
is repeated until scufing occurs. An illustration of the procedure
starting from level 7 is shown in Fig. 8.

The actual overloading experienced by a ship’s shafting line when
that ship is sailing through icy seas for example, is challenging to
reproduce. With its propeller constantly striking ice fles and the
subsequent loading on the gear contacts, it is difcult to be certain what
is actually going on in a real-life situation. In addition, the measure-
ment data achieved from such a situation is often considered installa-
tion-specifc, depending on the dynamic behaviour of the mechanical
system. Therefore, only coarse estimate of potential loading schemes is
utilised for both series B and C. The loading during the overloading

---

**Fig. 6.** First 10 steps of a stepwise loading structure for the scufing tests.

**Fig. 7.** Illustration of stepwise increasing overload cycle series of test series B. Series of
overload cycles is applied after a series of nominal load cycles at the chosen level.

**Fig. 8.** Principle of stepwise increasing overload cycle of test series C. Each overload
cycle regime is followed with a nominal load cycle regime at the particular level.
The actual scuffing initiation is manifested as a rapid increase in the torque signal. This is regarded as a stop criterion for the testing.

5. Results and discussion

Each test was repeated three times with the same loading scheme. The lubricant temperature at the contact inlet remained within the range of 60 ± 1.5 °C for all the tests.

The surface roughness for all the test specimens was measured at 60 ± 1.5 °C for all the tests.

Table 2

<table>
<thead>
<tr>
<th>Test</th>
<th>R_a [μm]</th>
<th>R_z [μm]</th>
<th>Radius [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>A1, curved</td>
<td>0.294</td>
<td>2.524</td>
<td>98.0</td>
</tr>
<tr>
<td>A1, flat</td>
<td>0.291</td>
<td>2.356</td>
<td>N/A</td>
</tr>
<tr>
<td>A2, curved</td>
<td>0.295</td>
<td>2.011</td>
<td>97.8</td>
</tr>
<tr>
<td>A2, flat</td>
<td>0.300</td>
<td>2.503</td>
<td>N/A</td>
</tr>
<tr>
<td>A3, curved</td>
<td>0.313</td>
<td>2.982</td>
<td>101.3</td>
</tr>
<tr>
<td>A3, flat</td>
<td>0.301</td>
<td>2.253</td>
<td>N/A</td>
</tr>
<tr>
<td>B1, curved</td>
<td>0.313</td>
<td>2.993</td>
<td>97.4</td>
</tr>
<tr>
<td>B1, flat</td>
<td>0.273</td>
<td>1.960</td>
<td>N/A</td>
</tr>
<tr>
<td>B2, curved</td>
<td>0.301</td>
<td>2.281</td>
<td>95.0</td>
</tr>
<tr>
<td>B2, flat</td>
<td>0.300</td>
<td>2.479</td>
<td>N/A</td>
</tr>
<tr>
<td>B3, curved</td>
<td>0.292</td>
<td>1.839</td>
<td>98.1</td>
</tr>
<tr>
<td>B3, flat</td>
<td>0.274</td>
<td>1.976</td>
<td>N/A</td>
</tr>
<tr>
<td>C1, curved</td>
<td>0.308</td>
<td>2.636</td>
<td>102.6</td>
</tr>
<tr>
<td>C1, flat</td>
<td>0.307</td>
<td>2.433</td>
<td>N/A</td>
</tr>
<tr>
<td>C2, curved</td>
<td>0.300</td>
<td>2.225</td>
<td>98.1</td>
</tr>
<tr>
<td>C2, flat</td>
<td>0.343</td>
<td>3.645</td>
<td>N/A</td>
</tr>
<tr>
<td>C3, curved</td>
<td>0.308</td>
<td>3.678</td>
<td>97.1</td>
</tr>
<tr>
<td>C3, flat</td>
<td>0.290</td>
<td>2.209</td>
<td>N/A</td>
</tr>
</tbody>
</table>

5.1 Surchaing failure

An averaging algorithm was applied on the actual measured signals. A moving average of changing length, depending on the observed load step, was calculated for the series of data points over the full data set. This was done in order to remove any electrical distortion caused by the measurement equipment and/or geometrical imprecision originating from the manufacturing tolerances or deformation of the components. The oscillation around the average for normal force was from ± 1.5... ± 5%, which corresponds to ± 0.5... ± 1.7% in maximum Hertzian contact pressure just before scuffing occurred. The upper limit of ± 1.7% means there is a ± 40 MPa oscillation in the maximum Hertzian pressure. This indicates that the step size of 150 MPa is enough to produce a distinct difference between the steps.

In the torque measurements the oscillation was ± 6... ± 8% respectively. The upper limit of ± 8% leads to ± 0.002 oscillation around the mean for the evaluated friction coefficient, which also shows that the tests are of sufficient accuracy.

The results from the test cases are shown in Table 3. The tests within each groups A, B and C are broadly the same, although there are minor differences between them. Case A3 was tested with a step length of 1500 s and reversed shaft speeds, i.e. the fast shaft was programmed to run at the slow shaft's speed, and vice versa. This experiment showed results in- line with A1 and A2. The starting level for the overload cycles for Case C1 was one level higher than for C2 or C3.

The amount of torque originating from the bearings and seals was calculated with Eq. (2). These values were subtracted from the measured signal in order to estimate the torque and friction between the discs at different loads. Typical result curves for the altered test types are shown in Figs. 10, 11 and 12.

The newly ground surfaces, along with the torque from the bearings and shaft seals produce inaccuracies in the determination of the contact torque and the ensuing friction coefficient. These are between 0 and roughly 1000 s, as can be seen in Figs. 10, 11 and 12. One reason for this is that the torque between the discs is at the same level as the torque rising from the bearings and seals. This leads to the partly labile behaviour of the system at low load levels.

5.1.1. Findings related to scuffing initiation

As Table 3 shows, in general, the initiation of scuffing requires high loads in terms of contact pressure and bulk temperature. Although the results show good repeatability within each test type, there are distinctive differences between the three series of tests, i.e. A, B and C. For example, in series B, with the overload cycles, the scuffing was initiated at a one-step lower load level and with about a 30% lower bulk temperature than it was for series A. Since the friction coefficient in series B was approximately 30% higher, the scuffing initiation may be explained by localised, instantaneous, high heat generation and subsequent high (flash) temperatures on the surfaces while they are settling. In this regard, Castro et. al. [29] have proposed an approach to predicting gear scuffing safety by dividing the analysis into global and local parts depending on the state of the running-in and the contact pressure.

A comparison of test series A and C shows that it is possible to apply loading up to 18% higher in terms of maximum Hertzian pressure without initiating scuffing. This is calculated by comparing the highest levels without scuffing, i.e. 9 and 12 respectively, with corresponding max Hertzian pressures of roughly 2.45 GPa and 2.90 GPa, respectively. It is also worth of noting, that the bulk temperatures at the moment of scuffing initiation do not differ significantly. Therefore, and because of the lower scuffing level reached with test C3, a need for further research is obvious.

When looking at the utilised normal forces, the most interesting difference can be seen between series B and C. If C3 is discounted, the required force for scuffing is nearly 94% higher in series C than it is in series B, and about 58% higher than in series A.
Table 3 shows that the bulk temperature measurement can only provide a rough estimate of scuffing initiation. The gear’s loading history and its consequent surface condition have a significant impact on the development of a scuffing failure. The effects of these factors on the calculation methods for evaluating scuffing risk will be studied in more detail in the near future.

5.1.2. The effect of sudden overload cycles

According to the test results in Figs. 10, 11 and 12, each increase in the load is followed by a settling period of about 200–400 s most clearly manifested at 4000 to 7000 s as a reduction in the torque signal. The risk of scuffing seems to be high during this period of stabilisation.

Table 3 shows that the bulk temperature measurement can only provide a rough estimate of scuffing initiation. The gear’s loading history and its consequent surface condition have a significant impact on the development of a scuffing failure. The effects of these factors on the calculation methods for evaluating scuffing risk will be studied in more detail in the near future.

5.1.2. The effect of sudden overload cycles

According to the test results in Figs. 10, 11 and 12, each increase in the load is followed by a settling period of about 200–400 s most clearly manifested at 4000 to 7000 s as a reduction in the torque signal. The risk of scuffing seems to be high during this period of stabilisation.

### Table 3

<table>
<thead>
<tr>
<th>Test</th>
<th>Overload starting level</th>
<th>Scuffing level</th>
<th>Normal force [kN]</th>
<th>Bulk temperature [°C]</th>
<th>Hertzian pressure [GPa]</th>
<th>Fric. coefficient [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>A1</td>
<td>N/A</td>
<td>10</td>
<td>9.67</td>
<td>178</td>
<td>2.59</td>
<td>1.73 0.022</td>
</tr>
<tr>
<td>A2</td>
<td>N/A</td>
<td>10</td>
<td>9.45</td>
<td>177</td>
<td>2.57</td>
<td>1.71 0.026</td>
</tr>
<tr>
<td>A3</td>
<td>N/A</td>
<td>10</td>
<td>9.73</td>
<td>175</td>
<td>2.60</td>
<td>1.73 0.022</td>
</tr>
<tr>
<td>B1</td>
<td>7</td>
<td>9</td>
<td>7.88</td>
<td>136</td>
<td>2.42</td>
<td>1.61 0.031</td>
</tr>
<tr>
<td>B2</td>
<td>7</td>
<td>9</td>
<td>7.89</td>
<td>133</td>
<td>2.42</td>
<td>1.61 0.030</td>
</tr>
<tr>
<td>B3</td>
<td>7</td>
<td>9</td>
<td>7.75</td>
<td>127</td>
<td>2.41</td>
<td>1.61 0.033</td>
</tr>
<tr>
<td>C1</td>
<td>8</td>
<td>13</td>
<td>15.0</td>
<td>169</td>
<td>3.04</td>
<td>2.03 0.023</td>
</tr>
<tr>
<td>C2</td>
<td>7</td>
<td>13</td>
<td>15.5</td>
<td>159</td>
<td>2.76</td>
<td>1.84 0.024</td>
</tr>
<tr>
<td>C3</td>
<td>7</td>
<td>11</td>
<td>11.6</td>
<td>-</td>
<td>-</td>
<td>- 0.021</td>
</tr>
</tbody>
</table>

Fig. 10. Calculated Hertzian maximum contact pressure, torque at the contact and friction coefficient between the discs for test A1. Scuffing is initiated after step increase at 9000 s.

Fig. 11. Results for test B1. Series of overload cycles are applied after step increase at 6000 s leading to scuffing failure.

A comparison of series A and B shows that although an increase of about 150 MPa in the maximum Hertzian contact pressure is not damaging, 300 MPa during the settling period clearly leads to failure under high loading conditions. However, if the contacting surfaces are run-in properly, an overload of roughly 450–750 MPa can be tolerated. This tendency is clearly shown in the results from test series C, despite of the deviations due to local surface roughness or material variation.

5.1.3. The evolution of friction

The friction coefficient rises along with the load from 1000 s until it reaches a turning point at between 3000 and 4000 s. Thereafter the friction begins to decline as shown in Figs. 10–12.

During the first period, the development of friction is dominated by
increasing viscosity along with the loading as well as by asperity contacts in the mixed lubrication region as the flattening of the contacting surfaces progresses. Once the test set-up is at the level of ~1550 MPa in maximum Hertzian contact pressure, there is a clear change in the process as the friction begins to decline. There are two possible scenarios, or combination of them, which could explain this. The first is that at this point the aggressive asperities are considerably levelled and the system is moving towards an elasto-hydrodynamic lubrication regime, where the definition of the friction coefficient is dictated by the shear stresses in the lubricant. This period is characterised by the tendency of the friction coefficient to slightly drop along with increases in the load. This produces a consequent reduction in the lubricant’s limiting shear stress along with the increasing temperature, as noted in Ref. [30]. It is also noticeable that during this period the introducing of overload cycles has no significant effect on the friction coefficient. In the second scenario, the additive system of friction improvers in the lubricant become active, which would also have a strong influence on the system’s behaviour.

5.1.4. The evolution of the bulk temperature

The bulk temperature climbs to a new level after each step-increase in the load, as shown in Fig. 13. This is due to an increase in the generation of frictional heat in the contact. After reaching the local maximum, in roughly 300 s, the heat production and the cooling via the transfer of heat energy within the lubricant and to the surrounding structures stabilises. A small drop in the temperature can be observed, indicating a reduction of the friction coefficient. This is also supported by a simultaneous drop in the torque signal.

The comparison of the temperature curves for tests A1 and C3 in Fig. 13 show, that for example between 7000 and 8000 s the overload-type of loading to one step upper level produces approximately half of the temperature rise that it does under full-time loading on this level. After stabilisation and return to full-time loading at the previous load level with the overload procedure, the bulk temperature is observed to be declining, which indicates reduction in corresponding friction. This trend seems to continue during the process of applying the overload cycles in series C. This suggests that the heat generation can be kept under control, decreasing the risk of scuffing, without losing the advantage gained from the surface flattening with this procedure.

Fig. 12. Results for test C2. Series of overload cycles are visible in all the curves. Scuffing initiation occurs in the beginning of the 5th series.

Fig. 13. Bulk temperature for test cases A2, B1 and C2. Scuffing initiation is seen as a rapidly rising peak in the temperature signal.

6. Conclusions

This paper has presented the results of an approach to investigating scuffing initiation under overloading by means of a twin-disc test device. The main conclusions can be summarised as follows:

- The twin-disc test arrangement is well suited to study scuffing under dynamic loading.
- The mechanical and thermal loads are efficiently controlled and their influence on the behaviour of the lubrication film are well captured.
- A wide range of applied loads and surface conditions as a result of running-in can be studied effectively.
- Scuffing initiation due to overloading is substantially dependent on the occurrence of the loading.
- Overload cycles during the running-in period significantly increase the risk for scuffing.
- The introduction of overload cycles in a controlled manner, taking into account the settling of the surfaces and the increase in surface temperature, may improve the component’s resistance to scuffing initiation under heavy loading.
- The load history of the gears and their subsequent surface condition caused by running-in have a considerable effect.
- Using the bulk temperature to predict the risk of scuffing in a material-lubricant pair only gives a rather coarse result.

The improved performance against scuffing failure gained by introducing overloading is not only dependent on the magnitude of the applied load cycles or the sequence, but the prevalent temperature and the contact pressure. In addition, also other variables such as the
gear’s material, its initial surface condition and the lubricant, which impacts were not studied within the tests, may have a clearly noticeable effect. Because of this complexity, further research is required to enable the results of these tests to be utilised in dimensioning and more precise calculation of the risk for scuffing. In practice the achieved knowledge could be also applied to, for example, running-in processes of gears in marine applications.

Acknowledgements

This study is carried out under the auspices of the ArTEco (Arctic Thruster Ecosystem) project, which is part of the Maritime Technologies II Research Programme (MARTEC II) under the European Commission ERA-Net scheme. The authors wish to gratefully acknowledge financial support from Tekes decision number: 40303/14 and other partners in the project. The funders have had no influence on the content of this paper.

References

[26] Sodhi RV, Dhuppalla AK, Evans HF. Scuffing performance of a hard coating under EHL conditions at sliding speeds up to 16 m/s and contact pressures up to 2.6 GPa. J Tribology 2008;130.
An experimental investigation of scuffing initiation due to axial displacement in a rolling/sliding contact

M Savolainen, A Lehtovaara

Tribology International, 119 (2018), 688-697

Publication reprinted with the permission of the copyright holders.
An experimental investigation of scuffing initiation due to axial displacement in a rolling/sliding contact

Matti Savolainen *, Arto Lehtovaara
Group of Tribology and Machine Elements, Laboratory of Materials Science, Tampere University of Technology, PO BOX 527, 33101 Tampere, Finland

ARTICLE INFO

Keywords:
Scuffing
Twin-disc
Axial displacement
Friction

ABSTRACT

This paper presents an investigation into the effect of axial displacement on the initiation of scuffing in a rolling/sliding contact by means of a twin-disc test device. Firstly, an existing twin-disc test device is modified to be suitable for applying axial displacement cycles and a rolling/sliding motion to the disc contact. Secondly, the results of two test series are analysed. Both series are subjected to a combined step-wise increasing normal force and constant rolling/sliding loading. The first series is without axial displacement, while in the second series, continuously reciprocating axial displacement cycles are applied starting at a specific normal force level. The results show that a stabilized axial displacement loading can increase the scuffing load carrying capacity.

1. Introduction

Ship shafting lines usually transfer power through bevel gears. These mechanical systems experience occasional overload cycles caused, for example, by the ship's propeller hitting hard obstacles such as ice floes when operating in polar regions. The resulting deformation of the supporting structures can cause axial displacement of one or the other of the gear wheels located between the shafts. Consequently, additional sliding between interacting gear teeth may occur, which may disturb stabilized behaviour of the contact during operation.

There appears to be a close relationship between the initiation of scuffing and the lubricant film breakdown [1]. The thickness of the elastohydrodynamic lubrication (EHL) film may decrease as its viscosity decreases due to extensive frictional heating, which in turn is caused by the high surface sliding velocity and the contact pressure. This may lead to damaging asperity contacts. According to Jacobson [2], if the shear stresses in the lubricant reach the shear strength of the lubricant in the prevailing condition, the oil is no longer dragged along the surfaces straight into the contact. The oil loses its grip on the surface and slides down into the valleys between the surface asperities. This can cause the surface asperities to break through the oil film and come into direct contact with the opposite surface. On the other hand, if most of the limited shear stress of the lubricant is utilised in the main direction of the sliding movement, even relatively minor sliding in another direction may allow the lubricant to slip out sideways. This results in a thinner oil film or film breakdown [2]. Moreover, Larsson et al. have reported that a high sliding speed also seems to increase the risk for oil film breakdown [3] and Castro et al. have proposed an approach for predicting the likelihood of scuffing by dividing the problem in global and local parts and by considering the state of the running-in of the surfaces [4]. Additionally in a study by Li et al. the interactions between the gear dynamics and the gear tribological behaviour was researched [5]. It was found that not only the line-of-action dynamic response but also the off-line-of-action vibratory motion affects the surface flash temperature. For the reason that in extreme conditions scuffing can initiate after only a short period of severe load cycles, axial displacement may well have an effect on the lifetime of a gear mechanism.

The traditional experimental approach to assessing the scuffing load-carrying capacity of a lubricant-material pair has been standardized and is conducted with FZG tests [6]. Additionally, bevel-gear-based test devices have been utilised successfully over the years. In such studies, the focus has been on varying the lubricant, the oil temperature, the applied loading, the amount of oil supplied and the effect of a coating [7,8]. However, these studies have used a fixed gear geometry, which limits the extent of the variations in the types of loads that can be applied.

A twin-disc test device can be utilised to generate an elliptical pressure distribution by specifically defining the geometries of the discs [9,10]. It is widely accepted that the pressure trace mimics the contact between the gear teeth. However, the usage of the results in real gears demands attention of several factors such as formation of the lubricant film and friction coefficient as well as surface roughness and topography in the contact location [11]. In several papers, research using the
The twin-disc test device has focused on examining friction behaviour, pitting formation and the prediction of wear as well as the effect that the condition of the surface has on scuffing initiation [12–16]. In this regard, Holm et al. concluded that in an EHL contact, surfaces with circumferential grinding show decreasing film thickness in line with an increasing surface roughness [17]. In addition, Li proposed the scuffing resistance performance to decrease as the difference between the roughness lay direction angles of the two surfaces increases [18]. In another study, Holm et al. varied the direction of the velocity vectors (skew angle) of the rotating discs in contact [19]. This led them to the conclusion that increasing the skew angle increases the thickness of the oil film for constant absolute values of the sum velocity. In addition, Pu et al. studied a load situation, where the entraining and sliding velocity directions do not coincide with the principal axis of the contact ellipse by using an ellipsoid-on-disc arrangement. They concluded that increase in the entraining velocity angle causes rise of flash temperature and changes in the friction coefficient can be observed depending on the increase in the angle and the prevalent lubrication regime [20].

There is not a lot of research data available related to scuffing initiation and additional displacement loading in an EHL contact. Thus, this paper presents an investigation into the relationship between axial displacement cycles and the initiation of scuffing in a rolling/sliding contact by using a twin-disc test device. The fundamental idea of this approach is to disturb the stable rolling/sliding contact behaviour with an additional displacement loading in controlled conditions.

Much of the information about this test device, including the hydraulic and lubrication systems and the control and measurement equipment have been presented in Ref. [21]. That publication also describes the test discs, the reference test results and the pertinent conclusions. Therefore, the main focus of this paper is on the equipment used to create an axial displacement in the contact and the associated test results. This paper also presents previously unpublished analysis of the surfaces of the discs used in the earlier test series and a comparison to the discs used in the tests introduced in this paper.

2. Test equipment

The twin-disc test device, which was used in this research, is shown in Fig. 1 and is also described in detail in Refs. [21,22]. For this study, the device was updated in order to enable the possibility of introducing axial displacement cycles to one of the discs. This axial movement is generated with a hydraulic cylinder, which is attached to the bearing housing of the rotating fast shaft as shown in Fig. 2. The housing of the moving bearing and the radial load cylinder are mounted on the frame with joints that permit axial displacement. A through-rod-type cylinder was chosen for the axial displacement cylinder, because it allows the position of the disc to be monitored with a proximity sensor, which can sense the free end of the cylinder rod. The cylinder is driven with a high-response directional control valve with electrical position feedback. In addition the valve control unit was reprogrammed to cope with upgrades to the mechanical and hydraulic systems.

The speed of the rotation and the direction of the shafts are adjusted with frequency converters, which are connected to electric motors. These motors can reach a maximum speed of 6000 RPM and are coupled to the shafts via flexible couplings, which allow for minor misalignments between the connected axles.

During the tests a separate hydraulic unit supplies lubricant to the disc contact and to the shaft bearings. The lubricant flow is directed to the contact from the inlet side. A microcontroller unit automatically regulates the inlet temperature of the lubricant. This can be set to maintain any constant temperature between 25 and 120 °C. The flow of lubricant can be set anywhere between 0.5 and 20 l/min. This value can be adjusted using a hydraulic pump driven by a frequency-converter in combination with manual valves.

2.1. Measurements

The measurement system not only records all of the parameters for the lubricant as described above, but it also records the normal force, the torque of the slow shaft and the applied axial displacement of the fast shaft. In addition, the bulk temperature of the fast shaft is logged with a thermocouple, which is located approximately 3.5 mm beneath the surface at the disc contact. The measured temperature data is transferred from the rotating shaft to the stationary measurement equipment with a telemetry system.

The friction coefficient between the discs is calculated from the measured normal force and the torque. In order to obtain the actual friction between the discs, the torque originating from the seals and the bearings of the shaft, has to be out-calibrated from the measured signal. This is done using the technique explained in Ref. [21].

3. Test specimen and lubricant

The test discs were made from 18CrNiMo7-6 (EN 10084) and were case hardened to a surface hardness of 59–61 HRC. The diameter of the disc at the contact surface is 70 mm, and the width of the possible contact area is 10 mm. In order to generate the elliptical contact trace, the discs on the fast rotating shaft are ground to a radius of 100 mm in the axial...
direction, while the discs on the slow rotating shaft are flat. This grinding is done with a specially designed grinding device. This device can be used to manufacture a surface topography comparable to the flank of a real gear tooth, whose grooves are perpendicular to the sliding/rolling direction. Fig. 3 illustrates the specimen surface after grinding.

All the tested discs are ground after they have been assembled on a shaft. This minimizes the risk of any eccentricity error between the contact surface and the locations of the bearing's mountings. The requirement for the average surface finish (Rz) ranges from 0.28 to 0.32 μm on each disc and the average distance of the highest peak from the lowest valley (Rk) is designed to be below 3 μm. This grinding procedure is explained in more detail in Ref. [22].

The lubricant used in the tests is a mineral oil-based commercial gear oil that is commonly used in marine propulsion systems. The oil has an EP-additive system and is specifically designed for heavy-duty industrial gears. The specifications for the lubricant are shown in Table 1.

### 4. Experimentation procedure and test cases

The temperatures of the lubricant and both of the shafts were stabilized to a constant level during the first hour of every test. This was done by rotating the shafts at the same speed of 175.8 RPM with lubricant circulation but without the normal force loading. During test runs, the fast shaft was rotated at 2039.3 RPM and the slow one at 175.8 RPM. These speeds result in a slide-to-roll ratio of 1.68, which supports scuffing initiation at an achievable radial force level. The resulting sliding speed of 6.83 m/s is slightly higher than the sliding speed at the tooth tip (5.56 m/s) for the FZG scuffing test with gear type A [6]. Therefore, the used sliding speed can be considered relatively high in comparison to values present in real gear applications.

In a real situation, it is often challenging to predict the cooling effect of the lubricant flow at a contact point. However, the oil flow to the contact inlet was set to 6 l/min. Preliminary tests showed that this value was able to deliver an adequate oil supply and a clear bulk temperature increase during a load step rise. The lubricant inlet temperature was set at a contact inlet was set to 6 l/min. Preliminary tests showed that this value was able to deliver an adequate oil supply and a clear bulk temperature increase during a load step rise. The lubricant inlet temperature was set at 60 °C. Although slightly below average, this can be considered as a typical value for the common temperature range of 40 °C-100 °C found in industrial gears for power transmission applications [23]. The lubricant flow and the temperature were kept constant during the tests.

The test matrix comprises two different types of test runs:

A. Reference test
B. Addition of series of axial movement cycles
   a. Starting simultaneously with radial load increase
   b. Starting in the middle of the execution of a radial load step (stabilized radial load condition)

Both tests follow a common procedure of stepwise increases in the normal load, as presented in Fig. 4. The load is increased once every thousand seconds (1000 s). During each 1000 s period at a particular load level, the aggressive asperities on the contact surfaces are levelled. The temperature of the discs is simultaneously increased and stabilized at a higher level. The magnitude of the step is defined according to the Hertzian maximum contact pressure $p_{\text{max}}$ for elliptical contact between the discs:

$$p_{\text{max}} = \frac{3F_N}{2\pi ab}$$

where $a$ and $b$ are the semi-major and semi-minor axes of the contact ellipse respectively. With the current set-up the steps from zero to step 1 and from step 1 to step 2, are greater than the following steps. Steps 2 to 6 are about 230 MPa each. These steps can be considered as running-in steps, which are needed before the device enters the actual testing range. From load level 6 and upwards, the step increase is 150 MPa. The difference between the A series of tests and the Ba series is that in the latter series the continuous axial movement cycles are all applied at the same time when starting the increase to load level 7, which occurs after 6000 s. The aim is to analyse the combined influence of a normal load increase and axial displacement cycles on scuffing initiation. In the Bb series, however, the axial movement begins after completing the stabilization process of the surfaces, which is approximately 500 s after the radial load increase. This test series was added to the testing plan in order to observe the effect that the settling of the surfaces has on scuffing initiation. In all cases, the axial movement is resumed along with the stepwise increases in the normal force loading.

It is extremely difficult to predict the possible occurrence and magnitude of any axial displacement cycles between the gear teeth in the gearbox of a real shafting line. The loading may vary significantly depending on the excitation from the propeller and the mechanical

---

**Table 1**

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Kin. viscosity @40 °C</td>
<td>150 [mm²/s]</td>
</tr>
<tr>
<td>Kin. viscosity @100 °C</td>
<td>15 [mm²/s]</td>
</tr>
<tr>
<td>Density @15 °C</td>
<td>897 [kg/m³]</td>
</tr>
<tr>
<td>VG class</td>
<td>[-]</td>
</tr>
<tr>
<td>Flash point (COC)</td>
<td>240 [°C]</td>
</tr>
<tr>
<td>Pour point (COC)</td>
<td>-24 [°C]</td>
</tr>
</tbody>
</table>

---

![Fig. 3. Perpendicular grinding of a test specimen.](image)

![Fig. 4. Stepwise loading procedure for the scuffing tests.](image)
behaviour of the entire shafting system. For this reason the measurement data for any such situation is often considered to be installation-specific. However, a rough estimate of a potential loading scheme is applied to the tests conducted within this study. The magnitude of the axial oscillation is ±1 mm around the middle line of the discs. The amplitude of the oscillation is roughly 50% of the length of the ellipse’s semi-major axis at load level 7, that corresponds to about 2.1 GPa in maximum Hertzian contact pressure. The comparable value at 3 GPa is 35%. The frequency of the movement is approximately 12 Hz and the shape represents a saw-tooth pattern. Fig. 5 illustrates the centre point of the contact travel path on the disc of the slow rotating shaft. The saw-tooth path is roughly 0.3% longer in comparison to straight-line path causing insignificant change in sliding speed.

The stepwise normal load addition is continued until either scuffing failure occurs or 3 GPa in maximum Hertzian pressure is reached. If scuffing is initiated, it is recognised as a rapidly increasing torque.

5. Results and discussion

The type A test series was repeated three times and the type B series, four times. The lubricant inlet temperature was within the range of 60 ± 1.5 °C for every test. The surface condition of all the test discs was measured at the middle of the contact at three separate points in the circumferential direction of the disc. The angle between the measurement points was approximately 120°. Table 2 presents the average of the three surface roughness’s and the radius of the curvature measurements for the ground test discs. The numbers reveal only small deviations between the discs for the Rz values. However, the scatter is slightly higher for the Ra values especially in the B test series; the Ba1 discs can be considered to be slightly rougher than the Ba3 and Bb1 discs.

A typical surface profile of a single measurement before testing is show in Fig. 6. The length of the measurement is 4 mm, which equals to five sampling lengths each of 0.8 mm. The round curvature of the disc was removed from the measured signal.

![Disc width [mm]](image)

![Disc circumference [mm]](image)

**5.1. Scuffing failure**

The results of the tests are shown in Table 3 and the typical measurement curves are shown in Figs. 7–9. Furthermore, all the result curves expect the test A3, are represented in Fig. 10 to observe the dispersion between the tests. The curves for the test A3 were left out, since the step duration was significantly longer in comparison to the other tests. The measured torque curves were used to evaluate the corresponding friction coefficient results. Before evaluation, the torque induced by the seals and bearings was calibrated out from the measured signals by using the experimentally found equation presented in Ref. [21]. In addition, all the measured signals were worked out with an algorithm for the moving average. The length used for the averaging is changing according to the processed load step. This procedure is required in order to remove any interference from the signals emanating from the electrical measuring equipment or any geometrical imprecision due to manufacturing tolerances and/or deformations in the components. The oscillation of the normal force and the maximum contact pressure for the B series results were observed to remain within the same ranges as for series A. These are ±1.5 ± 5% for the normal force at the scuffing load level, which corresponds to ±0.5… ±1.7% in maximum Hertzian contact pressure.

Testing was stopped for all the tests in series A and for the tests in Ba3 and Bb1 immediately after scuffing was detected in the form of the rapidly increasing torque. However, after scuffing initiation, the shafts continued to rotate for a short period as can be seen clearly for example in torque curves in Fig. 10. No scuffing initiation was observed for the Ba1 and Ba2 test series, and therefore, the testing continued until 3 GPa in maximum Hertzian contact pressure was reached. The final step increase to this load level raised the torque between the discs to a level at which the test disc had become detached. This effect also occurred in the Bb1 series. However, Table 3 shows the highest recorded bulk temperature values before the disconnection for the tests Ba1, Ba2 and Bb1.

Figs. 7–9 show inaccuracies in the torque and the ensuing friction coefficient signals between 0 and roughly 2000 s. This is because at low load levels the torque originating from the bearings and seals is at the same level as the contact torque between the discs, although they are in opposite directions. However, this phenomenon can be discounted when observing scuffing behaviour at higher load levels. Clearly noticeable dispersion between the tests can be seen in between the friction curves in Fig. 10. For instance even though the difference between torque curves is very small at the second load level, the evaluated friction coefficients in the disc contact vary roughly from 0.025 to 0.0375. This may be originating from the bearings and seals as their effect to the friction coefficient calculation is the highest especially at the

### Table 2

Average surface properties for the ground test discs.

<table>
<thead>
<tr>
<th>Test</th>
<th>Ra [μm]</th>
<th>Rz [μm]</th>
<th>Radius [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>A1, curved</td>
<td>0.29</td>
<td>2.52</td>
<td>98.0</td>
</tr>
<tr>
<td>A1, flat</td>
<td>0.29</td>
<td>2.36</td>
<td>N/A</td>
</tr>
<tr>
<td>A2, curved</td>
<td>0.30</td>
<td>2.01</td>
<td>97.8</td>
</tr>
<tr>
<td>A2, flat</td>
<td>0.30</td>
<td>2.50</td>
<td>N/A</td>
</tr>
<tr>
<td>A3, curved</td>
<td>0.31</td>
<td>2.98</td>
<td>101.5</td>
</tr>
<tr>
<td>A3, flat</td>
<td>0.30</td>
<td>2.25</td>
<td>N/A</td>
</tr>
<tr>
<td>Ba1, curved</td>
<td>0.34</td>
<td>3.11</td>
<td>101.4</td>
</tr>
<tr>
<td>Ba1, flat</td>
<td>0.33</td>
<td>3.29</td>
<td>N/A</td>
</tr>
<tr>
<td>Ba2, curved</td>
<td>0.33</td>
<td>3.43</td>
<td>101.3</td>
</tr>
<tr>
<td>Ba2, flat</td>
<td>0.31</td>
<td>2.12</td>
<td>N/A</td>
</tr>
<tr>
<td>Ba3, curved</td>
<td>0.30</td>
<td>1.71</td>
<td>100.0</td>
</tr>
<tr>
<td>Ba3, flat</td>
<td>0.30</td>
<td>1.77</td>
<td>N/A</td>
</tr>
<tr>
<td>Bb1, curved</td>
<td>0.29</td>
<td>1.49</td>
<td>102.9</td>
</tr>
<tr>
<td>Bb1, flat</td>
<td>0.31</td>
<td>1.73</td>
<td>N/A</td>
</tr>
</tbody>
</table>
Fig. 6. A typical surface roughness profile in the disc circumferential direction with $R_a = 0.29$ and $R_z = 2.47$.

Table 3

<table>
<thead>
<tr>
<th>Test</th>
<th>Axial load starting level</th>
<th>Scuffing level</th>
<th>Normal force [kN]</th>
<th>Bulk temperature [°C]</th>
<th>Hertzian pressure [GPa]</th>
<th>Fric. coef. [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>A1</td>
<td>N/A</td>
<td>10</td>
<td>9.67</td>
<td>178</td>
<td>2.59</td>
<td>1.73</td>
</tr>
<tr>
<td>A2</td>
<td>N/A</td>
<td>10</td>
<td>9.45</td>
<td>177</td>
<td>2.57</td>
<td>1.71</td>
</tr>
<tr>
<td>A3</td>
<td>N/A</td>
<td>10</td>
<td>9.73</td>
<td>175</td>
<td>2.60</td>
<td>1.73</td>
</tr>
<tr>
<td>Ba1</td>
<td>7</td>
<td>N/A</td>
<td>15.4</td>
<td>189*</td>
<td>3.03</td>
<td>2.02</td>
</tr>
<tr>
<td>Ba2</td>
<td>7</td>
<td>N/A</td>
<td>15.2</td>
<td>199*</td>
<td>3.02</td>
<td>2.01</td>
</tr>
<tr>
<td>Ba3</td>
<td>7</td>
<td>7</td>
<td>5.21</td>
<td>140</td>
<td>2.12</td>
<td>1.41</td>
</tr>
<tr>
<td>Bb1</td>
<td>7</td>
<td>12</td>
<td>12.2</td>
<td>216*</td>
<td>2.80</td>
<td>1.87</td>
</tr>
</tbody>
</table>

Fig. 7. Maximum Hertzian contact pressure, torque originating from the disc contact and friction coefficient between the discs for test series A1. Scuffing initiation can be seen as a rapid increase of the torque signal.

Fig. 8. Results for the test series Ba1. Scuffing failure was not initiated, but the test was stopped after reaching 3 GPa in maximum Hertzian contact pressure.
This effect decreases towards higher load levels, where the scuffing failures occurred. Also, minor deviations in the initial surface topography and running-in may have an influence.

### 5.1.1. Findings related to scuffing initiation

The results in Table 3 show clear uniformity in the scuffing initiation for the A test series. The scuffing failure occurred for all the cases when load level 10 was reached, which corresponds to approximately 2.6 GPa in maximum Hertzian contact pressure. When compared to the scuffing level of the B test series, the test discs in series Ba1 and Ba2 were able to tolerate approximately 60% higher normal force without initiating scuffing. In these tests, the maximum Hertzian contact pressure was about 3.0 GPa and the difference to A series is thus about 17%. On the other hand, scuffing failure occurred at level 12 for the case Bb1. That is a 12% increase in normal force and 8% in maximum Hertzian contact pressure in contrast to the series A results. In addition, scuffing failure occurred instantly after applying axial displacement movement for the Ba3 series.

A minor self-healing scuffing initiation was recorded in the case of Ba1 at 6000 s, which is the moment at which the axial displacement loading starts. This is seen as a sharp peak in the torque curve in Fig. 8. A similar peak can also be seen in Fig. 9 for the Bb1 test, which corresponds with the start of the axial displacement movement at about 6500 s. This may indicate an instantaneous decrease in the thickness of the EHL film. One possible explanation for this is that the lubricant flow into the EHL contact is disturbed because of the start-up of the axial displacement cycles and less oil is dragged into the contact, which results in a thinner film of oil. This increases the risk of metal-to-metal contact, which in turn may lead to a minor scuffing initiation. In addition, the similarities between the self-healing peaks of tests Ba1 and Bb1 suggest that increase of the radial load is not the dominant factor for the scuffing initiation when axial displacement cycles are applied.

After a self-healing event, the EHL contact behaviour seems to stabilize. The increases in the torque are roughly the same for the following radial load steps between A1 and Ba1 as illustrated in Figs. 7 and 8. The performance of the EHL contact also improves after applying the axial displacement cycles, since higher normal load levels are tolerated as is shown in Table 3. The B series results also indicate the challenging nature of scuffing, where a minor variance in either the lubrication or the components could cause failure or result in a self-healing scuffing initiation. However, the differences between the test discs found through the surface roughness values in Table 2 do not seem to have a clear correlation with the scuffing initiation. It should also be noted, that the rapid increase at the end of the torque curve in Fig. 8 is caused by incorrect shutdown procedure of the system.

### 5.1.2. Friction coefficient

It is challenging to see the effect of the axial displacement from the friction curve since there is such a small difference between any two consecutive load steps in the tests. However, larger changes are visible. As can be seen from Figs. 7–9, after 3000 s the friction coefficient decreases as the load increases. This decrease could be caused by any combination of several possible behaviours in the contact system. Firstly, the roughness of the contacting surfaces continues to flatten out along with any increases in the load possibly leading to decrease in the thickness of the oil film. Simultaneously the contact system working in the mixed lubrication regime continues to move towards an elasto-hydrodynamic lubrication regime, where the shear stresses in the lubricant may dominate in defining the friction coefficient. Moreover, the limiting shear stress tends to drop slightly along with increasing temperature [24], which also leads to a decrease in the friction coefficient. The continuous decrease in the friction coefficient may also result from the influence of the additive system of friction improvers, which become active after about 3000 s.

A momentary rise in the friction coefficient can be noted after a normal load increase. This is probably caused by the increased amount of low load levels [21].
asperity contacts on the interacting surfaces, which can occur under normal loads. After a settling period of 200–400 s the signal value stabilizes to a slightly lower level in each load step because of the flattening of the surface asperities. In addition, rapid peaks are seen in the friction coefficient curves in Fig. 8 (at 6000 s) and 9 (at 6500 s). These peaks are caused by the self-healing scuffing initiation and originate from the start-up of the axial movement cycles as discussed earlier. However, there is a drop in the friction coefficient after a self-healing event as is clearly visible for test Bb1 shown in Fig. 9. Similar behaviour can also be seen in Fig. 8 although not as clearly, since the radial load increase is influencing the friction coefficient at the same time. The fundamental reason for this reduction in the friction coefficient requires further research.

5.1.3. Bulk temperature

The typical bulk temperature curve follows the stepwise increasing path of the loading scheme as is shown in Fig. 11. The scuffing initiation is seen as a rapidly rising peak at the end of the curve for the series A2 and Bb1. On the other hand, the disconnection of the thermocouple from the test disc is shown by the falling temperature signal for Bb1 at about 12000 s. The difference between series A and Bb1 bulk temperatures is about 22% just before scuffing initiation. In addition, a roughly 10% higher bulk temperature was observed for the tests A1 and Bb2 in comparison to the series A values (without scuffing initiation) as presented in Table 3.

Fig. 11 shows a separation of the temperature curves for A2 and Bb1 at 6000 s. This was the moment, at which the axial displacement loading was applied in test Bb1. The increase in temperature for Bb1 after this load step is approximately 50% of the increase in A2. There are at least two possible explanations for this behaviour. Firstly, the axial displacement cycles spread the heat energy generated by friction over a wider area under the contact point, which leads to a smaller amount of heat transmitted to the measurement point of the thermocouple. Another explanation is that the generated heat energy is smaller in magnitude due to behaviour of the friction coefficient as discussed above. However, as there is no clearly visible reduction in temperature increase for the following load steps, it seems more likely that the temperature drop only occurs when the axial displacement cycles are first applied. This was also noted for the Bb1 test series.

The overall level of the bulk temperature is higher for Bb1 than for the other tests in Fig. 11. This is probably caused by variations in the surface topography, which lead to differences in the friction coefficient and consequently in the bulk temperature. Another explanation for the difference is that the measurement point may be simply slightly closer to the surface and therefore suffering higher heat load. It is also worth noting that in the Bb1 series the lubricant comes from another manufacturing batch, which may also have an effect.

According to the temperature results, an accurate prediction of scuffing initiation in advance or evaluation of the temperature on the contact surfaces requires information also on other parameters in addition to the bulk temperature. This is noted in Ref. [21] as well. However, stabilization of the temperature at the measurement location can be considered as a clear indication on reaching equilibrium in heat generation in the contact after a change in the load condition.

5.2. Analysis of the surfaces

The contact surfaces were analysed with an optical profilometer. Typical images of the tested surfaces and measurement line locations for the surface profiles are shown in Figs. 12 and 13. Fig. 14 illustrates the extent of plastic deformation and material removal from the surfaces along the measurement lines. The curves in Fig. 14 were produced by subtracting the measured profile of a tested disc, from an ideal profile curve of the particular test disc. For the curved discs, the ideal profile was generated according to the measured value of the radius in Table 2 and for the flat discs, a straight line was used. The comparison is shown in Fig. 15 for the fast rotating curved discs.

Scuffing failure leads to damaged contact surfaces caused by welding and tearing of the material. The darker colour of the surface indicates that the surface temperature has been high. This is particularly evident in the middle of the contact for the scuffed surfaces of series A2, as shown in Fig. 12. Large particles of the material have been torn off at the edges of the contact area. The dark area on the surface of the slow-rotating disc in A2. However, from a distance of 2 mm apart from the contact area, it is challenging to tell exactly how much combined wear and deformation has occurred compared to the original profile shape of the fast rotating disc in A2. However, from a distance of 2–8 mm, the depth of the maximum profile dip is approximately 9 μm in the middle of the contact area. In addition, a deformation of about 2 μm can be noticed outside the contact area, but in the opposite direction. The corresponding values for the Bb2 are roughly 2 μm and 5 μm. Fig. 15 illustrates the comparison between ideal and tested profile curves, which was the basis for the evaluation of the deformation and the material removal. This comparison shows the tendency of material loss in the middle of the contact for the A2 fast-rotating disc and plastic deformation for the Bb2 test disc.

![Fig. 11. Bulk temperature for tests A2, Bb1 and Bb1. The initiation of scuffing failure is seen as a rapid increase at the end of the curve for A2. The sudden decrease at about 12000 s for Bb1 indicates disconnection of the thermocouple from the test disc.](image-url)
Fig. 12. Typical contact surface after scuffing failure (series A2). The fast-rotating disc with curved grinding is on the left and the flat slower-rotating disc is on the right. The profile depth was measured along the horizontal line in the middle of the figure.

Fig. 13. Typical test disc surface for the Ba2 test series. The curved fast-rotating disc is on the left, where the axial displacement cycles were added during the testing. The flat slow-rotating disc is on the right. The profile depth was measured along the horizontal line in the middle of the figure.

Fig. 14. Disc surface profile depth after testing.

Fig. 15. Comparison of the measured surface profile after testing and the ideal surface curvature of curved discs over the contact area in the axial direction.
The slow rotating disc from the A2 test has suffered about 7 μm profile height reduction as shown in Fig. 14. Due to the material loss a flat plateau was formed in the middle of the contact area. In contrast, only about a 4 μm dip can be observed in the middle of the contact area for the slow rotating disc from Ba2, which is mainly due to the plastic deformation of the surface under high contact pressure. However, the grinding grooves remain visible as can be seen in Fig. 13.

5.3. Material hardness

The material hardness (HV0.2) was measured on a cross section of the test cases from A2 and Ba1 in the middle of the contact area. The reference curve of the untested disc was measured from a hardened, but unground test specimen. In order to be comparable with the tested discs, the starting location of the measurement, indicating the zero depth, was shifted to a depth of 0.25 mm from the actual surface. This corresponds to the average amount of material that would have been removed by grinding. The results for the fast rotating curved discs are presented in Fig. 16 and for the slow rotating discs, which were ground flat, in Fig. 17.

The hardness of the tested discs is generally roughly 15–20% lower after testing over the measurement range. The high temperature and consequent tempering of the material is believed to be the cause of this. The relative decrease in hardness is the greatest at the surface especially for the scuffed disc in Fig. 17, but a slight drop can also be observed for the fast rotating disc from the A2 test in Fig. 16. The reduction in hardness at the depth of 0.05 mm was roughly 22% for the A2 series fast rotating disc and about 34% for the slow rotating disc. The equivalent values for the Ba1 series discs are 14% and 22% respectively. The significant softening of the material indicates a decrease in the load-carrying capacity of the components. This is applicable for the series B discs as well, even though scuffing was not initiated.

6. Conclusions

In this study, a twin-disc test device was used to investigate scuffing initiation under axial displacement loading in a rolling/sliding contact. The main conclusions that can be drawn are as follows:

- Axial displacement cycles resulting in a change in the sliding path and direction, do have an effect on the scuffing initiation and the performance of the EHL.
- The introduction of reciprocating axial displacement cycles seem to instantaneously reduce the thickness of the oil film and thus momentarily increase the risk for scuffing initiation. This can be seen as a peak in the evaluated friction coefficient curve. However, after stabilization of the EHL system the friction coefficient decreases.
- In a stabilized condition with the axial displacement cycles, the EHL contact can tolerate significantly higher loading without scuffing initiation than it could in a similar load situation without axial displacement cycles.
- Even though scuffing is not initiated at a high load level with axial displacement cycles, high temperature and pressure loads lead to plastic deformation and material softening around the contact area.

It appears that a highly localized frictional temperature build-up is required to generate a scuffing failure. As the performed tests showed, small deviations in the contact path are capable of influencing the process of scuffing initiation by increasing the tolerance for the normal load. However, further research is required in order to find the fundamental reason for changes in the risk of scuffing under axial displacement cycles. In addition, a specific disc design in combination with the axial displacement loading may also enable an efficient method for studying undesired loading and contact load distribution on the tooth surface. The component material, the initial surface condition of the interacting surfaces and the lubricant, can also have an effect.

Acknowledgements

This study was carried out under the auspices of the ArTeCo (Arctic Thruster Ecosystem) project, which is part of the Maritime Technologies II Research Programme (MARTEC II) under the European Commission ERA-Net scheme. The authors wish to gratefully acknowledge the financial support of Tekes decision number: 40303/14 and other partners in the project. The funders have had no influence on the content of this paper.

Appendix A. Supplementary data

Supplementary data related to this article can be found at https://doi.org/10.1016/j.triboint.2017.12.007.

References


[15] Siddle RW, Dhulipalla AK, Evans HP. Scuffing performance of a hard coating under EHL conditions at sliding speeds up to 16 m/s and contact pressures up to 2.0 GPa. J Tribol 2008:130.


An approach to investigating subsurface fatigue in a rolling/sliding contact

M Savolainen, A Lehtovaara

International Journal of Fatigue, 117 (2018), 180-188
doi:10.1016/j.ijfatigue.2018.08.014

Publication reprinted with the permission of the copyright holders.
An approach to investigating subsurface fatigue in a rolling/sliding contact

Matti Savolainen*, Arto Lehtovaara

Group of Tribology and Machine Elements, Laboratory of Materials Science, Tampere University of Technology, PO BOX 527, 33101 Tampere, Finland

A R T I C L E   I N F O
Keywords:
Fatigue
Twin-disc
Subsurface

A B S T R A C T
This paper presents an approach to studying subsurface fatigue failures in a rolling/sliding contact with a twin-disc test device. A series of surface hardened test discs were tested with different load levels. A destructive inspection method was utilized showing subsurface cracks beneath the surface after a large number of load cycles with high loading. In addition, an elastoplastic finite element model considering the effects of increased hardness and residual stresses was created. The calculated results showed critical locations in the discs beneath the surface, which coincided with the experimentally found.

1. Introduction

There are two fatigue failure modes, which are well considered in standards for dimensioning gear wheels; tooth root bending fatigue and contact fatigue at the tooth flank [1,2]. With these failure modes, the initiation of the cracks leading to fatigue failure often start from or close to the surface. Because gear wheels are usually surface hardened, the failure may initiate from a subsurface zone beneath the hardened surface layer. In this region, the material strength is lower than at the surface and the compressive residual stresses induced by the hardening are not effective. However, the material is affected by the balancing tension residual stresses, which may accelerate the growth of the crack. Consequently, a new failure mode called Tooth Interior Fatigue Fracture (TIPF) has recently been proposed to account for the fatigue cracks that are initiated inside a gear tooth [3].

The material strength of a gear wheel has long been defined by conducting push-pull or bending tests for specimen made out of a specific material and with a specific surface treatment [4,5]. The results of these tests can be used to predict tooth root bending strength when dimensioning a gear wheel. Another commonly used approach is to conduct a full-scale bending test on a single gear tooth. In this way, the effects of the actual geometry and the surface treatment as well as robustness of the tooth are accurately reflected in the results [6–8].

Traditionally, the durability of the tooth flank has been defined with the aid of an FZG test device [9] by observing the formation of pitting or micropitting marks on the surface of the tooth flank. Several studies have focused on the effect of demanding thermal conditions, comparison of lubricant types and the effect of lubricant viscosity and the surface roughness [10–13]. In addition, the twin-disc test device has been successfully utilised in research into the effects that the lubricant type, surface roughness and surface treatment have on the formation of micropitting [14]. In their twin-disc tests, Oila et al. concluded that the initiation of micropitting is mostly controlled by contact pressure, while its progression is, in turn, mainly driven by the operating speed and slide-to-roll ratio [15]. In addition, Seo et al. have reported that higher material ductility and fracture toughness leads to higher resistance to contact fatigue, but a lower resistance to wear [16]. In another study by Seo et al., they evaluated the growth mechanism of fatigue cracks under the condition of lubrication [17]. They concluded that the cracks grew continuously due to the contact pressure, while its growth rate was accelerated by the effect of hydrostatic pressure.

Many finite element–based calculations of a rolling contact have focused on analysing the wheel-rail contact of trains [18–20]. Ringsberg developed a strategy for prediction of rolling contact fatigue (RCF) crack initiation [21]. He concluded that strain-life approach with elastic–plastic FE analysis makes a powerful combination in predicting the initiation of fatigue cracks. Kráčálek et al. used 2D finite element models to see whether the crack growth predictions from twin-disc tests could be transferred to full scale wheel/rail experiments. They concluded that the crack lengths must be scaled by a factor of approximately 30 to produce identical results [22]. Miyashita et al. studied the failure mode of spalling under rolling contact fatigue of a sintered alloy in a two-cylinder contact. Their experimental and finite element analysis results showed that the distance of the subsurface crack from the surface coincided with the depth of the maximum shear stress range [23]. Quite recently, there have been made progress on the simulation of gear contact fatigue. He et al. concluded according to their analysis that the contact fatigue of a gear is dominated by the elastic damage [24]. In addition, Liu et al. reported that the contact fatigue subsurface and the case-core transition area should both be
evaluated to estimate the risk of pitting and tooth flank fracture [25].

Many of the above-mentioned studies have concentrated on fatigue failure modes that are initiated either at or very near to the surface. In addition, rolling contact fatigue of ball and rolling element bearings have been widely investigated over the years as listed in Ref. [26]. However, quite often the studied components had been through-hardened. Nonetheless, the recent RCF studies have well considered also the surface region [27–29] and fatigue testing of case-hardened steel under bending loading have been researched [4,5]. However, there is not much data on subsurface-initiated cracking for surface-hardened components under rolling/sliding loading.

This paper presents an approach to investigating subsurface crack initiation in case-hardened components by using the twin-disc test device. In addition, an approach of analysing the test system with an elastoplastic finite element model of the arrangement of the discs is presented.

3. Test discs

The test discs were made from 18CrNiMo7-6 (EN 10084). After turning, they were case hardened to a surface hardness of 59–61 HRC. The outer diameter of the discs at the contact surface is 70 mm, and the width at the ground contact area is 10 mm. The discs on the fast-rotating shaft were ground to a radius of 100 mm in the axial direction (illustrated as R2 in Fig. 1). However, the discs on the slow-rotating shaft were ground to be flat, meaning infinite radius R1 in Fig. 1. As a result, an elliptical contact trace is generated to the contact.

The test discs were ground after they had been assembled on the shafts by means of a shrink-fit. This was done in order to minimise the risk of any eccentricity error between the contact surface and the bearings. The grinding grooves are set to be perpendicular to the sliding/rolling direction in order to mimic the flank of a real gear tooth. The average surface finish (Rz) had to be between 0.28 and 0.32 μm and the average distance of the highest peak from the lowest valley (Ra) had to be between 0.02 and 0.03 μm in earlier experiments [31] and therefore the effect on the stresses in the subsurface region is considered to be small.

2. Test equipment

The test device is highly automated allowing unmanned operation. The normal force FN is applied to the disc contact using a hydraulic cylinder, which is driven by a fast regel valve. The valve is controlled with a system that uses the measured load signal from a force transducer situated behind the cylinder to adjust the loading to the discs. The torque in the slow rotating shaft is also measured and this signal can be used to calculate the friction coefficient between the discs. The rotation speed and the direction of the shafts is controlled with frequency converters, which are connected to electric motors. These motors are coupled to the shafts via flexible couplings, which can tolerate any slight misalignment between the connected axles. The maximum speed of the motors is 6000 RPM.

A separate hydraulic unit controls the pressurised lubrication of the disc contact and the rolling bearings on the shafts. The lubricant is directed onto the disc contact from the inlet side. A microcontroller automatically controls the lubricant temperature, while the lubricant flow is controlled by means of frequency-converter-driven pump motor and manual valves. The lubricant temperature can be set anywhere between 25 and 120 °C and the flow at between 0.5 and 20 l/min. The test device is highly automated allowing unmanned operation.

The principle of the twin-disc test device.
insignificant.

3.1. Hardness and residual stresses

In order to find the effect that the case hardening has on the material, the hardness and the residual stresses were measured according to the depth. The results for the hardness (HV3) of an unground disc are shown in Fig. 3. This measurement indicates case hardening depth of about 1.1 mm for an unground disc. Since the grinding procedure typically removes approximately 0.25 mm from the surface of a disc, the resulting case hardening depth of the disc to be tested is about 0.85 mm and the surface hardness of such a disc is approximately 660 HV.

Residual stress measurements for two different unground test discs were taken with two separate methods. Firstly, the X-ray diffraction method was utilised to measure the surface of the disc in the middle of the contact area. This method is limited to a depth of roughly 5 μm. Therefore, electrochemical etching was used to remove a layer of material after each measurement in order to get readings from further below the surface. However, the material removal by etching may lead to relaxation of the residual stresses and changes in the shape of the measured surface, after each etching cycle, can have an influence on the results. These consequences are probably to affect more the deeper the measurements are done in the subsurface region.

Since the X-ray-based approach didn’t show any tension residual stresses, another measurement was performed throughout the thickness of the disc using the deep hole drilling (DHD) method [36]. Deficiency of this approach is that the error in the results tends to be greater near to the component surface than in its interior [37].

Due to uncertainties in the measurement results of both of the methods, the results values were combined to present the residual stress state through the heat effected region for further analysis. The measured residual stress curves in circumferential direction are shown in Fig. 4. Measurements were also taken in the axial direction of the disc, but the difference between these measurements curves and the ones taken in the circumferential direction was insignificant. The shear residual stresses were zero in both measurements.

4. Experimentation

The data acquisition system of this twin-disc test device collects information on the lubricant temperature and flow, the normal force, the rotation speed of the shafts and the torque on the slow rotating shaft. The load cycles were calculated from the rotation speed of the shafts and the testing time. The lubricant used in the tests is a mineral oil-based commercial gear oil specifically designed for heavy-duty industrial gears. The oil is equipped with EP-additive system and the specifications are shown in Table 1.

4.1. The test procedure

Every test began by rotating both of the shafts for about an hour at the same speed, 175.8 RPM, with lubrication circulation but without the normal force loading. This allows the temperatures of the shafts and the lubricant to be stabilized. The lubricant temperature was set at 60 °C, which is a typical value for the common temperature range of 40–100 °C found in industrial gears for power transmission applications [38]. The oil flow of 6 l/min was defined according to preliminary tests to ensure a sufficient supply of oil to the contact. The lubricant flow was directed onto the contact from the inlet side. The lubricant temperature and flow were kept constant during all the tests.

Throughout the tests the rotation speed for the fast-rotating shaft was 2039.3 RPM and for the slow-rotating shaft 1447.4 RPM. These speeds result in a sliding velocity of roughly 2.2 m/s, a mean rolling velocity of 6.4 m/s and a slide-to-roll ratio of 0.34. The surface sliding speed was set at a relatively low value as this is often the case midway up a bevel gear tooth; the point at which the highest contact load occurs during operation and where the rolling load dominates [39].

The normal load was ramped up for roughly 60 s at the beginning of every test. Once the desired loading was reached the load was kept at the same level until the end of the test. The load cycles during ramp-up were ignored when counting the overall number of load cycles for each test. The applied load was determined according to the Hertzian maximum contact pressure $P_{\text{max}}$ for elliptical contact between the discs:

$$P_{\text{max}} = \frac{3F_{\text{N}}}{2\pi ab}$$

where $a$ and $b$ are the semi-major and semi-minor axes of the contact ellipse. As soon as a particular number of load cycles had been accumulated, the test was stopped.

4.2. Preparation for crack inspection

After testing all the discs were removed from the shafts with a hydraulic press and checked with ultrasonic inspection equipment to find the potential locations of any subsurface cracks. Then the discs were cut into four pieces at crack-free locations. A typical resulting quarter piece is shown in Fig. 5.

The discs were first cut in the axial direction and then in the circumferential direction. This enables the cross section of the disc to be examined directly in the middle plane of the elliptic pressure trace of the contact area. Only two of the four quarter pieces of each disc were examined, since the circumferential cutting removes a lot of material from the other two pieces, in where cracks may possibly have been located. The remaining two quarter-pieces were ground and polished to

![Fig. 2. The load frame.](Image 263x113 to 438x242)

![Fig. 3. The hardness measurement results in a cross section of an unground disc.](Image 61x487 to 237x609)
the level of surface roughness at which the cracks are visible with an
optical microscope. The heat generation was minimised by allowing
adequate circulation of cooling-fluid during the preparation process.
Before inspection with an optical microscope, the surfaces were etched
with a 4% solution of nitric acid and ethyl alcohol (Nital).

5. Fatigue test results and discussion

During all the tests the lubricant inlet temperature was within the
range of 60 ± 1.5 °C. The surface condition of all the test discs was
measured before testing. These measurements were done in the middle
of the contact area at three separate points in the circumferential di-
rection of the disc. The angle between the three points was roughly
120°. The Ra values ranged from 0.27 to 0.34 and the
Rz from 1.16 to 2.29. The radius of the fast-rotating discs’
curvature varied between 101.0 and 102.9 mm. It can be assumed that the scatter in the values
has little e
ff
e
ct on the crack initiation and propagation beneath the
surface of the disc. The results of the fatigue tests are presented in
Table 2 and Fig. 6. Figs. 7–9 show typical subsurface cracks. The surface
of the disc in the figures is horizontal and above the crack.

According to the results in Table 2, a relatively high contact pres-
sure and a large number of load cycles (above 2 GPa and 16 millions of
cycles) are required to generate visible subsurface cracks with the
method described in this study. No propagation of the cracks to the
surface was observed in the test cases even after 72.5 millions of load
cycles. This indicates that the cracks grew underneath the hardened
layer where the material strength is signi
fi
cantly lower. It is also worth
of noting that the maximum of the max shear stress occurs much closer
to the surface (z in Table 2) in comparison to the depth where the cracks
were found. However, the approach of cutting and grinding may not

Table 1
Lubricant specification provided by the supplier.

<table>
<thead>
<tr>
<th></th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Kin. viscosity @40 °C [mm²/s]</td>
<td>150</td>
</tr>
<tr>
<td>Kin. viscosity @100 °C [mm²/s]</td>
<td>15</td>
</tr>
<tr>
<td>Density @15 °C [kg/m³]</td>
<td>897</td>
</tr>
<tr>
<td>VG class</td>
<td>150</td>
</tr>
<tr>
<td>Flash point (COC) [°C]</td>
<td>240</td>
</tr>
<tr>
<td>Pour point [°C]</td>
<td>−24</td>
</tr>
</tbody>
</table>

Table 2
Fatigue test results at different load levels showing the accumulated load cycles and whether the cracks were found from a fast-rotating or from a slow-rotating disc.

<table>
<thead>
<tr>
<th>Max Hertzian pressure [GPa]</th>
<th>Contact ellipse axes</th>
<th>Max shear stress [GPa]</th>
<th>Load cycles [millions]</th>
<th>Cracks found</th>
<th>Crack location depth [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>a [mm]</td>
<td>b [mm]</td>
<td>τmax [GPa]</td>
<td>x [mm]</td>
<td>Fast</td>
<td>Slow</td>
</tr>
<tr>
<td>A 1.8</td>
<td>1.60</td>
<td>0.51</td>
<td>0.58 0.36</td>
<td>7.0</td>
<td>5.0</td>
</tr>
<tr>
<td>B 1.8</td>
<td>1.60</td>
<td>0.51</td>
<td>0.58 0.36</td>
<td>7.0</td>
<td>5.0</td>
</tr>
<tr>
<td>C 2.0</td>
<td>1.80</td>
<td>0.58</td>
<td>0.65 0.41</td>
<td>1.1</td>
<td>3.6</td>
</tr>
<tr>
<td>D 2.2</td>
<td>2.00</td>
<td>0.64</td>
<td>0.73 0.45</td>
<td>7.0</td>
<td>5.0</td>
</tr>
<tr>
<td>E 2.2</td>
<td>2.00</td>
<td>0.64</td>
<td>0.73 0.45</td>
<td>72.5</td>
<td>51.5</td>
</tr>
<tr>
<td>F 2.2</td>
<td>2.00</td>
<td>0.64</td>
<td>0.73 0.45</td>
<td>23.0</td>
<td>16.3</td>
</tr>
<tr>
<td>G 2.5</td>
<td>2.30</td>
<td>0.74</td>
<td>0.84 0.52</td>
<td>21.1</td>
<td>15.0</td>
</tr>
</tbody>
</table>
reveal all the cracks and the locations of their initiation in the disc, as they are not directly connected to the visible surface. In addition, smaller cracks may have been removed during the preparation process.

6. Finite element analysis

The stresses and strains in the discs were calculated with a finite element model of the twin-disc assembly. The software used was Abaqus R2017x. The entire model and a detailed cross-section at the contact of the discs is illustrated in Fig. 10.

This model has four parts; the two shafts and the two discs that have been shrink-fitted on to them. The fitting overclosure is set at 0.015 mm. The dense part of the mesh of the discs at the contact in Fig. 10 is attached with tied-contact to the surrounding elements. The model is roughly 2.9 MDOF in size and is constructed with first-order hexahedron elements (C3D8R) using the reduced-integration method. The surface nodes on the shafts, where a bearing would be installed in reality are connected to a centre node using rigid beam elements. There are a total of four centre nodes each located at the rotation axis of the shafts in the middle of the width of a bearing. Boundary conditions and loads representing a real load situation are applied to these nodes. Both shafts are allowed to rotate around their longitudinal axis while the fast-rotating shaft is permitted to move in the radial direction in order to make contact with the slow-rotating shaft when a radial load is applied. In addition, the rotary degrees of freedom around the two other axes are free in order to realistically simulate the behaviour of the real bearings.

The contact between the discs is modelled as finite sliding with friction coefficient of 0.024, which is based on earlier measurements. The contact type of small sliding was used in the shrink-fit between the shafts and the discs. The friction coefficient in this case was 0.12.

The surface layer of the discs is modelled with small elements in order to take account the surface hardening. There are 0.1 mm-thick element layers in the depth direction up to a depth of 3 mm as shown in Fig. 10. In other directions, the size of the elements nearby the contact of the discs is 0.2 mm. The accuracy of the model was checked by comparing the analytically evaluated maximum Hertzian contact pressures to the results calculated with an elastic finite element model.

The finite element model gave a maximum of 4% lower pressure values between 1.5 and 2.5 GPa and was thus considered to be sufficiently accurate.

6.1. Material model

The basic parameters for the finite element analysis, the Young’s modulus, the Poisson’s ratio and the thermal expansion, were set at 210 GPa, 0.3 and $1.2 \times 10^{-5} \text{K}^{-1}$, respectively. Due to the surface hardening, the material properties at the surface differ significantly from the properties of the material in the core. Therefore, an elastoplastic material model that utilises the Ramberg-Osgood relationship between stress and strain [40] was used:

$$
\varepsilon = \frac{\sigma}{E} + \frac{\sigma}{E} \left( \frac{\sigma}{\sigma_0} \right)^{n-1}
$$

(2)

where $\varepsilon$ is strain, $\sigma$ is stress, $E$ is Young’s modulus, $\sigma_0$ is the yield strength of the material and $\alpha$ and $n$ are the material parameters. The $\alpha$ can be evaluated from the yield strength by assuming a 0.2% yield offset and the $n$ can be found by fitting with the tensile data. Therefore, the yield and the tensile strength values were calculated as a function of the material hardness through the hardened layer, as demonstrated by Pavlina et al. [41]. Both the resulting curves and the hardness (which was fitted to the measured data by taking into account the material removed during the grinding process) are illustrated in Fig. 11. Eq. (2) utilised this data to define the stress-strain curves for each element layer in the material affected by the hardening process and the bulk
6.2. Residual stresses

Case hardening increases the volume of the material near the surface and compressive residual stresses occur in this region. In the finite element analysis (FEA) this was modelled by applying a temperature load to the element layers, which are affected by the case-hardening. Due to thermal expansion, compressive residual stresses were generated near the surface and tension residual stresses beneath the hardened region.

The residual stress measurements were conducted on an unground test disc. The grinding before the tests removes roughly 0.25 mm of the disc surface, which was taken into account in the generation of the residual stress field in the calculation model. However, the residual stresses caused by the grinding are considered to be insignificant and are therefore not included in the model. Fig. 12 shows a comparison between the measured residual stress curves and the calculated approximation of the residual stress in the finite element model. The measured curves illustrate the residual stresses in the circumferential direction of the disc. Since the shear residual stress is zero, the principal stresses in the FE-model can be utilised to extract the compressive and tension residual stresses in the model. The finite element-based stress curve in Fig. 12 is a combination of the minimum principal stress on the compressive side (negative values) and the maximum principal stress on the tension side (positive values) through the region, which was affected by the hardening. The presented stress values were taken from nodes, meaning that they were transferred from element integration points to related nodes with Abaqus (UNIQUE NODAL).

7. FEA results and discussion

The finite element analysis was conducted as a static calculation. Firstly, the shrink-fit was calculated and then the residual stresses in the discs were solved. Secondly, the normal load was applied and the sliding effect was introduced to the discs as forced rotation of the shafts. The calculation step of the rotation was conducted in several smaller increments in order to simulate sliding/rolling load in the contact.

Four load levels were analysed. These radial loads generated maximum Hertzian contact pressures of 1.8, 2.0, 2.2 and 2.5 GPa between the discs. The calculated element results were extracted from element integration points for the related nodes with Abaqus (UNIQUE NODAL).

7.1. Stress results

The stresses were examined in a cross sectional plane (in the circumferential direction), which coincides with the location of the highest contact pressure point in the middle of the elliptic surface contact on the disc. Typical stress contours in this cross section are shown in Fig. 13. The first plot illustrates the compressive stress near the disc contact. The second contour shows the stress concentration due to shearing at the sides of the centre point of the disc contact.

The resulting curves from the cross-section cut are presented in Fig. 14. This figure first shows the stress history during one load cycle at a depth of 0.6 mm with a radial load resulting in 2.5 GPa maximum Hertzian contact pressure. The highest amplitude values are calculated for the stress components S11 and S12 for both the fast rotating disc...
(FD) and the slow-rotating one (SD). The figure also shows the maximum and minimum values for these stress components captured during one load cycle as a function of depth and load. These curves indicate that the maximum range for the S11 stress component can be found at the surface and the stress levels clearly decrease as a function of depth and load. The maximum range values for the evaluated load levels for S11 are roughly 1.75 GPa, 1.97 GPa, 2.18 GPa and 2.52 GPa and for S12 they are 0.81 GPa, 0.90 GPa, 1.04 GPa and 1.19 GPa.

The decrease in the stress range according to depth is slightly lower...
for S12 component than it is for S11. For the load case of 2.5 GPa the decrease between the depths of 0.5 mm and 1.0 mm is about 33% for the S11 component (2.04 GPa to 1.36 GPa) and ~27% for the S12 (1.16 GPa to 0.85 GPa).

7.2. Plastic strain results

The equivalent plastic strain (PEEQ) was derived for the calculated cases. In both discs, plastic deformation was only found for the load case that produced 2.5 GPa contact pressure on the surfaces. Fig. 15 shows the maximum of the PEEQ variable at different depths and the fitted hardness curve. A concentration can be seen at a depth of roughly 0.4–0.8 mm for both discs. This is the location, where the maximum range of the stress component S12 occurs. This indicates that when the load is high enough, plastic deformation is produced at the depth, where the shear stress range is highest.

Another plastic deformation peak for both of the discs can be seen at a depth of roughly 1.1–1.7 mm. This is the depth where the material strength is significantly lower than at the surface. In addition, the residual stresses are either close to zero or on the tension side, which can be considered as detrimental to the strength of the component. The experimental results in Table 2 also indicate that this depth is a potential area for crack initiation even at lower load levels than the 2.5 GPa in maximum Hertzian pressure. It is also worth noting that according to Fig. 14 the stresses are clearly lower in this region than at the disc surface. Additionally, the similarity of the plastic strain curves in Fig. 15 suggests that difference in the surface curvatures of the interacting discs has insignificant effect on the subsurface plastic deformation.

8. Conclusions

In this study, a twin-disc test device was used to investigate subsurface fatigue in a rolling/sliding contact. The main conclusions that can be drawn are:

- The twin-disc test device is well-suited for studying subsurface crack initiation in high loaded areas and beneath the hardened layer in a rolling/sliding contact. However, the proposed approach of destructive crack inspection may blot out smaller cracks and their initiation locations.
- An elastoplastic finite element model of the discs that takes into account the case-hardening effects, such as the increase of the material hardness and the residual stresses, can be used to predict crack initiation:
  - The crack locations in the experimental part of this study were found to be at the depths, which were also considered to be critical according to the finite element analysis.
  - The difference in the curvatures of the surfaces in contact seems to have an insignificant effect on the subsurface plastic deformation.

It appears that relatively high loading and a large number of load cycles is required to generate cracks that largely propagated in the subsurface region during testing. An uncertainty in the results of the detection process, especially concerning smaller cracks, originates from the fact that the used inspection method may have removed some of them from the specimen before examination. However, detailed research of crack growth behaviour in the material is left for future studies.

It was also shown that the presented finite element-based approach was able to find weak locations in the subsurface region. Nonetheless, the criteria for judging the fatigue safety, the effect of material work hardening and the relaxation of the residual stresses under cyclic loading were not studied in this paper. Therefore, further research is recommended.

Acknowledgements

This study was carried out under the auspices of the ArTeCo (Arctic Thruster Ecosystem) project, which is part of the Maritime Technologies II Research Programme (MARTEC II) under the European Commission ERA-Net scheme. The authors wish to gratefully acknowledge the financial support of Business Finland (formerly Tekes, decision number 40303/14), and other partners in the project. The funders have had no influence on the content of this paper. The authors would like to express their gratitude to A4 Gears for the case hardening of the test discs, Taru Karhula (Tampere University of Technology) for the X-ray based residual stress measurements and Veget Ltd. for the residual stress measurements using the Deep-Hole Drilling technique.

Appendix A. Supplementary material

Supplementary data associated with this article can be found, in the online version, at https://doi.org/10.1016/j.jfatiug.2018.08.014.

References

An investigation into subsurface fatigue in a rolling/sliding contact

M Savolainen, A Lehtovaara

Submitted manuscript.
An investigation into subsurface fatigue in a rolling/sliding contact

Matti Savolainen, Arto Lehtovaara
Group of Tribology and Machine Elements, Laboratory of Materials Science, Tampere University of Technology, Tampere, Finland

Abstract

This paper presents aspects of subsurface fatigue analysis and the testing of surface hardened components under rolling/sliding load. Firstly, different approaches to determine fatigue damage are compared by utilizing an elastoplastic finite element model of a twin-disc device as an example. Secondly, models of the twin-disc test assembly are solved by altering case-hardening depth and contact load due to curvature modification. The results show that a stress-strain-based damage model is sensitive to changes in the discs, although small deviations in the contact surface curvature or case-hardening depth do not have a significant effect on the damage in the subsurface region. In addition, the effect of tension residual stresses in the analysis was examined showing a slight increase in damage.

Keywords
Fatigue, subsurface, twin-disc, case-hardening, FEA
1. Introduction

Gear wheels are common case-hardened components subjected to rolling/sliding loading under high contact pressure. The strength of the material in case-hardened gear wheels has been often defined by performing push-pull or bending tests for a specimen made out of the interested material with a particular surface treatment [1, 2]. The achieved results provide support for the dimensioning of real gear wheels to prevent tooth root bending fatigue. Another popular method is to perform a bending tests on a single gear tooth. In this case, the effects of the tooth geometry, the surface treatment and the surface roughness are directly reflected in the results [3-5]. Additionally, the strength of the tooth flank has long been studied by conducting tests with the FZG device [6] by noting the formation of micropitting or pitting marks on the tooth surface. Several papers have focused on examining the failure formation in harsh thermal conditions, with different types and viscosities of lubricant as well as different surface conditions of the gear tooth [7-10].

In the above-mentioned gear failure modes the possible cracks initiate often from, or close to, the surface eventually leading to fatigue failure. However, the failure may also start deeper in the material leading to fracture of the gear tooth. In addition, recently a gear failure mode of Tooth Interior Fatigue Fracture (TIFF) has been suggested [11]. In this mode, the fatigue cracks initiate in the interior of a gear tooth possibly beneath the hardened layer of a surface-hardened gear tooth. The material strength in this region is significantly lower in comparison to the surface and the compressive residual stresses caused by the hardening are not active. Instead, the material is influenced by the tension residual stresses, which can promote crack initiation and growth.

An elliptical pressure distribution can be generated between the discs in the twin-disc test device with specifically defined discs [12, 13]. It is broadly accepted that this pressure distribution resembles conditions between gear teeth, although using the achieved test results in dimensioning real gears requires close attention to a number of factors as stated in Ref. [14]. Several studies, using the twin-disc test device, have focused on examining the behavior of friction, the formation of pitting, the prediction of wear and the effects of the surface condition and the loading have on scuffing initiation [15–21].
A recent paper by the authors [22] proposed a technique to studying cracking in the subsurface region with the twin-disc test device. This paper also introduced an elastoplastic finite element model of the test configuration, whose material model
accounted for effect of the case hardening by including hardness change and residual stresses through the heat affected region. It was found that this FE-model was able to reveal critical locations beneath the hardened layer.

One research question in dimensioning of case-hardened gear wheels is, to what extent do tolerances in manufacturing, which lead to small changes in the case-hardening depth and curvature of the surfaces in contact, affect the subsurface fatigue of a component. In addition, the residual stresses may decrease during operation or there may be uncertainties in the description of the residual stress distribution in the component at the design phase. This may also have an effect on the durability of the component.

This paper presents an investigation of the effects of the above-mentioned manufacturing issues related to subsurface fatigue. The FE-models of the twin-disc test assembly are utilized as an example in the analysis. The paper begins by comparing different approaches to the calculation of fatigue damage. Then, a stress-strain-based damage criterion is used to distinguish differences between the discs with varying case-hardening depths and axial radius of the surface curvature. Finally, the influence of the occurrence of the residual stresses is observed.

2. Twin-disc test arrangement

The principle of the twin-disc system is illustrated in Figure 1. A pressure trace between the discs can be generated by pushing the discs together with a normal force $F_N$. The twin-disc test arrangement used in this study has been presented in Refs. [15, 20, 21]. The analysed discs were manufactured out of 18CrNiMo7-6 (EN 10084) and case-hardened to 59-61 HRC at the surface and to a depth of approximately 1.1 mm (before grinding). The outer diameter of the discs is 70 mm ($R_1$ and $R_2$ in Fig. 1) and the nominal contact area width is 10 mm.

![Fig. 1. Principle of the twin-disc test device.](image-url)

The contact surfaces have to be specifically ground to produce an elliptical trace between the discs in contact, which mimics the condition between real gear teeth in a
bevel gear set. This grinding process is described in Ref. [23]. The process results in a certain radius of curvature in the axial direction on one of the discs, while the other is ground flat or with an infinite radius ($r_1$ and $r_2$ in Fig. 1). The discs are mounted on shafts before grinding using a shrink-fit procedure in order to minimize any eccentricity errors due to manufacturing.

3. Fatigue of discs

Generally, fatigue life can be divided into three stages; crack initiation, crack growth and the propagation or instability stage. Typically, cracks nucleate due to a dislocation movement generating slip bands, which have slip steps in the order of 0.1 μm. These slip steps, which are caused by reversed slip due to load cycles, may produce intrusions and extrusions in the material. The intrusions are regarded as a source of crack initiation and are often in parallel with the maximum shear plane, especially in ductile-behaving materials. In brittle-behaving materials, the slip may occur at discontinuities, such as at the matrix-inclusions interface, although they can also nucleate along slip systems due to shearing. Crack growth, in turn, can be divided into two stages. Often the early phase of the growth occurs along the maximum shear plane, while in the second phase the maximum principal normal stress plane dominates [24, 25].

Rolling contact fatigue (RCF) is considered as a fatigue failure mode occurring in machine components, which are suffering from localized high Hertzian surface pressure under rolling loading. This phenomenon starts below the surface where a crack propagates producing a pit or a spall on the surface of the component. According to Miyashita et al, the critical location for such cracks to originate is in the region of maximum shear stress beneath the surface [26]. Among the machine components, which typically suffer from this phenomenon are roller and ball bearings, rail-wheel contacts and gears.

In rolling contact fatigue, the stress history beneath the subsurface region is usually non-proportional (out-of-phase) meaning the stress components do not rise and fall in time with each other. In addition, the state of stress is multiaxial and governed by Hertzian contact theory. It is also commonly accepted that, rather than being randomly orientated, most of the cracks nucleate and grow in certain planes, controlled by the material characteristics and the state of loading. This is indicated in several studies [27-29], which support and substantiate the use of the critical plane approach in investigations into multiaxial fatigue. The direction of the crack growth is believed to follow either the maximum shear stress or strain plane or the maximum principal stress or strain failure plane depending on the characteristics of the material and loading. Therefore, several critical plane damage criteria are based on observing the maximum shear or maximum principal planes [30-37]. These can be categorised to stress- or strain-based, stress-strain-based or energy-based models.

It is often believed that, in general, stress-based approaches may work effectively in the high cycle fatigue (HCF) regime where plastic deformation is usually insignificant. However, in the low cycle fatigue (LCF) regime or under loads containing overload cycles for instance, significant plastic deformations may occur and in these cases strain-
based criteria may suit the situation better. In addition, even if shearing dominates the crack growth, tensile normal stress in the shear plane acts as decreasing crack closure and hence results in reduced fatigue life. This is supported by the data in Ref. [38], which showed that crack growth rate was increased when maximum normal stress increased leading to a shorter fatigue life as stated in Ref. [39].

3.1. Calculation of damage

A component’s fatigue life is often defined by comparing the damage due to load cycles to the material strength. In this study, the focus is on investigating fatigue damage formation in the discs through the heat-affected region. It is assumed that there is no significant plastic deformation during the loading cycles. It is also well known that the shear stresses are significant in the subsurface region near the contact. Hence, three damage calculation methods combining shearing and normal stress were utilized in the following analysis. Findley [30] originally proposed the first approach:

\[ D_F = \left( \frac{\Delta \tau_F}{2} + k_F \sigma_F \right)_{max} \]  

(1)

Where \( \Delta \tau_F \) is the shear stress range, \( \sigma_F \) is the maximum of the normal stress in the shear stress plane at any moment of time during the load cycle and \( k_F \) is a factor related to the material’s sensitivity to normal stress.

Fatemi and Socie [37] have proposed another damage calculation model, also used in the analysis:

\[ D_{FS} = \frac{\Delta \gamma_{max}}{2} \left( 1 + k_{FS} \frac{\sigma_{n,max}}{\sigma_0} \right) \]  

(2)

Where \( \Delta \gamma_{max}/2 \) is the maximum amplitude of shear strain in any plane and \( \sigma_{n,max} \) is the maximum normal stress acting on the maximum shear strain plane over the load cycle. The \( \sigma_0 \) is the yield strength and \( k_{FS} \) is a material constant coupling the normal stress and the shear strain parts.

Quite recently, Ince et al. proposed a fatigue damage parameter based on generalized strain energy (GSE) [40]. This damage parameter \( W_{GSE} \) includes the normal and shear strain energy terms in the plane where the maximum generalized strain energy is experienced:

\[ W_{GSE} = \left( \tau_{max} \frac{\Delta \gamma^e}{2} + \Delta \tau \frac{\Delta \gamma^p}{2} + \sigma_{n,max} \frac{\Delta \varepsilon^e}{2} + \Delta \sigma_n \frac{\Delta \varepsilon^p}{2} \right)_{max} \]  

(3)

Where \( \tau_{max} \) is the maximum shear stress, \( \Delta \gamma^e/2 \) is the elastic shear strain amplitude, \( \Delta \tau/2 \) is the amplitude of the shear stress and \( \Delta \gamma^p/2 \) is the plastic shear strain amplitude. The \( \sigma_{n,max} \) is the maximum normal stress acting on the maximum shear strain plane over the load cycle, \( \Delta \varepsilon^e/2 \) is the elastic normal strain amplitude, \( \Delta \sigma/2 \) is the normal stress amplitude and \( \Delta \varepsilon^p/2 \) is the plastic shear strain amplitude.
In order to find the critical plane for the maximum damage, the stress tensors needed to be rotated at each examined location. The rotation was done around two axes \((x\) and \(y\) or 1 and 2) from 0 to 180 degrees in increments of 5 degrees for both axes. A smaller increment of 2 degrees that required longer calculation time was tested, but there was no significant change in the results.

3.2. Evaluation of shear stress/strain amplitude and normal stress

The shear stress amplitude can be found utilizing several methods as presented in Ref. [41]. Maybe the most commonly used approach is the Minimum Circumscribed Circle (MCC) method. In this method the smallest possible circle is defined around a path, which is drawn by the shear stress (strain) components \(\tau_{12}\) and \(\tau_{13}\) \((\gamma_{12}\) and \(\gamma_{13}\)) during the load cycle in the critical plane in the \(\tau_{12}/\tau_{13}\) system. The radius of the circumference defines the shear stress amplitude \(\Delta \tau_{ij}/2\) \((\Delta \gamma_{ij}/2)\). The concept of the minimum circle takes into account the non-proportionality of the loading and was originally proposed by Dang Van [35] and later by Papadopoulos [36].

In this study, the normal stress value is taken on the shear plane at any moment of time during the load cycle. Due to the absence of test data that could have been utilized to define the \(k_F\) and the \(k_{FS}\) in Equations 1 and 2, a series of calculations was conducted in which the values of these factors were varied.

4. Finite element analysis

The twin-disc test assembly was modelled using Abaqus version R2017x. The finite element model along with the cross-section of the region in contact is presented in Fig. 2.

![Finite element model](image)

Fig. 2. Finite element model of the test configuration on the left; a cut section of the contact region on the right.

The size of the entire FE-model is about 2.9 MDOF and was built with first-order elements (C3D8R). The model consists of two test discs, which have been assembled on two shafts using shrink-fit with an overclosure of 0.015 mm. The detailed part of the mesh, illustrated in Fig. 2, was connected to the discs using tied-contact. There are four nodes, each placed on the longitudinal axis of the shafts, at the center of the bearing
width. These nodes are connected to nodes on the shaft surface at the bearing locations with rigid beam elements. They are used as locations for the loads and the boundary conditions. Sideways movement of one of the shafts is allowed in order to engage contact with the other shaft, while both of the shafts can rotate around the rotational axis. The rotational degrees of freedom are free, thus mimicking realistic bearing behaviour.

A finite sliding type of contact was used between the discs with a coefficient of friction of 0.024. This value was derived from earlier measurement results [20, 21]. A small sliding type of contact was applied in the shrink-fit joining discs to the shafts. The friction coefficient there was set at 0.12. There are element layers near the surface up to a depth of 3 mm with a thickness of 0.1 mm as shown in Fig. 2. The size of the elements is roughly 0.2 mm in the other directions.

The quality of the element mesh was examined by comparing the maximum Hertzian contact pressure data of an analytical evaluation to the finite element-based results. The differences between the pressure values of the elastic finite element model and the analytically found ones were a maximum of 4% between 1.5 to 2.5 GPa while the FE-based values were lower. Thus, the model was considered to be sufficiently accurate.

4.1. Surface hardening effects

The material characteristics near the surface are substantially different from the properties of the core material because of the hardening. This means that they need to be defined individually at different depths. Firstly, the elastic modulus, the Poisson’s ratio and the thermal expansion were set at 210 GPa, 0.3 and $1.2 \cdot 10^5$ K$^{-1}$ respectively at any depth of the material. Secondly, the elastoplastic behaviour of the material was evaluated for each element layer through the hardened region as a function of hardness. The relationship between stress and strain was modelled according to Ramberg-Osgood [42]:

$$
\varepsilon = \frac{\sigma}{E} + \alpha \left(\frac{\sigma}{\sigma_0}\right)^{n-1} 
$$

(4)

where $\varepsilon$ is strain, $\sigma$ is stress, $E$ is elasticity and $\sigma_0$ is the yield strength. The $\alpha$ and $n$ are specific material parameters. The $n$ can be evaluated by fitting with the tensile data and the $\alpha$ can be found from the yield strength by setting the offset for yielding to be 0.2%. The values for the yield and the tensile strength were estimated as proposed by Pavlina et al [43].

Another consequence of the case hardening is that compressive residual stresses are produced at the surface and beneath the hardened region balancing tension residual stresses occur. This was formulated in the finite element analysis (FEA) by introducing a temperature load to the layers of elements, which are affected by the hardening process. The residual stresses occurred in the model due to thermal expansion of these element layers.
4.2. Analyzed cases

The basis and the reference for the calculations is the analysis presented in Ref. [22], where the radius of the curved disc was set at 100 mm in the axial direction of the discs and the case-hardening depth (CHD) was roughly 0.85 mm. This study aimed to examine the effect of tolerances in the manufacturing process. Firstly, the curvature of the fast rotating disc was modified. Secondly, the case hardening depth of one of the discs in contact was varied. Thirdly, the influence of the residual stresses was studied by solving a theoretical case, where the residual stresses are not present in the discs. The aim of this was to shed light on how uncertainties in the definition of the residual stresses may affect the results. The normal load for all the solved cases was set at 5.9 kN, which corresponds to 2.53 GPa in maximum Hertzian pressure between the discs. The calculated variants are shown in Table 1 and the variation in the radius and the CHD were chosen so that they would roughly cover a typical range of change in real gears due to tolerances.

Table 1. Calculated cases with varied parameters. “FD” stands for fast-rotating disc and “SD” slow-rotating disc.

<table>
<thead>
<tr>
<th>#</th>
<th>Radius of curvature (FD) [mm]</th>
<th>Radius of curvature (SD) [-]</th>
<th>CHD (FD) [mm]</th>
<th>CHD (SD) [mm]</th>
<th>Residual stresses</th>
</tr>
</thead>
<tbody>
<tr>
<td>A1</td>
<td>100</td>
<td>Flat</td>
<td>0.85</td>
<td>0.85</td>
<td>Yes</td>
</tr>
<tr>
<td>B1</td>
<td>100</td>
<td>Flat</td>
<td>0.85</td>
<td>0.85</td>
<td>No</td>
</tr>
<tr>
<td>C1</td>
<td>90</td>
<td>Flat</td>
<td>0.85</td>
<td>0.85</td>
<td>Yes</td>
</tr>
<tr>
<td>C2</td>
<td>110</td>
<td>Flat</td>
<td>0.85</td>
<td>0.85</td>
<td>Yes</td>
</tr>
<tr>
<td>D1</td>
<td>100</td>
<td>Flat</td>
<td>0.95</td>
<td>0.75</td>
<td>Yes</td>
</tr>
</tbody>
</table>

The measured hardness profile curve (CHD 0.85) from Ref. [22] was modified in order to simulate case hardening depths of 0.75 mm and 0.95 mm. Based on the hardness values, corresponding yield and tensile strength curves were produced according to Pavlina et al [43]. The resulting data is illustrated in Fig. 3 and was used in defining elastoplastic material behaviour with Eq. 4 for the calculation models.
Fig. 3. Hardness and yield and tensile strength curves for calculated cases. Additionally, the occurring residual stress data were generated for a CHD of 0.75 and a CHD of 0.95 using the measured curve (CHD 0.85) from Ref. [22] as a starting point. Figure 4. Illustrates the resulting residual stress curves in the circumferential direction of the disc according to the principal stresses in the FE-model.

Fig. 4. Residual stress curves in the finite element models for different values of the case hardening depth.

5. FEA results and discussion

The calculation of the model contains four steps, which were solved as static cases. First, the shrink-fit between the shafts and the discs was calculated, which was then followed by evaluating the residual stresses near the surface. The normal force of 5.9 kN was applied to the system in the third step. The fourth step consists of rotation of the shafts in small increments. This approach simulated sliding/rolling condition between the discs. The total rotation was roughly 2 degrees for the shaft with the flat disc and about 2.9 degrees for the shaft with the curved disc. The magnitude of the rotation of the shafts and the normal force were the same for each calculation case. The element results were transformed from the integration points of the elements to adjacent nodes within Abaqus (UNIQUE NODAL) for further analysis. The logarithmic strain data was used instead of
engineering strain due to the nature of the parameter giving the correct value of the final strain when deformation takes place in a series of increments [44].

Typical stress and logarithmic strain distributions in the cross section are shown in Fig. 5. The results are from a cross-section, which coincide with the highest contact pressure point of the elliptic pressure trace at the surface. The stresses/strains in the x-axis direction are on the left while the right plots illustrate the shear stress/strain distribution in the xy-plane. No plastic strains were found in the model.

![Stress and strain distributions](image)

Fig. 5. Stress components S11 and S12 and strain component LE11, LE12 at the cross section of the contact for case A1.

The stress and strain data at the cross-section cut for case A1 are shown in Figure 6. The uppermost plots illustrate the stress/strain history at a depth of 0.4 mm during one load cycle for the fast-rotating (FD) disc. The largest values can be observed for the components S11 and S12 as well as LE11 and LE12 during a load cycle. According to Fig. 5 and Ref. [22] the curves for the slow-rotating disc (SD) do not differ significantly from the curves of the fast-rotating disc and are therefore not studied separately in this investigation.
Fig. 6. The first two graphs show the stress history over a load cycle for case A1 at the depth of 0.4 mm. The lower graphs presents the maximum and minimum values for components S12, LE11 and S11 as a function of depth during a load cycle. “FD” refers to the fast-rotating disc.

Fig. 6 also illustrates the minimum and the maximum values as a function of depth for components S11, LE11, S12 and LE12 captured during one load cycle. According to this data, the maximum range for the S12 stress and the LE12 strain components can be found on a plateau approximately between the depths of 0.3 mm and 0.5 mm. The maximum range for the S11 stress component is at the surface and a nearly linear decrease can be seen in the stress levels the deeper one goes into the material. The maximum strain range for the component LE11 can be found at a depth of about 0.3 mm.
In contrast to the curves for the component S11, the effect of the residual stresses is visible in this coordinate direction (x). This is reflected as a substantial change in the maximum values for LE11 between 1.0 mm to 1.5 mm, much like the residual stress curve in Fig. 4.

5.1. Damage

The shear stress amplitude was evaluated with the MCC method for stress and strain data over a load cycle. The FE-results of case A1 were utilized in the comparison of the methods according to Eq. 1, 2 and 3. Since fatigue data to calculate values for the factors $k_F$ and $k_{FS}$ is not available, a series of analyses was conducted by varying these parameters. A range, which covers typical values for steels [45-47], was tested: 0.1, 0.3 and 0.8 for the $k_F$ and 0.5, 1.0 and 2.0 for the $k_{FS}$. The resulting damage curves with all the methods are presented in Fig. 7.

![Fig. 7](image)

Fig. 7. The damage $D_F$ calculated according to Eq. 1 and by varying the factor $k_F$ as a function of depth is on the left. The graph in the middle illustrates the damage $D_{FS}$ according to Eq. 2 and variation of $k_{FS}$. On the right is the damage parameter $W_{GSE}$ results according to Eq. 3.

According to the resulting curves in Fig. 7, all three methods showed that the maximum damage was at the same depth where the maximum shear stress or strain ranges occur in Fig. 6. The variation of the factors $k_F$ and $k_{FS}$, which are used to emphasize the effect of normal stress, do not have a significant influence on the results. This is because the tension normal stress values, used by the tested methods, are low in the examined region. However, a small but clearly visible deviation between the result curves can be seen in the region from roughly 1.3 mm to 1.8 mm, especially for the $D_{FS}$ curves. At this depth, the shear stresses or strains are lower in comparison to the normal stress than they are near the surface and the residual stress values are on the tension side as shown in Fig. 4.

5.2. Effect of residual stresses

One uncertainty in analysing the fatigue life of case hardened components may originate from the description of the residual stress distribution in the material. In addition, relaxation of these stresses may occur due to the use of the component.
Therefore, a theoretical comparison of two calculation cases was conducted: in the first one (A1), the residual stresses are included in the model and in the second (B1), the stresses are left out. The resulting strain and stress curves of maximum and minimum values for the fast-rotating disc during the load cycle, along with the damage, are shown in Fig. 8.

All the damage calculation methods in Fig. 8 show a difference between the damage curves for cases A1 and B1. For the shear strain based approach \((D_{FS}, W_{GSE})\) this seems to be caused by a slight deviation between the strain curves of LE12. In addition, a step-up is visible for the damage curve \(D_{FS}\) for case A1 starting at a depth of about 1.3 mm, which originates from the tension residual stresses at this depth as shown in Fig. 4. This can also be realized in the stress curves for the component S11 as slightly higher tension stress values for case A1, which is probably also the reason for the difference in the damage curves according to Eq. 1 \((D_F)\) in that subsurface region.

All of the above indicates that the residual stresses in the subsurface region slightly increase the magnitude of the damage. In addition, the method with Eq. 2 seems to be lightly more sensitive to changes in the FE-model, and therefore it was used in further analyses with the \(k_{FS}\) set at 1.0.
5.3. Effect of surface curvature

A change in the disc’s curvature in the axial direction has an obvious effect on the maximum contact pressure and it also affects the subsurface stresses and strains. The maximum Hertzian contact pressure $p_{\text{max}}$ between the discs for elliptical contact can be calculated as follows:

$$p_{\text{max}} = \frac{3F_N}{2\pi ab}$$  \hspace{1cm} (5)

where $a$ and $b$ are the semi-major and semi-minor axes of the contact ellipse. The resulting pressure values according to Eq. 5 for cases A1, C1 and C2 are 2.53 GPa, 2.60 GPa and 2.47 GPa respectively (FEA based values are 2.53, 2.59 and 2.47 GPa). The values show that a decrease of roughly 11% in the radius down to 90 mm leads to an increase of about 2.8% in the maximum Hertz pressure. Correspondingly, an increase of 10% in the radius up to 110 mm results in a decrease of 5% in the maximum Hertzian contact pressure.

The variation of the maximum and minimum LE12 strain and S11 stress values along with the resulting damage are illustrated in Fig. 9 for the fast-rotating disc. The decrease of the radius to 90 mm leads to a roughly 2.5% increase in the maximum strain amplitude in comparison to case A1, where the radius was 100 mm (A1: 6.497E-3 mm, C1: 6.657E-3 mm). Consequently, increasing the radius to 110 mm results in a decrease of the maximum strain amplitude by a magnitude of roughly 2.8% (A1: 6.497E-3 mm, C2: 6.315E-3 mm). The increase of the stress component S11 is roughly 2.8% at the surface for case C1 in comparison to A1 and the decrease is about 2.3% for case C2 (2.24 MPa for C1 and 2.13 MPa for C2 compared to 2.18 MPa for A1).

A similar trend can also be seen in the damage values for the analysed cases, especially at a depth of from 0.3 to 0.4 mm, where the strain component LE12 reaches its maximum value. However, if one goes slightly deeper beneath the hardened layer (starting from about 1.0 mm) the difference between cases A1, C1 and C2 is insignificant. This indicates that a relatively large variation in the axial curvature of the disc’s contact surface can be allowed without having any substantial effect on the strains or stresses at a depth of about 1.0 mm and deeper in the material.
5.4. Effect of the depth of the case-hardening

After case-hardening, the contact surface needs to be ground in order to generate sufficient surface curvature and roughness. Usually, the material hardness at the surface does not change much when the grinding depth varies between ±0.1 mm. However, this variation does lead to a change in the case hardening depth. Fig. 10 illustrates the effect of a small change in the grinding depth on the maximum and minimum values of the strain and stress components LE12 and S11, as well as the damage during a load cycle.

The results in Fig. 10 show that minor variations in the CHD do not have a clear effect on the strain, stress or damage values. Any small differences between the curves are probably due to varying residual stress fields between the cases. The effect of change in the case-hardening depth is probably more visible in the definition of the material strength through the heat-affected region.
6. Conclusions

In this study, different considerations related to subsurface fatigue of case-hardened components under rolling/sliding loading and high contact pressure were investigated. The focus was on any uncertainties which may originate from manufacturing tolerances. In addition, fatigue analysis of case-hardened components was discussed and the effect of the residual stresses on the formation of fatigue damage was examined using the twin-disc test arrangement as an example. The main conclusions can be drawn as follows:

- Shear stress- and shear strain-based fatigue criteria show similar distributions of damage in the subsurface region, and the maximum damage was found at the depth where the shear stress or strain amplitudes were also at their highest. However, the shear strain-based approach (Fateme-Socie) seems to be more sensitive to the occurrence of the residual stresses.

- Two of the examined criteria (Findley and Fateme-Socie) showed a relatively small increase in the damage beneath the hardened layer due to the increased consideration given to normal stress through the factors $k_F$ and $k_F$. This seems to be mainly caused by the tension residual stresses in this region.

- A small variation in the radius of curvature of the contact surface has an effect on the contact pressure and the magnitude of the maximum shear stress or strain directly beneath the surface. However, further under the hardened layer, starting from a depth of about 1.0 mm, the effect disappears.

- A small variation in the CHD, possibly originating from a deviation in the grinding depth, has no significant effect on stress, strain or damage distribution in the subsurface region.

According to the all three of the utilised calculation methods, it appears that small manufacturing-induced deviations in the disc only have a small or insignificant effect on the formation of fatigue damage in the subsurface region. It was also found that the values of the tension normal stress were relatively low in the discs only having a minor effect on the extent of the damage. However, in real gears the bending of the tooth may slightly change the stress distribution in the subsurface region. The calculation of the fatigue life of the discs and its relation to real case-hardened components from the viewpoint of subsurface fatigue is left for future research.

7. Acknowledgements

The authors wish to gratefully acknowledge the financial support of TTY Foundation. The funder has had no influence on the content of this paper.
References


[18] Snidle RW, Dhulipalla AK, Evans HP. Scuffing performance of a hard coating under EHL conditions at sliding speeds up to 16 m/s and contact pressures up to 2.0 GPa. *J Tribol* 2008: 130.


[46] Everitt CM, Alfredsson B. Contact fatigue initiation and tensile surface stresses at a point asperity which passes an elastohydrodynamic contact. *Trib Int* 2018; 123: 234-255.
