Grease Lubrication Mechanisms in Bearing Seals

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I have no special talent. I am only passionately curious.
Albert Einstein

Preface

This thesis presents the results of my 4 years Ph.D. studies at the Division of Machine Elements at Luleå University of Technology in Sweden. I am very grateful that I got the opportunity to study the fundamentals of “grease lubrication mechanisms in bearing seals” and to develop models that will help to increase our knowledge to improve sealing performance in industry.

The work has mainly been conducted at SKF Engineering & Research Centre in the Netherlands and I would like to acknowledge my gratitude to SKF for their financial support and for giving me the opportunity to work and study in a very challenging and knowledgeable industrial environment.

I want to thank my supervisor Prof. Braham Prakash for his good guidance and encouragement of my work. I also send my gratitude to all the other people in the Division of Machine Elements and the Division of Fluid and Experimental Mechanics for working together and giving me some times in Luleå. I will never forget the beauty of the Norrbotten area and the joyful outdoor activities.

I would especially like to thank my supervisor Prof. Piet Lugt from SKF for his enthusiastic guidance and for sharing his passion for grease lubrication with me. I like to thank Dr. Bas van der Vorst and Dr. Marco van Zoelen for the many fruitful discussions we had on grease lubrication. I send my gratitude to all my colleagues at SKF for giving me a very good time and the opportunity to work with them. I would like to thank Prof. Stathis Ioannides and Mr. Alexander de Vries for their kind permission to publish the work presented in this thesis.

Finally, I thank my loving family for their support and the interest they showed in my work.

Pieter Baart
Nieuwegein, September 2011

Abstract

Rolling bearings contain seals to keep lubricant inside and contaminants outside the bearing system. These systems are often lubricated with grease; the grease acts as a lubricant for the bearing and seal and improves the sealing efficiency. In this thesis, the influence of lubricating grease on bearing seal performance is studied. Rheological properties of the grease, i.e. shear stress and normal stress difference, are evaluated and related to the lubricating and sealing performance of the sealing system. This includes the seal, grease and counterface.

The grease velocity profile in the seal pocket in-between two sealing lips is dependent on the rheological properties of the grease. The velocity profile in a wide pocket is evaluated using a 1-dimensional model based on the Herschel-Bulkley model. The velocity profile in a narrow pocket, where the influence of the side walls on the velocity profile is significant, is measured using micro particle image velocimetry. Subsequently, the radial migration of contaminants into the seal pocket is modelled and related to the sealing function of the grease. Additionally, also migration in the axial direction is found in the vicinity of the sealing contact. Experimental results show that contaminant particles in different greases consistently migrate either away from the sealing contact or towards the sealing contact, also when the pumping rate of the seal can be neglected.

Lubrication of the seal lip contact is dependent on several grease properties. A lubricant film in the sealing contact may be built up as in oil lubricated seals but normal stress differences in the grease within the vicinity of the contact may result in an additional lift force. The grease, which is being sheared in the vicinity of the contact, will also contribute to the frictional torque.

It is important to maintain a lubricant film in the sealing contact to minimize friction and wear. Here the replenishment of oil separated from the grease, also referred to as oil bleed, is of crucial importance. A model is presented to predict this oil bleed based on oil flow through the porous grease thickener microstructure. The model is applied to an axial sealing contact and a prediction of the film thickness as a function of time is made.

The work presented in the thesis gives a significant contribution to a better understanding of the influence of lubricating grease on the sealing system performance and seal lubrication conditions.

Samenvatting

Wentellagers bevatten afdichtingen om smeermiddel binnen en vuildeeltjes buiten te houden. Deze systemen worden meestal gesmeerd met smeervet dat als smeermiddel dient voor zowel het lager als de afdichting en daarnaast een afdichtende werking heeft. In dit proefschrift wordt de invloed van het smeervet op de werking van de lagerafdichting bestudeerd. De rheologische eigenschappen, d.w.z. de afschuifspanning en normaalspanning, van het vet worden bestudeerd en gerelateerd aan de smerende en afdichtende werking van het totale systeem dat bestaat uit een afdichting, smeervet en loopvlak.

Het snelheidsprofiel van de vetstroming in de kamer tussen twee afdichtingslippen hangt af van de rheologische eigenschappen van het vet. Het snelheidsprofiel in een brede kamer wordt benaderd met een 1-dimensionaal model gebaseerd op het Herschel-Bulkley model. Het snelheidsprofiel in een smalle kamer, waar de wanden een significante invloed hebben op de stroming, wordt gemeten met de “micro particle image velocimetry” methode. De migratie van vuildeeltjes in radiale richting in de met vet gevulde kamer tussen de twee afdichtingslippen wordt gemodelleerd en gecorreleerd met de afdichtende werking van het vet. Ook de migratie van vuildeeltjes in axiale richting in de buurt van het lip contact wordt bestudeerd. Experimentele resultaten laten zien dat vuildeeltjes in verschillende vetten consistent of weg van het lip contact of naar het lip contact toe migreren, ook als de pompwerking van de afdichting verwaarloosd kan worden.

De smering van het lipcontact hangt af van verschillende eigenschappen van het vet. Een smeerfilm in het contact kan worden gegenereerd zoals in olie gesmeerde afdichtingen maar de normaalspanning in het vet kan resulteren in een extra kracht op de afdichtingslip. De afschuiving van het vet in de buurt van het lipcontact zal ook bijdragen aan het wrijvingskoppel.

Het is belangrijk een smeerfilm in het lipcontact te behouden om wrijving en slijtage te minimaliseren. Hier is de afscheiding van olie uit het smeervet, ook wel olie bloeden genoemd, van cruciaal belang. Een model om de olie bloeding te voorspellen wordt gepresenteerd en is gebaseerd op stroming door de poreuze microstructuur van het vet. Het model wordt toegepast op een axiale afdichtingslip en de filmdikte in het contact wordt voorspeld als een functie van de tijd.

Het werk in dit proefschrift geeft een significante bijdrage aan een beter begrip van de invloed van smeervet op de afdichtende werking van het afdichtingsysteem en de smeringscondities in het contact.

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Appended papers

Below is a list of papers which are included in the thesis.

Paper A: Baart, P., Lugt, P.M. and Prakash, B. ‘Review of the lubrication, sealing and pumping mechanisms in oil and grease lubricated radial lip seals,’ *Proc. Inst. Mech. Engrs., part J: J. Engineering Tribology*, 2009, 223, 3, 347-358

Paper B: Baart, P., Lugt, P.M. and Prakash, B. ‘Non-Newtonian effects on film formation in grease lubricated radial lip seals,’ *Tribology Transactions*, 2010, 53, 3, 308-318

Paper C: Baart, P., Green, T.M., Li, J.X., Lundström, T.S., Westerberg, L.G., Höglund, E., and Lugt, P.M., ‘The influence of speed, grease type, and temperature on radial particle migration in a double restriction seal,’ *Tribology Transactions*, accepted for publication

Paper D: Baart, P., Lugt, P.M. and Prakash, B. ‘Contaminant migration in the vicinity of a grease lubricated bearing seal contact,’ *Journal of Tribology*, accepted for publication

Paper E: Baart, P., Van der Vorst, B., Lugt, P.M. and Van Ostayen, R.A.J., ‘Oil bleeding model for lubricating grease based on viscous flow through a porous microstructure,’ *Tribology Transactions*, 2010, 53, 3, 340-348

Paper F: Baart, P., Van Zoelen M.T., and Lugt, P.M., ‘Film thickness model for grease lubricated bearing seals with an axial contacting lip,’ to be submitted for publication

About the author

Pieter was born in the Netherlands in November 1982 and received his bachelors' degree in mechanical engineering at Delft University of Technology in The Netherlands. He continued his studies with a specialization in Tribology and received his Masters degree (with honours) at Delft University of Technology in 2007. Subsequently, he started his PhD studies at Luleå University of Technology in Sweden while being employed and working at SKF Engineering and Research Centre in the Netherlands.

Below is a list of other publications, including conference contributions, which are not included in the thesis as appended papers.

Abstract: Baart, P., Lugt, P.M. and Prakash, B. 'Review of the lubrication, sealing and pumping mechanisms in oil and grease lubricated radial lip seals,' *Proceedings of the 35th Leeds-Lyon symposium on Tribology*, 2008

Extended abstract: Baart, P., Lugt, P.M. and Prakash, B. 'Non-Newtonian effects on film formation in grease lubricated radial lip seals,' *Proceedings of the STLE 64th annual meeting*, 2009

Licentiate thesis: Baart, P., 'Grease lubrication in radial lip seals', *Luleå University of Technology*, ISBN 978-91-86233-50-1, 2009

Poster: Baart, P., Lugt, P.M. and Prakash, B. 'Grease lubrication in radial lip seals,' *14th Nordic Symposium on Tribology*, 2010

Paper: Green, T.M., Baart, P., Westerberg, L.G., Lundström, T.S., Höglund, E., and Lugt, P.M., 'A new method to visualize grease flow in a double restriction seal using micro particle image Velocimetry,' *Tribology Transactions*, in press.

Paper: Li, J., Höglund, E., Westerberg, L.G., Green, T.M., Lundström, T.S., Lugt, P.M., Baart, P. 'Measurements of grease flow in channels with restrictions using μ PIV,' *Proceedings of the 14th Nordic Symposium on Tribology*, 2010

Extended abstract: Baart, P., Lugt, P.M. and Prakash, B. 'The sealing function of grease: contamination migration in grease lubricated seals,' *Proceedings of the STLE/ASME International Joint Tribology Conference*, 2010

Nomenclature

A	Cross section	m^2
A	Grease constant	$(m \cdot \ln(T))^{-1}$
A_r	Cross section grease reservoir	m^2
a	Particle radius	m
B	Grease constant	m^{-1}
b	Seal contact width	m
C_1	Constant	$(kg \cdot m)^{-1}$
C_2	Constant	$kg^{-1} \cdot m$
D_f	Soap fiber diameter	m
d_s	Sealing contact diameter	m
F_{body}	Specific body force	N/m^3
F_c	Centrifugal force	N
F_d	Drag force	N
$F_{friction}$	Friction force	N
F_{lip}	Specific lip force	N/m
F_N	Normal force	N
f_o	Initial soap mass fraction	-
G	Duty parameter	-
H_o	Initial height grease reservoir	m
h	Film thickness	m
h	Gap height	m
h_{max}	Maximum film thickness	m
K	Consistency constant	$Pa \cdot s^n$
K_c	Consistency constant	$Pa \cdot s^n$
k	Boltzman constant	$J \cdot K^{-1}$
k	Permeability	m^2
$M_{friction}$	Frictional moment	$N \cdot m$
m	Power law index	-
N_1	First normal stress difference	Pa
N_2	Second normal stress difference	Pa
N_x	Normalized relative intensity	-
n	Rotational speed	min^{-1}
n	Power law index	-
n_c	Power law index	-
Pe	Peclet number	-
∇p	Pressure gradient	Pa/m
Q_{body}	Flow rate oil loss - body force	m^3/s
Q_{feed}	Flow rate oil feed	m^3/s
Q_{loss}	Flow rate oil loss	m^3/s
Q_{pump}	Flow rate oil loss - pumping	m^3/s

R	Rim radius	m
R_c	Radius axial contact	m
R_o	Radius grease reservoir	m
r	Radius	m
r_i	Inner radius	m
r_o	Outer radius	m
T	Temperature	°C
T	Torque	N·m
Ta	Taylor number	-
Ta_c	Critical Taylor number	-
t	Time	s
U_s	Shaft surface speed	m/s
u	Velocity	m/s
u_r	Radial velocity	m/s
V_o	Initial volume oil in contact	m ³
V_{oil}	Volume oil in contact	m ³
V_{loss}	Oil loss volume	m ³
W	Width grease reservoir	m
Wi	Weissenberg number	-
x	Coordinate direction	m
y	Coordinate direction	m
z	Coordinate direction	m
α	Lip angle air side	°
β	Lip angle lubricant side	°
$\dot{\gamma}$	Shear rate	s ⁻¹
η	Oil viscosity	Pa·s
η_o	Low shear rate viscosity plateau	Pa·s
η_{40}	Base oil viscosity at 40°C	Pa·s
η_{100}	Base oil viscosity at 100°C	Pa·s
η_{∞}	High shear rate viscosity plateau	Pa·s
η_{bo}	Base oil viscosity	Pa·s
θ	Angle	rad
λ_l	Relaxation time constant	s
ρ_g	Grease density	kg/m ³
ρ_p	Particle density	kg/m ³
σ_{2l}	Shear stress	Pa
τ_y	Yield stress	Pa
Ω	Angular velocity	rad/s
ω	Angular velocity	rad/s

Chapter 1

Introduction

1.1 Machine elements

Industrial machines, automotive vehicles and many other products contain parts moving relative to each other. These parts typically transfer a high mechanical load in rolling or sliding motion and have to perform reliably. The interaction between the surfaces of these parts is the science of tribology. More generally, tribology is the study of the principles of lubrication, friction, and wear.

Knowledge about the tribological behaviour of contacting surfaces helps to design more reliable machine elements with higher performance and reduced frictional losses. At present, there is an especially large focus on reducing energy consumption and CO₂ emissions. Here, tribology is the key element in designing new systems which have lower friction, and increased system performance and reliability.

Figure 1 Cylindrical roller bearing: 1- outer ring, 2- inner ring, 3- roller and 4- cage separating the rollers.

Figure 2 Rotary shaft seal: 1- seal contact with rotating shaft, 2- outer static seal, 3- metal reinforcement and 4- garter spring for constant lip force.

A well known machine element containing many tribological contacts is the rolling bearing, which enables rotation between two parts with low friction while transmitting high mechanical loads. Figure 1 shows a Cylindrical Roller

Bearing (CRB) containing cylindrical rollers. To make these bearings perform reliably, effective lubrication is required to prevent the direct contact of the rollers and the ring surfaces. The lubricating medium can be oil or grease but today, in 80-90% of all rolling bearings, grease is the preferred lubricant. In order to prevent lubricant leakage into the environment and to prevent contaminants from entering and damaging the bearing, elastomer seals are often being used. Figure 2 shows a typical rotary shaft seal which is used in many types of machines and automotive applications. The rolling bearing, lubricant, and seal are often integrated into one system, called a “sealed and greased for life” unit. Figure 3 shows an example of a Truck Hub Unit (THU), which enables the rotary motion of the wheel relative to the truck axle. This unit contains the complete system of two Tapered Roller Bearings (TRB), grease lubricant, and two elastomer seals. By supplying such a complete system to the market, the truck manufacturer can maximize performance and reliability of the product while reducing the number of parts and mass.

Figure 3 Truck Hub Unit (THU) containing the complete system of rolling bearings, grease lubricant, and seals.

1.2 Bearing seals

From first appearances, a rotary shaft seal looks like a very simple machine element. It consists of a lip which is in contact with the rotating shaft surface and is supported by a metal reinforcement. A garter spring may be used to apply a constant lip force. Oil seals are designed with asymmetric lip angles as shown in Figure 4. The angle α between the shaft and lip on the air side is smaller than that the angle β on the lubricant side to prevent lubricant from leaking out of the system. In ‘grease seals’, the difference in angles is less pronounced and often reversed to provide better sealing against contaminants from the environment. Leakage is less of a concern here due to the consistency

of grease, which prevents excessive loss of lubricant. Therefore, the main purpose of the grease seal is to keep contaminants out.

Figure 4 Rotary shaft seal running on a shaft with lip angle $\alpha < \beta$ for sealing oil

Between the soft elastomer seal lip and the hard metal shaft there is pure sliding motion. When looking at this sliding contact in more detail, it is surprising that there is hardly any wear of the elastomer and that the frictional torque is low. This can only be explained by the presence of a thin lubricant film which is sheared with low friction. The presence of this lubricant film was first reported in 1957 by Jagger [25]. He has shown that the frictional torque of a seal is much lower when lubricated than under dry conditions. Even when increasing the load, the lubricant film remains at the interface. From 1957 onwards, the focus has mainly been on the mechanisms of lubrication and the sealing action (no leakage). Many hypotheses have been published during the years on how these seals work. Hirano and Ishiwata [19] suggested in 1965 that the film formation is a result of micro-hydrodynamic lubrication between the rough shaft surface and a smooth seal. However, only one year later, Jagger and Walker [26] claimed that the seal surface is rough rather than the shaft surface. Nowadays, it is believed that both the seal and shaft roughness are important. Numerical models support this hypothesis, such as those by Gabelli and Poll [14], Shen and Salant [41], Hajjam and Bonneau [17]. Horve [22] has shown that a high seal roughness and a shaft roughness of 0.25-0.5 R_a are critical for long seal life and good sealing performance. Sealing performance is also referred to as the pumping action of the seal and has been studied extensively. Several hypotheses have been proposed in an attempt to explain why these seals do not leak. They include various physical aspects such as surface tension, capillary forces, Weissenberg effect, vortex flow, seal lip dynamics, and tangential deformations in the seal contact. The latter was proposed by Kuzma [31] in 1969 and was further developed by Kammüller [27] in 1986. Today, the tangential deformation theory is widely accepted as the primary sealing mechanism. The other 'secondary' mechanisms can still be very important under certain operating conditions.

All the fundamental work on rotary shaft seals was done on systems where oil was used as the lubricating medium. This oil is assumed to behave as a Newtonian fluid. However, Schulz et al. [40] claim that (multigrade) oil behaves in a non-Newtonian manner at the very high shear rates experienced in seals and its normal stress component can generate the load carrying capacity, as well as explaining the sealing action. Unfortunately, no experimental results have been published yet that can verify these hypotheses. An extensive review on these mechanisms was part of this PhD study and reported in paper A.

1.3 Grease lubrication

Grease is a complex lubricant which consists of base oil (65-95%), additives (0-10%) and thickener (3-30%) [5]; the thickener forms a network that acts as a reservoir for the oil. The most widely used grease thickeners are lithium, calcium or aluminium soaps. The soap gives the grease a semi-solid appearance and enables lubrication in applications where a continuous supply of oil is not possible.

The semi-solid composition of the grease, including solid and fluid phases, results in complex flow behaviour. Below a certain ‘yield’ stress, the grease is almost stiff like a solid and deforms elastically. Above this yield stress, the grease will flow like a fluid and the viscosity will decrease with increasing shear rate. This semi-solid behaviour is also called viscoelastic behaviour. There is no exact defined yield stress value but an apparent yield stress is often used in the non-Newtonian flow models describing the grease shear stress.

As previously mentioned, in 80-90% of all rolling bearings, grease is the preferred lubricant. Grease is easy to apply and it does not leak easily. Low friction is obtained after an initial running period where the grease is redistributed through the bearing and moves to the sides of the running track, Figure 5. The grease is retained in the bearing using a simple shield or seal and it continuously supplies oil to the contacting surfaces for lubrication [13]. This mechanism where oil is slowly released from the grease is called oil bleeding and is of great importance for bearing performance and reliability.

In grease lubricated bearings, it is not only the small gap between the seal lip and the shaft that provides sealing against contaminants. The grease in the vicinity of the seal lip and grease in between two sealing lips also provide a sealing action, Figure 6. This additional “grease sealing mechanism” is expected to be present but its existence and functionality has yet to be ascertained.

Figure 5 Grease lubrication in a deep groove ball bearing containing a RS seal.

Figure 6 SKF Mudblock seal with grease in-between the sealing lips providing additional sealing action.

1.4 Objectives

Elastomer lip seals have been successfully used since the 1940's to seal lubricated systems. Despite extensive experimental and theoretical research, it is still not fully clear how these seals function. Especially concerning grease lubrication, very little is known. In the work presented in this thesis, the focus is on grease lubricated bearing seals and the interaction between the grease as a lubricant and the sealing contact.

The aim of this work is to better understand grease lubrication in bearing seals and to develop models for analyzing and predicting lubricating conditions, as well as the sealing function of grease. This thesis therefore essentially consists of two parts, describing the sealing function of grease and the lubricating conditions.

Sealing function of grease: To determine the grease velocity profiles in a seal pocket geometry and contaminant migration in the vicinity of the seal lip contact in order to explain the sealing function of the grease.

Lubricating conditions: To investigate the non-Newtonian flow behaviour of grease and to identify whether the normal stress effect can significantly contribute to lubricant film formation. To investigate the oil bleed mechanism of grease, driven by a centrifugal body load, and relate this to seal lip contact replenishment.

The knowledge provided in this work is intended to be included in advanced seal friction and seal life models. Such models have been previously developed for oil lubricated seals but need to be reviewed to make them applicable to grease lubricated bearing seals.

1.5 Outline

The Chapters in part 1 of the thesis provide an overview and discussion of the main results and conclusions. More detailed information can be found in the appended papers which are included in part 2 of the thesis.

PART 1

Chapter 1 briefly introduces the subject and describes the motivation for carrying out the research. In Chapter 2, the non-Newtonian flow behaviour, or rheology, of grease is discussed and it is explained how the grease rheology is evaluated through rheometer experiments. The grease's rheological behaviour determines the grease velocity profile in sealing geometries and these aspects are discussed in Chapter 3. Here the 3-dimensional velocity field in a seal pocket is measured and the influence of grease type, speed and temperature on the velocity profile is evaluated. The grease rheology and flow profile are important for understanding the sealing function of the grease. Chapter 4 discusses contaminant migration in the radial direction of a grease pocket between sealing lips. Subsequently, contaminant migration in the vicinity of the sealing contact is evaluated. Another result of the non-Newtonian flow behaviour of the grease is the existence of normal stress differences that generate an additional lift force on the seal lip. The contribution of this effect to film formation is discussed in Chapter 5. The replenishment of oil to the sealing contact for maintaining the lubricant film, which occurs from oil bleed of the grease, is discussed in Chapter 6. In Chapter 7 the main conclusions and recommendations for future research are presented.

PART 2

Paper A contains a literature review on the lubrication, sealing and pumping mechanisms in oil and grease lubricated radial lip seals. Paper B discusses grease rheology and presents a film formation model based on normal stresses in the grease. In Paper D the grease flow in seal pockets between seal lips is studied. The influence of grease type, speed, and temperature are measured and the radial migration of contaminant particles is predicted. Contaminant migration in the axial direction in the vicinity of the sealing contact is studied in Paper D. Paper E presents the oil bleed model based on oil flow through the porous soap microstructure. This oil bleed model is applied on an axial sealing geometry to predict the film thickness in the sealing contact as a function of time.

Chapter 2

Grease rheology

Grease consists of liquid base oil, additives and solid thickener. This makes grease a complex semi-solid fluid. The grease flow behaviour is characterized by rheological models that describe the viscosity, shear stress and normal stress differences as a function of the shear rate. The various model parameters are obtained from flow curves measured on rheometers as described in paper B. Additional results from paper C and paper D are also included.

2.1 Grease properties

Lubricating grease consists of 65-95% base oil, 0-10% additives, and 3-30% thickener [11], where the latter gives the grease a semi-solid structure. The typical structure of a metallic soap type thickener looks like a fibrous network and is shown in Figure 7. Other types of networks can be expected for greases based on other thickener systems like organic clay or polymers. The interaction between the oil and thickener system determines the flow properties, or rheology, of the grease.

Figure 7 SEM image of a lithium hydroxystearate soap network after washing out the base oil.

The semi-solid character of lubricating grease prevents it from flowing at low stresses, i.e. a minimum stress has to be overcome before flow starts. This stress is referred to as the yield stress (Barnes [3]) or the “apparent” yield stress since grease actually shows creep flow at stresses below the apparent yield stress. Separation of the thickener and the oil may result in even more complex flow behaviour at low shear rates, such as shear banding or wall slip (Keentok [29]). Consequently, when the viscosity or shear stress is measured as a function of the applied shear, a non-linear relationship is found.

Another consequence of the thickener structure is that it introduces elasticity, which results in additional stress components in the grease orthogonal to the direction of shear. These stresses are referred to as normal stress differences and are of the same order of magnitude as the shear stresses (Hutton [24]).

Furthermore, the grease properties are strongly temperature dependent. The viscosity of the base oil reduces significantly with temperature, which also reduces the grease viscosity and apparent yield stress. The rheology, together with the type of base oil, the type of thickener, and the selection of additives, are the main parameters that determine the performance of greases in many engineering applications. A general description of the greases and other lubricants used in this thesis can be found in Appendix A.

2.2 Rheology models

Many models are available to describe rheological properties of complex fluids. Some specific models find their application in grease lubrication and can be used to describe the viscosity and shear stress as a function of shear rate. An additional model for the normal stress difference in grease is presented in this thesis.

2.2.1 Viscosity

The flow of lubricating grease is strongly non-linear due to the decrease in viscosity with increasing shear rate, called shear thinning. In general, the viscosity reaches a maximum plateau at low shear rates and a minimum plateau at high shear rates. There is a transition region in-between low and high rates, as shown in Figure 8. Equations to describe this general flow curve need at least four parameters. A general model that can be used here is the Cross model (Barnes et al. [4])

$$\eta = \frac{\eta_o - \eta_\infty}{1 + (K_c \dot{\gamma})^{n_c}} + \eta_\infty, \quad (1)$$

where $\dot{\gamma}$ is the shear rate, η_o gives the low shear rate viscosity plateau, η_∞ gives the high shear rate viscosity plateau, K_c is the consistency constant and n_c the shear thinning constant.

Figure 8 Cross model plot with constant viscosity plateaus at low and high shear rates. The dashed line indicates the boundary from which the Sisko model can be used.

In seal applications, the focus is mainly on the high shear rate part of Figure 8 because of the high shear rates caused by the narrow sealing gaps in combination with high sliding velocities. Consequently, Eq. (1) can be simplified for high shear rates to obtain the Sisko model shown below (Barnes et al. [4]):

$$\eta = K\dot{\gamma}^{n-1} + \eta_\infty, \quad (2)$$

with K the consistency constant, n the shear thinning constant and η_∞ the minimum viscosity plateau. Eq. (2) is only valid for shear rates from the dashed line in Figure 8 onwards since it no longer includes the low shear rate viscosity plateau. At very high shear rates, the viscosity in the Sisko model approaches η_∞ . For lubricating greases, η_∞ is assumed to be equal to the base oil viscosity.

2.2.2 Shear stress

The shear stresses in the grease can be calculated by multiplying the viscosity in Eq. (2) with the shear rate. The grease's apparent yield stress behaviour is obtained by incorporating an additional term:

$$\sigma_{21} = \tau_y + K\dot{\gamma}^n + \eta_\infty\dot{\gamma}, \quad (3)$$

where 1 is the main flow direction and 2 is the direction in which the velocity is varying, τ_y is the apparent yield stress, K the consistency constant, n the shear thinning constant and η_∞ the base oil viscosity. Eq. (3) can be recognized as the Herschel-Bulkley model but including an additional base oil viscosity term (Palacios and Palacios [37]). The rheological parameters in Eq. (3) have to be determined by fitting the model to flow curve measurements (shear stress versus shear rate) on a rheometer.

2.2.3 Normal stress difference

The elasticity of the grease results in the presence of normal stress differences, as shown by Hutton [24], due to anisotropic deformation of the grease microstructure. A first normal stress difference is defined in Eq. (4). A second normal stress difference may also exist but is typically much smaller than the first normal stress difference (Barnes et al. [4]).

$$N_1 = \sigma_{11} - \sigma_{22}, \quad (4)$$

The White-Metzner model is used to model the first normal stress difference which follows power-law behaviour over a range of shear rates and includes the lubricant viscosity (Barnes et al. [4]). The White-Metzner model makes use of the Sisko model, Eq. (2), for the viscosity, giving

$$N_1 = 2\lambda_I (K\dot{\gamma}^{n-1} + \eta_\infty)\dot{\gamma}^m, \quad (5)$$

where λ_I is the relaxation time constant and m the power-law constant. The parameters in Eq. (5), except for λ_I and m , can be found by fitting Eq. (3) to shear stress measurements. The parameters λ_I and m can subsequently be found by fitting Eq. (5) to normal stress difference measurements on a rheometer.

2.3 Experimental rheometry

The values for the various parameters in the models presented in Section 2.2 follow from fits to experimentally obtained flow curves that are measured on a rheometer. For this purpose, different types of rheometer experiments can be carried out. In this section, the measurement method that is used in this thesis is presented, including the limitations of these rheometer experiments.

2.3.1 Rheometer types

Different types of rheometers are available to measure grease rheology and to determine the fitting parameters for the rheology models. Most popular are the rotational rheometers with different geometries, such as the cone and plate (Figure 9), parallel plates (Figure 10), concentric cylinder (Figure 11), and vane (Figure 12). In the rotary cone and plate rheometer, it is assumed that due to the angle of the cone, the shear rate is constant over the plate radius and gap height. The maximum shear rate that can be reached is limited by the maximum rotational speed of the rheometer or by grease fracture or grease loss that may occur at the edge. The advantage of the rotary parallel plate rheometer is that higher shear rates can be reached by decreasing the gap height, and consequently, edge effects are less likely to occur. Here, the shear rate is assumed to be constant over the gap height but is no longer constant over the plate radius. In the concentric cylinder type rheometer, a small gap is formed between two concentric cylinders, one of which is rotating. Again, a constant shear rate in the gap is assumed, which is a good approximation if the gap is sufficiently small compared to the cylinder radius. To reduce wall slip (or phase separation at the wall) in all of these geometries, a high surface roughness is required. An extreme version of such high roughness is obtained in the vane geometry, where the inner cylinder in the concentric cylinder rheometer is replaced by a vane.

Figure 9 Cone and plate geometry for a rotational rheometer

Figure 10 Parallel plate geometry for a rotational rheometer

Figure 11 Concentric cylinder geometry for a rotational rheometer

Figure 12 Vane geometry for a rotational rheometer with 6 vanes

The experimental results are usually presented as the viscosity or shear stress as a function of the shear rate. These are indirect results since most rotational rheometers apply a torque and measure the angular velocity. In all rotary rheometers, the shear stress is calculated from the torque and geometry size, and the shear rate is calculated from the angular velocity and gap height. For calculating the shear rate, the grease velocity profile is assumed to be linear in a sufficiently small gap which does not necessarily have to be true since wall slip, shear thinning and yield stress behaviour may result in more complex flow profiles (Davies and Stokes [11]). These disturbances are also present in non-rotational rheometers like the capillary tube rheometer, where the viscosity is determined by measuring the flow rate as a function of a pressure difference over a pipe.

2.3.2 Thin film rheometry

For the work presented in this thesis, a rotational rheometer with parallel plate geometry is used for measuring the rheological properties of the grease. To simulate the very high, continuous shear rates as found in bearing seal applications, high rotational speeds and small gap heights are required. This method will introduce several measurement errors due to the non constant shear rate distribution, inertia on the grease sample, errors in gap setting, viscous heating of the sample, and edge fracture. These errors are discussed in e.g. Davies and Stokes [11] as the main errors in narrow gap parallel plate rheometry, and are discussed in more detail in Paper B.

More difficulties may arise due to the semi-solid nature of the grease. To measure the bulk properties of the grease, as modelled in Section 2.2, the gap has to be at least 10 times larger than the characteristic dimension of the grease structure. Hurley and Cann [23] found a typical grease thickener fibre length of 10 μm , which means that the parallel plate gap setting must be at least 100 μm .

At very high shear rates, the grease may also mechanically age. This mechanical aging is caused by an irreversible breakdown of the thickener structure, resulting in a significant reduction in viscosity. At very low shear rates, wall slip at the parallel plate surface takes place, which results in a reduced shear rate in the bulk grease and misleading experimental results from the rheometer data.

2.3.3 Measurement method

A 25 mm diameter parallel plate geometry is used in an AR1000N rheometer from TA Instruments. Measurements are done at continuous shear. Alternatively frequency sweep experiments can be done where the grease sample is deformed with only very small amplitudes at a range of frequencies. However, this does not represent the flow conditions in bearing seal applications, and therefore, this method was not used.

Before starting an experiment the rheometer environmental chamber is heated to the required temperature and the gap setting error is determined using a Newtonian oil, as described in Paper B. An excessive amount of grease is loaded into the geometry, which is subsequently set to the required gap height. Since the excess of grease has to be pushed out, some internal stresses are introduced into the sample. This surplus of grease is carefully removed from the edge. Next, the internal stresses are removed with a pre-shear program before the actual measurement is done. This pre-shear program should not reach too high speeds to prevent unintended grease loss from the edges or edge fracture. After the pre-shear program the grease is left to rest for some time and the flow curve is measured from a low angular velocity up to a high angular velocity. After the experiment the edge is examined for grease loss and edge fracture.

The measurement data has to be corrected for the errors mentioned in Section 2.3.2. To correct for the non-constant shear rate over the plate radius the shear stress is calculated from the measured torque as (Macosko [35])

$$T = 2\pi \int_0^R \sigma_{21} r^2 dr, \quad (6)$$

$$\sigma_{21} = \frac{T}{2\pi R^3} \left(3 + \frac{\partial \ln T}{\partial \ln \dot{\gamma}_R} \right), \quad (7)$$

where σ_{21} is the shear stress, $\dot{\gamma}_R$ the shear rate at the rim, T the measured torque, and R plate radius. From Eq. (6) or Eq. (7) parameters in the rheology model, Eq. (3), can be derived. Note that for the cone and plate geometry Eq. (7) reads $\sigma_{21} = 2T/(\pi R^3)$.

Similarly, the normal stress difference can be calculated as (Macosko [35]) as

$$F_N = \pi \int_0^R (N_1 - N_2) r dr, \quad (8)$$

$$N_1 - N_2 = \frac{F_N}{\pi R^2} \left(2 + \frac{\partial \ln F_N}{\partial \ln \dot{\gamma}_R} \right), \quad (9)$$

where $N_1 - N_2$ is the normal stress difference and F_N is the measured normal force. Since N_2 is typically much smaller than N_1 it can be assumed that $N_1 - N_2 \equiv N_1$. Subsequently, the λ_I and m parameters in Eq. (5) can be derived. However, it should be noted that in this case Eq. (5) is actually a fit to $N_1 - N_2$. For the cone and plate geometry Eq. (9) reads $N_1 = 2F_N/(\pi R^2)$. Since on the parallel plate rheometer $N_1 - N_2$ is measured and on the cone and plate rheometer N_1 is measured, measurements using both rheometer types enable determination of N_2 up to the maximum shear rate that can be obtained in the cone and plate rheometer.

In order to be consistent in the fitting procedure, a computer calculation scheme is used. Here the yield stress and base oil viscosity are first measured at the correct temperature, and subsequently, entered in the code manually. The other parameters are determined based on a best R^2 fit.

2.4 Experimental flow curve

The rheological behaviour of grease is represented in flow curves. In this section, the flow curves of a multi-purpose rolling bearing grease are measured on a parallel plate rheometer. Based on these flow curves, fits are made to determine the parameters for the rheology models.

2.4.1 Shear stress

The shear rheology of a multi-purpose rolling bearing grease (grease D in paper B) is measured at different gap heights of 500, 250, 100, 50 and 25 μm at 25 $^\circ\text{C}$. Shear rates up to $\dot{\gamma} = 5 \cdot 10^4 \text{ s}^{-1}$ are obtained at the smallest gap of 25 μm . The experimental results for the shear stress are shown in Figure 13 and have been corrected for a positive gap error of $\varepsilon = 38 \mu\text{m}$. The measurement results at different gap settings agree well at high shear rates, but at low shear rates, some deviation in the results is observed which is due to wall slip.

Figure 13 Flow curve for the shear stress of grease D at 25 °C

The shear stress model from Eq. (3) has been fitted to the measurement data in Figure 13 and is plotted as a solid line. The correction for the non-constant shear rate over the plate radius is incorporated in the model. The dashed line represents the shear stress based on the base oil viscosity, which is considered an asymptote to the model at high shear rates. The same experiment has been repeated for four different temperatures and the values for the rheological parameters that are obtained are presented in Table 1. Here, τ_y directly results from the measurements and η_∞ is taken as the base oil viscosity at the correct temperature. Subsequently, K and n are used for fitting the curve.

	25 °C	70 °C	100 °C	120 °C	Unit
τ_y	350	60	25	10	Pa
η	0.17	0.030	0.0095	0.0044	Pa·s
K	20	10	5.0	3.0	Pa·s ⁿ
n	0.50	0.49	0.48	0.48	-
λ_I	5.6	4.6	4.2	4.0	s
m	0.71	0.71	0.71	0.71	-

Table 1 Rheology model parameters for a multi-purpose grease (grease D).

2.4.2 Normal stress difference

In addition to the shear stress, the normal stress difference is also measured, which is derived from the normal force in the parallel plate geometry. Here it is assumed that the second normal stress difference is very small and can be neglected. The term ‘normal stress’ is used for the stress that results in the normal force on the plates in the rheometer. The measurements are corrected for the gap error and inertia effects and the results are presented in Figure 14. The relaxation time constant λ_I and the power law index m in the normal stress difference model, Eq. (5), are fitted to the measurement results at shear rates above 500s^{-1} and are also presented in Table 1. The model therefore shows a good agreement with the experimental results at shear rates between $3 \cdot 10^2 < \dot{\gamma} < 10^4 \text{s}^{-1}$. However, at shear rates below 300s^{-1} , the spread in the measured data is large due to the yield stress behaviour of the grease as explained by Hutton [24]. Further data presented in Paper B shows that at higher temperatures, Eq. (5) also fits the lower shear rates well. The results from Figure 14 confirm that the normal stress differences at higher shear rates are in the same order of magnitude as the shear stress in Figure 13.

Figure 14 Flow curve for the normal stress difference of grease D at 25°C

2.4.3 Temperature

The effect of temperature on the rheology of grease D has been studied at four different temperatures: 25, 70, 100 and 120 °C. The experimental results are presented in Paper B and the parameters for the rheology model can be found in Table 1. All parameters τ_y , K , n , η_∞ , and λ_I seem to be temperature dependent except for the power-law exponents n and m .

2.4.4 Yield stress

The yield stress or apparent yield stress is used in the shear stress model, Eq. (3), and can be determined and defined in several ways. One way is to read its value from the flow curve in Figure 13 as the stress at the constant stress plateau at low shear rates. However, due to wall slip in the rheometer geometry, such a plateau is not always clearly present. In this case, the vane geometry can be used and an experiment where the torque on the vane geometry is slowly increased while measuring the angular velocity could be performed. Typical results are shown in Figure 15. The yield stress is then calculated from the torque where the angular velocity starts to deviate from linearity using the following equation

$$\tau_y = \frac{T}{2\pi R^2 \left(h + \frac{R}{3} \right)}, \quad (10)$$

where T is the torque, R the vane radius and h the penetration depth of the vane into the cup. For greases with a high yield stress, the penetration depth h is reduced, which consequently reduces the required torque.

Figure 15 Yield stress (angular velocity) measurement results NLGI1 and NLGI2 grease.

2.5 Rheology in the vicinity of the contact

In this section, the grease viscosity and normal stress difference in the vicinity of the sealing contact are determined from the flow curves and presented as a function of the distance from the sealing contact.

2.5.1 Viscosity

Due to the angle of the seal lip, the shear rate dU_y/dz is a function of the distance from the sealing contact. Consequently, the viscosity is also a function of the distance to the sealing contact and depends on the grease's shear thinning properties. The viscosity at any distance from the sealing contact can be determined from the rheometer experiments (flow curves). Figure 16 shows the local viscosity for some greases, a solution of 0.5 wt% Poly(Ethylene) Oxide (PEO, $M_w \sim 8,000,000$) in water, and the LG1 base oil, all at a shaft surface velocity of 0.1 m/s, a 12° lip angle and ambient temperature (25°C). These greases and fluids are further used and discussed in Chapter 4.

Figure 16 Apparent viscosity in the vicinity of the sealing contact at a shaft surface velocity of 0.1 m/s, a 12° lip angle and ambient temperature (25°C). The points represent the rheometer results and the line a linear fit (viscosity gradient).

Figure 16 shows the different absolute viscosities of the greases, with the LG2 grease having the highest viscosity and the LG3 grease having the lowest viscosity. A linear fit (solid line in Figure 16) has been made to be able to determine the viscosity gradient in the vicinity of the sealing contact. The

constant viscosity gradient is only valid for the region where $x < 2$ mm and will change further away from the contact where $x \gg 2$ mm. The LG4 grease has the highest viscosity gradient here and reaches the LG2 viscosity for $x > 1.8$ mm. As expected, the PEO fluid and LG1 base oil have a constant viscosity.

2.5.2 Normal stress difference

Similar to the viscosity, the normal stress difference is also a function of the distance from the sealing contact and is determined from the rheometer experiments (flow curves). Figure 17 shows that the highest normal stress difference is present when closest to the sealing contact. The LG2 grease, which had the highest viscosity in Figure 16, also has the highest normal stress difference here. The LG4 grease has a relatively low normal stress difference compared to the relatively high viscosity in Figure 16. The normal stress difference in the PEO fluid is similar to the normal stress differences in the greases while the viscosity is significantly lower. As expected, the LG1 base oil shows no presence of any normal stress difference at all. The viscosity and normal stress difference data will be discussed further in Chapter 4.

Figure 17 First normal stress difference in the vicinity of the sealing contact at a shaft surface velocity of 0.1 m/s, a 12° lip angle and ambient temperature (25 °C).

2.6 Conclusion

The grease rheology or flow behaviour is very complex due to the semi-solid composition of grease. Equations have been presented to model the grease viscosity, shear stress and normal stress difference. The parameters in these models are determined from fits to measurements on the parallel plate rheometer. This type of rheometer has been used because it is the most suitable for measurements at the high shear rates that can be expected in bearing seals. Also, a methodology to measure the apparent yield stress is presented.

Finally, the viscosity and normal stress difference for a selection of greases, as a function of the distance from the sealing contact, are presented. The viscosity, viscosity gradient and the normal stress difference vary between greases when being sheared in the vicinity of the sealing contact.

Chapter 3

Grease flow in a seal pocket

The velocity profile of grease in a sealing geometry not only depends on the shaft speed and geometry, but also strongly depends on the grease rheology and temperature. First, the velocity profile in a wide pocket is evaluated. This is approached as a 1-dimensional flow problem. Secondly, the 3-dimensional velocity profiles in a narrow pocket are measured using a so-called “Double Restriction Seal (DRS)” setup in conjunction with a Micro Particle Image Velocimetry (μ PIV) technique. The influences of shaft speed, grease type and temperature on the grease velocity profile are evaluated. Details can be found in paper C.

3.1 Seal pocket geometry

The seal pocket is defined as the space in-between two sealing lips as found in many multi-lip seal designs. More generally, the seal lips are considered as sealing restrictions, which can be in or out of contact with the counter surface. As an illustration, Figure 18a shows a radial shaft oil seal, which has a contacting lip, loaded by a spring, and a non-contacting dust lip. The pocket in-between the lips is filled with grease. Here the pocket width is large compared to the height and is therefore referred to as a wide pocket. In such a case, the velocity profile is considered to be 1-dimensional in the centre of the pocket.

Figure 18 Seals including a seal pocket: (a) oil seal with dust lip and a wide pocket in-between the sealing lips, and (b) bearing seal with a narrow pocket in-between the sealing lips.

Figure 18b shows a non-contacting bearing seal. Here the pocket width is about equal to the pocket height and it is therefore referred to as a narrow pocket. In this case, the velocity profile is significantly influenced by the side walls and requires a 2-dimensional analysis.

3.2 Velocity profile in a wide pocket

The velocity profile in the wide pocket is evaluated for a Newtonian fluid and for the non-Newtonian grease by using the 4 parameter Herschel-Bulkley model.

3.2.1 Newtonian fluid

For a Newtonian fluid in a Couette geometry or concentric cylinder geometry, the velocity profile only depends on the shaft speed and gap geometry, i.e. the gap height relative to the shaft radius. In the case of a gap that is large compared to the shaft radius, a non-linear velocity profile may be expected. The 1-dimensional tangential velocity for Newtonian fluids in this geometry is given as (Batchelor [6])

$$u(r) = U_s \left[\frac{r_o/r - r/r_o}{r_o/r_i - r_i/r_o} \right], \quad (11)$$

where U_s is the shaft surface velocity, r_i the inner radius or shaft radius and r_o the outer radius or housing radius. For sufficiently small gaps where the gap height is small relative to the shaft radius, i.e. $(r_o - r_i)/r_i \ll 1$, Eq. (11) approaches a linear velocity profile. This is the case for a Newtonian fluid in the situation where the shaft radius is 20 mm and the gap is 1.5 mm as shown in Figure 19.

3.2.2 Herschel-Bulkley fluid

Lubricating greases generally show a non-linear flow behaviour, and consequently, Eq. (11) can not simply be applied. The fluid velocity profile is therefore calculated from the 4-parameter Herschel-Bulkley rheological model, as presented in Eq. (3), using the methodology from Kelessidis and Maglione [30]. Figure 19 shows the results of the model for NLGI1 and NLGI2 greases, both at 2 shaft speeds: 0.02 m/s and 0.075 m/s at ambient temperature (25 °C). The velocity profile of the Newtonian oil is approximately linear since $(r_o - r_i)/r_i$ is small. The greases show a small deviation from the Newtonian case due to shear thinning. However, when the shaft velocity or temperature is increased, this deviation reduces and the velocity profiles become more linear.

Figure 19 1-dimensional velocity profile in a wide pocket for a Newtonian oil and for NLGI1 and NLGI2 greases at ambient temperature (25 °C).

3.3 Velocity profile in a narrow pocket

In this section, the grease velocity profile in a narrow pocket is measured using micro Particle Image Velocimetry (μ PIV). The influence of different operating conditions on the velocity profile is also studied.

3.3.1 Measurement methodology

To measure the grease velocity profile in a narrow seal pocket, where the velocity profile is significantly influenced by the side walls, an experimental setup is used that allows for optical measurements inside the seal pocket or grease chamber. For the setup used, shown in Figure 20, the walls are made of thin transparent glass windows to obtain appropriate access with a microscope lens. To initiate an axial grease flow, grease is fed from the pressure chamber through the radial restriction into the grease chamber. Such a grease flow may occur in sealed bearing applications where internal pressures are generated; e.g. by heating up or by a relubrication action.

Figure 21 shows a close-up of the grease chamber and indicates the planes in which the grease velocity profiles are measured. The F planes show the tangential grease velocity as a function of the radial z -position. The B planes show the tangential grease velocity as a function of the axial x -position and the axial grease velocity as a function of the tangential y -position.

Figure 20 Double Restriction Seal setup for simulating a narrow seal pocket for grease velocity measurements. Detail A is shown in Figure 21.

Figure 21 Detail A from Figure 20, showing grease chamber dimensions and the measurement planes which are indicated with F and B.

The velocity profiles in the F and B planes are measured with a micro Particle Image Velocimetry (μ PIV) system which consists of a high speed camera, an optical microscope, and a pulsed laser light source. The grease is seeded with small fluorescent particles that light up in the laser light. A single measurement consist of two images of the flow, separated by a time step Δt . A cross correlation method is used to calculate the displacement of small clusters of particles (interrogation areas) during the time step and calculate the vector velocity field. An averaged vector velocity field is created over a series of single measurements and some further filtering and averaging is done to improve the quality of the results. The system and μ PIV method are described in more detail in Green et al. [16].

Figure 22 NLGI2 Grease velocities in the tangential y-direction in the grease pocket in the three different measurement planes indicated with F in Figure 21 at ambient temperature (25 °C).

3.3.2 3D velocity profile

The grease velocity profiles have been measured in two orthogonal directions, as indicated with the F and B planes in Figure 21. First, Figure 22 shows the results of the velocity measurements in the tangential y-direction, at ambient temperature (25 °C) and a shaft surface speed of 0.05 m/s for the three different measurement planes indicated by F in Figure 21. Here, the velocity in the F1 plane, which is close to the glass window, does not reach the shaft velocity of 0.05 m/s because it enters into the axial restriction. The results for the F2 and F3 planes are very similar even though they are measured at different positions. The grease velocity is close to zero for $r > 21$ mm, suggesting that the grease

apparently does not yield in this area. At smaller radii, the grease velocity increases exponentially towards the rotating shaft. No indication of wall slip or a grease velocity in the radial direction was found.

Measurements in the planes indicated with B in Figure 21 are also carried out. As expected, the measured velocity components in the tangential y -direction, at the points where the B and F plane intersect, coincide with the results in Figure 22. The velocity components in the axial x -direction in the three B planes are presented in Figure 23. This axial flow is driven by the pressure applied in the pressure chamber, as indicated in Figure 20. It is found that the axial flow mainly takes place close to the rotating shaft. This type of axial flow was further studied by Li et al. [33] where it was found that, as well as in the case of a flow without a rotating shaft, the flow depth into the seal pocket is very limited.

Figure 23 NLGI2 Grease velocities in the axial x -direction in the grease pocket at ambient temperature (25 °C) in the three different measurement planes indicated by B in Figure 21.

3.3.3 Influence of speed, grease type, and temperature

The velocity profile of the grease in the narrow pocket depends on the shaft speed, grease type and temperature. The effects are studied experimentally using the μ PIV method again but with a higher measurement resolution by using smaller fluorescent seeding particles and a higher microscope magnification. However, due to this higher magnification, it is not possible to look as far into the grease chamber and so the measurement plane is set at 0.1 mm behind the end face of the shaft, in-between plane F1 and F2 in Figure 21.

The setup has been equipped with a hot air box to examine the influence of temperature on the grease velocity profiles. Further details of the experimental method can be found in paper C.

Figure 24 shows the velocity profile as a function of shaft speed for the various grease types. All the grease velocity profiles deviate significantly from the 1-dimensional Newtonian case in Figure 19.

Figure 24 Grease velocities in the tangential y-direction in the narrow grease pocket for NLGI2, NLGI1, and NLGI00 grease at different shaft velocities at ambient temperature (25 °C).

The velocity profile also depends on temperature. At elevated temperatures, the shear thinning effect decreases, as shown in Figure 25 for the NLGI2 grease.

Figure 25 Grease velocities in the tangential y-direction in the grease pocket for an NLGI2 grease at different temperatures.

By plotting the data from Figure 24 and Figure 25 on a logarithmic scale for the velocity, it was found that the velocity profile can very well be described by an exponential model where the slope depends on the grease type and temperature but not on the shaft velocity. Additionally it became clear that in the apparent unyielded area, some very slow creep flow takes place. A very good model fit, as presented in Figure 24 and Figure 25 with the continuous lines, was made with the equation

$$u(r, T) = U_s e^{[A \ln(T) - B](r - r_i)} \left[\frac{r_o/r - r/r_o}{r_o/r_i - r_i/r_o} \right], \quad (12)$$

where T is the temperature and A and B are grease-type dependent constants. Values for these constants were determined for the NLGI1 and NLGI2 greases and are presented in Table 2. As a consequence, the shapes of the velocity profiles are not unique and equal velocity profiles for different greases can be obtained at different temperatures.

Grease type	A [(m·ln(T)) ⁻¹]	B [m ⁻¹]
NLGI2	1160	-6720
NLGI1	790	-4540

Table 2 Grease-type dependent parameters to describe the velocity profile in a narrow seal pocket

3.4 Conclusion

The velocity profile of grease in a *wide* seal pocket is modelled as a 1-dimensional flow. In a *narrow* seal pocket, the 3D velocity profile has been measured under different operating conditions using Micro Particle Image Velocimetry.

By using the 4-parameter rheology model, the grease velocity profile in the wide pocket shows only a small deviation from the Newtonian velocity profile. In the narrow pocket, a significant influence due to the pocket side walls is present, reducing the grease velocity. The velocity profile in the narrow pocket, at the measurement position, can be modelled with a very simple exponential model. Equal velocity profiles are obtained here for different NLGI grade greases at different temperatures. This indicates that for this case, the non-Newtonian grease shear thinning behaviour can be reflected in a single parameter.

It is found that a pressure difference over the narrow pocket results in grease flow in the axial direction, which mainly takes place close to the rotating shaft.

Chapter 4

Contaminant migration in grease

The grease flow in sealing geometries can be highly non-linear due to the non-Newtonian rheology of the grease. Therefore, contaminant particles will migrate differently in grease than in lubricating oil. Moreover, the grease provides an additional sealing function to the bearing and sealing system as presented in paper C and paper D.

4.1 Sealing function of grease

Oil seals and bearing seals can be equipped with additional sealing lips to increase the sealing performance against contaminants from the environment. Examples of such seals are shown in Figure 18 and Figure 26. The additional seal lips are lubricated with grease by applying some grease in the seal pocket between the sealing lips. Many textbooks on (grease) lubrication state that the grease also increases the sealing performance. For example, the Automotive Lubricants Reference Book [18] gives as an advantage of grease that it prevents the ingress of dirt into the mechanism. Lansdown [32] states that “grease can form a very effective seal against ingress of dirt and other contaminants into the system”. According to them, contaminants are less likely to be transported through the bearing due to the semi-solid flow behaviour of the grease. However, detailed knowledge on the mechanisms and the specific grease properties that provide this “sealing function of grease” is lacking.

Figure 26 Seals equipped with additional lips for better contaminant exclusion

Three mechanisms have been identified that provide additional sealing functions from the grease and are discussed in this Chapter. The mechanisms are:

- 1) Migration of contaminant particles in a seal pocket.
- 2) Migration of contaminant particles in the vicinity of the sealing contact.
- 3) Pressure difference and a limited flow depth into a seal pocket.

Input to the three “sealing function of grease” mechanisms is the work on grease rheology from Chapter 2 and on the grease flow profiles in the seal pocket from Chapter 3.

4.2 Migration in a seal pocket

Contaminant particles that enter into the seal pocket will move with the grease flow and migrate in the radial direction due to centrifugal forces. Consequently, particles migrate away from the sealing contact which reduces the probability of entering the bearing.

4.2.1 Migration model

Solid contaminant particles that have entered into the grease pocket are assumed to move with the same circumferential velocity as the grease. Consequently, centrifugal forces act on particles and the particles migrate to a larger radius. This centrifugal force is given as

$$F_{c,r} = \frac{4}{3}\pi a^3 (\rho_p - \rho_g) \frac{U_\theta^2}{r}, \quad (13)$$

where a is the particle radius, $(\rho_p - \rho_g)$ density difference between the particle and the grease, r the radial position and U_θ is the circumferential velocity of the grease which has been evaluated in Chapter 3. The radial particle migration is slowed down by drag forces as the particles migrate through the grease. The drag force is predicted using the Stokes drag equation (Batchelor [6])

$$F_{d,r} = -6\pi a \eta_r U_{p,r}, \quad (14)$$

which has been defined for a spherical particle moving through a stationary or quiescent fluid with a Reynolds number $Re \ll 1$. Here the grease viscosity η_r is a function of the local shear rate and $U_{p,r}$ is the particle velocity in the radial direction. Assuming that Eq. (13) and Eq. (14) are in equilibrium and particle acceleration or inertia is small enough to be neglected, the local particle velocity in the radial direction reads

$$U_{p,r} = \frac{2}{9} a^2 \frac{1}{\eta_r} (\rho_p - \rho_g) \frac{U_\theta^2}{r}, \quad (15)$$

where η_r and U_θ are calculated from the grease velocity profiles in Chapter 3. Due to the non-linear grease rheology model, including the shear thinning of the grease, it is not possible to solve Eq. (15) analytically and a numerical integration is used to calculate the radial position of the contaminant particles as a function of time. More details of the model can be found in Paper C.

4.2.2 Contaminant migration

To calculate the radial migration of a solid contaminant particle in a grease pocket, the velocity profile U_θ is required. For this, the grease flow velocity models from Chapter 3 can be used, i.e. the Herschel-Bulkley model for the wide pocket and the exponential model from Eq. (12) for the narrow pocket. Calculations are done for the wide and narrow seal pockets and for three different greases using Eq. (15). Figure 27 shows the radial migration results as a function of time. Here, after approximately 10 minutes a difference between the different pocket types and grease types becomes significant. In the wide pocket, the radial migration of a particle in all of the greases is very similar due to the very small differences in the grease velocity profile; see also Figure 19. In the narrow pocket, this migration takes significantly longer for all greases and clear differences between greases are observed. Here, it takes more than 2 hours for the NLGI00 grease to migrate to half of the height of the narrow pocket. It takes 20 hours and 70 hours respectively for the NLGI1 grease and NLGI2 grease at a shaft speed of 1 m/s. The contaminant particles in the NLGI2 grease migrate relatively fast in the first few minutes but significantly slow down at larger radial positions where the grease is apparently unyielded, as shown in the velocity profile in Figure 24.

The radial migration is discussed in more detail in Paper C and it is shown how the radial position of a particle after 100 hours strongly depends on the shaft's rotational velocity and the particle size. However, the results for the different greases in the wide pocket are very similar due to the almost equal velocity profiles in the 1-dimensional geometry.

Figure 27 Radial migration as a function of time for the wide and narrow gap geometry; position of a 14 μm diameter particle for different greases at $U_s = 1$ m/s and $T = 25^\circ\text{C}$

4.3 Migration in the vicinity of the contact

Contaminant particles that are present in the vicinity of the contact, where the grease is continuously being sheared, experience hydrodynamic forces and migrate either away or towards the sealing contact. This phenomenon is discussed below and is experimentally studied in various greases.

4.3.1 Migration theory

Solid contaminant particles in a sheared fluid flow may migrate orthogonal to the direction of fluid motion. Karnis and Mason [28] showed in an experimental study that for an elastic fluid in a pipe flow at low Reynolds numbers, neutrally spherical buoyant particles migrate in the radial direction to the pipe centre. In a Couette flow with a rotating inner cylinder the particles migrate to the outer cylinder. Gauthier et al. [15] found that neutrally buoyant particles in a shear thinning fluid, migrate to the pipe centre in a pipe flow and to the inner cylinder wall in a Couette flow. In the pipe flow and Couette flow, the shear rate is not constant in the radial direction, which is an important requirement for the particle migration in these fluids to take place. In a shear thinning fluid, the particles migrate consistently to the location where the shear rate is high. In an elastic fluid, the particles consistently migrate to the location where the shear rate is low. This migration behaviour is independent of the

particle shape since similar results were found for small disk and rod shaped particles. Ho and Leal [20] showed with an analytical study how spherical particles in an elastic fluid may migrate to the location where the shear rate is low. Normal stress differences in the fluid drive the particle migration here. This clearly applies to grease flow where the fluid rheological behaviour, i.e. shear thinning and normal stress differences, drives the particle migration.

The vicinity of the sealing contact, in which contaminant particle migration is considered, is schematically presented in Figure 28. The lower plane in the Figure represents the shaft, which rotates with a surface velocity U_s , and the top plane represents the seal lip, with a lip angle α . In this geometry, a spherical contaminant particle with radius a is assumed to be suspended in the grease. Here, gradients in the grease viscosity and normal stress difference in the axial x -direction are present since the shear rate dU_y/dz is a function of the gap height and thus the axial x -position.

Figure 28 Particle in a fluid velocity field in the vicinity of the sealing contact. Migration in the x -direction is studied.

In line with the experimental results from Karnis and Mason [28], and Gauthier et al. [15], it will be shown that particle migration in the vicinity of the sealing contact takes place in the axial x -direction, i.e. either towards or away from the sealing contact. In paper D, the effects of hydrodynamic forces, the pumping action of the seal, Brownian motion, particle inertia, and vortices, on particle migration in the vicinity of the sealing contact are discussed. It is concluded that hydrodynamic forces on the particles due to the grease rheology, dominate particle migration. Wall effects are not considered and it is found that only in the first few tenths of a millimetre from the sealing contact, where the Reynolds number is large, significant inertia forces may be present.

4.3.2 Measurement methodology

The measurement setup to evaluate the migration of contaminant particles in the vicinity of the sealing contact, consists of two FKM bearing seals which are positioned back-to-back, to create a seal pocket as shown in Figure 29. The seals are mounted on a hollow and transparent sapphire shaft. Spherical

fluorescent MF-Rhodamine B-particles, with a diameter of 10 μm , are used as contaminant particles. The particles are excited by laser light, and subsequently, they emit light with a different wavelength which is filtered using a small band pass filter to make them easily observable. Images are made with a digital camera before and after each measurement using a 100x optical magnification, so that only the fluorescent particles are visible, as shown in Figure 29. The images show the individual fluorescent particles and are analyzed using a software program that calculates the ‘normalized relative light intensity N_x ’, which represents the number of particles, and is defined as

$$N_x = \frac{n_{x,t=45} - n_{x,t=0}}{\sum_x n_{x,t=0}}, \quad (16)$$

where $n_{x,t=45}$ is the cumulative light intensity at distance x from the contact at $t = 45$ min, $n_{x,t=0}$ is the cumulative light intensity at distance x from the contact at $t = 0$ min, and the summation represents the total light intensity at $t = 0$ min in the whole domain. More details of the experimental setup and analysis can be found in Paper D.

Figure 29 Schematic top view of the setup with the digital microscope for observation of the particles in the grease in the vicinity of the seal contact. The insert contains a typical image of a particle distribution where the dashed line indicates the sealing contact.

A number of relatively transparent greases, which are used for lubricating rolling bearings, are studied and presented here. The greases are all Lithium based greases with mineral or PolyAlphaOlefin (PAO) base oils as shown in

Table 3, and have different rheological properties. The grease tackiness or elasticity is indicated by the “finger test”; for tacky greases long strings are formed when pulling the grease apart between two fingers. In addition to the greases, the base oil, that was separated from the LG1 grease, is used. The base oil shows Newtonian behaviour, i.e. no shear thinning and no normal stress differences. Also, a solution of 0.5 wt% Poly(Ethylene) Oxide (PEO, $M_w \sim 8,000,000$) in water is used. This fluid has a very low viscosity but is very elastic, with extremely high normal stress differences, and a limited amount of shear thinning. The rheological behaviour of the different lubricants is presented in more detail in section 2.5 and Paper D.

Grease type	Thickener/base oil	$\eta_{\text{base oil at } 25^\circ\text{C}} [\text{Pa}\cdot\text{s}]$	Tacky [finger]
LG1	Lithium/Mineral	0.25	+
LG2	Lithium/Mineral	0.45	++
LG3	Lithium/PAO	0.03	-
LG4	Lithium/Mineral	0.21	-
LG5	Lithium/Mineral	0.33	+

Table 3 Grease types and properties: chemical composition, base oil viscosity, and tackiness as experienced in finger test.

4.3.3 Contaminant migration

The concentration of contaminant particles in the vicinity of the sealing contact is measured for the different lubricants and shaft speeds. The results for the base oil and PEO are not presented here, but can be found in Paper D. Figure 30 shows the migration results for different greases at 23 rpm, corresponding to a shaft surface velocity of 0.1 m/s. Here, a positive ‘Normalized relative intensity N_x ’, as defined in Eq. (16), means that, locally, the number of particles has increased during the experiment. A negative value means that the number of particles has decreased.

In Figure 30, the LG4 grease has a positive value for $x < 0.8$ mm and a negative value for $x > 0.8$ mm, meaning that contaminant particles have migrated towards the sealing contact. In all the other greases, the contaminant particles migrate away from the sealing contact. The LG2 and LG3 greases show relatively large negative values, meaning that here a larger number of particles have migrated away from the sealing contact. Similar results were found at a higher shaft speed of 233 rpm, corresponding to a shaft surface velocity of 1 m/s. Here, the LG1 grease shows extremely large positive values, indicating that severe migration of particles towards the contact has taken place. This also shown in Figure 32 where images of the particle distributions at $t = 0$ min and t

= 45 min are shown respectively. This severe migration is caused by the leakage of LG1 grease from the seal pocket. For the other greases, such significant grease loss was not observed.

Figure 30 Particle migration in different greases at 23 rpm. A positive value means that the number of particles has increased at the end of the experiment.

Figure 31 Particle migration in different greases at 233 rpm. A positive value means that the number of particles has increased at the end of the experiment.

In order to predict the contaminant particle migration in the grease, a qualitative analysis is done, including the grease shear thinning behaviour and the grease elasticity, i.e the viscosity gradient and normal stress difference respectively.

Figure 16 showed the grease viscosity in the vicinity of the sealing contact. Here, a viscosity gradient in the x -direction, due to shear thinning, is clearly present. Additionally, a temperature gradient, due heat generation from viscous friction in the sealing contact, increases the viscosity gradient further. The viscosity gradient is expected enhance particle rotation, and bring contaminant particles towards the high shear rate and low absolute viscosity zone (Gauthier et al. [15]).

Snijkers et al. [42] showed that particle rotation is slowed down by the fluid elasticity when the Weissenberg number is larger than ~ 0.5 . They define the Weissenberg number as the normal stress difference $N_1 - N_2$ divided by the shear stress as

$$Wi = \frac{N_1 - N_2}{\sigma_{21}}, \quad (17)$$

which can be calculated for the greases using the rheological data from section 2.5. The normal stress differences can induce particle migration to low shear rate zones, i.e. away from the contact, as shown in Gauthier et al. (16) and Ho and Leal (20).

The LG4 grease has the largest viscosity gradient in Figure 16, without including a temperature gradient, and shows positive migration of contaminant particles towards the sealing contact in Figure 30 and Figure 31. The normal stress differences in the LG4 grease, in Figure 17, are relatively low, resulting in $Wi < 0.5$ as shown in paper D. For the LG1, LG2 and LG5 greases, $Wi > 0.5$ in the first millimetre from the sealing contact, meaning that particle rotation is slowed down significantly, and particles migrate away from the sealing contact. An exception is the LG3 grease, which seems not to fit the theory presented above. An explanation can be found in the very low base oil viscosity. The low base oil viscosity is expected to result in low viscous friction in the sealing contact. Consequently, also the temperature gradient is expected to be small, resulting in only a small increase of the viscosity gradient. This is different for the other greases, where the relatively high base oil viscosity results in a much larger temperature gradient and viscosity gradient, enhancing particle migration towards the sealing contact.

Although no quantitative correlation between the rheological properties and the contaminant particle migration was found, it is shown qualitatively that the viscosity gradient and Weissenberg number are important. A large viscosity gradient results in migration towards the sealing contact and a large

Weissenberg number results in migration away from the sealing contact. For quantitative predictions, more detailed information about the fluid velocity profile, grease rheology, temperature gradient, wall effects, and the influence of the contaminant particles on the velocity profile are needed, which requires expensive numerical models. Due to the complexity of the geometry, the nature of the grease, and other effects, like shear degradation and aging, such models are not yet available today.

a) LG1 $t = 0$ min

b) LG1 $t = 45$ min

c) LG2 $t = 0$ min

d) LG2 $t = 45$ min

Figure 32 Particle migration images as taken before and after the experiment at 233 rpm for LG1 and LG2 greases. The sealing contact is indicated with the dashed line; the arrow indicated the grease meniscus after grease loss.

4.4 Pressure difference

Relubrication, heating up or cooling down of the bearing system during operation cycles may result in a pressure difference over the sealing system. In such a case, the pressure can be released through the sealing contact and the seal pocket, which is known as “breathing” of the bearing.

It was shown in Figure 23, and is discussed in paper C, that in the narrow DRS pocket, a pressure induced axial grease flow mainly takes place in the first few tenths of a millimetre close to the rotating shaft. The thickness of the grease layer that axially flows in-between two restrictions is defined as the “flow depth” and was studied in more detail by Li et al. [33]. They evaluated the flow depth under static conditions, i.e. no shaft rotation, in an experimental setup consisting of a straight channel with two flow restrictions. Their results show that, in-between the two restrictions, part of the grease does not flow, which means that the distance between the restrictions is too small for the flow to fully develop again. Their results also show that, after a restriction, a minimum distance is required for the grease flow to fully develop again. The flow depth was measured as a function of flow rate and grease type and was found to be significantly lower for the NLGI1 and NLGI2 grease than for the NLGI00 grease.

Similar to the experimental setup by Li et al. [33], in the narrow seal pocket the distance between the restrictions is too small for the axial grease flow to fully develop. This means that part of the grease in-between the restrictions, i.e. the grease close to the outer wall, does not flow. Consequently, particles that have migrated in the radial direction, as shown in section 4.2, may not be picked up in an axial grease flow. In a wide seal pocket, where the width is much larger than the height, the grease flow will fully develop and pick up particles in the axial grease flow, except for the small volumes close to the corners.

4.5 Conclusion

The radial migration of solid contaminant particles has been predicted in both narrow and wide seal pockets. It is concluded that in the *wide* pocket contaminant particles migrate, during the same time period, to a larger radius than in a *narrow* pocket. The migration depends on the grease type and operating conditions.

In a narrow seal pocket, the radial migration reduces the probability of contaminant particles flowing into the bearing. Here, contaminant particles migrate due to centrifugal forces to a larger radius in the pocket and stay here, also when rotation stops, due to the yield behaviour of the grease. In the case that an axial grease flow takes place, which can be caused by breathing of the

bearing, only contaminant particles close to the shaft move and contaminant particles inside the narrow seal pocket stay trapped.

In a wide seal pocket, an axial grease flow has time to fully develop again, and consequently, contaminant particles may flow with the axial flow.

Grease in the vicinity of the sealing contact may also have a sealing function. Here, grease can be selected in which contaminant particles migrate to lower shear rates, to reduce the probability of contaminant particles reaching the sealing contact. Consequently, the possibility of contaminant particles moving through the sealing contact is also reduced. This sealing action is caused by the grease's rheological properties, i.e. shear thinning and elasticity, causing viscosity gradients along the particles in the vicinity of the sealing contact and normal stress differences respectively.

Chapter 5

Lubrication of the seal lip contact

Compared to the lubrication mechanisms of oil lubricated rotary shaft seals, in grease lubricated seals, additional mechanisms have to be considered. In bearing seals, the rheological properties of the grease may result in additional lift and friction from normal stress differences and shear stresses respectively. Details can be found in paper A and paper B.

5.1 Lubrication mechanisms

For oil lubricated rotary shaft seals, it is well accepted that the formation of a lubricant film between the seal and shaft is due to micro-elastohydrodynamic lubrication between micro-asperities or micro-undulations at the seal surface and shaft roughness. Gabelli and Poll [14] modelled the lubricant film formation between two parallel rough surfaces using Reynolds equation and finite difference techniques. Later, Shi and Salant [41] used a more realistic quasi-random surface roughness geometry for their simulations. Hajjam and Bonneau [17] studied the influence of different mathematical roughness models on performance features like film thickness, power loss, pumping rate and lip temperature. Their results show that the choice of roughness model largely influences the numerical results. The micro-elastohydrodynamic lubrication mechanism has been the topic for a large number of numerical studies, and is discussed in more detail in Paper A.

Schulz et al. [40] introduced a new mechanism. They assumed the oil in the seal contact to behave non-Newtonian at very high shear rates and claimed that the normal stress differences in the oil provide a load carrying capacity that is able to separate the seal lip from the shaft without the need for micro-elastohydrodynamic lubrication. For greases, Binding et al. [8] showed that the first normal stress difference in the grease is of the same order of magnitude as the shear stress. This means that normal stress differences in the grease may give a significant contribution to the force balance on the seal and shaft surfaces, i.e. formation of a lubricant film. At the same time, this grease will generate additional friction due to viscous shear.

5.2 Lubrication conditions

The presence of grease in the vicinity of the sealing contact depends on how grease is initially applied to the system, and on the shaft speed, grease type, and temperature. Consequently, the sealing function of grease as described in section 4.3 and the forces resulting from normal stress differences in the grease according to section 5.1, strongly depend on how grease is distributed in the vicinity of the sealing contact. Typical grease distributions for a bearing seal lubricated with the NLGI 2 grease are shown in Figure 33. Here grease is applied to the vicinity of the sealing contact, and subsequently, the rotational velocity of the 82 mm steel shaft is increased from 0 to 3000 rpm.

Figure 33 Lubricating conditions as a function of rotational shaft speed: (a) start initial filling, (b) below separation speed, (c) above separation speed, and (d) above critical speed.

The results in Figure 33 indicate that at low shaft speeds, below a so defined “separation speed”, the whole grease volume is being sheared in the vicinity of the sealing contact. Above this separation speed, part of the grease moves onto the stationary seal due to centrifugal forces, and part of the grease sticks to the shaft. The latter can be identified as the Weissenberg rod climbing effect where grease moves in axial direction due to normal stress differences. When the shaft speed is increased further, most the grease will move onto the stationary seal. However, still some grease may be present in the vicinity of the sealing contact.

The different lubricating conditions, as identified in Figure 33, also depend on the grease type and operating temperature and are difficult to predict due mechanical aging of the grease, wall slip, and adhesion properties.

5.3 Grease lift force

In bearing seals the rheological properties of the grease may result in additional lift from normal stress differences in the grease in the vicinity of the sealing contact.

Grease in the vicinity of the sealing contact is being sheared, and consequently, normal stress differences in the grease may result in a normal stress on the seal lip. This normal stress is similar to the normal stress in the parallel plate rheometer that leads to the normal force on the plates as discussed in Chapter 2. Such normal stresses are also present in a concentric cylinder rheometer and act in radial direction on the cylinder walls, Macosko [35]. However, in this rheometer geometry, which is quite similar to the bearing seal application, the normal stresses acting on the walls can not be determined.

Figure 34 Grease in the vicinity of the sealing contact.

In paper B, it is assumed that a normal stress, derived from normal stress difference measurements using the parallel plate rheometer, can be directly used to predict the pressure on the seal lip. The specific lift force, i.e. lift force per unit circumferential length, is then calculated using the normal stress:

$$F_{lift} = \int_{x_1}^{x_2} N_1 dx, \quad (18)$$

where N_1 is the normal stress, assumed to be equal to Eq. (5), and x_1, x_2 define the integration boundaries in the axial direction with $x_1 = 0$.

However, a new derivation, based on a recently developed view on the problem, suggests that the method to predict the specific lip force by solving Eq. (18) (paper B) is incorrect. Appendix B presents this new derivation, leading to

$$F_{lift} = - \int_{x_1}^{x_2} N_2 dx, \quad (19)$$

where N_2 is the second normal stress difference, which is negative and much smaller than the first normal stress difference N_1 as shown in Appendix C. Consequently, the lift force predicted in Eq. (19) is smaller than the lift force predicted in paper B.

The assumptions that were used in Appendix B for the derivation of Eq. (19), have not been validated. Further study on how normal stress differences in the grease generate a lift force is required before firm conclusions can be given.

5.4 Grease friction

The grease in the vicinity of the sealing contact is being sheared which results in frictional losses. The frictional torque calculation is based on the rheology model of the grease.

5.4.1 Friction model

Seal friction is the result of friction in the sealing contact and friction from the grease, which is being sheared, in the vicinity of the contact. In the sealing contact, where the shear rates are very high, the grease viscosity approaches its base oil viscosity and friction is expected to be equal to the friction in oil lubricated seals. Additional friction from shear stresses in the grease, in Figure 34, is studied here. The increase in frictional torque due to the grease in the vicinity of the sealing contact can be calculated by integrating the shear stresses as

$$M_{friction} = 2\pi r^2 \int_{x_1}^{x_2} \sigma_{zy}(x) dx, \quad (20)$$

where σ_{zy} is given by Eq. (3), and x_1, x_2 define the integration boundaries in the axial direction with $x_l = 0$. Eq. (20) can be integrated, as described in Paper B, when a linear grease velocity profile is assumed, giving

$$M_{friction} = 2\pi R^2_1 \left[\tau_y (\hat{x}_2 - \hat{x}_1) + \frac{K}{(1-n)} \left\{ \frac{U_s}{\varphi} \right\}^n (\hat{x}_2^{(1-n)} - \hat{x}_1^{(1-n)}) + \eta_\infty \left\{ \frac{U_s}{\varphi} \right\} (\ln|\hat{x}_2| - \ln|\hat{x}_1|) \right]. \quad (21)$$

Here K , n , m , λ_1 , η_∞ are the grease rheological parameters as defined in 2.4, U_s is the shaft surface velocity, and φ is the lip angle. The definitions for $\hat{x}_1 = h_o/\varphi$, and $\hat{x}_2 = (x_2 - x_1) + h_o/\varphi$ follow from the integration boundaries where h_o is the film thickness in the contact. For more details, the reader is referred to paper B.

5.4.2 Friction from the grease

Eq. (21) shows that the frictional torque is the result of three terms: the apparent yield stress of the grease, the shear thinning term and the base oil viscosity. Predictions of the frictional torque at different operating conditions, similar to those in section 5.3, are shown in Figure 35. Note that in a real seal application, the temperature will be a function of speed and will depend on the thermal characteristics and environment of the application. The results shown here are for a constant temperature.

The results in Figure 35 are non-linear due to the non-Newtonian behaviour of the grease at very high shear stresses found close to the sealing contact. At some distance away from the sealing contact, the shear rates decrease significantly, which results in an increase in viscosity. Consequently, at high shaft speeds, i.e. high shear rates, grease loss or edge fracture may occur here, like in the cone and plate and parallel plate rheometers as described in section 2.3. This grease loss or edge fracture will subsequently result in a reduction in frictional torque.

The model presented in Eq. (21) can be used to predict frictional losses as a function of the amount of grease and operating conditions when the rheological properties of the grease at the operating temperature are known.

Figure 35 Frictional torque predictions for different temperatures and lip angles α - β , $h_o = 2 \mu\text{m}$ and $U_s = 10 \text{ m/s}$. The grease height h_g indicates the amount of grease present in the vicinity of the contact as defined in Figure 34.

5.5 Conclusion

Normal stresses in the grease, due to the high shear rates in the vicinity of the sealing contact, may generate additional lift forces on the seal lip in addition to the lift generated by micro elastohydrodynamic lubrication. However, further work is required to be conclusive on the significance of a lift generated by the grease in the vicinity of the sealing contact. Frictional losses due to the grease are calculated based on the grease rheology.

The models presented in this chapter can easily be included in existing film thickness models for oil seals to make them applicable to grease lubricated seals as well.

Chapter 6

Sealing contact replenishment

In a grease lubricated seal, there is not a continuous supply of lubricant to the sealing contact, as would be the case for an oil lubricated seal. Therefore, the lubricating conditions in the sealing contact depend very much on the lubricant supply, or lubricant replenishment. Details can be found in paper E and paper F.

6.1 Replenishment mechanisms

The replenishment mechanisms of lubricating greases are very different from those of lubricating oils due to its semi-solid behaviour. Unlike oil, grease can not freely flow to the sealing contact for replenishment. Consequently, less lubricant is available for lubrication, resulting in very thin lubricant films in the contact, higher contact temperatures, and increased seal aging and wear (Dürnegger and Haas [13]). This limited contact replenishment is very similar to what happens in grease lubricated rolling bearings, as described by Lugt [34], where grease is pushed to the sides, away from the moving contact surfaces, forming an oil reservoir. For bearing seals, three ways of sealing contact replenishment are defined below.

Firstly, system dynamics like vibrations or shock loads exert forces on the grease reservoirs that are positioned on the seal and counterface. Consequently, the grease may creep or parts of the grease reservoir may break loose and fall into the sealing contact to provide replenishment.

Secondly, when grease is in contact with a moving surface, shear degradation or mechanical aging of the grease may take place as explained by e.g. Mérieux et al. [36]. Here, it is assumed that a grease layer which touches a moving surface is being sheared, and locally the thickener structure degrades. Consequently, base oil is released that may flow into the contact for lubrication.

Finally, oil can separate from the grease due to diffusion, wetting, or body forces. Body forces, resulting from centrifugal forces, especially cause oil separation from the grease. This replenishment mechanism is further explained in the following sections.

6.2 Oil bleed from grease

The solid thickener in the grease is assumed to form a microstructure which contains the oil as shown in Figure 7. The grease microstructure can release oil, which is referred to as “oil bleed”. Baker [1] found that the oil bleed rate increases with temperature and strongly depends on the grease type. Today, the standard for the oil bleed rate of greases is the DIN5187 test [12], where the amount of oil bled under a static load is measured after 168 hours. For the replenishment of the sealing contact oil bleeding will be mostly driven by centrifugal forces induced by the rotational speed of the surface the grease reservoir is positioned on. A theoretical model is developed that relates the rotational speed, base oil viscosity, thickener structure deformations, and permeability to the volumetric oil flow out of the thickener structure.

6.2.1 General flow equation

For predicting the oil bleed rate, the oil flow through a porous thickener structure is calculated based on Darcy’s law [7] which is given in the general form as

$$Q = A \frac{k}{\eta} \nabla p, \quad (22)$$

where Q is the volumetric flow rate through a cross section area A . The ability for the oil to flow through the porous thickener structure is described with the permeability constant k and the oil viscosity η . A pressure gradient ∇p drives the oil flow here. Unlike in Darcy’s original experiment, the permeability k is not constant in the oil bleed process and a body load is forcing the oil flow instead of a pressure gradient. The force balance on a small grease volume reads

$$F_{body} + F_{friction} = 0 \quad (23)$$

where F_{body} is the centrifugal body force on the oil due to the rotational speed and $F_{friction}$ is the friction force depending on the base oil viscosity, i.e. temperature, and the grease microstructure permeability.

6.2.2 Grease microstructure model

The grease microstructure has been visualized with a Scanning Electron Microscope (SEM) in Figure 7 and an Atomic Force Microscope (AFM) in Figure 36a. For the oil bleed model, the measured grease microstructure is simplified to circular, randomly ordered fibres suspended in the oil as shown in Figure 36b. Subsequently, the structure is further simplified for fresh grease as

fibres ordered in an orthogonal arrangement as shown in Figure 36c. Differentiation between greases is made by the characteristic dimensions of the microstructure, i.e. fibre diameter, fibre volume fraction in the grease, and the base oil viscosity. This is described in further detail in paper E.

Figure 36 Grease microstructure, (a) AFM image of a lithium complex soap network, (b) 1st simplification to rigid fibres, and (c) 2nd simplification to orthogonally arranged rigid fibres.

As soon as oil starts to bleed out of the microstructure, the total grease volume must decrease. This means that also the microstructure has to be compressed, which can be done in two ways. Firstly, the distance between fibres can be reduced while the orthogonal arrangement of fibres is maintained, i.e. by maintaining the isotropic microstructure. Secondly, it can be assumed that the fibres that are initially parallel to the oil flow gradually tilt and finally become perpendicular to the oil flow, resulting in an anisotropic microstructure. The deformation of the microstructure is modelled as

$$k = \frac{2}{3}k_{\perp} + \frac{1}{3}k(\theta), \quad (24)$$

where $k_{\perp}$ is the permeability perpendicular to a grid of uniaxial aligned fibres and $k(\theta)$ is the permeability of a grid of uniaxial aligned fibres which are tilted with an angle θ . The tilting angle depends on the amount of oil that has already bled out of the grease. k is the total permeability of the grease microstructure. For fresh grease, or for the case where the microstructure is assumed to be isotropic, the term $k(\theta)$ in Eq. (24) is constant, as for a grid of uniaxial fibres aligned parallel to the flow.

6.2.3 Oil bleed model

With Eq. (24) substituted into (22) and including the centrifugal body load, the oil bleed rate can be calculated as a function of the fibre volume fraction f of the grease and base oil viscosity as

$$Q_{bleed} = -A \frac{k(f)}{\eta(T)} \rho \omega^2 r, \quad (25)$$

where A is the outflow area and R_o and r are the radii of the grease reservoir. A numerical integration scheme is used to predict the volumetric oil separation over time as a function of rotational speed and temperature. The oil bleed model is validated with experiments. The experimental setup consists of a small 20 mm diameter cylindrical cup that is filled with 10 grams of grease. This is shown in Figure 37. The cup is rotating and oil is forced out through the flat outer end of the cylinder. A filter and sieve make sure that only oil flows out and thickener remains in the cup.

Figure 37 Schematic view of the grease cup in the experimental oil bleed setup for dynamic measurements

Oil bleed experiments and model predictions are done for a lithium complex / mineral oil bearing grease LGM. Initial values for the model calculation are given in Table 4. Figure 38 shows the experimental results and the model predictions. The percentage of oil loss is plotted against time. Good agreement is found between the model predictions, at different temperatures and rotational speeds, and the experimental results.

The model results show that the oil bleed rate is largest at the start of the experiment, when the grease still contains most of its oil. Due to the oil loss, the fibre volume fraction will increase and the grease microstructure becomes denser, reducing the permeability and slowing down the oil flow.

Consequently, the bleeding rate reduces and stops when the grease has lost approximately 60% of its oil. In the model, this is due to the fact that the fibres in the microstructure start to touch and completely block the oil flow. Physically, this will not happen in the real soap microstructure but the microstructure will become so dense that capillary forces hold the last 30-40% of the oil inside. Consequently, the grease is no longer able to lubricate the system as also described by Cann et al. [10].

Parameter	Value	Unit
D_f	$0.2 \cdot 10^{-6}$	m
f_o	0.26	-
η_{40}	0.10	Pa s
η_{100}	0.011	Pa s
ρ	900	Kg/m ³

Table 4 LGM grease properties for oil bleed predictions

(a)

(b)

Figure 38 Oil loss from LGM grease due to oil bleed of an anisotropic grease microstructure where fibres that are initially parallel then tilt to become perpendicular to the pressure gradient or oil flow, (a) at varying temperature and constant speed of 3000 rpm and (b) at varying speed at a temperature of 120 °C. The symbols represent the experiments, and the lines represent the model.

6.3 Film thickness in axial seal contact

The oil bleed model is applied to grease in the vicinity of an axial seal contact and the film thickness in the contact is predicted over time. The model for the film thickness is highly simplified, but the results can be used to predict the time at which the seal starts to run in the mixed lubrication regime.

6.3.1 Film thickness model

In grease lubricated seals, the lubricating condition or film thickness is assumed to be determined by the availability of lubricant near the sealing contact. After an initial churning phase, the grease may form an oil reservoir on the seal and/or counter surface as indicated in Figure 39, and slowly release oil for lubrication towards the sealing contact.

Figure 39 A grease lubricated seal with; axial sealing lip contact indicated in box a, and a grease reservoir indicated in box b.

A model is developed to predict the film thickness in the sealing contact as a function of time. The model solves a mass balance of oil feed to and loss from the contact. Here, the oil loss from the contact is due to seal pumping and centrifugal forces acting on the oil film in the gap as indicated in Figure 40. Oil feed to the contact is the result of oil bleed from the grease reservoir on the rotating part, and schematically presented in Figure 41. The resulting film thickness is given by

$$h = \frac{\int_0^t (Q_{feed} - Q_{loss}) dt + V_o}{2\pi R_c b}, \quad (26)$$

where V_o is the initial volume of oil in the contact and Q_{feed} and Q_{loss} the flow rates of the oil feed and oil loss respectively. R_c is the radial position of the contact and b is the contact width assuming a uniform oil film thickness.

Figure 40 Dimensions of the sealing contact with (a) oil loss due to seal pumping and (b) ingested meniscus and oil loss due to centrifugal body load.

Oil loss due to natural pumping of the seal is predicted using the empirical equations from Horve [21]. These equations have been derived from extensive experimental work on oil lubricated radial shaft seals, of which both sides of the sealing contact are fully flooded with oil. It is assumed here that these equations can also be applied to an axial seal contact given that the seal geometry, i.e. lip angles, and seal material are similar to the ones in [21]. Horves equation for the seal pump rate reads

$$Q_{pump} = 1.04 \cdot 10^{-8} R_c^3 \frac{n}{60} G^{-1/3}, \quad (27)$$

where R_c is the radius of the sealing contact, n the shaft speed (rpm), and $G = 2\pi b n \eta / (60 F_{lip})$ a dimensionless duty parameter. From the pump rate in Eq. (27) the film thickness can also be calculated. This is the film thickness under fully flooded conditions, and consequently, the maximum film thickness possible. Stakenborg [43] and Salant [38] showed that as soon as all the oil is pumped from the air side to the oil side of the seal, the oil meniscus will ingest into the sealing contact as shown in Figure 40b. When this happens, pumping due to the asymmetric pressure distribution, as described by Kammüller [27], will no longer take place. The oil loss from seal pumping is found to be dominant as long as the sealing contact is fully flooded with oil. However, as soon as the meniscus is ingested into the sealing contact, seal pumping will no longer occur and subsequent oil loss only takes place due to centrifugal forces on the oil film in the sealing contact.

Figure 41 Dimensions of the grease reservoir on the rotating part, after the churning phase.

The oil film in the sealing contact is sheared in the circumferential direction, and consequently, the centrifugal forces on the oil vary from zero at the stationary surface to the maximum value at the rotating surface. The body force and flow equation are defined in Paper F which, after integration over the contact height, give the oil loss from body forces as

$$Q_{body} = 2\pi R_c \frac{\rho \omega^2 R_c}{40\eta} h^3, \quad (28)$$

with ρ the oil density, ω the angular shaft velocity and R_c the radial position of the contact.

To summarize, in the theoretical model, three phases of operating conditions can be identified.

- Phase 1: In the churning phase, the grease is being churned and the grease reservoirs are formed. The film thickness in the contact is equal to the maximum film thickness h_{max} .
- Phase 2: Oil bleed from the grease reservoir supplies oil to the sealing contact. At the same time, oil is lost from the contact due to seal pumping as long as fully flooded conditions remain.
- Phase 3: The maximum film thickness can not be maintained due to insufficient replenishment of the contact. The oil meniscus is ingested into the sealing contact and seal pumping stops. Oil loss from the contact is continued by the body forces and the film thickness decreases slowly over time.

6.3.2 Film thickness predictions

Film thickness predictions are done for the same lithium complex / mineral oil grease LGM as used in section 6.2. To predict the film thickness in the sealing contact as a function of time Eq. (26) is numerically integrated. Typical model results, based on the grease properties in Table 4 and the initial parameter settings from Table 5, are presented in Figure 42 for various operating conditions. Here, the maximum film thickness depends on the rotational speed and the operating temperature, which affects the base oil viscosity. For the case that a grease reservoir of $AH_o = 0.5 \text{ mm}^2$ is present, the maximum film thickness remains constant for a certain period of time. At some moment, the film thickness starts to decrease due to the limited replenishment of the sealing contact. The curves for the two speeds at 70°C show that at lower speeds, the maximum film thickness is smaller but can be maintained for a much longer period of time. After all, at low speed, the oil loss from the sealing contact is much lower than at high speed. Figure 42 also shows the predicted film thickness without the presence of a grease reservoir. Here, the film thickness starts to decrease instantaneously and it takes a significantly shorter time to reach the critical film thickness, $h_{crit} = 0.25 \text{ }\mu\text{m}$, equal to the counterface surface roughness. The critical film thickness may be used as an indication for the transition from full film lubrication to mixed lubrication regimes where some direct contact between the seal and counterface may occur.

Figure 42 Oil film thickness in the sealing contact with LGM grease: starting from fully flooded conditions.

Parameter	Value	Unit
A_r	$0.5 \cdot 10^{-6}$	m^2
b	$0.4 \cdot 10^{-3}$	m
F_{Lip}	20	N/m
H_o/W	2	-
R_c	$44 \cdot 10^{-3}$	m
R_o	$42 \cdot 10^{-3}$	m

Table 5 Initial values for the grease and seal parameters for film thickness predictions

A simple engineering model is made, based on a characteristic parameter set in the oil bleeding model in Eq. (25) and in the oil loss model in Eq. (28). The characteristic term, $\eta/(n^2 \cdot d_s)$, is defined where $d_s = 2R_c$ is the diameter of the sealing contact. In the engineering model, the influence of the size of the grease reservoir is also included by fitting the engineering model to results from the physical model of Eq. (26). Consequently, the engineering model reads

$$t_c = [C_1 A_r + C_2] \left[\frac{\eta}{(n^2 \cdot d_s)} \right], \quad (29)$$

where the two fitting parameters for this particular grease type read $C_1 = 2 \cdot 10^{15} (\text{kg} \cdot \text{m})^{-1}$, $C_2 = 6 \cdot 10^6 \text{ kg}^{-1} \cdot \text{m}$, and $A_r = WH_o$ provided that $H_o/W = 2$. The seal size is included in d_s , the base oil viscosity and operating temperature in η and the rotational speed in n . Figure 43 shows how the results of the physical model from Eq. (26) become a straight line when scaled with the characteristic term $\eta/(n^2 \cdot d_s)$. In this Figure, the continuous line is the engineering model. The simplified engineering model can be used to predict the time until the mixed lubrication regime is reached.

For a radial seal contact, oil feed and oil loss will take place according to the current model. Consequently, the film thickness that is formed during the churning phase will remain for an infinite time. Here addition oil feed and loss mechanisms have to be included like e.g. loss due to evaporation of oil. These mechanisms are expected to act on a much longer time scale and are therefore assumed to be insignificant for the film thickness in the axial seal contact.

It is possible to include a more advanced pumping model in the physical model, which includes the actual seal lip geometry. However, since the pumping phase is short compared to the phase where the film thickness decays, and since the maximum film thickness is at least a few times higher than the critical film thickness, the pumping rate has little effect on the predicted critical time until

mixed lubrication is reached. Therefore, the present engineering model can also be used for other seal materials and geometries.

Figure 43 Operating time until critical film thickness is reached based on the $\eta/(n^2 \cdot d_s)$ parameter for $A_r = 0.5 \text{ mm}^2$.

6.4 Conclusion

The oil bleeding of grease has been modelled as viscous flow through a porous microstructure which is subjected to centrifugal body forces. The permeability of the grease microstructure has been assumed homogeneous and has been derived for both an isotropic and an anisotropic fibre arrangement. The model results show good agreement with experimental results.

A model has been presented to predict the lubricant film thickness in the sealing contact, taking into consideration replenishment of the contact from grease reservoirs. Here, the grease reservoir can only supply oil for a limited period of time, and subsequently, the film thickness decreases due to centrifugal forces acting on the oil layer. A simplified engineering model has been derived to calculate the time until mixed lubrication conditions are reached. The models presented in this Chapter can be used as part of seal life prediction models or grease selection.

Chapter 7

Conclusions and future work

The lubrication mechanisms and sealing function of grease in bearing seals have been studied. The main conclusions are summarized in this Chapter and recommendations for future research are also given.

7.1 Main conclusions

Detailed conclusions regarding the various subjects can be found at the end of each Chapter. In this section, the main conclusions are summarized.

Grease lubrication in seals:

- In the past, research has mainly been focussed on oil lubricated seals, and very little is known about the grease lubrication mechanisms in bearing seals.
- The semi-solid character and rheological properties of the grease have a significant impact on the lubrication conditions relative to oil lubricated seals.

Grease flow in seal pocket:

- In a *wide* seal pocket, the grease flow can be considered 1-dimensional and the velocity profile is approximately linear due to the small pocket height relative to the shaft radius.
- In a *narrow* seal pocket, the side walls significantly influence the velocity profile and some grease is unyielded at the outer radius.
- Identical velocity profiles are obtained in the measurement plane for different NLGI grade greases at different temperatures, indicating that a single parameter can be used to describe the shear thinning behaviour of grease.
- In the *narrow* seal pocket, grease flow in the axial direction, due to a pressure difference, mainly takes place close to the shaft surface.

Contaminant migration in grease:

- In the *wide* seal pocket, radial contaminant migration does not significantly depend on grease type. In the *narrow* seal pocket, radial contaminant migration is smallest for the highest NLGI grade grease.
- In the vicinity of the sealing contact, contaminant particles in the grease consistently migrate away or towards the sealing contact depending on the grease's rheological properties.
- A pressure gradient in the *wide* seal pocket forces all contaminant particles in the grease volume to move in the axial direction. A pressure gradient in the *narrow* seal pocket only forces contaminant particles which are close to the shaft to move in axial direction.
- The sealing function of grease can be ascribed to the migration behaviour of contaminant particles through the grease.

Lubrication of the seal lip contact:

- Normal stresses in the grease, due to the high shear rates in the vicinity of the sealing contact, may generate an additional lift force on the seal lip which is in addition to the lift generated by micro elastohydrodynamic lubrication.
- The grease in the vicinity of the sealing contact contributes to the total frictional torque.

Sealing contact replenishment:

- Oil separation, or oil bleed, from the grease can be modelled as viscous flow through a porous grease microstructure subjected to centrifugal body forces. Here, the grease microstructure changes from an isotropic network to an anisotropic network of soap fibres and oil bleed stops when $\sim 60\%$ of the oil is lost.
- The time until the mixed lubrication regime is reached, in an axial sealing contact, can be modelled with a simplified engineering model, which scales with the characteristic term $\eta/(n^2 \cdot d_s)$. The size of the "grease reservoir" has a large impact on the lubricating conditions while the impact of the natural pumping action of the seal is negligible.

7.2 Recommendations for future work

Based on the results obtained from this PhD work, some recommendations for future research are given in this section.

Grease lubrication in seals:

- Aging or mechanical degradation of the grease in the sealing application could significantly change the rheological properties of the grease in time. Further research taking in to consideration the mechanical degradation of grease is recommended to be able to understand the performance of grease over longer periods of time.

Grease flow in seal pockets:

- The grease velocity profile in the narrow grease pocket has been studied experimentally. However, the 3D velocity of the grease also depends on the geometrical shape of the grease pocket. Further experimental studies are recommended to better understand this geometrical effect. The experimental results of the grease velocity measurements, can also be used for validation of 3-dimensional numerical models to predict flow properties of the grease in complex geometries.

Contaminant migration in grease:

- Experiments are recommended to validate the radial migration model. More advanced numerical models, which include the complexity of the geometry and the grease, are recommended to be able to do a quantitative prediction of axial migration of contaminant particles in the vicinity of the contact.

Lubrication of the seal lip contact:

- Further work is required to be conclusive on the significance of a lift generated by the grease in the vicinity of the sealing contact. The amount of grease, present in the vicinity of the contact, and the rheological properties change due to mechanical degradation of the grease. Further work on the grease distribution and operating conditions are recommended for reliable predictions over a longer period of time.

Sealing contact replenishment:

- Oil bleed from the “grease reservoir” initiates replenishment to the sealing contact. It is recommended to further study the formation of these grease reservoirs and how they are maintained over time, depending on operating conditions. Further improvement of the model can be achieved by extending the bleeding model with secondary effects.

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Appendix A

This appendix explains some general grease properties and summarizes the lubricants that are used in this study.

Grease properties

Most greases selected for this work are typical rolling bearing greases. These greases have a consistency of the NLGI 1, NLGI 2 or NLGI 3 grade. Additionally a much softer grease of the NLGI 00 grade is used. The description and use of the NLGI grades is indicated in Table A1.

Consistency	Description	Application Example
NLGI 00	Semi-fluid grease for (open) gears	Open gear boxes
NLGI 1	Soft grease for low temperature and improved pumpability	Systems including centralized lubrication systems
NLGI 2	Normal grease for rolling bearings	Multi-purpose grease, wheel bearing units
NLGI 3	Firm grease for high temperature and high speed	Electrical motor bearings

Table A1 Definition of directions in the parallel plate rheometer and lip seal.

Beside the consistency there are several other properties that may vary between greases as indicated below

- *Thickener type*; e.g. lithium, lithium complex, calcium, polyurea for different maximum (melting) temperatures and different water resistance properties.
- *Base oil viscosity*; typically around 0.2-0.5 Pa·s at 25 °C but can be selected different for extreme conditions.
- *Base oil type*; e.g. mineral oil, PAO, ester oil for different temperature conditions.
- *Additives*; e.g. extreme pressure, anti-wear, friction modifiers, anti-oxidant, viscosity improvers for reliable lubrication in extreme conditions, to extend bearing/lubricant life, and reduce friction.
- *Yield stress*; minimum stress that has to be overcome before the grease starts to flow.

- *Tackiness*; adhesive and cohesive properties of grease, often evaluated with the “finger test” where a tacky grease generates long strings when pulled apart between two fingers.
- *Oil bleed*; separation of the base oil from the thickener structure which depends on the composition of the grease and operating conditions, e.g. temperature.

These properties determine the performance of a grease in a specific application. Consequently, many different greases for specific applications and operating conditions exist. Some of the properties mentioned above are specified or can easily be measured. Other properties, e.g. the exact formulation of the additive package, are only known by the grease manufacturer.

Grease lubricants

The properties of the greases that are used in the thesis are summarized in Table A2.

Name	Thickener/base oil	NLGI grade	$\eta_{\text{base oil at } 25^\circ\text{C}}$ [Pa·s]	Tacky [finger]
D ¹	Lithium/Mineral	2	0.17 ²	+
NLG100	Lithium-silicate/mineral	00	0.89	+
NLG11	Lithium/mineral	1	0.49	+
NLG12 ¹	Lithium/Mineral	2	0.25	+
LG1 ¹	Lithium/Mineral	2	0.25	+
LG2	Lithium/Mineral	2	0.45	++
LG3	Synthetic/PAO	2	0.03	-
LG4	Lithium/Mineral	1-2	0.21	-
LG5	Lithium/Mineral	2-3	0.33	+
LGM	Lithium/Mineral	2-3	0.24	-

Table A2 Greases used in this work and their main properties. The base oil viscosity at 25 °C is calculated using the Walther equation [39].

¹ This is the same grease and is used as reference grease.

² Base oil viscosity calculated with the Arrhenius model [2].

Oil lubricants

The base oil, which was extracted from the LG1 grease, using a centrifuge and filter, is specified in Table A3. Therefore, base oil including additives was obtained.

Name	Base oil	η at 25 °C [Pa·s]	Tacky [finger]
Base oil LG1	Mineral	0.25	--

Table A3 Oils used in this work and their main properties.

Other lubricants

A very elastic fluid, i.e. fluid which generates very high normal stress differences, was made by dissolving 0.5 wt% Poly(Ethylene) Oxide (PEO, $M_w \sim 8,000,000$) in water and is specified in Table A4.

Name	Composition	η at 25 °C [Pa·s]	Tacky [finger]
PEO	Water/PEO	0.04	++

Table A4 Other lubricants used in this work and their main properties.

Appendix B

This appendix presents the derivation of the predicted lift force by the grease that is being sheared in the vicinity of the contact.

Equations of motion

For the seal lip geometry the directions of motion are defined in Table B1.

Direction	Description	Parallel plates	Lip seal
1	Main flow direction	Circumferential direction	Circumferential direction, θ
2	Direction of varying velocity	Axial direction	Radial direction, r
3	Remaining direction	Radial direction	Axial direction, x

Table B1 Definition of directions in the parallel plate rheometer and lip seal.

The equations of motion in cylindrical coordinates are defined as

$$r: \quad 0 = \frac{1}{r} \frac{\partial}{\partial r} (r \sigma_{rr}) - \frac{\sigma_{\theta\theta}}{r}, \quad (\text{B1})$$

$$\theta: \quad 0 = \frac{1}{r^2} \frac{\partial}{\partial r} (r^2 \sigma_{r\theta}) + \frac{\partial \sigma_{\theta x}}{\partial x} + \frac{\sigma_{\theta r} - \sigma_{r\theta}}{r}, \quad (\text{B2})$$

$$x: \quad 0 = \frac{1}{r} \frac{\partial}{\partial r} (r \sigma_{rx}) + \frac{\partial \sigma_{xx}}{\partial x}, \quad (\text{B3})$$

where it is assumed that flow variations in circumferential direction are zero and no inertial effects are present. Inertia effects can relatively easily be included but are left out in the derivation for the lift force due to normal stresses. The normal stress differences are defined as

$$N_1 = \sigma_{\theta\theta} - \sigma_{rr}, \quad (\text{B4})$$

$$N_2 = \sigma_{rr} - \sigma_{xx}, \quad (\text{B5})$$

Specific lift force

To calculate the pressure that is acting on the seal lip the stress σ_{rr} will be derived.

It is assumed that the main flow takes place in the circumferential direction θ and the main shear stress is $\sigma_{r\theta}$. It is further assumed that the lip angle is sufficiently small that no other shear stresses are present such that

$$\sigma_{\theta\theta} = \sigma_{\theta r} = \sigma_{rx} = 0. \quad (\text{B6})$$

Consequently, according to Eq. (B3), the gradient of σ_{xx} in the x direction is zero. Since at the free boundary of the grease with the air σ_{xx} is balanced by the ambient pressure, and if it can be assumed that there is no surface tension or other edge effect, the stress $\sigma_{xx} = 0$ throughout the grease volume. Substituting this in Eq. (B7) gives the stress σ_{rr} as

$$\sigma_{rr} = N_2, \quad (\text{B7})$$

To calculate the specific lift force, the fluid pressure $p = -\sigma_{rr}$ can be integrated over the seal width resulting in

$$F_{lift} = - \int_{x_1}^{x_2} N_2 dx, \quad (\text{B8})$$

where x_1 and x_2 are the boundaries between which the grease is present. Here N_2 is a function of the shear rate and therefore varies in the x direction where the gap height between the shaft and the seal is varying. Note that the second normal stress difference N_2 is negative and typically much smaller than the first normal stress difference N_1 . Consequently a small positive lift force is generated.

Appendix C

This appendix presents the determination of the second normal stress difference N_2 based on rheology experiments on the cone and plate and parallel plate rheometer.

Experimental method

The second normal stress difference N_2 can be determined from experiments on the parallel plate and cone and plate geometry. The cone and plate geometry can be used to measure N_I up to shear rates limited by the maximum rotational velocity of the rheometer and/or by grease loss due to a relative large gap at the rim. The parallel plate geometry can be used to measure N_I - N_2 up to much higher shear rates by reducing the gap between the plates. Subtracting the cone and plate result from the parallel plate result then gives the second normal stress difference N_2 .

The NLGI 2 grease from Appendix A Table A2 is used. For the experiment with the cone and plate geometry a 40 mm diameter cone is used with an angle of 4°. Consequently, the gap at the edge of the geometry is ~1.4 mm and edge effects, e.g. grease loss, may be expected at high shear rates. For the parallel plate geometry 25 mm diameter plates are used with a 0.5 mm gap setting. Consequently higher shear rates can be obtained than with the cone and plate geometry. Two experiments are done with each rheometer geometry to check the reproducibility of the results.

Experimental results

In Figure C1 the results of the shear stress measurement for both rheometer geometries are plotted to check that both geometries give the same result. Subsequently, in Figure C2 the normal stress difference results, i.e. N_I for the cone and plate geometry and N_I - N_2 for the parallel plate geometry, are presented. The results show some noise for the parallel plate measurements which is introduced by the differential term in the equation to calculate N_I - N_2 from the normal force. In the cone and plate measurements the slope of N_I decreases at high shear rates which is due to edge effects, mainly grease loss at the edge.

Figure C3 shows the first normal stress differences where it is assumed that $N_2 = -0.15 N_I$ in the parallel plate geometry such that N_I can be calculated from the N_I - N_2 measurement. From the experimental results it can be concluded that N_2 exists and that N_2 is about 15% of $-N_I$.

Figure C1 Shear stress as a function of shear rate at a temperature of 25 °C.

Figure C2 Normal stress differences as a function of shear rate at a temperature of 25 °C.

Figure C3 First normal stress differences as a function of shear rate at a temperature of 25 °C.

Paper A

Review of the lubrication, sealing, and pumping mechanisms in oil- and grease-lubricated radial lip seals

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Abstract: Radial lip seals are successfully used since the 1940s to seal lubricated systems. Despite extensive experimental and theoretical research in the field, it is still not fully clear how these seals function. Experimental studies, found in the public literature, show that the relatively high surface roughness of the seal lip is very important for good and reliable performance. In addition, the pressure distribution under the lip seems to be a critical factor. Six fundamental hypotheses are presented on the lubrication, pumping, and sealing mechanisms to explain the working principles of these seals. It is generally accepted that lubrication results from micro-elastohydrodynamic film build up between the rough seal surface and the shaft. Non-symmetrical tangential deformations of the lip surface are observed during experiments and assumed to act like spiral groove bearings that generate a pumping action and lubricant film. Another hypothesis suggests that the lubricant will behave non-Newtonian under the very high shear rates experienced in operating conditions. This will reduce friction because of shear-thinning and enhances sealing. Macroscopic aids, like hydrodynamic pumping aids and engineered asperity patterns on the shaft, do improve seal performance. Almost all public literature discusses oil-lubricated radial lip seals while many seals are grease-lubricated, especially in certain technical fields. Due to the non-Newtonian behaviour of grease, the lubrication, sealing, and pumping mechanisms are assumed to differ from the oil-lubricated seals. Lower friction and improved protection against contamination are measured, and it is expected that the interest in grease lubrication will rapidly grow in future.

Keywords: radial lip seal, grease lubrication, pumping, sealing, non-Newtonian

1 INTRODUCTION

Radial lip seals are widely used in industry to retain lubricant and exclude contamination in rotating shaft and bearing applications. Perfect sealing can be obtained by full contact between the seal and the shaft or bearing inner ring. However, with a rotating shaft, this full contact will lead to high friction and wear. A lubricant film between the seal and the shaft will reduce both friction and wear but may lead to leakage of the lubricant into the environment. Well-designed radial lip seals show low friction, low wear, and no

leakage with a small lubricant film between the seal lip and the shaft. The thickness of this film should be large enough to minimize wear but thin enough to ensure proper sealing against contamination. Although radial lip seals are used since the 1940s, the mechanisms of lubrication and sealing are still poorly understood.

A typical example of a radial lip seal is shown in Fig. 1. The seal is mounted on a shaft with an interference fit. A garter spring is used to ensure that the seal lip is pressed against the shaft with a constant load throughout the seal life. The lip is characterized by two angles: the air-side angle α and the lubricant-side angle β . Seal designers know by experience that by selecting the air-side angle smaller than the lubricant-side angle, the seal keeps the lubricant inside the system.

In the public literature, three mechanisms have been identified to explain the performance of radial

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Fig. 1 Arrangement of a radial lip seal on a rotating shaft

lip seals: the lubrication, sealing, and pumping mechanism [1]. The lubrication mechanism generates a thin lubricant film between the lip and the rotating shaft. This film separates the lip from the shaft and will minimize friction and wear. The sealing mechanism provides static and dynamic sealing and keeps the lubricant inside and contamination outside the system despite the small lubricant film. The pumping mechanism ensures that the lubricant will be pumped from the air side back to the lubricant side. This pumping compensates for natural leakage.

In the public literature, many hypotheses can be found that explain these three mechanisms [2]. However, none of them gives a conclusive understanding of how radial lip seals work. Especially in relation to grease lubrication, very little is known.

The aim of this article is to critically review the lubrication, sealing, and pumping mechanisms for oil- and grease-lubricated radial lip seals. Therefore, first the performance of oil seals is discussed with the results from public experimental research and observations. Next, different hypotheses on the microscale contact level are identified to explain the lubrication, sealing, and pumping mechanisms. On the macroscale of the lip, mechanisms have been found that enhance the pumping action of radial lip seals. Grease-lubricated seals are introduced and studied in comparison with oil-lubricated seals. Finally, the various hypotheses are discussed with respect to oil and grease lubrication.

2 EXPERIMENTAL RESEARCH AND OBSERVATIONS

Although rubber seals are used since the 1940s, the research on these seals just started in 1957 with the work of Jagger [3]. Jagger [3] and Hirano and Ishiwata [4] tried to identify the mechanisms that explain the sealing action and the long life of the seal. This early work was mainly experimental and was continued by Brink and Horve [5], Nakamura and Kawahara [6], Horve [7], and Müller [8]. In the 1990s, the first numerical studies were done by Gabelli [9] and Gabelli and Poll [10]. Salant and co-workers [11–17] continued with many other numerical studies to identify the radial lip seal operating mechanisms.

Despite all the intensive work of many researchers, no conclusive mechanisms have been identified that fully explain the lubrication, sealing, and pumping mechanisms of radial lip seals. However, there seems to be consensus on a few seal properties that were observed during experiments and they will be discussed in this section:

- (a) there may exist a lubricating film under the lip;
- (b) successful seals pump lubricant from the air side to the lubricant side;
- (c) the microgeometry of the lip surface plays an important role;
- (d) the macrogeometry of the lip plays an important role;
- (e) seal life depends on lip temperature.

(a) In 1957 Jagger [3] proved the existence of a lubricant film between the lip and the shaft. He noticed that the dry friction was much higher than the friction when the seal was lubricated with oil. Further experiments showed that the lubricant film remains when increasing the radial load of the lip on the shaft. The existence of the lubricant film has subsequently been confirmed by many other researchers. Gabelli and Poll [18] measured a film thickness of 2.5–5.5 μm at 0.035–0.35 m/s. Later, Van Leeuwen and Wolfert [19] measured a (smaller) film thickness of 1–3.5 μm in the same speed interval. The classical lubrication theory cannot explain this lubricant film and the load-carrying ability of the lubricant film since there is no 'wedge' between the parallel shaft surface and the seal that may cause hydrodynamic pressure build up [20].

(b) One would expect the seal to leak lubricant from the high-pressure lubricant film between the lip and the shaft to the air side. However, a successful seal does not leak and is even able to pump the lubricant from the air side back to the lubricant side of the seal. This phenomenon is called the pumping mechanism and increases with shaft speed, shown by Kawahara and Hirabayashi [21] and Horve [22]. Horve [22] has also shown that this pumping or sealing ability is very much related to the surface roughness or microgeometry of the seal (see Fig. 2).

(c) Horve [22] showed that a successful seal has a high surface roughness that is much higher than the roughness of the shaft. A new moulded seal does not have this high roughness and needs a running-in time to wear-off the rubber skin and become rough. A shaft roughness of 0.25–0.50 μm in averaged height (Ra) is critical here; too low shaft roughness does not run in the seal and a too high shaft roughness gives an excessive amount of wear and decreases seal life. The high seal roughness after running-in is critical for reliable seal performance.

The seal surface after running-in is characterized by the resulting roughness pattern. Müller [8] found that some seals show axially extended undulations, while

Fig. 2 Seal surface roughness: a rough non-leaking seal (left) and a new leaking seal (right) [22] (reprinted with permission from SAE paper 910530 © 1991 SAE International)

Fig. 3 Roughness geometries: irregular microasperities (left) and axially extended undulations (right). The arrow indicates the sliding direction

other seals show irregular microasperities over the contact surface (see Fig. 3). The development of this roughness pattern depends very much on the choice of seal material. Paige and Stephens [23] showed that the specific conditions under which the seal operates also have a large influence on the formation of the roughness pattern and the formation of microasperities. They also showed that it is not only the rubber skin that wears-off during running-in, but it is also the tip of the lip that will wear-off and flatten. Horve [22] already showed that the contact width must be wide enough to have a sufficiently large amount of undulations or microasperities to seal the gap.

(d) The macrogeometry of the lip is designed such that the location of the maximum pressure under the lip is closer to the lubricant side than the air side. This is accomplished by choosing the right air-side and lubricant-side angles and the location of the garter spring (if present). The elasticity of the rubber material allows for some shaft eccentricity or dynamic run-out. Finite-element models are often used to calculate global lip deformations and the pressure distribution under the lip.

(e) Lip temperature is a very important parameter in determining seal life. High temperatures reduce seal life because of ageing, swelling, and coking of the rubber material. High temperatures are enhanced by high sump temperatures and frictional heat between the lip and the shaft. For example, high oil

viscosity, misalignment of the seal, and high speeds result in high friction and heat development resulting in a shorter seal life. The seal life can be expected to double for every 14 °C decrease in sump temperature [7]. Moreover, heat dissipation from the seal contact and heat generation from other machine components are very important. This explains why identical seals, in different locations on the same shaft, may exhibit considerable differences in sealing performance and life [24].

3 MICROSCOPIC LUBRICATION, SEALING, AND PUMPING

A number of hypotheses have been presented in the public literature to explain the lubrication, sealing, and pumping effect of conventional oil-lubricated radial lip seals. These hypotheses are listed in this section and subsequently described in some detail.

3.1 Surface tension and contact angle

Jagger [3] developed a theory on the effect of surface tension to understand the sealing action of seals. In his experiments, he measured the leakage of the seal against a static head of oil. He discussed the formation of a lubricant meniscus at the air side of the seal to be the sealing mechanism. This meniscus is a direct result of the surface tension at the lubricant–air interface. Jagger [3] added chemicals to reduce the surface tension of the lubricant but found no change in the sealing ability of the seal. Then he flooded both the lubricant side and air side of the seal with oil to remove the meniscus. This gave rise to leakage and he concluded that the meniscus is the mechanism for sealing. In a later study by Jagger and Wallace [25], they also discussed the contact angle between the lubricant, air, and seal. They investigated the variation in leakage

using an oil–water interface instead of oil–air interface. They then assumed that the observations also apply to an oil–air interface. As in the previous study by Jagger [3], they again concluded that the formation of the meniscus explains the sealing action of the seal.

More recent studies by Müller [8] and Horve [22] show that the seal actually pumps lubricant from the low-angle (air) side to the high-angle (lubricant) side when both sides of the seal are flooded with oil and no meniscus is present. Jagger and Wallace [25] did observe this pumping in their experiments but they were not able to explain it.

Rajakovics [26] tried to explain the effect of surface tension with an analytical study. He calculated the capillary force in the gap. The capillary force is determined by the surface tension, contact angle, and gap height. He claimed that the fluid viscosity and contact width are irrelevant to sealing since they do not contribute to the capillary force expression. Müller [8] and Horve [22] showed that the fluid viscosity and gap width after running-in are of great importance on seal performance. Rajakovics [26] and Jagger and Wallace [25] could not explain the difference in performance of a new seal and a seal that had been run in.

Stakenburg [24] observed the wetted area under the seal lip through a stationary shaft where the seal was rotating. He distinguished the contact area, wetted area at the lubricant side, and the wetted area at the air side of the contact. Under static conditions, the wetted area at the air side is much bigger than that at the lubricant side because of the capillary forces. With a rotating seal, the wetted area at the air side decreases due to the pumping action of the seal. Now, under steady-state conditions, there will be a balance between the pumping action of the seal and the capillary force. Increasing the rotational speed will finally cause cavitation at the sides of the contact area. Stakenburg [24] performed the experiments under realistic operating conditions. Most other studies were carried out under pumping conditions with a reservoir of oil at the low-angle side of the seal but that system does not represent the normal steady-state running conditions of the radial lip seal.

3.2 Micro-elastohydrodynamic lubrication

In 1965 Hirano and Ishiwata [4] suggested microhydrodynamic lubrication between a rough shaft and a smooth seal to be the lubrication mechanism. They applied the foil-bearing theory in a microscopic scale, taking into account the effect of shaft surface roughness. A hydrodynamic film will build up when the roughness asperities move relative to the smooth elastic seal surface. One year later, in 1966, Jagger and Walker [27] claimed that the seal surface is rough rather than the shaft surface. Also in 1966 Hamilton *et al.* [28] presented an experimental and

analytical study and confirmed that load support can be obtained with this microhydrodynamic lubrication theory. They manufactured surfaces with different microasperity heights and modelled the lubricant film using Reynolds equation.

The first significant numerical studies were performed by Gabelli [9] and Gabelli and Poll [10]. Gabelli [9] modelled the lubricant film formation between two parallel rough surfaces, the seal and the shaft, using the Reynolds equation and finite-difference techniques. In his model, the load is carried by both hydrodynamic pressure and microasperity contact. Gabelli [9] has shown that his model is an efficient tool for evaluating the influence of roughness geometry on the lubrication of radial lip seals. Gabelli and Poll [10] also take into account the visco-elastic bulk effects of the rubber lip on the lubricant film pressure. Here, the load-carrying capacity is the sum of the hydrodynamic pressure component, the microasperity contact, and a squeeze-film component, which includes the dynamic response of the rubber lip.

Salant and co-workers [11–17] published a series of papers including deformation of the microasperities and deformation of the lip. In the papers by Gabelli [9], Gabelli and Poll [10], and Salant the microasperities are represented by regular periodic structures given by sinusoidal functions. The flow between the lip and the shaft is considered laminar. The lubricant is assumed to be an incompressible Newtonian fluid of constant dynamic viscosity. Since the lubricant film is very thin, the Reynolds equation is applied, which is given for a two-dimensional geometry as

$$\frac{\partial}{\partial x} \left(h^3 \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left(h^3 \frac{\partial p}{\partial y} \right) = 6\eta_0 U_x \frac{\partial h}{\partial x} \quad (1)$$

where h is the film thickness equation, p the film pressure, η_0 the dynamic viscosity, x tangential position, y the axial position, and U_x the shaft surface velocity in the tangential direction. The film thickness equation is expressed in more detail as

$$h(x, y) = h_o + h_{\text{lip}}(y) + h_{\text{asp}}(x, y) \quad (2)$$

where h_o is the nominal film thickness, $h_{\text{lip}}(y)$ the macroscopic axial geometry of the lip, and $h_{\text{asp}}(x, y)$ the seal surface roughness geometry. The seal surface roughness geometry is modelled by simple periodic sinusoidal functions.

Shi and Salant [12] used a more realistic quasi-random surface roughness geometry. They generated a surface that consists of random roughness heights with a specified average roughness, correlation length, and Gaussian distribution. They made the simplification of a perfectly smooth shaft and assumed pure elastic deformation of the lip. The results of their quasi-random model are in agreement with experimental results published in the public literature.

It is well known that the seal surface roughness after running-in is much higher than the shaft roughness [22]. From experimental work, it is also known that the shaft roughness has a significant influence in lip seal performance. Salant and Shen [13] and Shen and Salant [14] developed numerical models and demonstrated that small fluctuations in shaft roughness can produce large hydrodynamic effects. This is due to the non-linearity of the Reynolds equation. Hajjam and Bonneau [29] showed the importance of selecting appropriate roughness models for the seal surface in numerical modelling. They studied the influence of different mathematical roughness models on performance features as power loss, film thickness, pumping rate, and lip temperature. The results show that the choice of roughness model highly influences the numerical results. Unfortunately, they were not able to validate their models with experimental data.

To solve the numerical models mentioned above, equations (1) and (2) have to be solved simultaneously: The Reynolds equation for the lubricant film pressure and the film thickness equation that takes into account the lip deformation and surface roughness geometry. This results in a highly non-linear problem and in certain zones in the contact, normally close behind the microasperities, cavitation occurs. Here the Reynolds equation calculates non-realistic negative pressures. In the numerical models, this negative pressure is then set to an assumed cavitation pressure that ranges from zero to atmospheric pressure. These numerical models are solved for steady-state situations.

Solving the numerical models including transient behaviour, where a dynamic load is applied, has not been done yet. Hajjam and Bonneau [30] studied the dynamic load behaviour including cavitation and compared their model to oscillating parallel plate models. When these parallel plates move away from each other, the pressure falls to vapour pressure in the centre between the plates. Their model can also be applied to dynamically loaded radial lip seals.

3.3 Micro-elastohydrodynamic lubrication: pumping

The micro-elastohydrodynamic lubrication (micro-EHL) theory, as discussed above, explains the film formation between the seal microasperities and the shaft but does not explain the pumping action of the seal. Müller [8] described a mechanism called 'the cascade pumping side flow concept'. This hypothesis includes micro-EHL and explains both the lubrication and pumping mechanism of the seal.

The elastic microasperities on the seal contact area will elastically deform and the top will flatten because of the contact pressure under the lip. This contact pressure under the lip is asymmetric because of the unequal lip angles α and β and the location of the

Fig. 4 The cascade pumping side flow concept

garter spring. The circumferential line with maximum pressure is called the pressure equator (see Fig. 4). The footprints of the asperities on the smooth shaft are therefore largest around this pressure equator. When the shaft rotates, a micro-EHL film will develop between the elastically deformed microasperity and the shaft that provides the load-carrying capacity. However, a certain amount of lubricant will not flow through the contact but flows along both sides of the microasperity. Here, the flow along one side of the asperity is against the global (contact) pressure gradient. Considering the microasperities close to the pressure equator, flow passes the pressure equator in both directions continuously. At the low-angle side α , the global (contact) pressure gradient is lower and the density of microasperities is higher than that at the high-angle side β . Müller [8] suggests that therefore a net pump flow is generated from the low-angle side α to the high-angle side β .

3.4 Non-symmetrical tangential deformations

In 1969, Kuzma [31] presented a theory including tangential deformation of a face seal surface due to viscous shear forces. He thought that the theory could also be applied to radial lip seals. Kammüller [32] visualized this tangential deformation and he related the pumping rate to the observed roughness pattern. The hypothesis of Kammüller [32] is shown in Fig. 5. Axially extended undulations were observed with the shaft at rest. When the shaft rotates, the highest viscous friction force will be where the contact pressure is the highest, thus at the pressure equator. The seal surface will then tangentially deform and the axially extended undulations will form a V-shape, like in an asymmetrical spiral groove bearing, and pump lubricant to the centre of the contact. Due to the asymmetric groove pattern with the pressure equator closer to the lubricant side, there will be a net pump flow from the air side to the lubricant side of the seal.

Kammüller [32] related the lubricant-side angle and air-side angle geometry to the sealing performance of the seal. He concluded from his measurements

Fig. 5 Tangential deformations and pumping effect

that a significant difference in angles is needed to generate the asymmetric tangential deformations that are required for pumping. The seal will function well when this pumping rate is higher than the natural leakage.

Tønder and Salant [11] numerically studied this non-symmetric groove pattern and modelled the oil flow under the lip. Their model predicts a reduction in leakage flow with increasing viscosity, speed, and lip width. This will finally lead to total suppression of the leakage flow. They also performed a parametrical study on the parameters describing the lip surface pattern with axially extended undulations. Their work shows that the assumed non-symmetrical tangential deformation theory is able to generate a non-leaking situation under realistic conditions.

However, most radial lip seals have been found to have irregular microasperities, rather than axially extended undulations. Therefore, Salant and Flaherty [15] studied the non-symmetrical tangential deformation in combination with different types of seal surface roughness: axially extended undulations and microasperities with different densities. Their numerical model clearly showed the highest pumping rates for the axially extended undulations but also the microasperities showed positive pumping rates and sufficient load support. Salant and Flaherty [15] showed that the pumping rate strongly depends on the shape, density, and orientation of the microasperities. It appeared to be the crossflow between the microasperities that reduces their pumping effectiveness compared to the axially extended undulations.

Bavel *et al.* [33] performed a study very similar to that of Salant and Flaherty [15]. They modelled both the axially extended undulations and the microasperity surface geometry. They evaluated the axially extended undulations first, called in their study the one-dimensional roughness structure, with and without tangential deformations. They showed that without tangential deformations, a negative pumping rate, thus leakage, occurs. When including the

tangential deformations, the calculated radial load capacity and the pumping rate became more consistent with experimental data. Next, they evaluated the microasperity surface geometry, called in their study two-dimensional roughness structure, including tangential deformations and found realistic values for the pumping rate, load capacity, and friction.

An interesting step in the analysis of flow under the lip has been performed by Salant and Rocke [16] and Rocke and Salant [17]. Instead of a deterministic approach, they used a method using flow factors in the Reynolds equation. This greatly reduces the computation time. Here, the lip geometry properties (surface roughness pattern and lip elasticity) are studied with microasperities that stretch in the direction of shear. Also, the pumping rate and load support were calculated. An interesting result is that the maximum pumping rate was found for a film thickness of $2.5\text{ }\mu\text{m}$. They have shown that, if the point of maximum deformation is located further towards the oil side, the pumping rate will increase enormously. This means that the point of maximum radial pressure should be moved towards the oil side, which can be realized by designing the lubricant-side angle β much larger than the air-side angle α .

3.5 Non-Newtonian fluid behaviour

In non-Newtonian fluids, fluid viscosity is a function of the shear rate. High shear rates also introduce a normal stress perpendicular to the direction of shear [34]. Due to the normal stress, non-Newtonian fluids have the tendency to flow to the location with the highest shear. This is demonstrated with a rotating rod in a stationary vessel filled with a non-Newtonian fluid. Here, the fluid climbs up the rotating rod, called the rod climbing effect or Weissenberg effect.

Schulz *et al.* [1] introduced a new and innovative hypothesis. They assumed a Newtonian lubricant to behave in a non-Newtonian manner in a thin film at very high shear rates of $\sim 10^6\text{ s}^{-1}$. This exhibits anisotropic behaviour like the Weissenberg effect and a normal stress in the radial direction will occur (see Fig. 6). They claimed that the normal stress provides a load-carrying capacity that can separate the lip from the shaft, without the need for micro-EHL. Davies and Stokes [35] did experiments with narrow-gap parallel-plate rheometry to measure the rheological properties of different types of non-Newtonian fluids, including complex multiphase fluids like grease. They measured the viscosity and normal forces up to shear rates of 10^5 s^{-1} in gaps between 5 and $100\text{ }\mu\text{m}$ and showed that the normal force indeed increases with increasing shear rate. They also showed a dependence on gap height and surface roughness of the plates.

Schulz *et al.* [1] performed experiments to show the significance of non-Newtonian fluid behaviour

Fig. 6 Normal stress effect due to non-Newtonian fluid behaviour [36]

inside the thin lubricant film. They compared two fluids, one Newtonian fluid and one non-Newtonian fluid. Equal viscosities were obtained by selecting the right temperature. In their experiments, the non-Newtonian fluid showed a lower frictional torque value under steady-state conditions. They concluded that the lower friction is due to non-Newtonian behaviour. Later, Wiehler and Wollesen [36] discussed the use of polymer additives to enhance the non-Newtonian behaviour of the lubricant under the lip to decrease friction. Hajjam and Bonneau [37] showed a similar reduction in friction for a non-Newtonian fluid with a theoretical approach where the viscosity is a function of the shear rate. They used the Gecim law for non-Newtonian fluids to show that friction reduces due to shear thinning. The viscosity decrease also reduces the film thickness and therefore leakage.

3.6 Vortexes

Radial lip seals can operate without leakage even under unfavourable conditions where a relatively large gap is formed. Müller and Ott [38] explained this dynamic sealing mechanism with the formation of a secondary oil flow, called Görtler-Taylor vortex flow. They showed in their experiments that a rigid Plexiglas sleeve seal, with an inner radius larger than the outer radius of the shaft, is able to seal the gap at sufficiently high shaft speeds, called the 'sealing speed'. At speeds lower than the sealing speed, the leakage rate increases with increasing gap height. In their experiments, with the shaft highly flooded with oil, a flow component pointing radially inwards is clearly visible. The oil flow close to the sealing gap must now turn axially away from the seal. As a result, the oil-air meniscus is pulled closer to the sump.

Qu [39] continued with measurements very similar to those of Müller and Ott [38]. He showed that the leakage reduces continuously with increasing shaft speed (see Fig. 7). The sealing speed is found to occur at a critical Taylor number where the oil head, the viscosity of the oil, and the negative vortex pressure are

Fig. 7 Oil leakage rate for different shaft speeds and oil levels. $H = 38.1$ mm corresponds to an oil level equal to the shaft centre [39] (reprinted with permission from SAE paper 930528 © 1993 SAE International)

in dynamic balance. The dynamic sealing will sustain with increasing speed until a shaft speed is reached where the rotational flow near the gap turns turbulent. If this sealing speed is not attained before turbulent flow occurs, the dynamic sealing will never happen.

Müller and Ott [38] reported that the sealing speed is independent of the gap height. This would mean that the oil would seal itself at a certain shaft speed even when the gap height is a few millimetres. Qu [39] showed in his study that the minimum sealing speed does depend on the gap height. The experiments of both Müller and Ott [38] and Qu [39] show that seals can still function properly even when there is a relatively large gap as long as the operation parameters are carefully chosen.

4 MACROSCOPIC MECHANISMS TO IMPROVE PUMPING

To improve the pumping performance of the conventional radial lip seal macroscopic mechanisms can be used. The main macroscopic mechanisms are listed in this section.

4.1 Hydrodynamic pumping aids

Seals with hydrodynamic pumping aids, also called hydrodynamic seals, have asperities on the lip surface close to the contact area. These asperities can be helix ribs, triangles, wedges, or pads. The asperities act as vanes and pump lubricant from the air side to the lubricant side of the seal. They generate an additional pumping action that is especially useful in case of high natural leakage or when not even the smallest amount of leakage is allowed.

Hydrodynamic pumping aids often work in one direction of rotation only. This makes this type of seals

only applicable in applications such as engines where the shaft rotates in only one direction. Hydrodynamic seals that are direction independent do also exist, like seals with triangular-shaped asperities. Other disadvantages can be the tendency to pump dust inwards because of the forced pumping and the seal can pump too much and run dry, which will end seal life quickly [5].

4.2 Seal cocking

Cocking is the result of misalignment of the seal where the seal is installed under a small angle compared to the perpendicular position to the shaft (see Fig. 8). The dynamic seal contact will create a sinusoidal pattern on the shaft. When observing one fixed spot on the shaft, a reciprocating motion of the seal wiping oil back to the oil side can be observed. The seal 'scrapes' towards the lubricant side and 'floats' back towards the air side, which can be modelled with straightforward hydrodynamic equations.

Horve [7] showed that this reciprocating motion increases the pumping action. However, because of the misalignment, cocked seals have regions of high and low pressure between the lip and the shaft. As a result, these seals will run hotter and the increased lip temperature will reduce the seal life. Horve [7] also measured this increase in underlip temperature. Because of the shorter seal life, increase in friction, and power consumption, seal cocking should be minimized.

4.3 Wave seals

Brink and Horve [5] used the reciprocating motion, as in seal cocking, to develop a new type of seal, the wave seal. Wave seals have a smooth moulded lip that makes a wave-like path on the shaft surface (see Fig. 8). The same reciprocating motion as with cocking occurs that increases the pumping rate. The big difference is that the inner radius of the wave seal is a true round

Fig. 8 Cocked seal and wave seal that make a wavelike path on the shaft

circle. This avoids the pressure variation and thus temperature increases so that long seal life is maintained. Contrary to most hydrodynamic seals, wave seals perform well in both directions of shaft rotation.

Figure 8 shows the dynamic running path of the seal on the shaft (shaded area). The area covered on the shaft is much bigger than that on conventional oil seals. Since most frictional heat is dissipated through the shaft by conduction, the lip temperature is reduced and seal life increases.

Brink and Horve [5] performed an experimental study to investigate the properties of wave seals. They compared the wave seal design with the conventional seal design and with seals with hydrodynamic pumping aids. They were all made of the same rubber compound. The experiments showed 25 per cent decrease in underlip temperature, 30 per cent less frictional torque, and an increase in life of 45 per cent for the wave seal. Pumping rates are higher than that for conventional seals but not as high as for the hydrodynamic seals. Brink and Horve [5] have also shown that wave seals ingest about 30 per cent less dust compared to hydrodynamic seals.

4.4 Asperities on the shaft

Otto and Patterson [40] patented engineered asperities on a shaft to improve lubrication and pumping. Figure 9(a) shows such an asperity that acts as a pump impeller and forces the lubricant flow to the left and partly over the asperity itself. Due to the shape of this triangular asperity, a pumping action is generated, which is independent of the direction of rotation. Otto and Patterson [40] claim that this type of asperity also increases the load capacity and life of the seal surface because of improved lubrication.

Because of the lack of (economical) manufactory methods for such surface textures, the engineered surfaces were never used. However, recently, the bearings and seals laboratory at the University of Kentucky

Fig. 9 Asperities on the shaft: (a) triangular asperity that results in a pumping action and (b) hexagonal asperity

has developed a manufacturing technique based on X-ray lithography and UV photolithography to fabricate these engineered surface textures. Hadinata and Stephens [41] conducted a numerical analysis to investigate the elastohydrodynamic effect of various geometries of microasperities. They used the Reynolds equation to calculate the load capacity, friction, and pumping of triangular, square, hexagonal, and circular microasperity geometries. The hexagonal asperity is shown in Fig. 9(b).

Optimal performance was predicted at asperity heights of 1–3 μm . This height is about the same order of magnitude as the roughness of the rubber seal lip, which they assumed to be smooth in their model. The calculated results for the triangular asperity show 23 per cent reduction in friction and 83 per cent increase in pumping rate compared to the microundulation model of Salant and Flaherty [15]. These engineered microasperities show good theoretical performance, but experiments are needed to verify this.

5 GREASE-LUBRICATED SEALS

Eighty to ninety per cent of all rolling element bearings are grease-lubricated. These systems are all sealed to keep contamination out and lubricant in. Grease seals can be integral with the bearing or mounted outside the bearing in the housing. The main function of grease seals is to keep contamination out, whereas the main function of oil seals is to keep the oil in. These contrary functions result in two fundamental differences in seal design. In oil seals, with the air-side angle α smaller than the oil-side angle β , the seal pumps from the air to the oil side. Hydrodynamic aids can be used to increase this pumping rate. In grease seals the difference between the air-side angle α and grease-side angle β is less pronounced. The pumping rate is therefore much lower or in the opposite direction. This prevents pumping contamination like water or mud into the bearing system.

In heavy duty applications, where systems have to perform in water or muddy environments, special seals have been designed. Figure 10 shows the SKF Mud Block seal [42]. This seal contains beside the

Fig. 10 Grease-lubricated seals: a conventional seal (left) and the SKF Mud Block grease seal with extra lips (right)

conventional lip with the garter spring, several inner lips, which form barriers against contamination. These inner lips are lubricated with highly water-resistant grease that fully fills the space in between the lips. The grease not only lubricates the lips but also makes it more difficult for contamination to pass the free space in-between lips. Also, in more conventional seal designs that are grease-lubricated, grease improves the sealing against contamination. Here, a stationary dam of grease is formed at the lubricant side of the seal close to the contact. This grease dam withholds contamination from entering the system (see Fig. 10).

Grease has fundamentally different characteristics than liquid lubricants like oil. Grease contains beside oil an amount of solid thickener also. This gives the grease a consistency and a strong non-Newtonian behaviour. It is not clear what the role of the solid thickener is in the lubrication and sealing mechanism. Seal manufacturers claim that the life of grease-lubricated seals is around 50 per cent of the life of oil-lubricated seals [43]. They give three reasons for this reduction in seal life.

1. Because of the stationary grease dam, there is less circulation of the lubricant around the lip. This results in less heat dissipation and an increase in lip temperatures, which will age the rubber lip material.
2. Due to the reduced circulation of lubricant, less fresh lubricant will be supplied to the contact and the contact could run (partly) dry. Subsequently, wear of the lip will increase because of poor lubrication.
3. Contamination particles that come under the lip will be blocked by the grease dam and now accumulate close to and in the lip contact. This will increase wear of both the rubber seal and the shaft.

The stationary grease dam might be bad from a lubrication point of view but the smaller lubricant film provides better sealing against large contamination particles. Small contamination particles that do come under the lip will now be caught by the grease dam and do not enter the bearing contacts. This will finally increase bearing life despite the reduction in seal life.

Dürnegger and Haas [43] measured the frictional torque of conventional radial lip seals lubricated with grease. They measured frictional torques of 0.6 Nm (3000 r/min) to 0.7 Nm (500 r/min) and showed that the frictional torque decreases when speed increases (see Fig. 11). This correlates with an increase of the measured shaft temperature. For oil-lubricated seals, higher frictional torques of 0.8 Nm (3000 r/min) to 0.9 Nm (500 r/min) were measured. These oil-lubricated seals run at 5°C (1000 r/min) higher temperature than the grease-lubricated seals.

Fig. 11 Frictional torque in grease-lubricated seals [43]

It is therefore surprising that a higher frictional torque is measured since a lower viscosity would be expected here. It does not become clear from their paper why this happens.

The reduced friction and increased sealing ability for grease-lubricated seals give this type of lubricant big advantages in contaminated environments. However, special designed grease seals show some leakage of grease because of the lower pumping rates than that in oil seals.

6 CONCLUDING REMARKS

In the previous sections, several hypotheses on the lubrication, sealing, and pumping mechanisms in radial lip seals were presented. These hypotheses can be divided into primary and secondary mechanisms and will be discussed for oil-lubricated seals. Next, the differences between oil and grease-lubricated seals will be discussed.

Early experimental work showed that high seal surface roughness is required for reliable seal operation, and the non-symmetrical pressure distribution under the lip seemed to be critical for zero leakage. The microhydrodynamic lubrication theory and the non-symmetrical tangential deformation theory include both these seal surface roughness and non-symmetrical pressure distribution effects. Salant and Flaherty [15] used both theories to model the seal performance quite successfully. Their model showed the difference between the surface roughness geometries of axial extended undulations and different types of microasperities. They also confirmed the fundamental features of the non-symmetrical tangential deformations. Today, these two theories have been accepted to be the primary mechanisms describing lubrication, sealing, and pumping in radial lip seals.

The secondary mechanisms include surface tension, non-Newtonian fluid behaviour, vortexes, and macroscopic mechanisms to improve pumping. In

1957, Jagger [3] concluded that a lubricant meniscus under the lip, which is a result of surface tension, is the main sealing mechanism. However, this meniscus cannot explain the pumping action and therefore the surface tension theory has been abandoned as the primary sealing mechanism. Stakenborg [24] explained the lip seal operation to be a balance between the pumping action and the capillary force under the lip. This means that a small amount of lubricant is kept under the lip by the capillary force. Due to the converging wedge at the air side, this capillary force will increase when the meniscus is pulled under the lip towards the contact area. The steady-state sealing can also be explained by the non-symmetrical tangential deformation theory only. Here, the oil will be pumped towards the lubricant side and the meniscus will be pulled under the lip. Now the tangential deformed surface will run partly dry and a balance will be reached where the pumping rate is equal in both directions.

Vortexes at the lubricant side of the seal will occur at high shaft speeds and provide a sealing action even when the gap is relatively large. At high shear rates, non-Newtonian behaviour of the lubricant might influence lubrication and friction. These very high shear rates can generate a normal stress that provides additional load capacity. Lubricant shear thinning might reduce viscous friction. The macroscopic aids to enhance lubrication and pumping are also secondary mechanisms and can be explained with basic hydrodynamic models. Here, the performance of engineered asperities on the shaft can be questioned. Horve [22] showed that high shaft surface roughness increases wear of the rubber lip. A shaft with engineered surface asperities, as suggested by Hadinata and Stephens [41], is extremely rough and may cause very high wear rates of the seal, especially when operating in the mixed lubrication regime during start/stop and low speeds.

In practice, the formation of the seal surface roughness geometry depends on the seal material and running-in conditions. This can make the same seal to perform differently in different applications. It is therefore very challenging to develop theoretical models that describe radial lip seal performance. It also has to be noted that the theoretical models often model a single steady-state situation only. At very high speeds, vortexes and non-Newtonian behaviour might become important and have to be considered. These secondary effects have been observed in experiments but are not included in the type of models Salant and Flaherty [15] made.

All theoretical models in the public literature deal with oil-lubricated seals and only a few include non-Newtonian fluid behaviour. Grease lubrication is relatively unexplored here. Dürnegger and Haas [43] showed in their experiments that grease behaves differently from lubricating oils. They measured lower

fictional torques for grease than for oil-lubricated seals. They also observed a sudden reduction in frictional torque just after the start of most experiments. After ~6 h the frictional torque has increased to a constant level somewhere between the starting torque and the minimum torque (see Fig. 11). They explained this behaviour with a gradual loss of lubricant in the contact, which would result in a reduction in film thickness and an increase in frictional torque. This correlates with the observed increase in temperature. Their explanation suggests that the seal now operates in a mixed lubrication regime but with a lower frictional torque than the oil-lubricated seals. Other studies suggest that oil-lubricated seals operate in the full film regime at the speeds evaluated here. It would, therefore, be more convenient that the grease seals operate in the full film regime as well or other effects play a critical role here. One effect could be shear thinning of the strong non-Newtonian grease. Or the normal stress effect could enhance film formation and subsequently reduce friction. These secondary effects are expected to be more significant in grease than in oil lubrication and could explain the lower frictional torque for grease-lubricated seals measured by Dürnegger and Haas [43].

Another major difference between oil- and grease-lubricated seals is that grease-lubricated seals are better in sealing against contaminated environments. Here the stationary grease dam behind the lip forms an extra barrier for contamination to enter the system. Oil seals are designed to retain oil and therefore high pumping rates are preferred. This means that contamination will also be pumped into the system. Due to the consistency of lubricating grease, lubricant leakage will be rather small. Grease-lubricated seals are therefore primarily designed to prevent contamination from entering the system and have low or negative pumping rates.

With the knowledge presented in the public literature, models can be made to predict the lubricant film thickness and friction in case of oil lubrication. The literature shows that surface roughness plays an important role here. This roughness is generated during running-in and seal operation. Therefore, input to a film thickness and friction model should come from running-in or wear models. Today, those models do not exist. For grease-lubricated seals similar mechanisms as for oil lubrication could apply. However, the secondary effects are assumed to play a significant role here in lubrication, sealing, and pumping.

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Paper B

IMPORTANT NOTE

After this paper has been published, an error in the calculations regarding the normal stress difference was found, which leads to different values of the relaxation time constant, and to different results regarding the predicted lift force on the seal. The correct equations are used in the thesis Chapter 2. The predicted lift force is discussed in Chapter 5.

Non-Newtonian Effects on Film Formation in Grease-Lubricated Radial Lip Seals

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In existing models, the only lubricant property used for predicting film thickness in radial lip seals is the (base) oil viscosity. Lubricating greases show non-Newtonian behavior, and additional normal stress components develop that may contribute to the load-carrying capacity. This study investigates the shear rheology of greases and determines whether this “normal stress effect” in grease can significantly contribute to film formation in radial lip seals. First, the rheological behavior of grease is studied in a rotary plate-plate rheometer at small gaps of 25–500 μm up to shear rates of $5 \cdot 10^4 \text{ s}^{-1}$. The rheology measurements are used for a rheology model that predicts the first normal stress difference in the grease. Second, a seal lip model was developed to predict the lift force generated by the normal stress effect that separates the seal from the shaft. The model results show that the load-carrying capacity depends very much on the operating conditions: lip geometry, speed, and temperature. The model predicts a lift force that is over 50% of the seal specific lip force for low-contact pressure-bearing seals. The model can easily be used in existing oil seal models and makes it possible to optimize seal design by utilizing the normal stress effect.

KEY WORDS

Non-Newtonian Behavior; Lip Seals; Grease; Normal Stress; Lubrication

INTRODUCTION

Radial lip seals are lubricated by a thin film of lubricant to minimize friction and wear. The presence of this lubricant film is explained by micro-elastohydrodynamic lubrication and tangential deformations of the seal surface (Baart, et al. (1)). Theoretical models have been developed to predict film thickness, pumping, and friction in oil-lubricated radial lip seals (Gabbelli (2); Salant

and Flaherty (3); Hajjam and Dominique (4)). In these types of models, the load-carrying capacity is the result of the micro-hydrodynamic oil pressure, viscosity, and seal roughness. Schulz, et al. (5) published a hypothesis that says that the load-carrying capacity is generated by normal stresses in the oil that would behave in a non-Newtonian way at very high shear rates. The significance of this normal stress effect has not yet been identified.

Normal stresses are a result of the viscoelastic properties of a non-Newtonian fluid and have been observed in polymer solutions, emulsions, and particle suspensions (Barnes, et al. (6)). Tanner (7) derived a model for normal stresses in lubricating oil based on the general Oldroyd models (8). Oldroyd (8) showed that his general model predicts experimental results for polymer solutions quite well. However, there is no evidence that these general equations are also suitable for oils. Tanner (7) theoretically investigated whether the normal stresses could explain the seal behavior. He concluded from the models and the experimental evidence on polymer solutions that it is unlikely that the non-Newtonian behavior of oil at high shear rates explains the sealing function and load-carrying capacity. However, no experimental data were available for oils to verify his model. Later, Wiliamson, et al. (9) studied the viscoelastic properties of multigrade engine oils in a journal bearing. They concluded that polymer additives in the oil increase the load-carrying capacity at high eccentricity ratios. The normal stress effect seemed to be large enough to be practically important here.

Hutton (10) used grease as an example of fluids with a yield stress to identify the existence of a normal stress. His results showed the existence of significant normal stress components in grease. No exact values for the normal stress could be measured on the Weissenberg rheogoniometer because of the dependence on the yield stress and zeroing problems of the normal force. Therefore, Binding, et al. (11) continued the work on a torsional balance rheometer, and here the normal stress was balanced by an external load, to determine absolute values of the normal stress. They showed that the normal stress in grease is of the same order of magnitude as the shear stress.

This normal stress effect in grease and the relatively low contact pressures between the seal lip and the shaft might result in a significant contribution to film formation. Figure 1 shows how

NOMENCLATURE

a	= vicinity width air side	R	= rim radius
b	= contact width	r	= radius
c	= position minimum film height	T	= temperature
d	= vicinity width lubricant side	u	= shaft surface speed
$\Delta F_{\text{inertia}}$	= inertial force	x_1	= integration boundary
F_{lift}	= specific lift force	x_2	= integration boundary
F_{lip}	= specific lip force	α	= lip air side angle
F_{rheology}	= specific normal stress force	β	= lip lubricant side angle
$F_{\mu\text{EHL}}$	= specific micro-elastohydrodynamic force	$\dot{\gamma}$	= shear rate
h_1	= film height at edge contact	$\dot{\gamma}_m$	= measured shear rate
h_2	= film height at $c/b = 1/3$	ε	= gap error
h_g	= grease height	η	= viscosity
h_o	= minimum film height	η_m	= measured viscosity
$h(x)$	= film thickness function	η_o	= low shear rate viscosity
K	= constant	η_r	= real viscosity
K_c	= constant	η_{∞}	= high shear rate viscosity
M_{friction}	= frictional torque	η_{∞}	= base oil viscosity
m	= power law index	λ_1	= relaxation time constant
N_1	= first normal stress difference	ρ	= grease density
N_2	= second normal stress difference	σ_{zy}	= shear stress
n	= power law index	τ_y	= yield stress
n_c	= power law index	φ	= lip angle
		ψ_1	= normal stress constant
		Ω	= angular velocity

the normal stress effect may load the seal lip and generate a lift force.

In Part I of this study, a rheology model for the grease is developed. In addition to the viscosity and shear stress, a model for the normal stress is introduced. These models are verified with experiments on the rheometer at shear rates up to $5 \cdot 10^4 \text{ s}^{-1}$. In Part II, a seal lip model is presented to investigate the significance of the normal stress effect on film formation in grease-lubricated radial lip seals.

PART I: GREASE RHEOLOGY—EXPERIMENTS AND MODEL

Grease Rheology Model

Grease consists of liquid base oil, additives, and solid thickener, which make it a complex semisolid fluid. The grease rheol-

ogy is usually described by a yield stress and shear rate dependent shear stress. Also, elasticity and normal stresses could be important in the seal lubrication problem.

Equations to describe a general flow curve need at least four parameters. A general model that can be used here is the Cross model (Barnes, et al. (6)),

$$\eta = \frac{\eta_o - \eta_{\infty}}{1 + (K_c \dot{\gamma})^{n_c}} + \eta_{\infty}, \quad [1]$$

where η_o is the low shear rate viscosity plateau, η_{∞} is the high shear rate viscosity plateau, and K_c and n_c are constants. In this study, the focus is on the high shear rate side of Eq. [1] because of the high shear rates experienced in the seal applications. Therefore, it can be assumed that $\eta \ll \eta_o$, and Eq. [1] can be written as

$$\eta = \frac{\eta_o}{(K_c \dot{\gamma})^{n_c}} + \eta_{\infty}, \quad [2]$$

which can be rewritten to obtain the Sisko model (Barnes, et al. (6)), which reads

$$\eta = K \dot{\gamma}^{n-1} + \eta_{\infty}. \quad [3]$$

This model for the viscosity can be used to model the shear stress. However, grease shows a yield stress behavior such that an additional term for the yield stress τ_y has to be included:

$$\sigma_{zy} = \tau_y + K \dot{\gamma}^n + \eta_{\infty} \dot{\gamma} \quad [4]$$

Equation [4] can be recognized as the Herschel-Bulkley model, including an additional term as presented by Palacios and Palacios (12). The original Herschel-Bulkley model should not be used at the high shear rates experienced in seals because the bulk viscosity would then have dropped below the base oil viscosity, which is very unlikely. Palacios and Palacios (12) derived the

Fig. 1—Normal stresses in the grease generating a lift to separate the seal from the shaft.

Fig. 2—Weissenberg rod-climbing effect in a non-Newtonian fluid.

four-parameter model in Eq. [4], where η_{∞} is the base oil viscosity, such that the bulk viscosity at high shear rates will approach the base oil asymptote. Also, Einstein's formula for suspensions (Barnes, et al. (6)) can be used here for η_{∞} , but then more details on the amount of thickener and thickener structure that are often not known for grease would be necessary.

Secondary flow effects have been observed in non-Newtonian fluids when the fluid is sheared. Here normal stresses can develop to be orthogonal to the direction of shear. This may result in the Weissenberg rod-climbing effect, where a fluid climbs up a rotating rod (Fig. 2). Newtonian fluids will move to the wall of the vessel because of inertia forces. Normal stresses in a polymer solution occur when polymer molecules deform due to shear stresses and subsequently want to move back into their original shape because of their elasticity. Normal stresses have also been measured in emulsion and particle suspensions (Barnes, et al. (6)). Lubricating grease contains a microstructure of solid thickener and shows strong non-Newtonian behavior. The Weissenberg rod-climbing effect has also been observed for grease by the authors but is rather small. This is expected to be due to the yield stress behavior.

The first normal stress difference N_1 acts in a direction that is orthogonal to the plane of shear. The second normal stress difference N_2 acts in a direction that is parallel to the plane of shear and perpendicular to the direction of shear. N_2 is negative, normally much smaller than N_1 , and therefore in many viscoelastic fluids negligible. In the radial lip seal contact, N_1 will tend to lift the seal from the shaft (see Fig. 1) and enhance lubricant film formation. N_2 will force lubricant to move from the axial direction into the direction of the highest shear rate and could be a source for lubricant supply into the contact and could also play a role in the pumping action of the seal. N_2 is assumed to be small compared to N_1 and is neglected in the present study.

In classic constitutive models, a relation between shear rate and the first normal stress is available (Barnes, et al. (6)). This relation follows a power law behavior over a range of shear rates and reads

$$N_1 = \psi_1 \dot{\gamma}^m \quad [5]$$

In classic models like the second-order Maxwell model or the Oldroyd-B model, the constant $m = 2$. More complex behavior is

often observed where $m \neq 2$, and it is even possible that different parts of the normal stress curve follow different power law functions. The ψ_1 in Eq. [5] is approximated by the White-Metzner model (Barnes, et al. (6))

$$\psi_1 = 2\eta(\dot{\gamma})\lambda_1, \quad [6]$$

where $\eta(\dot{\gamma})$ is the shear rate dependent viscosity and λ_1 is the first relaxation time constant. This relation was also used by Williamson, et al. (9) for multigrade oils. The new parameters in the first normal stress equation are the relaxation time constant λ_1 and power law constant m . Substituting Eq. [6] into Eq. [5] and using Eq. [3] for the viscosity, the first normal stress difference can be written as

$$N_1(\dot{\gamma}) = 2\lambda_1 (K\dot{\gamma}^{n-1} + \eta_{\infty}) \dot{\gamma}^m. \quad [7]$$

The constants, except for λ_1 and m , can be found by fitting the shear stress equation, Eq. [4], to shear stress measurements. Subsequently, the constants λ_1 and m can be found by fitting the normal stress equation, Eq. [7], to normal force or normal stress measurements.

Thin-Film Rheometry

The rotational rheometer with plate-plate geometry is used for measuring the rheological properties of the grease. To simulate very high shear rates as in radial lip seal applications, high rotational speeds and small gap heights are required. Since the maximum rotational speed is limited, the gap height has to be reduced to obtain the high (rim) shear rates. However, this method will introduce several errors in the measurements due to shear rate distribution, inertia, gap settings, viscous heating, and edge fracture. These errors were identified by Davies and Stokes (13) as the main errors in narrow-gap parallel-plate rheometry and will be shortly discussed here.

The shear rate distribution in parallel-plate rheometry results in the highest shear rates at the plate's rim since the shear rate increases with the radius. The rheometer measures the total torque and total normal force as functions of the rotational speed. Both torque and normal force are modeled for the non-Newtonian grease by integrating Eq. [4] and Eq. [7] over the radius of the parallel plates. By fitting the model to the measurements, the parameters in Eq. [4] and Eq. [7] can be determined. Subsequently, the torque and normal force are converted into shear stress and normal stress by using constant factors for both the measurements and the model. These constant factors were derived for Newtonian fluids and therefore result in slightly lower values (Davies and Stokes (13)).

Inertia effects in the grease may result in a lower reading of the normal force because the plates will be pulled together as the grease is spun outward. A correction for this effect can be done by (Davies and Stokes (13))

$$\Delta F_{\text{inertia}} = -\frac{3\pi\rho\Omega^2 R^4}{40}, \quad [8]$$

where ρ is the grease density, Ω is the angular velocity, and R is the rim radius. This correction will be included in the measurement data.

A gap error between parallel plates can be particularly noticeable at small gap heights below 100–200 μm . This error can arise from nonparallelism, nonconcentricity, edge effects, wall slip, nonflatness of the plates, and the gap-zeroing procedure. By assuming that the gap error arises from the gap height measurement, it is relatively easy to include a correction for this error in the experimental results. A method using a Newtonian fluid as a reference to measure the gap error is used as described by Davies and Stokes (13). The measured gap error is used for a correction on the shear rates. The real shear rate corresponding to the measured shear rate can be calculated as

$$\dot{\gamma} = \dot{\gamma}_m \frac{h}{h + \varepsilon}, \quad [9]$$

where $\dot{\gamma}$ is the actual shear rate, $\dot{\gamma}_m$ the measured shear rate, h the gap height setting, and ε the gap error. A gap error ε of 10 μm will already shift the measured data to 30% lower shear rates when the gap height setting h is only 25 μm . A small gap error can have big impact on the accuracy of the results when one is looking at very small gaps.

Edge effects include rim fracture (Hutton (10)) and radial migration of grease. These edge effects give an important contribution to the shear stress and normal stress because shear rates and thus shear stresses and normal stresses are highest at the rim.

Viscous shear heating leads to an increase in temperature of the grease sample and a decrease in viscosity. The measured shear stress and normal stress will thus decrease. The viscous heating effect should be distinguished from shear aging, which results from mechanical working of the grease where the microstructure of the thickener in the grease has broken down. The experimental results have shown that an increase in temperature gives only a temporary effect, which is fully reversible. Heat generated in the sample will be rapidly dissipated into the plates because of the small gap height (Davies and Stokes (13)). Therefore, no temperature gradients in the rheometer sample are expected.

Experimental Method

The non-Newtonian behavior of grease is measured in a rotational rheometer with plate–plate geometry (TA instruments AR1000N). The experiments include torque and normal force measurements by varying

- gap height;
- temperature;
- grease type.

Grease D is measured at different gap heights of 500, 250, 100, 50, and 25 μm . Maximum shear rates up to $\dot{\gamma} = 5 \cdot 10^4 \text{ s}^{-1}$ can be obtained with the smallest gap of 25 μm , while the 500 μm gap height is less sensitive to errors like geometrical errors of the plates. The measurements with grease D have been done at 25°C, 70°C, 100°C, and 120°C. The largest normal stresses are expected at 25°C, where the grease viscosity is highest. However, the higher temperatures give a more realistic representation of the seal contact. The rheology is also measured for other greases: A, B, C, E, and F at a constant gap height of 250 μm at 70°C. These greases are all greases used in rolling element bearings and have an NLGI number of ~2. Table 1 shows the grease charac-

TABLE 1—GREASE CHARACTERISTICS: BASE OIL TYPE, BASE OIL VISCOSITY, THICKENER TYPE, AND NLGI NUMBER

Grease Characteristics			
Grease	Base Oil (cSt@100°C)	Thickener	NLGI
A	Ester (9.4)	Polyurea	2-3
B	Mineral (17)	Lithium	2
C	Mineral (10.5)	Polyurea	2-3
D	Mineral (12)	Lithium	2
E	PAO (14.5)	Lithium	2
F	Mineral (8)	Lithium	2-3

teristics; i.e., thickener structure, base oil type, base oil viscosity as given by the manufacturer, and consistency.

In the rheometer, the parallel-plate geometry is used with a diameter of $2R = 25 \text{ mm}$. The gap zero is set by using the standard rheometer gap-zeroing procedure. The gap error is determined before and after each series of experiments from measurements with Newtonian oil (viscosity = 15 Pa.s at 25°C) at three different gap heights according to the method described by Davies and Stokes (13). The average gap error is used to correct the measured shear rate according to Eq. [9]. An error due to wall slip cannot be identified with Newtonian oil and has to be identified from the grease measurement results themselves after correcting for the gap error. After each test, the grease at the rim has to be checked for edge fracture and radial migration of grease.

For a typical experiment, the grease sample is taken from the grease can with a spatula and is put in the center between the rheometer plates. The plates are moved to the required gap height, and the grease that is squeezed out at the rim is carefully removed. Subsequently, a preshear program is run to remove initial stresses that have been introduced by decreasing the gap. In this preshear program, the angular velocity is increased from 0 to 100 rad/s in 10 min and then decreased to 0 rad/s in 10 min. For the measurement, the angular velocity is increased from 0 to 100 rad/s in 10 min, and the torque, normal force, and temperature are measured. The environmental chamber was used to control temperature at 70°C, 100°C, and 120°C within 0.5°C. Temperature could not be controlled at 25°C, but due to the short test time the temperature never increased more than 2°C except at the highest shear rates at the end of the experiment. The error introduced here was found to be less than 10% of the normal force measured.

In case measurements are done at different gap heights, the same sample has been used each time. The first measurement is done at the largest gap, and subsequently the gap is narrowed to a smaller height. Grease that has been squeezed out at the rim is removed, and the preshear program is run. It is important that the environmental chamber reaches a stable equilibrium temperature before the measurement is started.

Experimental Results and Discussion

Newtonian oil is used to determine the gap error at 100, 50, and 25 μm at the required temperature before and after a series of experiments. Figure 3 shows the results of such a gap-error

Fig. 3—Shear stress measurement with Newtonian oil at 25°C up to shear rates of $\dot{\gamma} = 5 \cdot 10^4 \text{ s}^{-1}$. The insert contains data that corrected for a gap error of 27 μm .

measurement. It can be seen here that a decrease in shear stress is measured for a decreasing gap. This error comes from an error in the gap height measurement, which can be corrected using Eq. [9]. The exact gap error can be determined by using a graphical method, explained by Davies and Stokes (13). The following equation is used here to determine the real viscosity and subsequently the gap error:

$$\frac{h}{\eta_m} = \left(\frac{1}{\eta_r} \right) h + \left(\frac{\varepsilon}{\eta_r} \right), \quad [10]$$

where η_m is the measured viscosity (slope in Fig. 3) and η_r is the real viscosity. The quotient h/η_m can be plotted against the gap setting h . The slope of the straight line, the fit through the points, determines the real viscosity $1/\eta_r$. The intercept at $h = 0$ gives the ε/η_r term from which the gap error can be determined. For the measurements in Fig. 3, a gap error $\varepsilon = 27 \mu\text{m}$ was found. The insert in Fig. 3 shows the results after the gap-error correction. It can be seen here that the data points now overlap as expected.

The shear rheology of grease D has been measured at different gap heights of 500, 250, 100, 50, and 25 μm at 25°C. Shear rates up to $\dot{\gamma} = 5 \cdot 10^4 \text{ s}^{-1}$ have been obtained at the smallest gap of 25 μm . The shear stress results are shown in Fig. 4 and have been corrected for the gap error. Here the curves overlap almost perfectly at the higher shear rates. At lower shear rates the curves do not overlap exactly, which could be due to effects like wall slip. In this study the focus is on high shear rates, as found in seal applications, and therefore no further attention is given to the low shear rate errors. After each measurement the rim was checked for edge fracture and radial migration of grease, but no such edge effects were observed. Viscous heating of the sample was checked by measuring temperature and found insignificant.

The shear stress model, Eq. [4], has been fit to the measurement data in Fig. 4 using the parameter values in Table 2. Here τ_y directly results from the measurements, and η_∞ is taken as the base oil viscosity at 25°C. Subsequently, K and n are used for fit-

Fig. 4—Shear stress grease D at 25°C for different gap heights including correction for the gap error.

ting the curve. The dashed line in Fig. 4 represents the shear stress in pure base oil to show that the grease model reaches the base oil asymptote at high shear rates.

Figure 5 shows the results for the normal stress, which have been corrected for the gap error and inertia effects. The inertia effects appeared to be rather small and could have been neglected. The relaxation time constant λ_1 and power law index m in the normal stress model, Eq. [7], have been fit to the measurement results at shear rates above 500 s^{-1} . The model therefore shows good agreement with the measurement results at shear rates between $3 \cdot 10^2 < \dot{\gamma} < 10^4 \text{ s}^{-1}$. However, at shear rates below 300 s^{-1} , the spread in the measured data is large due to the yield stress behavior of the grease as explained by Hutten (10) and Binding, et al. (11). The results show that the normal stress at higher shear rates is in the same order of magnitude as the shear stress in Fig. 4. This is confirmed by the aforementioned observations from Binding, et al. (11).

Temperature effects have been studied at four different temperatures: 25°C, 70°C, 100°C, and 120°C. Figure 6 shows the normal stress results for grease D at 100°C. The results show good consistency and no dependence on gap height. The large spread at lower shear rates as found at 25°C is not observed here due to the fact that the yield stress decreases with increasing temperature. The shear stress results for the different gap heights overlap

TABLE 2—RHEOLOGY MODEL PARAMETERS FOR GREASE D

Model Parameters Grease D					
Variable	25°C	70°C	100°C	120°C	
τ_y	350	60	25	10	Pa
η	0.17	0.030	0.0095	0.0044	Pa·s
K	20	10	5.0	3.0	Pa·s ⁿ
n	0.50	0.49	0.48	0.48	—
λ	2.8	2.3	2.1	2.0	s
m	0.71	0.71	0.71	0.71	—

Fig. 5—Normal stress grease D at 25°C for different gap heights including a correction for the gap error and inertia.

at high shear rates, but at low shear rates some deviation that seemed to depend on the gap height and could have been a result of wall slip was observed.

The parameters for the rheology model to fit the measurements at the four different temperatures can be found in Table 2. As expected, the parameters τ_y , K , n , η_∞ , and λ_1 depend very much on temperature except for the power law exponents n and m , which seem to be temperature independent.

Different greases were studied at a temperature of 70°C and a gap height of 250 μm . Figure 7 and Fig. 8 show the shear stress and normal stress results for these greases. The figures show that greases that have a higher shear stress also have a higher normal stress and show that the normal stress is in the same order of magnitude as the shear stress. This suggests that the normal stress and shear stress are related. Comparing Eq. [4] and Eq. [7] shows that the relaxation time constant λ_1 is important for the normal stress

Fig. 6—Normal stress grease D at 100°C for different gap heights including a correction for the gap error and inertia.

Fig. 7—Shear stress for different greases at 70°C and a 250- μm gap.

but not for the shear stress. At the same time the yield stress τ_y does not appear in Eq. [7] and seems not to influence the normal stress. The relation between shear stress and normal stress should therefore be found in the grease viscosity, which is a function of the shear rate, Eq. [3].

The rheology model in Eq. [7] is based on general equations that are used in polymer rheology. Similar physical phenomena are expected to occur in grease where part of the microstructure is elastically deformed during shearing. Therefore, it is likely that the general equations are also suitable for describing normal stress effects in grease. However, it is arguable whether for the very thin films found in seal applications this analogy is still valid because the film thickness will be in the same order of magnitude as the grease microstructure dimension. For larger film thicknesses, $h \geq 25 \mu\text{m}$, the rheology model is well supported by rheometer measurements, and it was possible to fit the rheology model to the normal stress measurements by only adding

Fig. 8—Normal stress for different greases at 70°C and a 250- μm gap.

a temperature-dependent relaxation time constant λ_1 and a constant power law index parameter m to the set of parameters describing the shear stress.

PART II: SEAL LIP MODEL—GREASE LUBRICATED

Seal Lip Model

The rheology model for normal stresses in the grease will be used as input for the seal lip model. This model includes the lip geometry, film thickness, surface speed, and temperature. Two lubrication areas in the seal have been identified here. First is the contact zone, which is the contact area of the seal that would land on the shaft when the rotational speed of the shaft is zero. The film thickness in this contact zone is very small, and extremely high shear rates are expected here. Secondly the vicinity of the contact is studied by assuming that some grease will sit close to the contact. This grease will be sheared and will contribute to the frictional torque and lift. Despite the fact that shear rates are much lower here than in the contact, the normal stress effect is expected to have a significant contribution.

To allow for the formation of a lubricant film between the shaft and the seal, the seal lip force has to be balanced by the conventional micro-elastohydrodynamic pressure and the normal stress effect

$$F_{lip} = F_{\mu EHL} + F_{lnt}, \quad [11]$$

where F_{lip} is the specific lip force, $F_{\mu EHL}$ the force resulting from the micro-elastohydrodynamic pressure, and F_{lnt} the force resulting from the normal stress effect. The seal lip model will predict the percentage of the normal stress effect contribution to the total lift that balances the seal lip force. The pumping action that avoids leakage of lubricant is not considered in this study. The second normal stress might play a role here, but this has not been investigated.

In the contact zone the shear rates are extremely high due to the small film thickness. Film thickness measurements on oil-lubricated seals show an oil film thickness of 1–5.5 μm (Van Leeuwen and Wolfert (14)). Thinner films are expected for grease-lubricated seals (Dürnegger and Haas (15)), but no absolute values have been found in the open literature. Therefore, the minimum film thickness measured in oil-lubricated seals is used. The contact zone geometry is defined by the parameters in Fig. 9b, where the film thickness $h(x)$ is defined over the contact width b by the heights h_1 and h_2 . Here three situations can be distinguished— $h_1 < h_2$, $h_1 = h_2$, and $h_1 > h_2$ —which all may occur in practice depending on seal design and running conditions. Consequently, h_2 is chosen as 2 μm such that the minimum film thickness is 1 μm when $h_1/h_2 = 2$. For the thin lubricant films and high shear rates in the contact zone, the rheology model has to be extrapolated to shear rates of $\dot{\gamma} = 1 \cdot 10^7 \text{ s}^{-1}$. This extrapolation has to be done with great care and will be reviewed in the discussion of the model results. It is assumed that the lubricating medium in the contact zone is grease. Additionally, the case is considered where only pure base oil is lubricating the contact zone, so that the normal stress effect does not apply here but only in the vicinity of the contact.

Fig. 9—Simplified lip geometry: (a) grease volume in the vicinity of the lip contributing to the lift, and (b) grease volume in the contact zone with contact width b and film thickness $h(x)$.

Maximum shaft surface speeds are depending on seal type, lubricant type, and lip force. Dürnegger and Haas (15) studied grease-lubricated 80-mm (inner diameter) garter spring seals up to shaft surface speeds of 10.5 m/s. Contacting bearing seals allow for maximum surface speeds of 15 m/s for the same seal inner diameter. Two phases in seal service can be distinguished: start-up and continuous running. During (first) start-up of the system there is grease under the seal lip at room temperature. Subsequently the system will heat up due to frictional heating in the contact. This frictional heating is not incorporated in the model, and steady-state situations with constant temperature have been considered only. Dürnegger and Haas (15) measured a temperature of 57°C on the shaft surface for a grease-lubricated seal at 10.5 m/s. This temperature was measured close to the contact, and temperatures in the contact itself are higher. Local temperatures of 70–120°C are expected in the lip contact.

The vicinity of the contact contains grease as well (see Fig. 9a). Shear rates are much lower here than in the contact zone, but due to the large width $a + d$, a lift force will be generated by the normal stress effect. The shear rate depends on the shaft speed u and the geometry that is defined by the air side and lubricant side angles α and β . The contact width b , as defined in Fig. 9b, is set at zero here so that the influence of the contact zone and the vicinity of the contact can be studied separately. To obtain the total picture of the lift generated, both effects can be combined.

The seal lip model includes the lip geometry $h(x)$, which is defined by the parameters in Fig. 9 and is used to calculate the local shear rate under the lip as

$$\dot{\gamma} = \frac{u}{h(x)}, \quad [12]$$

where u is the shaft surface speed. Equation [12] can be substituted into Eq. [4] and Eq. [7] to obtain the shear stress and normal stress as a function of the axial position x . The specific lift force F_{lift} , which is the result of the normal stresses in the contact and vicinity of the lip, can be calculated by integrating the normal stress difference:

$$F_{\text{lift}} = \int_{x_1}^{x_2} N_1(x) dx, \quad [13]$$

where x_1 and x_2 represent the zone of interest. For example, this can be the contact zone $x_2 - x_1 = b$ or the total width containing grease $x_2 - x_1 = a + b + d$. The seal lip model also predicts frictional torque M_{friction} resulting from the shear stresses in the grease as

$$M_{\text{friction}} = 2\pi r^2 \int_{x_1}^{x_2} \sigma_{xy}(x) dx, \quad [14]$$

where r is the shaft radius and $2\pi r$ is the shaft circumference. Note here that Eq. [13] and Eq. [14] include the rheology models, which have been defined for constant temperatures only. Temperature can be included in the rheology model by changing the constants in Eq. [4] and Eq. [7] into temperature-dependent functions. These functions can be found by fitting curves through the data obtained at different temperatures in Table 2. These functions make the rheology model applicable for more advanced seal lip models. To keep it simple here, the rheology parameters have been kept constant.

An analytical equation can be derived for calculating the normal stress in the zone $x_2 - x_1$ when the film thickness is described by a simple linear equation $h(x) = \varphi x + h_o$. Substituting Eq. [7] into Eq. [12] gives the integral equation for the lift force

$$F_{\text{lift}} = 2\lambda_1 \int_{x_1}^{x_2} \left[K \dot{\gamma}^{(n-1)} + \eta_{\infty} \right] \dot{\gamma}^m dx. \quad [15]$$

The film height at x_1 is defined as $h(x_1) = h_o$, and the slope of the lip is defined as $dh/dx = \varphi$. When the vicinity of the contact is considered, the slope φ will represent the lip angle α or β , and h_o will represent the film thickness in the contact zone. To solve the integral in Eq. [15], the integral has to be split into two parts, and new integration boundaries have to be defined

$$F_{\text{lift}} = 2\lambda_1 \left\{ \int_0^{\hat{x}_2} (K \dot{\gamma}^{(n-1)} + \eta_{\infty}) \dot{\gamma}^m d\hat{x} - \int_0^{\hat{x}_1} (K \dot{\gamma}^{(n-1)} + \eta_{\infty}) \dot{\gamma}^m d\hat{x} \right\}, \quad [16]$$

where $\hat{x}_1 = h_o/\varphi$ and $\hat{x}_2 = (x_2 - x_1) + h_o/\varphi$. Here the constant film height h_o only appears in the integration boundaries and is not in the integral. In this way, one obtains a very simple expression for the shear rate:

$$\dot{\gamma} = \frac{u}{\varphi \hat{x}}. \quad [17]$$

The difference of the two integrals in Eq. [16] describes the same zone $x_1 - x_2$ as defined by the integration boundaries in Eq. [15]. The integral equation in Eq. [16] can now be solved by keeping all the constants in the rheology model:

$$F_{\text{lift}} = 2\lambda_1 \left[\frac{K}{(2-n-m)} \left\{ \frac{u}{\varphi} \right\}^{(n+m-1)} (\hat{x}_2^{(2-n-m)} - \hat{x}_1^{(2-n-m)}) \right]$$

$$+ \frac{\eta_{\infty}}{(1-m)} \left\{ \frac{u}{\varphi} \right\}^m (\hat{x}_2^{(1-m)} - \hat{x}_1^{(1-m)}) \right]. \quad [18]$$

A similar analytical equation can be derived for the shear stress by substituting Eq. [3] into Eq. [14]:

$$M_{\text{friction}} = 2\pi R^2 \left[\tau_{\gamma}(\hat{x}_2 - \hat{x}_1) + \frac{K}{(1-n)} \left\{ \frac{u}{\varphi} \right\}^n (\hat{x}_2^{(1-n)} - \hat{x}_1^{(1-n)}) \right. \\ \left. + \eta_{\infty} \left\{ \frac{u}{\varphi} \right\} (\ln |\hat{x}_2| - \ln |\hat{x}_1|) \right]. \quad [19]$$

The analytical equations are a tool for including the non-Newtonian grease behavior in current film thickness and friction models that exist for oil-lubricated seals.

The model results include the frictional torque M_{friction} and lift force F_{lift} resulting from the normal stress effect in the grease for different seal (contact) geometries and surface speeds. Subsequently the frictional torque and lift force are also calculated for different values of the grease height h_g in the vicinity of the contact using the analytical model. These calculations are done for constant temperatures of 25°C, 70°C, 100°C, and 120°C. It is recognized that in reality temperature will rise with increasing speed, but no frictional heating is included in the current model.

The lift force F_{lift} , calculated with the model, is the result of the pressure generated by the normal stress only and will be compared to the seal lip force. A lip force F_{lip} is used of 6 N/m that represents a low-contact pressure-bearing seal and a lip force of 60 N/m that represents a shaft seal with garter spring. The elastohydrodynamic term F_{JEHL} is not considered in this study. The seal lip model will show which part of the lip force is balanced by the normal stress effect $F_{\text{lift}}/F_{\text{lip}}$.

Model Results and Discussion

The normal stress effect has been measured on the rheometer up to shear rates of $5 \cdot 10^4 \text{ s}^{-1}$. Much higher normal stresses were observed at low temperatures than at high temperatures. The rheology model is used in the seal lip model to predict the contribution of the normal stress effect to balance the specific seal lip force.

The contact zone dimensions depend very much on the seal wear mechanism. The contact width b of a new seal is $\sim 100 \mu\text{m}$ and can increase up to $400 \mu\text{m}$ when the lip wears off (Van Leeuwen and Wolfert (14)). For the model a contact width of a run-in seal of $200 \mu\text{m}$ is assumed. Three different contact geometries are studied: a flat contact zone with constant film thickness $h_1 = h_2 = 2 \mu\text{m}$ and a contact zone with a minimum or maximum film thickness at $c/b = 1/3$ such that $h_1 \neq h_2$. In case of the flat contact zone, the lift is proportional to the width b . The geometry parameters for the seal have been chosen based on realistic values found in open literature. Values for the initial choice of the seal geometry, Fig. 9, can be found in Table 3. Figure 10 shows the results of the specific lift force F_{lift} for different temperatures and contact zone geometries at 70°C. It can be seen here that small variations in contact geometry h_1/h_2 have a relatively small influence on the total lift force, and the effect of temperature appears to be more significant here. The model predicts a specific lift force of $\sim 2 \text{ N/m}$ for the flat contact at 70°C. This value can be

TABLE 3—INITIAL CHOICE OF SEAL GEOMETRY PARAMETERS

Seal Geometry Parameters		
b	200	μm
c/b	1/3	—
h_1	2	μm
h_2	1–3	μm
α	20–40	$^\circ$
β	30–60	$^\circ$
F_{lip}	6	N/m

related to the seal’s specific lip force, which can vary from 6 N/m for low-contact pressure-bearing seals to 60 N/m for shaft seals that have a garter spring. This means that for the low-contact pressure-bearing seals, 33% of the lip force can be balanced by the normal stress effect. For the shaft seals, this is only 3% and is therefore not significant.

Looking at the different temperature plots in Fig. 10, one can see that the lift force depends very much on temperature. This was expected, considering the rheometer experiments. The lift force of ~ 2 N/m reduces to 0.6 N/m at 100°C but can also increase to 13 N/m at 25°C. This means that at ambient temperature start-up or in low-temperature conditions, the lip force of the low-contact pressure-bearing seal can easily be balanced by the normal stress effect and that a lubricant film can be expected. Depending on the stiffness of the lip, the film thickness can increase and be rather large. This lift is independent of the seal surface roughness, which is normally generated during running-in, and therefore the normal stress effect may already generate a lubricating film when the seal is new.

In the vicinity of the contact the grease also contributes to the lift. Despite the relatively low shear rates here, the contribution of the normal stress effect is significant because of the relatively large widths a and d . The temperature in the vicinity of the contact is expected to be lower than in the contact zone, which increases the normal stress effect. Figure 9a defines the lip geom-

Fig. 10—Lip contact model for different temperatures and contact geometry $h_1 = 2 \mu\text{m}$ and $b = 200 \mu\text{m}$.

Fig. 11—Lift generated by the grease in the vicinity of the contact for different temperatures and lip angles α – β , $h_o = 2 \mu\text{m}$, and $u = 10 \text{ m/s}$.

etry where grease is present at both sides of the contact with a maximum height h_g . Figure 11 shows the model results for a shaft speed $u = 10 \text{ m/s}$ and a minimum film height $h_o = 2 \mu\text{m}$. The contact zone b is not included here such that $b = 0 \mu\text{m}$, and the calculated lift force is the result of the grease in the vicinity of the contact only.

Figure 11 shows that the grease in the vicinity of the contact generates a lift of $\sim 1.8 \text{ N/m}$ at $T = 70^\circ\text{C}$ if $h_g = 1 \text{ mm}$. In case h_g is smaller, e.g., $h_g = 0.3 \text{ mm}$, still a specific lift force of $\sim 1 \text{ N/m}$ is obtained. This means that 15–30% of the lip force can be balanced by the normal stress effect in the vicinity of the contact. This percentage will even be higher when one assumes the temperature in the vicinity of the contact to be lower than in the contact itself. Again, at ambient or low temperatures, e.g., 25°C , the lip force is completely balanced when $h_g > 0.6 \text{ mm}$, and a lubricant film can be expected in the seal contact without the need for micro-elastohydrodynamic lubrication. When the shaft speed reduces by 50% to 5 m/s, the lift force decreases by only 30%. Results for different lip angles are also shown in Fig. 11. It can be seen here that when the lip angles decrease, the lift force increases due to the increase in shear rate.

The lift force calculated for the vicinity of the contact can be added to the lift calculated for the contact zone. This means that for the low-contact pressure-bearing seal, the total normal stress effect can balance the lip force for 50–65% at 70°C . When one assumes the temperature in the vicinity of the contact to be lower than in the contact itself, this total percentage may be higher, and at low (ambient) temperatures the normal stress effect may completely balance the seal lip force.

It should be remembered that the rheology model for the thin film in the contact zone had to be extrapolated over two orders of magnitude. Moreover, it is assumed that a grease film including solid thickener is lubricating the contact zone. One can also anticipate that the thickener will move out of the contact such that the lubricant film consists of base oil only. In that case the

Fig. 12—Isothermal frictional torque as a result of shearing the grease in the contact zone for different temperatures and contact geometries: $h_1 = 2 \mu\text{m}$ and $b = 200 \mu\text{m}$.

non-Newtonian behavior and normal stress effect are not present in the contact zone. No evidence was found in the open literature regarding whether grease or only base oil is what lubricates the contact zone. Therefore, the results from the contact zone model have to be interpreted with care. When no normal stress effect in the contact zone is considered, the lift generated in the vicinity of the contact may still be 15-30% at 70°C, which is significant. For shaft seals, this percentage of lift generated by the normal stress effect is low and may not be significant due to the high lip forces.

Frictional torque at constant temperature is also an output of the seal lip model. Figure 12 shows that the frictional torque in the contact zone increases linearly with speed. This is because of the high shear rates already obtained at low shaft speeds due to the very thin lubricant film in the contact zone. Here the shear stress

Fig. 13—Isothermal frictional torque as a result of shearing the grease in the vicinity of the contact for different temperatures and lip angles $\alpha-\beta$, $h_0 = 2 \mu\text{m}$, and $u = 10 \text{ m/s}$.

approaches the base oil asymptote such that Fig. 12 describes the base oil behavior. Figure 13 shows the contribution to the frictional torque from the grease in the vicinity of the contact. Here the non-Newtonian behavior of the grease is clearly visible.

It can be seen that if the temperature in both the contact zone and the vicinity of the contact is 70°C, the frictional torque is $\sim 0.4 \text{ Nm}$, which is a realistic value. When the temperature in the contact zone increases to 100°C, the frictional torque decreases to $\sim 0.2 \text{ N/m}$ according to the model, where the film thickness remains constant. This might not be the case in reality, where the film thickness will decrease with increasing temperature. Again the plots in Fig. 12 and Fig. 13 are not “traction curves,” where the temperature depends on speed.

CONCLUSION

A rheology model was derived to model the shear stress and first normal stress difference in lubricating grease in a seal-like geometry. The model was fit to the measurements from a rheometer for different temperatures. The measurement results showed that the normal stress is of the same order of magnitude as the shear stress and strongly depends on temperature.

Subsequently, the rheology model was used as input for the seal lip model. Here the normal stress tends to lift the seal lip from the shaft and allows for the formation of a lubricant film. A lift force larger than the specific seal lip force was predicted for low operating temperatures. For more realistic operating temperatures, like 70-100°C, the model predicts a lift force of 0.6-3 N/m, which balances the seal lip force for 10-50% in case of a low-contact pressure-bearing seal. For shaft seals with much higher lip forces, the normal stress effect contributes for less than 5% and can be neglected. The primary lubricating mechanisms for oil-lubricated seals are dominant here.

To obtain the high shear rates in the contact zone, the rheology model had to be extrapolated for more than two orders of magnitude. It has been assumed here that a grease film lubricates the contact and that the rheology model can be applied to very thin films. No evidence was found to confirm these assumptions, and therefore the lip model for the contact zone has to be interpreted with care. At the same time, the seal lip model predicts lift resulting from the grease in the vicinity of the contact at shear rates and film heights similar to the ones measured on the rheometer. This part of the lip model gives reliable results and can easily be included in existing film thickness and friction models for oil seals to make them applicable to grease-lubricated seals. One can also use the seal lip model to optimize seal design by utilizing the normal stress effect.

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Paper C

The Influence of Speed, Grease Type, and Temperature on Radial Contaminant Particle Migration in a Double Restriction Seal

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Microparticle image velocimetry (μ PIV) is used to measure the grease velocity profile in small seal-like geometries and the radial migration of contaminant particles is predicted. In the first part, the influence of shaft speed, grease type, and temperatures on the flow of lubricating greases in a narrow double restriction sealing pocket is evaluated. Such geometries can be found in, for example, labyrinth-type seals. In a wide pocket the velocity profile is one-dimensional and the Herschel-Bulkley model is used. In a narrow pocket, it is shown by the experimental results that the side walls have a significant influence on the grease flow, implying that the grease velocity profile is two-dimensional. In this area, a single empirical grease parameter for the rheology is sufficient to describe the velocity profile.

In the second part, the radial migration of contaminant particles through the grease is evaluated. Centrifugal forces acting on a solid spherical particle are calculated from the grease velocity profile. Consequently, particles migrate to a larger radius and finally settle when the grease viscosity becomes large due to the low shear rate. This behavior is important for the sealing function of the grease in the pocket and relubrication.

KEY WORDS

Grease Flow; Microparticle Image Velocimetry; Particle Migration; Double Restriction Seal

INTRODUCTION

Lubricating greases are widely used for lubrication of rolling bearings, seals, and gears. Grease provides also a sealing function

in order to protect the system against contaminants. In a previous study the authors used a microparticle image velocimetry (μ PIV) method to measure the grease velocity profile in a double restriction seal (DRS) and evaluated the sealing function of the grease (Green, et al. (1)). They measured a nonlinear velocity profile in a grease-filled pocket as indicated in Fig. 1 and described the sealing function as the ability to capture contaminant particles in the pocket between two sealing restrictions. Depending on the grease velocity profile in the pocket, the contaminant particles will migrate in the radial direction, which will be further investigated in this study.

The problem of small contaminant particles moving in a quiescent Newtonian fluid is well described in, for example, Batchelor (2) and Kundu and Cohen (3). The flow of particles in non-Newtonian fluids like lubricating grease is more complex. Putz, et al. (4) and Tabuteau, et al. (5) studied the settling of spherical particles in a quiescent yield stress fluid. Spherical particles are found to fall down and reach a constant terminal velocity when their density is significantly larger than the fluid density. If the particle density is below a critical density, the particle comes to a complete stop. In a rotating fluid, centrifugal forces can be added to the gravitational forces, resulting in radial particle migration (Annamalai and Cole (6)). In order to calculate the centrifugal forces on the particle, the fluid velocity profile should be determined, which is nonlinear in Couette flow with large gaps and non-Newtonian fluids. Kelessidis and Maglione (7) presented a methodology to calculate the velocities and shear rates in a Couette rheometer based on the Herschel-Bulkley model.

In the current study, the grease velocity profile will be evaluated in two different sealing geometries, one with a wide pocket and one with a narrow pocket between two sealing restrictions. Such geometry can be found in many sealing applications; two examples are shown in Fig. 2. The velocity profile in wide pockets will be modeled as a one-dimensional problem using the Herschel-Bulkley rheology model based on Kelessidis and

Fig. 1—Typical velocity profile between a rotating shaft and a stationary housing. The dashed line represents a Newtonian fluid and the continuous line represents a shear thinning fluid.

Maglione's (7) research. The grease velocity in the narrow pocket will be measured using the DRS setup from Green, et al. (1) for different greases, different shaft speeds, and temperatures. Subsequently, the grease velocity profile models are used as input for a radial migration model that predicts how solid contaminant particles settle in the grease pocket. Finally, the sealing function of the grease will be discussed.

PART 1—GREASE VELOCITY PROFILE

The experimental setup and measurement method for μ PIV will be briefly explained and the obtained velocity profiles in the wide and narrow pocket will be discussed.

Methodology

Green, et al. (1) quantified three-dimensional grease velocity profiles with μ PIV in a grease pocket with a rotating inner cylinder and stationary outer cylinder (housing). A detailed description of the setup and (μ PIV) methodology can be found in Green, et al. (1). The setup was modified with a new grease supply system and a hot air box to examine the influence of elevated ambient temperatures on the grease velocity profiles. The elevated temperature was monitored with thermocouples that were inserted in the housing at the outer radius of the grease pocket. Follow-

Fig. 2—Seals including a labyrinth: (a) oil seal with dust lip and wide grease pocket and (b) double lip bearing seal with narrow grease pocket.

ing previous work, fluorescent seeding particles with a diameter of $3.23 \pm 0.006 \mu\text{m}$ were used in combination with a $10\times$ magnification. A multipass cross-correlation scheme was applied, with a decreasing interrogation window size from 128×64 pixels to 32×16 pixels and typically 55–62 data points were acquired in each velocity profile, resulting in a spatial resolution of $0.024\text{--}0.027 \text{ mm}$. Measurements were performed in a plane located 0.1 mm behind the end face of the rotating shaft; that is, approximately 0.2 mm away from the transparent window as seen in Fig. 3. This is not the same plane as that in previous work and the typical image that is observed is shown in Fig. 1. The shaft is driven at speeds from 0.01 to 0.1 m/s shaft surface velocity.

Greases with different rheological behaviors were used: a relatively stiff NLGI2 grease, a softer NLGI1 grease, and a very soft NLGI00 grease. These greases are transparent and it is possible to set the focal plane for measurements some distance into the grease pocket volume, F2' in Fig. 3. While the shaft is rotating, a small grease flow of 0.1 mL/min is forced to flow from the pressure chamber through the first sealing restriction into the grease pocket and leave the test rig through the second sealing restriction as indicated by arrows in Fig. 3. This grease flow ensures that the grease pocket is filled with grease throughout the measurement. As shown previously in Green, et al. (1), this grease flow takes place close to the rotating shaft and is sufficiently small that it does not significantly influence the tangential velocity measurements.

The rheology model for the grease used here is the Herschel-Bulkley model including an extra base oil viscosity term as presented by Palacios and Palacios (8) and used in Baart, et al. (9). This four-parameter Herschel-Bulkley model is defined as

$$\tau_{HB} = \tau_y + K\dot{\gamma}^n + \eta_{bo}\dot{\gamma}, \quad [1]$$

where τ_y is the apparent yield stress, K is the consistency parameter, n is the shear thinning parameter, and η_{bo} is the base oil

Fig. 3—Close-up of the grease pocket in the DRS setup with the focus plane of the camera indicated by F2'.

viscosity. The parameters τ_y and η_{bo} are measured directly and K and n result from fitting the model to the rheometer flow curve. The rheological properties were measured at 25°C using a rotating parallel plate rheometer according to the methods described by Baart, et al. (9). In the rheometer, wall slip at the plate surfaces is present below shear rates of $\sim 10 \text{ s}^{-1}$. The parameter fit for K and n is done for shear rates above this value and the parameters are presented in Table 1. The greases in Table 1 are equal to the greases used in previous studies by Westerberg, et al. (10) and Li, et al. (11). Because of the currently used four-parameter model and the use of an automatic best fit method, different values for the rheology parameters were found compared to the earlier manual fit used in Westerberg, et al. (10).

The base oil viscosity in Eq. [1] was calculated at the correct temperature from the viscosities at 40 and 100°C using the Walther equation (Sánchez-Rubio, et al. (12)). For determination of the apparent yield stress, a vane geometry was used on the rotational rheometer. This setup included a rotating vane with an outer diameter of 20 mm and length of 20 mm, including six vanes having a 0.1 mm thickness, shown in Fig. 4. The vane geometry was submerged 20 mm into a cup with a 25-mm inner diameter filled with grease. The wall of the cup was made of rough sandpaper to prevent wall slip (Keentok (13); Barnes (14); Barnes and Nguyen (15)). During the measurement, an increasing torque was applied on the vane and the angular velocity was measured. Figure 5 shows the results of the angular velocity against the applied torque for the NLGI2 and NLGI1 grease. Below a critical torque T_c , the angular velocity increased linearly with the applied torque, which was due to small elastic deformation at stresses below the apparent yield stress. Above the critical torque the angular velocity increased quickly and nonlinearly because the grease started to flow at stresses above the apparent yield stress. This point of critical torque can be visually observed in the graph and the apparent yield stress was subsequently defined as

$$\tau_y = \frac{T_c}{2\pi R^2 \left(h + \frac{R}{3}\right)}, \quad [2]$$

where T_c is the critical torque, R is the vane radius, and h is the penetration depth of the vane into the cup. For greases with a high yield stress, the penetration depth h can be reduced, which consequently reduces the required torque to reach the apparent yield stress point. The average apparent yield stress can be determined from Fig. 5 and is shown in Table 1 for the NLGI1 and NLGI2 greases at 25°C.

170 Velocity Profile in Wide Pocket

The grease velocity profile in the wide pocket, where any influence of the end walls can be neglected, is approached as a one-

Fig. 4—Schematic drawing of the vane geometry for apparent yield stress measurements.

dimensional case. For a Newtonian fluid in a Couette type of geometry, the velocity profile only depends on the shaft speed and geometry; that is, the gap height relative to the shaft diameter. In the case that the gap is large compared to the shaft diameter, a nonlinear fluid velocity (nonconstant shear rate) may be expected. This tangential velocity profile is given for Newtonian fluids as (Batchelor (2))

$$u(r) = U_s \left[\frac{r_o/r - r/r_o}{r_o/r_i - r_i/r_o} \right], \quad [3]$$

where U_s is the shaft surface velocity, r_i is the inner radius or shaft radius, and r_o is the outer radius or housing radius. For sufficiently small gaps where the gap relative to the shaft radius is small, that is, $(r_o - r_i)/r_i \ll 1$, Eq. [3] approaches a linear velocity profile.

Lubricating greases generally show a nonlinear flow behavior and consequently Eq. [3] cannot simply be applied. The fluid velocity profile is calculated based on the four-parameter Herschel-Bulkley rheology model using the parameters in Table 1. Kelesidis and Maglione (7) presented a methodology to calculate the tangential velocity through the gap height in a Couette rheometer based on the three-parameter Herschel-Bulkley model. To use the four-parameter Herschel-Bulkley model from Eq. [1], their flow equation was modified. Figure 6 shows the model results for a Newtonian oil according to Eq. [3] and for the NLGI2 and NLGI1 grease based on the four-parameter model at 0.01 and 0.05 m/s shaft speed, respectively. The velocity profile of the Newtonian oil is linear because $(r_o - r_i)/r_i$ is small. The greases show a small deviation from the Newtonian case due to shear thinning. However, when the shaft velocity is increased, this effect is reduced.

Velocity Profile in Narrow Pocket

The velocity profile in the narrow pocket is significantly influenced by the presence of the side walls; that is, left and right walls in Fig. 3. Consequently, lower velocities than those based on the

TABLE 1—GREASE RHEOLOGICAL PROPERTIES BASED ON THE FOUR-PARAMETER HERSCHEL-BULKLEY MODEL FIT TO A FLOW CURVE OBTAINED FROM A PARALLEL PLATE RHEOMETER AT 25°C

Grease Type	τ_{yield} (Pa)	K (Pa.s ⁿ)	n	η_{bo} (Pa.s)
NLGI 2	500	8.2	0.63	0.25
NLGI 1	260	61	0.42	0.49
NLGI 00	15	12	0.63	0.89

Fig. 5—Yield stress (angular velocity) measurement results for NLGI1 and NLGI2 grease.

one-dimensional analysis are expected. The DRS setup and μ PIV
method from Green, et al. (1) were used to evaluate the grease
tangential velocity profile at different shaft speeds and temper-
atures for three greases. Figure 7 presents the measured grease
velocity profile of the NLGI2 grease at several shaft speeds in the
F2' plane from Fig. 3. The velocity curves of the grease show sig-
nificant nonlinearity due to shear thinning effects. Consequently,
higher shear rates than for a Newtonian fluid case are present

close to the rotating shaft. Close to the stationary housing at 1.5
mm from the shaft (radial position 0.0215 mm) the grease seems
to stand still, indicating an apparent unyielded area.

Plotting the same data from Fig. 7 with the velocity on a log-
arithmic scale as in Fig. 8, shows that the data points approach a
straight line. This indicates an exponential velocity profile in the
form

$$u(r) = U_s e^{\beta(r-r_i)} \quad [4]$$

Fig. 6—One-dimensional tangential velocity profiles for a Newtonian fluid and for greases based on the four-parameter Herschel-Bulkley model.

Fig. 7—Grease velocity of the NLGI2 grease in a narrow seal pocket at different shaft speeds at $T = 25^\circ\text{C}$. The velocity profile is nonlinear and the grease at the outer radius seems to be unyielded.

where U_s is the shaft surface velocity, r is the radius from the center of rotation, and r_i is the radius of the shaft surface, where β is a shear thinning parameter of the grease. Equation [4] is plotted in Fig. 8 to show that Eq. [4] gives a non-zero velocity at the housing wall. Therefore, Eq. [4] is corrected with Eq. [3] to obtain

$$u(r) = U_s e^{\alpha(r-r_i)} \left[\frac{r_o/r - r/r_o}{r_o/r_i - r_i/r_o} \right], \quad [5]$$

where r_o is the outer radius of the gap and α is a new shear thinning parameter. The α -value was found to be constant ($\alpha = 225 - 3,000$) for the NLGI2 grease at all shaft speeds. Equation [5]

Fig. 8—Grease velocity of the NLGI2 grease in a narrow seal pocket at different shaft speeds at $T = 25^\circ\text{C}$. The straight lines for all experiments indicate an exponential velocity profile.

Fig. 9—Grease velocity of NLGI2, NLGI1, and NLGI00 greases in a narrow seal pocket at $T = 25^{\circ}\text{C}$. The different greases show different amounts of shear thinning.

was tested with data from the three different greases at different shaft speeds, and the results are shown in Fig. 9. Also, Eq. [5] is plotted here for each measurement condition and the α -values were determined for the different greases. This confirmed the idea that the α -value can be used as a grease property to describe its flow behavior or rheology in the DRS setup. The different α -values for the greases are presented in Table 2.

The shear thinning properties of grease are temperature dependent, and this behavior should reasonably be reflected in the α -value. Figure 10 shows flow measurement results of the NLGI2 grease at four different temperatures. At each temperature a fit was made using Eq. [5] with different α -values. The model fit (continuous line in Fig. 10) at $T = 25^{\circ}\text{C}$ appears to be a poor fit. However, this fit is based on the results at five different speeds in Fig. 8 where it fits very well with the other shaft speeds. New α -values were fitted for the other temperatures and are presented in Table 3. The table also contains the temperature-dependent α -values for the NLGI1 grease.

A model fit using the data in Table 3 was made to find an equation for the temperature dependence of the α -value, giving

$$\alpha = A \ln(T) + B, \tag{6}$$

TABLE 2—SHEAR THINNING PARAMETER α FOR DIFFERENT GREASE TYPES IN THE F2' PLANE IN THE NARROW POCKET AT 25°C

Grease Type	α -Value (m^{-1})
NLGI2	−3,000
NLGI1	−2,000
NLGI00	−1,000

TABLE 3—TEMPERATURE-DEPENDENT α -VALUES FOR NLGI2 AND NLGI1 GREASE IN THE F2' PLANE IN THE NARROW POCKET

Grease Type	25°C	50°C	70°C	100°C
NLGI2	−3,000	−2,200	−1,800	−1,400
NLGI1	−2,000	−1,450	−1,200	−900

where A and B are grease parameters given in Table 4 and T is temperature.

The temperature model, Eq. [6], for the shear thinning parameter α is substituted into Eq. [5] to obtain the equation for the grease velocity in the DRS F2' plane as a function of grease type, temperature, shaft speed, radial position, and geometry according to

$$u(r, T) = U_s e^{[A \ln(T) + B](r - r_i)} \left[\frac{r_o/r - r/r_o}{r_o/r_i - r_i/r_o} \right]. \tag{7}$$

Only the narrow pocket's radial dimensions are included in Eq. [7], and the axial dimension or width of the pocket is not included. This width and the position of the measurement plane should be included in the α -value, which has the dimension of m^{-1} . For the current work only the measurement plane at the position F2' is evaluated.

TABLE 4—TEMPERATURE PARAMETERS FOR α -VALUE MODEL IN THE F2' PLANE IN THE NARROW POCKET

Grease Type	$A \text{ [} (\text{mT})^{-1} \text{]}$	$B \text{ (m}^{-1}\text{)}$
NLGI2	1,160	−6,720
NLGI1	790	−4,540

Fig. 10—Grease velocity of NLGI2 grease in a narrow seal pocket as a function of temperature. Shear thinning decreases with increasing temperature.

DISCUSSION

Equations for predicting the grease velocity profile in wide and narrow pockets, as can be found in double restriction seals, have been presented. It is shown that the grease velocity profile in a wide pocket (Fig. 6) significantly deviates from the velocity profile in a narrow pocket (Fig. 9). The wide pocket model assumes a one-dimensional case where the effects of the side walls, that is, left and right walls in Fig. 3, are neglected. Because the width of the narrow pocket is almost equal to the height and because the measurement is performed at only 0.2 mm from the transparent window wall, the side walls do have a significant influence on the grease velocity profile here. This is due to the momentum equation, which includes the (shear) stress tensor τ_{ij} , and $\partial\tau_{ij}/\partial x_j$ in both spatial directions has to be accounted for. For a one-dimensional scenario, however, where the width is much greater than the height, the rate of change of the shear stress is zero in the axial direction. In the narrow gap situation, the walls slow down the flow and hence lower velocities are measured. Because the measurement method—that is, the time step in velocity measurement—was optimized for velocities close to the shaft, the signal-to-noise ratio at low velocities, <0.001 m/s, was relatively large, as shown in Fig. 8. The calculation method with interrogation windows may result in misleading results at the walls (Green, et al. (1)). The effect of the side wall on the velocity profile cannot be predicted by the one-dimensional model and an extension to two dimensions should be made if more detailed predictions are required in future work.

However, the grease velocity profiles in the F2' plane in the narrow pocket can be calculated using a simple analytical/empirical equation (Eq. [7]) where the shear thinning parameter α is temperature dependent. In Table 3 the α -values are found to overlap, meaning that equal velocity profiles in the pocket can be found for the two different greases at different temperatures.

For example, $\alpha = -1,450$ for the NLGI1 grease at $\sim 50^\circ\text{C}$, which equals $\alpha = -1,400$ for the NLGI2 grease at $\sim 100^\circ\text{C}$.

Furthermore, the Herschel-Bulkley model in Eq. [1] contains a yield stress, or apparent yield stress (Barnes (14)), below which the grease is assumed to stand still. In Fig. 6 such an unyielded area at the outer radius is visible at very low shaft speed; for example, 0.01 m/s. At higher shaft speed, for example, 0.05 m/s, such unyielded areas are no longer present and all of the grease flows. In the narrow pocket, where the side walls significantly influence the velocity profile, such an apparent unyielded area is clearly visible as shown in Fig. 7. However, here the unyielded area decreases with increasing shaft speed and is smaller for greases that are less shear thinning, as shown in Fig. 9. It is well known that the yield stress of greases depends on temperature (Gow (16)) and, consequently, a smaller unyielded area was measured at elevated temperatures, as shown in Fig. 10. However, although it appears that some grease was apparently unyielded in the narrow pocket in Fig. 7, in the experiment some very slow (creep) flow was measured. This becomes evident from Fig. 8, where the velocity is plotted on a logarithmic scale. Here it is shown that the grease does flow with a velocity that is an order of magnitude lower than the velocity close to the shaft. The occurrence of wall slip at the housing wall where shear rates are low may also explain that the velocity does not approach zero. Due to wall slip the bulk grease in the apparently unyielded area may move without being sheared. In combination with some creep flow as described above, this explains the non-zero velocity values at the outer radius in Fig. 8. The one-dimensional four-parameter Herschel-Bulkley rheology model would have predicted the grease to stand still in such a case due to the yield stress term in the model. In order to predict creep flow behavior, a rheology model with additional parameters should be used to include a maximum viscosity plateau at low shear rates.

Wall slip is expected to occur in the parallel plate rheometer experiments at low shear rates, as shown by Keentok (13) and Baart, et al. (9), and typically occurs at shear rates at the geometry wall below $\sim 10 \text{ s}^{-1}$. At the shaft surface in the DRS such wall slip was not observed from the velocity profile measurements. By using the derivative of Eq. [7] to calculate the shear rates in the DRS, it becomes clear that at the lowest shaft velocity of 0.01 m/s the shear rate at the shaft surface is 30 s^{-1} and therefore is large enough to avoid wall slip.

PART 2—CONTAMINANT MIGRATION

The sealing function of grease was discussed in Green, et al. (1); it was suggested that solid contaminant particles that pass through the first sealing restriction are captured by the grease in the pocket, or grease chamber, and therefore reduce the probability of particles passing through the second restriction. These particles typically have a larger density than the grease and consequently migrate to a larger radius in the pocket, due to centrifugal forces. In the second part of the article the migration of solid contaminant particles in the grease due to centrifugal body forces is simulated using the grease velocity profiles from Part 1.

Migration Model

Solid contaminant particles in the grease pocket are assumed to move with the same circumferential velocity as the grease. Consequently, centrifugal forces act on the particle and force the particle to migrate to a larger radius. The particle is slowed down due to drag forces as the particle migrates through the grease. Any hydrodynamic effects due to shear, shear thinning, or normal stresses, as discussed in fundamental work by, for example, Karnis and Mason (17) or Gauthier, et al. (18) for naturally buoyant particles, are neglected. The force balance is written as

$$F_{c,r} + F_{d,r} = m \cdot a_r, \quad [8]$$

where F_c is the centrifugal force, F_d is the drag force, m is the particle mass, and a_r is the particle acceleration, all in the radial direction. The particle mass can be calculated from the density ρ_p such that $m = 4\pi\rho_p a^3/3$ assuming that the contaminant particle can be approached as a sphere with an effective particle radius a . With the particle suspended in the grease, a correction for the difference in density between particle and grease has to be made. The equation for the centrifugal force then reads

$$F_{c,r} = \frac{4}{3}\pi a^3 (\rho_p - \rho_g) \frac{U_\theta^2}{r}, \quad [9]$$

where U_θ is the circumferential velocity and ρ_g is the grease density. The drag force is predicted using the Stokes drag equation (Batchelor (2)):

$$F_{d,r} = -6\pi a \eta_r U_{p,r} \quad [10]$$

for a spherical particle moving through a stationary or quiescent fluid with the Reynolds number $\text{Re} \ll 1$. Here η_r is the grease viscosity and $U_{p,r}$ is the particle velocity given that the grease velocity in the radial direction is zero. Although the grease viscosity and the grease viscosity will vary over the particle height, it is assumed that Eq. [10] can be used when the effective particle diameter is small. The local grease viscosity depends on the local

shear rate and is calculated for the one-dimensional case from the four-parameter Herschel-Bulkley model in Eq. [1] as

$$\eta_r = \tau_y \left(\frac{dU_\theta}{dr} \right)^{-1} + K \left(\frac{dU_\theta}{dr} \right)^{n-1} + \eta_{bo}. \quad [11]$$

In Part 1 it was shown that the grease velocity profile U_θ in the pocket is nonlinear and therefore the shear rate is a function of the radial position in the pocket. Consequently, the grease viscosity and drag force are a function of the radial position.

Because contaminant particles are small and the grease viscosity is high, the radial migration velocities will be very low. Particle accelerations a_r can therefore be neglected, which sets the right-hand side of Eq. [8] to zero. This assumption was checked and validated. Substituting Eq. [9] and Eq. [10] into Eq. [8] gives the local particle velocity in the radial direction as

$$U_{p,r} = \frac{2}{9} a^2 \frac{1}{\eta_r} (\rho_p - \rho_g) \frac{U_\theta^2}{r}, \quad [12]$$

where η_r and U_θ are calculated from the grease velocity profile. Due to the nonlinear rheology model including the shear thinning of the grease, it is not possible to solve Eq. [12] analytically, and a numerical integration is used to calculate the radial migration of the particle as a function of time. At each time step the particle velocity is calculated using Eq. [12] and multiplied by the sufficiently small time step $dt = 1 \text{ s}$ to calculate the radial migration distance.

Migration Results

To predict the radial migration position of a solid contaminant particle in a grease pocket, Eq. [12] and the velocity profiles obtained in Part 1 are used for the wide and narrow pocket and the default model parameters in Table 5. Figure 11 shows the results for the wide and narrow pocket in the same graph. It is only after approximately 10 min that a difference between the grease types and pocket width becomes significant. For the wide pocket the radial migration of the particle in the different greases is almost equal due to the very small differences in the grease velocity profile; see also Fig. 6. For the narrow pocket, differences become clear after approximately 1 h, where the particle migrates fastest in the lowest consistency grease. In the high-consistency NLGI2 grease, which has the highest viscosity at low shear rates, the particle velocity is lowest.

The radial migration also depends on the shaft speed, particle diameter, and temperature. This is shown in Figs. 12 and 13. Figure 12 first shows the radial position of a $14\text{-}\mu\text{m}$ -diameter particle after 100 h at different shaft speeds assuming a constant temperature of 25°C at all speeds. It is shown that at higher shaft speeds the particles migrate to a larger radius due to the larger centrifugal forces acting on the particle and reduced viscosity due to shear

TABLE 5—DEFAULT PARAMETERS FOR THE CALCULATION OF CONTAMINANT PARTICLE MIGRATION

Temperature	25	$^\circ\text{C}$
Particle diameter	14	μm
Particle density	2,100	kg/m^3
Grease density	930	kg/m^3
Shaft speed	1	m/s

Fig. 11—Radial migration as a function of time; position of a $14\text{-}\mu\text{m}$ -diameter particle for different greases at $U_s = 1\text{ m/s}$ and $T = 25^\circ\text{C}$.

thinning. Figure 13 shows a very similar trend where larger size particles migrate to a larger radius in a time period of 100 h. By increasing the temperature, the grease velocity profile becomes more linear and the viscosity decreases (see Part 1 and Fig. 10). Consequently, contaminant particles migrate further at elevated temperatures.

DISCUSSION

Results of the radial particle migration showed the settling of contaminant particles in a seal-like geometry. In a wide pocket, the three different greases give very similar results, where particles migrate within 2 h to half the height of the pocket; see Fig. 11. In the narrow pocket this migration takes significantly longer for

Fig. 12—Radial migration as a function of speed; position of a $14\text{-}\mu\text{m}$ -diameter particle for different greases after 100 h at $T = 25^\circ\text{C}$.

Fig. 13—Radial migration as a function of particle size; position of a particle for different greases after 100 h at $U_s = 1$ m/s and $T = 25^\circ\text{C}$.

Q6

all greases: just over 2 h for the NLGI00 grease, 20 h for the NLGI1 grease, and over 70 h for the NLGI2 grease at a shaft speed of 1 m/s.

It was shown in a previous study by Green, et al. (1) that in the narrow pocket an axial grease flow due to breathing of the bearing system or relubrication mainly takes place in the first few tenths of a millimeter close to the rotating shaft. Li, et al. (11) evaluated the flow depth for the three greases in more detail but in static conditions; that is, no shaft rotation. Their experimental setup contained two sealing restrictions in a pipe flow, similar to the narrow pocket, and the flow depth was measured as a function of flow rate and grease type. They found a significantly lower flow depth for the NLGI1 and NLGI2 grease than for the NLGI00 grease. Consequently, contaminant particles that have migrated further away from the shaft in low-consistency grease can still be picked up in an axial grease flow. They also showed that when there is just one restriction, the flow takes a length of the order of the pipe diameter to fully develop. This means that in a wide pocket the whole grease volume may flow in the axial direction due to a pressure gradient except for the volume close to the corners.

The observations described above are related to the operation cycle in a sealed and greased-for-life bearing unit; for example, Fig. 2b. Here contaminant particles may enter through the first sealing restriction into the grease pocket. During system operation, speeds and temperatures are relatively high and particles migrate more easily to a large radius as indicated in Fig. 11. At a later stage, when the rotational speed is reduced or stopped and temperature decreases, breathing or axial grease flow into the bearing system may take place. As shown in Green, et al. (1) and Li, et al. (11), this axial flow will mainly be close to the shaft surface in a narrow pocket and has a rather limited penetration into the grease pocket, especially when the temperature has

dropped. Consequently, particles that have migrated further than the flow depth will not flow into the bearing. In the wide pocket the whole grease volume flows, which increases the probability of contaminant ingress but also enables refreshment of the grease when relubrication is applied.

It has been assumed that the whole pocket between the sealing restrictions is filled with grease, as indicated in Fig. 14a, and that all of the grease is being sheared. These conditions will generally not take place in applications because the pocket is normally not fully filled with grease and grease may leak out in the case of noncontacting sealing restrictions. Additionally, it was shown in Part 1 (for example, Fig. 7) that in the narrow pocket the highest shear rates take place close to the shaft and subsequently the grease will mechanically age here. Consequently, the grease consistency decreases, which may also result in leakage. Finally, after some time, less grease will contact the shaft and only in the locations close to the contact restrictions can some grease meniscus be present, as indicated in Fig. 14b.

Q7

Fig. 14—Indication for grease position: (a) fully filled narrow pocket and (b) partly filled wide pocket.

The seal pocket geometry is important in order to hold a certain amount of grease that fulfills the sealing function as described in this study. The velocity profile in the grease meniscus will be influenced by the presence of the side wall and, consequently, the velocity profile, as measured in the narrow pocket, can be also expected here. This emphasizes the importance of understanding the nonlinear flow behavior of the grease as presented in Part I regarding the sealing function of the grease.

CONCLUSIONS

The grease velocity profile in the narrow pocket DRS geometry was measured for different shaft speeds, grease types, and temperatures. The grease shear thinning behavior or nonlinear correction on the Newtonian velocity profile in the narrow pocket does not depend on shaft speed and is defined by the α -value. This α -value is grease type dependent and changes with temperature, resulting in less shear thinning at higher temperatures. As a consequence, equal velocity profiles can be found for the different greases at different temperatures. The side walls of the small grease pocket geometry are found to influence the measured velocity profile significantly. This was proven by comparison with a one-dimensional model based on the Herschel-Bulkley model. The one-dimensional model was also used to predict the grease velocity profile in a wide pocket.

The radial migration of solid contaminant particles was simulated in a narrow and wide seal pocket. It was concluded that in the wide pocket contaminant particles migrate to a larger radius than in a narrow pocket. The migration also depends on the grease type and operating conditions. In a narrow pocket the radial migration reduces the probability for contaminants to flow into the bearing where an axial grease flow, for example, caused by breathing of the bearing, only takes place close to the shaft. In the wide pocket the whole volume of grease in the pocket flows in the axial direction and consequently also transports contaminant particles that are located at the outer radius. This explains the sealing function of the grease in the seal pocket in a double restriction seal.

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Paper D

Contaminant Migration in the Vicinity of a Grease Lubricated Bearing Seal Contact

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Abstract

Lubricating grease is commonly used for lubricating ‘sealed and greased for life’ rolling element bearings. This grease also provides an additional sealing function to protect the bearing against ingress of contaminants. In this work the sealing function of lubricating grease in the vicinity of the seal lip contact has been studied experimentally by measuring the migration of spherical fluorescent contaminant particles in the vicinity of the contact, as a function of shaft speed and lubricant type. The experimental results reveal that in some greases contaminant particles migrate towards the sealing contact where the shear rate reaches its highest value. However, for other greases, Newtonian base oils, and elastic fluids, this is not necessarily the case and contaminant particles consistently migrate away from the sealing contact. Various physical phenomena have been investigated to explain the difference in migration behavior. It is concluded that migration towards the sealing contact is driven by the viscosity gradient and migration away from the sealing contact is related to the Weissenberg number.

The sealing function of grease in the vicinity of the sealing contact is due to the migration of contaminant particles. The migration reduces the probability of particles to reach the sealing and bearing contacts.

Keywords

Radial lip seal, grease rheology, sealing function, contaminant particle migration.

INTRODUCTION

Radial lip seals are traditionally used to prevent loss of lubricant. However, in ‘sealed and greased for life’ bearings or bearing systems that are lubricated with grease, their main function is to prevent the ingress of contaminants. Such a system is shown in Figure 1. Here the very thin lubricant film between the seal lip and the shaft, or the bearing inner ring, is the main protection against contaminants entering the bearing. For a review on oil and grease lubricated seals, the reader is referred to Baart et al. [1].

Many textbooks on (grease) lubrication state that grease also increases the sealing performance. The Automotive Lubricants Reference Book [2] has mentioned an advantage of grease that it prevents the ingress of dirt contaminants into the system. Boner [3] recommends NLGI 1, 2 or 3 greases above the softer greases for better sealing. Also Miller [4] and the NLGI lubricating grease guide [5] state that grease

helps to seal out contaminants such as water and dirt but do not comment on the ‘sealing function’ mechanisms. Lansdown [6] states that grease can form a very effective seal against ingress of dirt or other contaminants into the system. Their explanation for this is that contaminants are less likely to be transported through the bearing due to the semisolid nature of grease. Unlike oil, grease does not continuously flow through the bearing and grease on the covers of the bearing can trap dirt and prevent it from reaching the contact surfaces.

Green et al. [7] measured the grease flow and grease velocity profile in a seal pocket in-between two seal lips in order to explain the sealing function of grease. Further experiments and a study on radial migration of contaminant particles in a double restriction seal were presented in Baart et al. [8]. They predicted the settling of solid contaminant particles inside a seal pocket filled with grease and discussed the sealing function of the grease in the presence of an axial grease flow. In addition to this, the grease in the vicinity of the sealing contact may play an important sealing role as well. After all, this is the first barrier for

contaminants after passing the narrow gap sealing contact. Even in the absence of an axial grease flow, contaminants may migrate in axial direction, toward or away from the sealing contact, when the lubricant in the vicinity of the contact is being sheared due to the rotation of the shaft or bearing ring. A pre-study on this subject has been presented in Baart et al. [9] where the migration of graphite particles in different lubricants was investigated. Their results show that in a Newtonian fluid, the particles migrate *away* from the sealing contact whereas in the case of grease, the particles migrate *towards* the sealing contact. They conclude that this migration is due to the rheology or shear thinning properties of the grease. It was found later that the graphite particles are too soft and break due to the high shear stresses in the high viscosity grease. Consequently, the increase in the number of particles in the vicinity of the contact is not due to migration but due to the fragmentation of the particles. For the current study, the experimental method has been improved and solid spherical fluorescent particles have been used instead. These particles can either represent dirt particles from the environment or internal wear particles from the bearing. Effects that may initiate migration are evaluated and experimental results on different greases are presented. The particle migration is correlated to the rheological properties of the grease.

Figure 1 ‘Sealed and greased for life’ Deep Groove Ball Bearing (DGBB) with contacting seals which are lubricated with grease.

BACKGROUND

Various publications are available on particle migration for several types of fluid velocity profiles, fluid rheology and particle types. Karnis and Mason [10] performed an experimental study on particle migration in a pipe flow and Couette flow at low Reynolds numbers. In the pipe flow, they found that for an elastic fluid, the particles migrate to the pipe centre where the shear rate is low. In the Couette flow,

the particles migrate to the outer cylinder, which was shown to be independent of the cylinder rotational direction and whether the inner or outer cylinder was rotating. They did not observe such migration in Newtonian fluids and ascribed the effect to the normal stress differences in the fluid. Their work was continued by Gauthier et al. for Couette flow [11] and Poiseuille flow [12], including also shear thinning fluids. It was found that in this case, the particles migrate in the direction where shear rate is high, i.e. to the inner cylinder and pipe wall, respectively. The migration behavior is independent of the particle shape since similar results were found for small disk and rod shaped particles. Another study including different particle sizes was done by Husband et al. [13]. They studied a homogeneous mixture of large and small naturally buoyant particles suspended in Newtonian oil in a Poiseuille pipe flow, Couette flow with rotating outer cylinder and free surface flow. In all cases, the particles migrate to the area where the shear rates are low.

In addition to the experimental work, theoretical studies have been published to explain the physics in the experimental observations. Naturally buoyant particles are considered so that the density difference between the particles and fluid does not have to be included. This brings the main focus on the fluid velocity profile, fluid rheology and particle size. Yang et al [14] compare and discuss the analytical equations of Saffman, McLaughlin, and Auton, which are based on a perturbing Stokes law with inertia or on a perturbing potential flow with a little vorticity. Ho and Leal [15] have derived analytical equations to calculate the particle trajectory in an elastic fluid where normal stress differences drive the particle migration. A numerical study has been done by Huang and Joseph [16]. Their simulations include different amounts of shear thinning and elasticity of the fluid in a pipe flow. They also did simulations for different Reynolds numbers. In a non-shear thinning Newtonian fluid at high Reynolds numbers, they predicted a small migration towards the wall of the pipe. By increasing the amount of shear thinning, the migration to the pipe wall becomes more significant. In a non-shear thinning but elastic fluid the particles migrate to the centre of the pipe. When including some shear thinning together with elasticity, part of the particles migrate to the wall while the majority of particles migrate to the pipe centre.

In a previous study, Baart et al. [17] considered the non-Newtonian grease rheology including shear thinning and normal stress differences. This grease flow behavior can be compared to the results found in e.g. Gauthier et al. [11] who suggest that due to the shear thinning behavior, the contaminants will

migrate to the high shear rate zone, i.e. towards the sealing contact where the gap is small and shear rates are high. Due to the normal stress differences, the grease contaminants are expected to migrate to the low shear rate zone, i.e. away from the sealing contact. Both these effects are present in the grease.

The typical seal geometry, which is considered, is shown in Figure 2. Here the lower plane is the shaft surface, which rotates with a surface velocity U_s . The top plane is the seal lip surface with a lip angle α . The space between the surfaces is filled with fluid, and a spherical particle, with radius a , is assumed to be present.

Figure 2 Particle in a fluid velocity field in the vicinity of the sealing contact. Migration in the x -direction is studied.

Before experimentally verifying the hypothesis that the grease rheology, i.e. hydrodynamic forces, cause the contaminant migration as discussed above, some other effects have been considered:

- a) Pumping action of the seal
- b) Brownian motion
- c) Particle inertia
- d) Vortices

Pumping action of the seal

Oil seals are designed to pump oil back into the system while bearing seals are designed to pump a little amount of lubricant outwards for better contaminant exclusion [1]. In case significant pumping is present, contaminants may flow with the grease and move towards the sealing contact. Horve [18] and Kammüller [19] showed that the pumping rate of an oil seal is a function of the lubricant viscosity, lip angle, seal material, surface roughness, and rotational speed, and is therefore rather complex. However, from the pre-study presented in Baart et al. [9], it was found that pumping is not significant in the experimental setup used and does not dominate the migration phenomena. The pumping rate of the seal is negligible at low shaft speed. At high shaft speed, some pumping may be present but its significance has to be

determined from the experimental results. Furthermore, contaminant particles that migrate in the grease have been found not to reach the sealing contact but do accumulate at a small distance away from the contact, which may be attributed to the grease rheology.

Brownian motion

The Peclet number is used to check the importance of hydrodynamic forces relative to Brownian forces. In the case that the Peclet number is larger than 1, particle migration is dominated by hydrodynamic forces. The Peclet number as defined in Snijkers et al. [20] reads

$$Pe = \frac{\eta \dot{\gamma} a^3}{k T} \quad (1)$$

where η is the viscosity, $\dot{\gamma}$ the shear rate, a the particle diameter, k the Boltzmann constant, and T the temperature. The Peclet number varies in the measurement domain and is estimated for conditions where a minimum Peclet number is expected. For example, with $\eta = 0.01$ Pa·s, $\dot{\gamma} = 10$ s⁻¹, $a = 10$ μm, and $T = 323$ K, the Peclet number is of the order $O(10^{-4})$ which is much larger than 1. Consequently, hydrodynamic forces are dominating over Brownian forces.

Particle inertia

The Reynolds number determines the significance of inertial forces relative to viscous forces on particle motion. In order to be in the viscous or hydrodynamic domain, the Reynolds number should be much smaller than 1. The Reynolds number for particles in a viscous flow, as defined in Snijkers et al. [20] reads

$$Re = \frac{\rho_f \dot{\gamma} a^2}{\eta} \quad (2)$$

where ρ_f is the fluid density. The Reynolds number varies in the measurements domain and is found to approach a number of the order $O(10^0)$ close to the sealing contact where the shear rate increases significantly and the viscosity reaches its lowest value. The Reynolds number is plotted in Figure 3 as a function of the distance to the sealing contact for conditions where the shaft speed is 233 rpm at the temperature is 70 °C. Here the Reynolds number approaches 1 in the first 0.1 mm from the sealing contact. Consequently, inertial forces on the particle

may dominate over viscous forces and different migration behavior can be expected locally. At higher speeds and temperatures where the grease viscosity drops, larger Reynolds numbers can be expected.

Figure 3 Reynolds number in the vicinity of the sealing contact at a shaft speed of 233 rpm and a temperature of 70 °C.

Vortices

In a Couette type of flow, Taylor-Görtler vortices may be developed above a critical fluid velocity. Qu [21] showed that in a seal application, such vortices are present at ~ 2000 rpm and this vortex flow is so strong that it can prevent oil leakage even when the sealing gap is relatively large. Wennehorst and Poll [22] observed particles migrating away from the sealing contact and ascribed this phenomena to the existence of vortices. Due to these vortices, the circulating flow might transfer particles through the fluid resulting in an effective migration of contaminant particles. The Taylor number is the ratio between inertia forces and viscous forces on the fluid flow. Above the critical Taylor number of $Ta_c = 1700$, Taylor-Görtler vortices can be expected. The Taylor number is calculated for a Couette flow as,

$$Ta = r_i(r_o - r_i)^3 \left(\frac{\rho_f \Omega}{\eta} \right)^2 \quad (3)$$

where ρ_f is the fluid density, Ω the shaft angular velocity, r_i the shaft radius, r_o is the seal radius and η the fluid viscosity. Realistic test conditions are chosen which would give the largest Taylor number with $r_i = 41$ mm, $(r_o - r_i) = 0.5$ mm, $\rho_f = 950$ kg/m³, $\Omega = U/r_i = 2.44$ rad/s, and $\eta = 0.024$ Pa·s (oil viscosity at 70 °C) resulting in a Taylor number of the order $O(10^{-3})$. This is much smaller than 1700 and therefore Taylor-

Görtler vortices are not expected to be present in the experiments.

Considering the different effects that may cause contaminant migration, it is found that hydrodynamic forces on the particle due to the grease rheology will dominate particle motion, except for the first few tenths of a millimeter where inertia forces may be present. This hypothesis is further evaluated with experiments on different fluids and greases.

METHOD

An experimental setup is presented to study the migration of contaminant particles in the vicinity of the sealing contact, where grease is applied in-between two contacting seal lips.

Measurement setup

The measurement setup consists of two FKM bearing seals that are positioned back-to-back to create a closed volume, see Figure 4. The lip angle α as indicated in Figure 2 is 12°. The seals are mounted on a hollow and transparent sapphire shaft which has an outer diameter of 82 mm. The sapphire shaft has thermal conductivity that is similar to that of steel and it is polished for excellent optical transparency. The alignment of the seals to the shaft is such that Dynamic Run Out (DRO) < 0.08 mm, Shaft To Bore Misalignment (STBM) < 0.1 mm and Seal Tilt (ST) < 0.5°. A digital camera equipped with a high magnification lens is used in combination with a 90° mirror to look through the hollow transparent sapphire shaft from the inside as indicated in Figure 4.

Spherical fluorescent MF-Rhodamine B-particles are used with an average diameter of 10.20 µm and a standard deviation of 0.17 µm. The particles are excited by 532 nm wavelength laser light to make them easily observable. The fluorescent particles emit light with a wavelength of 610 nm that is filtered using a small band pass filter. Images are made with the digital camera using a 100x optical magnification on which only the particles are visible. Figure 5 shows the experimental setup.

Figure 4 Schematic top view of the setup with the digital microscope for observation of particles in the grease in the vicinity of the seal contact including a typical image of a particle distribution. The dashed line indicates the sealing contact.

Figure 5 Setup with the digital microscope for observation of particles in the grease in the vicinity of the seal contact.

Grease type	Thickener/ base oil	$\eta_{\text{base oil}}$ 25°C [Pa·s]	Tacky [finger]
LG1	Lithium/ Mineral	0.25	+
LG2	Lithium/ Mineral	0.45	++
LG3	Lithium/ PAO	0.03	-
LG4	Lithium/ Mineral	0.21	-
LG5	Lithium/ Mineral	0.33	+

Table 1 Grease types and properties, chemical composition, base oil viscosity, and tackiness as experienced in finger test.

Lubricants

A number of relatively transparent greases, which are used for lubricating rolling element bearings, are studied. All are lithium greases with mineral or PolyAlphaOlefin (PAO) base oil, see Table 1. The greases have different rheological properties and show yield stress and shear thinning behavior with increasing shear rate. A first normal stress difference is present in the grease as shown previously by Baart et al. [17]. In addition to the greases, base oil that was separated from the LG1 grease is used. The base oil shows Newtonian behavior, i.e. no shear thinning and no normal stress differences. Also a solution of 0.5 wt% Poly(Ethylene) Oxide (PEO, $M_w \sim 8,000,000$) in water is used, which has a very low viscosity but is visco-elastic with extremely high normal stress differences and a limited amount of shear thinning. The rheological behavior of the different lubricants was measured at 25 °C and 70 °C according to the method described in Baart et al. [17]. The resulting viscosity and normal stress differences as a function of the axial distance from the sealing contact at 25 °C are plotted in Figure 6. The plots in Figure 6 can be readily used in the evaluation of the migration measurement results.

Measurement method

The lubricant sample is prepared by mixing the particles homogeneously through the lubricant. Part of the lubricant sample is used to fill the chamber between the seals before mounting them on the shaft. In the case air bubbles are present after mounting, the seal lip contact is locally lifted from the shaft and additional lubricant sample is applied using a syringe. It is then checked, before starting the experiment, that the particle concentration is homogeneous throughout the grease chamber.

White light from the external light source is used to identify the position of the seal lip contact. The laser light source is used to make images of both the front and rear seal lip at 100x magnification. A series of 8 images around the shaft circumference is made for both seal lips. The first image is made at a 0° shaft angle and each subsequent image after 45° shaft rotation. The migration experiments are done at shaft speeds of 23 and 233 rpm, corresponding to 0.1 and 1 m/s shaft surface velocity. After the experiment, again eight images are made as described above.

Figure 6 Rheological properties of greases in the vicinity of the sealing contact at 23 rpm: (a) apparent viscosity and (b) first normal stress difference

The images show the individual fluorescent particles and are analyzed using a software code. Here the light intensity is used as a high light intensity indicates the presence of a particle. The cumulative light intensity of each image pixel row parallel to the sealing contact is calculated and smoothened over a width of 50 pixels. Subsequently, background noise and differences in initial particle concentration between samples are eliminated by calculating the normalized relative migration of particles N_x , which is defined as

$$N_x = \frac{n_{x,t=45} - n_{x,t=0}}{\sum_x n_{x,t=0}} \quad (4)$$

where $n_{x,t=45}$ is the cumulative light intensity at distance x from the contact at $t = 45$ min, $n_{x,t=0}$ is the cumulative light intensity at distance x from the contact at $t = 0$ min, and the summation represents the total light intensity at $t = 0$ min in the domain. The cumulative light intensity is averaged over the 8

images that were made around the shaft circumference and the two seal lips.

Figure 7 Contaminant particle migration in LG1 grease at 23 rpm; results from one experiment

RESULTS AND DISCUSSION

The concentration of contaminant particles in the vicinity of the sealing contact is measured for the different lubricants and shaft speeds. The lubricants are specified by their rheological properties, i.e. viscosity and normal stress difference. The shaft speed is included to evaluate the significance of the hydrodynamic forces on particles relative to particle inertia effects and the natural pumping action of the seal.

Experiments at 23 rpm

Figure 7 shows the migration results of the eight individual measurements around the shaft circumference for the two seals with the LG1 grease. Despite the spread in the results, a clear trend is present where the normalized relative intensity, i.e. number of particles, in the first 0.1 mm from the contact is around zero and becomes negative further away from the sealing contact. This means that the particles that were initially here have migrated away from the contact. At some axial shaft positions, more particles are present, for $x > 1$ mm, which are the particles that were initially closer to the sealing contact. On average the normalized relative intensity is negative. The average line of the individual measurements in Figure 7 and the average line for other types of lubricants are shown in Figure 8.

Figure 8 Particle migration in different types of lubricants at 23 rpm

Figure 9 Particle migration in different greases at 23 rpm with fluorescent particles

Figure 10 Particle migration in different greases at 233 rpm with fluorescent particles

In Figure 8, the experiment with PEO has been repeated and good reproducibility is found. The normalized relative intensity, or particle migration, in PEO shows a consistent negative value equal to the results obtained in the pre-study [9]. Also for the base

oil similar results are obtained where the values are slightly negative meaning that also here some particles migrate away from the sealing contact. In the pre-study, a positive value, i.e. migration towards the contact, was measured for the LG1 grease due to fragmentation of particles. Here, the values are negative and particles therefore migrate away from the sealing contact. The contaminant particle migration is found to be much more significant in the grease than in the base oil and in PEO.

Figure 9 shows the results for different grease types at 23 rpm. The main differences between the greases from a rheological point of view can be found in Table 1 and Figure 6. Only in the LG4 grease particles are found to migrate towards the sealing contact where positive values are obtained in the first 0.8 mm from the contact. The migration in the other greases is negative and therefore away from the sealing contact where the LG2 and LG3 greases show the largest negative values meaning that here a larger number of particles have migrated away from the sealing contact.

In the experimental results presented in Figure 8 and Figure 9, the shaft velocity is very low. Consequently, the Reynolds number is very low and the seal pump rate is negligible. No leakage of lubricant from the volume in-between the seals could be identified, which means that the migration of the particles must be driven by hydrodynamic effects and the fluid rheology.

Experiments at 233 rpm

Experiments were also performed at a shaft speed of 233 rpm. The corresponding shaft surface speed of 1 m/s is still not really high but higher speeds are avoided to reduce the risk of damaging the fragile sapphire shaft. The experimental results for the different greases are shown in Figure 10. Here the LG1 shows extremely large positive values as severe migration of particles towards the contact has taken place. This is also shown in the images in Figure 11a and Figure 11b where the particle distributions at $t = 0$ min and $t = 45$ min are shown respectively. Leakage of LG1 grease from the pocket in-between the seals was observed which could be due to seal pumping. Figure 11b shows the high concentration of particles close to the contact. Further away from the contact, as indicated with the arrow, a grease meniscus lights up and only the volume between the meniscus and sealing contact contains grease that is being sheared. Therefore, it is concluded that for the LG1 grease at the high shaft speeds, leakage or seal pumping is dominating. However, for the other greases, such

significant grease loss was not observed and, consequently, no indication of seal pumping was found. Figure 10 also shows the results for the other greases for which the results are rather similar to those at 23 rpm in Figure 9. Again the LG2 grease shows negative values indicating that migration of particles takes place away from the sealing contact. This is also shown in Figure 11c and Figure 11d where the images show a reduced concentration of particles after 45 minutes.

Figure 11 Particle migration images before and after the experiment at 233 rpm for LG1 and LG2 greases. The sealing contact is indicated with the dashed line; the arrow indicated the grease meniscus after grease loss.

All the results show that the contaminant particles do not migrate into the sealing contact and neither get trapped between the seal lip and rotating shaft. Instead, particles are found to accumulate at a small distance, $x > 0.1$ mm for LG4 at 23 rpm and $x > 0.3$ mm for LG1 at 233 rpm, away from the sealing contact where the gap is still $> 2x$ and $> 8x$ the particle diameter respectively.

Migration mechanism

From the evaluation of the experimental results it is concluded that the difference in particle migration in the different fluids and greases has to be the result of the hydrodynamic forces and rheological properties. In the sealing system, the shear rates are high close to the sealing contact where the gap is small, but the shear rate decreases when moving away from the sealing contact. In line with the results from the literature, e.g. Gauthier et al. [11], particle migration in the shear thinning grease takes place in the direction of the highest shear rate i.e. towards the sealing contact. In conditions where particle inertia or fluid elasticity, i.e. normal stress differences, is significant, particles migrate away from the sealing contact instead.

To predict the exact contaminant particle migration using analytical equations, e.g. from Yang et al [14] and Ho and Leal [15], detailed information about the fluid velocity profile is required. The velocity profile in the vicinity of the sealing contact depends on the grease rheology, as shown in Baart et al. [8], but is disturbed by the presence of the contaminant particle. Consequently, quantitative predictions based on relatively simple equations are not possible and therefore a more qualitative analysis is done here.

Figure 9 and Figure 10 are correlated with the fluid rheology in Figure 6 and explain the differences in migration behavior between the greases. First, the grease shear thinning behavior, or viscosity gradient, is considered. In Figure 6a, a viscosity gradient in the x -direction is clearly present in the grease. This viscosity gradient is the result of shear thinning and is expected to bring contaminant particles towards the high shear rate, or low absolute viscosity, area. Differences in the fluid velocity along the particle and particle rotation are the causes for the migration. Shear thinning towards the sealing contact may be enhanced by a temperature gradient in the grease. Due to heat generation from viscous friction, the temperature will be highest in the sealing contact, resulting in a significant reduction of the viscosity. Consequently, a temperature gradient will be present and depends on the base oil viscosity of the grease.

In Figure 6a, a linear fit to the viscosity curve is made for $x > 0.4$ mm to determine the viscosity gradient due to shear thinning, without taking the temperature gradient into account. This viscosity gradient at 25 °C is considered constant for $0.4 > x > 2$ mm and is presented in Table 2. A qualitative ranking of the particle migration in the different greases is done by indicating the trends with “-“ when the normalized relative intensity is negative, i.e. migration away from

the sealing contact, and with “+” when positive, i.e. migration towards the sealing contact. Values for the qualitative ranking of the particle migration are also presented in Table 2.

Grease type	$d\eta/dx$ 23 rpm	migration 23 rpm	$d\eta/dx$ 233 rpm	migration 233 rpm
LG1	1.75	-	0.09	++*
LG2	1.65	--	0.12	--
LG3	1.60	--	0.11	-
LG4	2.10	+	0.15	o
LG5	1.50	-		-
Base oil	0.0	o		

Table 2 Viscosity gradient and migration in the vicinity of the sealing contact at 23 rpm 25 °C and 233 rpm 70 °C.

*grease leakage is the dominating mechanism here.

Table 2 shows that, at 23 rpm, LG4 has a significantly larger viscosity gradient than the other greases and only for this grease migration towards the sealing contact was observed in Figure 9. Also at 233 rpm LG4 has the largest viscosity gradient and again for this grease some migration towards the sealing contact was observed in Figure 10. The LG1 grease shows much larger migration at 233 rpm but this is due to grease leakage and therefore not taken into account in the evaluation. Consequently, it is concluded that contaminant particle migration in the vicinity of the sealing contact can be correlated to the viscosity gradient or shear thinning properties of the grease. The viscosity gradient contributes to the particle rotational velocity and consequently particle migration. However, no further differentiation between the other greases can be made based on the grease shear thinning and other effects need to be included.

Snijkers et al. [20] show that particle rotation is slowed down by fluid elasticity and relate this effect to the fluids Weissenberg number, which they define as

$$Wi = \frac{N_1 - N_2}{\sigma_{zy}} \quad (5)$$

where $N_1 - N_2$ is the normal stress difference as measured on a parallel plate rheometer and σ_{zy} is the shear stress. Wi can be calculated for the greases and other fluids using the rheological data from Figure 6. Results are presented in Figure 12. The experimental results from Snijkers et al. [20] show that particle rotation is not significantly affected if $Wi < 0.5$. They measure a decreasing rotational velocity of the particle

with increasing Weissenberg number when $Wi > 0.5$. In the sealing system considered in the current study the Weissenberg number is, like the viscosity, a function of the shear rate and consequently, decreases with increasing distance from the sealing contact.

Figure 12 Weissenberg number for different lubricants at 23 rpm and 25 °C based on the shear rate dU/dz from Figure 2.

Figure 12 shows that the Weissenberg number is very large for the PEO fluid, resulting in significant slowing down of the particle rotational velocity. Additionally, in such highly elastic fluid the normal stress differences can induce particle migration to low shear rate zones, i.e. away from the contact, as shown in Gauthier et al. [11] and Ho and Leal [15]. The Weissenberg number is zero for the base oil since no normal stress differences are present in a Newtonian fluid. Neither shear thinning nor a viscosity gradient is present resulting in no significant migration as shown in Figure 8. For the LG1, LG2 and LG5 grease, Figure 12 shows that $Wi > 0.5$ meaning that slowing down of the particle rotational velocity can be expected. For the LG4 grease, $Wi < 0.5$ and consequently no such slowing down will be present. Combining these results with the results from Table 2 gives some qualitative explanation for the contaminant particle migration results in Figure 9 and Figure 10. For example, at 23 rpm, LG4 shows some migration towards the sealing contact and has the largest viscosity gradient while $Wi < 0.5$, meaning that the effect of the normal stress difference is negligible. For the LG1, LG2 and LG5 grease, the viscosity gradient is much smaller while $Wi > 0.5$. Consequently, for these greases, migration takes place away from the sealing contact.

An exception in the current analysis is the LG3 grease, which seems not to fit the theory presented above. An explanation can be found in the very low base oil viscosity. The low base oil viscosity is

expected to result in low viscous friction in the sealing contact. Consequently, also the temperature gradient is expected to be small, resulting in only a small increase of the viscosity gradient. This is different for the other greases, where the relatively high base oil viscosity results in a much larger temperature gradient and viscosity gradient, enhancing particle migration towards the sealing contact.

It has been shown that the grease rheology has a significant influence on the migration of contaminant particles in the vicinity of a seal contact. Future experimental work should focus on a more quantitative analysis including a wider range of particle size and shape. For a theoretical quantitative analysis, detailed fluid velocity profiles around the particles should be calculated. Accurate numerical models should then be developed that include the 3D geometry with influence from the walls, fluid rheology, temperature gradient, and shear degradation or aging of the grease. Models including this level of complexity are not available today.

CONCLUSIONS

The migration of contaminant particles in grease has been studied experimentally with a view to explain the sealing function of the grease in ‘sealed and greased for life’ rolling element bearings. The contaminant particles are found to either migrate towards or away from the sealing contact and it is shown that the particle migration in the vicinity of the sealing contact depends on the lubricant rheology. Although no quantitative correlation between the rheological properties and the contaminant particle migration was found, it is shown qualitatively that the viscosity gradient and Weissenberg number are important. A large viscosity gradient results in migration towards the sealing contact and a large Weissenberg number results in migration away from the sealing contact. The sealing function of the grease in the vicinity of the sealing contact can be explained as follows. On the outside of the seal or in a seal pocket, a grease in which migration takes place away from the sealing contact reduces the probability of contaminant particles to reach the sealing contact. On the bearing side of the seal, a grease in which contaminant migration takes place towards the sealing contact reduces the probability for contaminants to flow into the bearing system.

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Paper E

Oil-Bleeding Model for Lubricating Grease Based on Viscous Flow Through a Porous Microstructure

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One of the criteria in selecting lubricating grease for rolling-element bearing applications is its ability to bleed oil, sometimes called “grease bleeding.” Oil bleeding is assumed to be the dominating mechanism supplying new oil to the rolling track for lubrication. In this study, a physical model has been developed to understand the relation between parameters that control oil bleeding. In the model, lubricating grease is described as a porous network, formed by the thickener fibers, that contains the base oil. This type of structure is confirmed by SEM and AFM images of a lithium complex grease showing a matrix of rigid fibers with random orientation. A relatively simple flow model based on Darcy’s law for viscous flow in porous media and an anisotropic microstructure deformation model was developed. The model relates the pressure gradient, oil viscosity, thickener structure deformations, and permeability to the volumetric oil flow out of the thickener network. The permeability depends strongly on the thickener microstructure. The model was verified with experiments at a wide variety of temperatures and rotational speeds.

KEY WORDS

Oil Bleed; Microstructure Permeability; Grease Lubrication; Rolling-element Bearing

INTRODUCTION

Grease is a complex lubricant consisting of a base oil, additives, and a thickener structure that gives the grease a consistency and acts as a reservoir for the base oil. The base oil is slowly released from the grease, in a process called “oil bleeding,” and

replenishes the lubricated contact. The mechanism by which this oil release is activated is not clearly understood. In fast rotating applications, a centripetal body force acting on the grease is assumed to be the driving force. Applications like this can be found in rolling-element bearings and axial seals where an axial lip runs against a rotating flinger or vice versa. See Fig. 1.

The solid thickener in the grease is assumed to form a microstructure containing the oil. This kind of structure with a solid matrix and a fluid in the free space is often referred to as a “porous medium” (Bear (1)). In most lubricating greases used in rolling-element bearings, the thickener is a lithium hydroxysulfonate or lithium complex, which is also referred to as a (metallic) “soap.” Lubricating greases show a yield stress flow behavior, i.e., they show elastic behavior below a certain stress and start to flow at higher stresses. The yield stress and elasticity of the grease depend on the strength of the fibrous network and consequently on the interaction between the oil and the thickener. Fresh grease contains a soap fraction of 3–30%, but the soap fraction increases during grease life, where oil is slowly bleeding out until a critical soap fraction of 50–60% is reached at the end of grease life (Cann et al. (2)).

Baker (3) measured the oil-bleed rate of several greases at different temperatures. He found that bleeding rates increase with temperature and strongly depend on the type of base oil and thickener. It appears in his research that bearings operating at high temperatures do not require significantly higher bleeding rates to ensure long life. The type of experiments that Baker did have been normalized in the DIN 51817 (4), where the amount of oil bleed under a static load is measured after 168 h. Dynamic tests simulating oil bleed in bearing conditions have not been found in literature.

A test rig has been developed to measure oil bleeding under a centripetal body load at elevated temperatures to obtain more realistic data. The thickener microstructure is studied, and the oil flow is modeled using porous media theory. The microstructure of a lithium complex soap containing a mineral oil will be modeled and studied in more detail in the current work.

NOMENCLATURE

η	= dynamic viscosity	h	= height of grease in cup
θ	= fiber tilting angle	k	= permeability
ν	= kinematic viscosity	L	= length
ρ	= density	L_o	= initial length
ω	= angular velocity	L_{min}	= minimum length
a	= acceleration	m	= mass
A	= cross-sectional area	p	= pressure
b	= soap fiber diameter	Δp	= pressure difference
f	= fiber-volume fraction	∇p	= pressure gradient
f_o	= initial fiber-volume fraction	q	= flow velocity
f_{max}	= maximum fiber-volume fraction	Q	= volumetric flow
f_{mo}	= initial fiber-mass fraction	r	= fiber radius
F_{body}	= body force	R	= radius
$F_{friction}$	= friction force	Re	= Reynolds number
g	= gravitational acceleration	t	= time
		V	= volume
		z	= coordinate in flow direction

THEORY

Grease Microstructure

Lithium hydroxystearate soap is the most common thickener type used in rolling-bearing greases. The microstructure of this type of thickener consists of long fibers. Hurley and Cann (5) showed that a lithium hydroxystearate soap has a fine and complex network of fibers slightly twisted in some areas. Lithium complex grease has a more simple microstructure formed by longer rope-like fibers with a higher degree of twist. They used several different techniques to examine the grease microstructure and found varying values for characteristic microstructure dimensions like fiber diameter and length. We will discuss in more detail two visualization techniques: first, Scanning Electron Microscopy (SEM); and second, Atomic Force Microscopy (AFM).

Figure 2 shows an SEM image of the microstructure of a lithium hydroxystearate soap. The fibrous microstructure is clearly visible here. Making such an SEM image requires high vacuum, and the oil has to be washed out of the grease so that only thickener is left. Since grease consists of 65–95% oil, interpretations based on SEM images have to be done carefully because the final microstructure might be a result of the washing technique and the physical microstructure might have collapsed (Salomonsson et al. (6)). The advantage of AFM is that wet sam-

ples, still containing the base oil, can be examined, and it produces a true three-dimensional surface profile. See Fig. 3. However there are some limitations regarding the scanning area and field of depth, and the scanning time is much longer than for the SEM. A comparison of washed and unwashed fiber dimensions on the AFM suggests that washed fibers are slightly thicker (Hurley and Cann (5)). This is expected to be the result of the soap-solvent interaction and swelling of the soap fibers.

Soap fiber diameters found in literature using SEM and AFM range from 0.1 to 0.5 μm . Hurley and Cann (5) found a typical fiber width of 0.1 μm for a lithium hydroxystearate grease and that of 0.15 to 0.2 μm for a lithium complex grease. These values were measured from washed samples using SEM.

The fiber diameter of the lithium complex grease used in this study was measured using AFM. This is expected to give the most reliable image of the microstructure and fiber dimensions since it measures the grease directly without a washing procedure. Figure 3 shows the AFM image of the lithium complex grease. The fiber diameter is $\sim 0.2 \mu\text{m}$, and the fibers clearly have a

Fig. 1—Grease subjected to centripetal body load: (a) near the axial lip of a seal and (b) on the cage of a rolling-element bearing.

Fig. 2—SEM image of a lithium hydroxystearate soap network after washing out the base oil.

Fig. 3—AFM image of the lithium complex grease 5 × 5 μm.

twisted rope-like shape but do not show the dense and bonded structure as in the SEM image in Fig. 2.

The lithium complex grease in Fig. 3 was measured on the Atomic Force Microscope from CSM instruments using the settings in Table 1.

General Flow Equation

Fluid flow through porous media was first studied and published by Darcy in 1856 (Bear (1)). From his experiments on water flow through a fully saturated column of sand, he found a relation between the pressure difference over the column of sand and the flow rate of the water as

$$\frac{Q}{A} = \frac{k}{\eta} \frac{\Delta p}{L}, \tag{1}$$

where Q is the volumetric flow rate through a cross-section area A . The ability of the fluid to flow through the porous media is described with the permeability constant k and the fluid viscosity η . A pressure difference Δp drives the fluid over length L in the flow direction.

Darcy’s law only applies when viscous shear forces dominate the friction such that inertia effects can be neglected. This means

that the Reynolds number must be small, i.e., $Re \ll 1$ as defined by Bear (1).

For the oil-bleeding model, the oil-bleed process is assumed to be a pressure driven flow through the porous microstructure of solid soap fibers. In order to identify the different parameters involved in oil bleeding, the force balance on a small grease volume is considered:

$$\vec{F}_{body} + \vec{F}_{friction} = \Delta m \cdot \vec{a}, \tag{2}$$

where F_{body} is the total of external bulk forces acting on a volume element such as gravity and acceleration, $F_{friction}$ is the friction force on the oil due to flow through the porous soap microstructure, and the right hand side is the inertial force. To apply Darcy’s law, the Reynolds number must be smaller than 1, which means that the inertial forces in Eq. [2] can be neglected. It was checked in the final model that the Reynolds number is indeed much smaller than 1 such that

$$\vec{F}_{body} + \vec{F}_{friction} = 0. \tag{3}$$

This equation is used to describe the oil bleeding of grease. The yield stress and elastic behavior of the grease are not considered here. To make a complete viscoelastic model, the geometry and deformation of the soap microstructure should be considered in too much detail to be included in the force balance. This would increase the complexity of the model enormously and is therefore not included at this time.

METHODS AND MODELING

Grease Microstructure Model

The grease microstructure has been visualized with the AFM in Fig. 3 and is shown again in Fig. 4 a. For the oil-bleed model, this measured structure is simplified to circular and randomly ordered fibers suspended in oil as shown in Fig. 4 b. For fresh grease, the fiber direction distribution is assumed to be uniform. Subsequently, to simplify the modeling of the permeability of the microstructure, the model is further simplified with the fibers ordered in an orthogonal arrangement, as shown in Fig. 4 c, with one axis parallel to the pressure gradient. This results in 1/3 of the fibers being parallel and 2/3 of the fibers being perpendicular to the oil flow. The orthogonal fiber arrangement has the advantage that anisotropy, i.e., fiber orientation, is relatively easy to model.

Base oil will flow out of the grease during oil bleed, which results in an increase in soap volume fraction. This means that the soap structure has to become denser and that the permeability will decrease, i.e., the resistance of the oil to flow through the soap microstructure increases. This mechanism makes the oil bleed decrease in time.

For the model, a homogeneous soap volume fraction over the whole grease volume is assumed. This is approximately true for sufficiently small grease volumes. Grease samples with bigger volumes that have been subjected to centripetal loads show an increase in soap fraction at the smaller radius. This can be explained with the oil flow in radial direction. For modeling the oil bleeding, the homogeneous soap distribution is assumed, and it will appear in the simulation results that this assumption is justified.

TABLE 1—SETTINGS FOR THE AFM

AFM CSM Instruments		
Operation mode	Noncontact	
Tip type	ACLA	
- Tip length	225	μm
- Oscillating frequency	160	kHz
- Free amplitude	150	nm
- Set point	30	%
Controller settings		
- P	0.06	
- I	200	μs ⁻¹
Scanning speed	0.1	line/s
Scanning resolution	514 × 514	pixels

Fig. 4—Thickener structure: (a) picture made with AFM, (b) 1st simplification to rigid fibers, and (c) 2nd simplification to orthogonally arranged rigid fibers where the structure consists of stacked cubic unit cells of volume $V_0 = L_0^3$ at $t = 0$.

Friction Force

The friction term in Eq. [3] is described using Darcy's law from Eq. [1], which reads in its general form as

$$\vec{q} = \frac{1}{\eta} \vec{k} \cdot \vec{\nabla} p, \quad [4]$$

where $\vec{q}$ is the fluid velocity vector, $\vec{k}$ the permeability tensor, and $\vec{\nabla} p$ the pressure gradient. From this equation, the friction force per unit volume is deduced as

$$\vec{\nabla} p_{friction} = \eta \vec{k}^{-1} \cdot \vec{q}. \quad [5]$$

This is the basic equation used in the force balance in Eq. [3] to calculate the oil flow velocity q . Eq. [5] requires values for the base oil viscosity η and the permeability k .

The base oil viscosity follows Newtonian behavior for the low oil flow velocities in the grease and is assumed to depend on temperature only. Walther's equation is used to describe the temperature-viscosity behavior (Sánchez-Rubio et al. (7)):

$$\log[\log(v + 0.7)] = A - B \log(T), \quad [6]$$

where v is the kinematic viscosity in cSt, and T is the temperature in Kelvin. A and B are constants that can be determined from two measured viscosities at known temperatures, normally 40°C and 100°C. Eq. [6] can be written for the dynamic viscosity as

$$\eta = \rho 10^{-6} \left[10^{(A - B \log(T + 273))} - 0.7 \right] \quad [7]$$

where η is the dynamic viscosity in Pa·s at temperature T in °C, and ρ is the oil density.

The permeability strongly depends on the porosity of the soap microstructure. In some cases, the permeability of a porous medium can be determined directly from experiments. However, in most cases, permeability models are required to calculate the flow as a function of the pressure gradient. The same applies to oil bleeding of grease. When a good model for the grease permeability is available, the oil bleeding can be predicted.

Gebart (8) derived analytical expressions for the permeability of flow that is parallel or perpendicular to a grid of

uniaxial aligned fibers. In the orthogonal arrangement of fibers as assumed in Fig. 4 c, the combined permeability is taken as a summation of these contributions as

$$k = \frac{2}{3} k_{\perp} + \frac{1}{3} k_{\parallel}, \quad [8]$$

where $k_{\perp}$ is the permeability perpendicular, and $k_{\parallel}$ is the permeability parallel to a grid of uniaxial aligned fibers. The expressions for permeabilities that are parallel and perpendicular as derived by Gebart (8) are respectively

$$k_{\perp} = \frac{16}{9\pi\sqrt{2}} \left(\sqrt{\frac{f_{\max}}{f}} - 1 \right)^{5/2} r^2 \quad [9]$$

$$k_{\parallel} = \frac{8}{57} \frac{(1-f)^3}{f^2} r^2, \quad [10]$$

where $f_{\max}$ is the maximum fiber-volume fraction, and r is the radius of the soap fibers. Eq. [9] has originally been derived for relatively high volume fractions where the pressure gradient over the complete volume is dominated by the pressure drop over the small gap region between two fibers. Gebart (9) showed that the error between his analytical equation, Eq. [9], and numerical simulations is smaller than 10% for fiber-volume fractions $f > 0.35$. For fresh grease with an initial soap volume fraction of, e.g., 0.10 and 0.20, the error is 30% and 20% respectively. The model simulation results will show whether this error at low fiber-volume fractions is acceptable for modeling oil bleeding of grease.

From Eq. [9] and Eq. [10], it is clear that as the volume fraction increases during oil bleed, the permeability decreases, which is the main reason why the oil-bleed rate tends to decrease over time. Change of the grease microstructure might cause additional reduction of the oil-bleed rate. The change of the grease microstructure can be modeled in several ways.

First the effect of fiber-volume fraction increase is studied assuming that the orthogonal arrangement is maintained during oil bleed. In this case the fiber-volume fraction increase can be achieved by increasing the fiber radius r or by decreasing the size L of the cubic unit cell. Only decrease of the cubic cell is taken into account since gradual growth of the fiber radius due to oil bleed is most unlikely to happen in reality. This means that the

Fig. 5—Two fiber arrangements for their maximum fiber-volume fraction: (a) the orthogonal arrangement and (b) the arrangement where the fibers that were initially parallel have tilted so as to be perpendicular.

fiber radius r in Eq. [9] and Eq. [10] remains constant while f increases in time. The maximum fiber-volume fraction possible for the orthogonal arrangement is reached when the fibers are touching (see Fig. 5 a), and $f_{max} = 3\pi/16$.

Secondly, a model is proposed that assumes that fibers initially parallel to the pressure gradient gradually tilt and finally become perpendicular to the pressure gradient and oil flow. Consequently, the microstructure becomes anisotropic during oil bleed, and a higher maximum volume fraction can be reached. Figure 5 b shows the situation where all fibers have tilted so as to be perpendicular at the maximum fiber volume fraction $f_{max} = \pi/4$, which is much higher than the case of the isotropic fiber arrangement in Fig. 5 a.

The anisotropic fiber arrangement induced that Eq. [8] has to be modified to include the tilting of fibers. Figure 6 shows one

Fig. 6—Fiber tilted with angle θ and flow q components perpendicular and parallel to the fiber.

individual fiber of the group of uniaxial aligned fibers that were initially parallel to the flow but that have now tilted to an angle θ . The oil flow around this group of fibers can be decomposed into two contributions: one for flow parallel to the fibers $q_{//}$ and one for flow perpendicular to the fibers $q_{\perp}$. The pressure drop dp/dz can subsequently be written as the summation of the parallel and perpendicular flow contributions:

$$\frac{dp}{dz} = \frac{\eta}{k_{\perp}} q_{\perp}^2 \sin^2 \theta + \frac{\eta}{k_{//}} q_{//}^2 \cos^2 \theta. \quad [12]$$

This can be written as Darcy's equation:

$$q_z = \frac{k(\theta)}{\eta} \frac{dp}{dz}, \quad [13]$$

with the permeability of the tilted fibers written as

$$k(\theta) = \frac{k_{//} k_{\perp}}{k_{//} \sin^2 \theta + k_{\perp} \cos^2 \theta}. \quad [14]$$

Assuming that in fresh grease, 1/3 of the fibers is orientated parallel, and 2/3 are orientated perpendicular to the oil flow, as in Fig. 4 c, one can calculate the total permeability including anisotropy as

$$k = \frac{2}{3} k_{\perp} + \frac{1}{3} k(\theta). \quad [15]$$

To evaluate the angle θ in Eq. [14], it is assumed that the volume reduction of the cubic unit cell comes only from a reduction in height $L(t)$ such that the base surface is constant and equals L_o^2 . Consequently $L_o - L_{min}$ is the height over which the fibers tilt during oil bleed, defined as the difference between the initial volume height L_o and the minimum volume height L_{min} of the cubic cell when the maximum volume fraction is reached. The height of the fibers parallel to the oil flow at time t equals $L(t) - L_{min}$, such that the angle θ can be written as

$$\cos \theta(t) = \frac{L(t) - L_{min}}{L_o - L_{min}}. \quad [16]$$

By expressing the unit cell volume $V(t)$ as $L_o^2 L(t)$ and using the relation $V(t) f(t) = V_o f_o$, Eq. [16] can be written as a function of

the soap volume fraction as

$$\cos \theta(t) = \frac{f^{-1} - f_{\max}^{-1}}{f_o^{-1} - f_{\max}^{-1}}. \quad [17]$$

Initially, when $f = f_o$, θ equals 0° and $k(\theta)$ becomes $k_{//}$, meaning that one third of the fibers is parallel to the oil flow and that Eq. [15] becomes equivalent to the isotropic case as in Eq. [8]. During oil bleed, the angle θ will increase and will finally become 90° when f equals the maximum fiber-volume fraction $f_{\max}$. Now all fibers have tilted perpendicular to the pressure gradient, and the oil bleed stops.

Body Force

The body force consists of gravity force and centripetal force due to the rotational speed. This force builds up the pressure in the grease that pushes out the oil. The body forces acting on grease rotating with angular velocity ω at distance R around a central point are the gravity force and the centripetal force due to rotation:

$$\vec{F}_{body} = m\vec{g} + m\omega^2\vec{R}. \quad [18]$$

For the grease in a rotating application like the axial seal lip, the gravity force is smaller by at least a factor of 100 than the centripetal forces and has therefore been omitted in the model.

The pressure gradient as derived by considering the body forces on an infinitesimally small volume element located at distance R from the center of rotation becomes

$$\nabla p_{body} = \rho\omega^2 R, \quad [19]$$

where ρ is the oil density. This expression is used as the body force that drives the oil flow in Eq. [3].

EXPERIMENTS AND VALIDATION

The standard test for measuring oil separation is described in DIN51817 (4). Here a dead weight is put on top of a cup filled with grease, and the oil is slowly pressed out through a sieve at the bottom side. The mass of the oil separated from the grease is measured after resting for 1 week in an oven at 40°C . A modified test is sometimes performed at elevated temperature or higher internal pressure.

The conditions in the DIN51817 test are not representative for grease experiencing high centripetal forces resulting from high rotational speeds. A more representative test rig was designed to measure oil separation from grease under centrifugal motion at different temperatures and rotating speeds. A schematic representation of this rig is shown in Fig. 7 and the main dimensions are given in Table 2.

The cylindrical cup is filled with grease, and four of these cups are mounted on a central rotating shaft. The oil bleeding out of the cup is collected in a small retrieval container that is screwed onto the cup. This container is not shown in Fig. 7. The mass of the oil in these containers determines the oil bleed, which is measured during a test period of 24 h at a defined test temperature and rotational speed.

In the test rig, the oil flow is parallel to the centrifugal body load resulting in a relatively simple one-dimensional flow. This one-dimensional flow is modeled with the force balance in Eq. [3].

Fig. 7—Sketch of the test rig to measure oil separation from grease under centrifugal motion.

The grease selected for this study is a lithium complex grease and has a NLGI 3 consistency. This grease has a simpler microstructure than a lithium hydroxystearate grease (Hurley and Cann (5)) and is expected to be described better by the microstructure model. The fresh grease has a high percentage of thickener content, which minimizes the error for low fiber-volume fractions in the analytical permeability model, Eq. [9].

Algorithm to Calculate Oil Bleeding

A numerical calculation is used to solve the equations and predict the oil bleed in time. In each time step Δt , the oil-bleed rate is calculated as follows:

1. The fiber-volume fraction f is calculated as $f = f_0 V(t)/V_0$.
2. The permeabilities $k_{//}$ and $k_{\perp}$ are calculated using Eq. [9] and Eq. [10].
3. The angle θ and the permeability $k(\theta)$ are calculated from Eq. [17] and Eq. [14] respectively ($\theta = 0$ in case of isotropic bleeding).
4. The grease permeability k is calculated from Eq. [15] as the combination of permeabilities parallel and perpendicular to the soap fibers.
5. The pressure gradient over the grease length h in the rotating cup (Fig. 7) is calculated from Eq. [19] as either

$$\nabla p_{body} = \frac{\omega^2 \rho}{h} \int_{R_o-h}^{R_o} R dR \quad [20]$$

or, when solving the integral,

$$\nabla p_{body} = \frac{\omega^2 \rho}{2h} [R_o^2 - (R_o - h)^2]. \quad [21]$$

6. Now the oil-bleed rate can be found from Eq. [1] as

$$\frac{dV}{dt} = -Q = -A \frac{k}{\eta} \nabla p_{body}. \quad [22]$$

7. The grease volume at time $t + \Delta t$ becomes

$$V(t + \Delta t) = V(t) + \frac{dV}{dt} \Delta t. \quad [23]$$

8. The total mass percentage of oil loss is calculated from the bleeding rate and will be compared with the experimental results.

TABLE 2—TEST RIG DIMENSIONS

Test rig dimensions			
R_o	67e-3	m	outer radius
A	3.14e-4	m ²	outflow area
h_{cup}	31e-3	m	cup length

The input for the model comes from the grease properties. The grease density and base oil viscosities are given by the grease manufacturer. The fiber diameter is determined from the AFM image. The fiber mass fraction and base oil density can be measured using standard measuring methods where the oil is separated from the soap. Values for the lithium complex grease are shown in Table 3.

For the experimental tests, four different ambient temperatures and two different rotational speeds are used.

RESULTS AND DISCUSSION

The oil-bleed model contains several equations for the permeability and the grease microstructure. The flow of the base oil through this microstructure is very sensitive to the permeability, which decreases with increasing fiber-volume fraction.

Figure 8 shows the different models proposed for the permeability. The boundaries of the models are given by the two cases where all fibers are uniaxially aligned and either perpendicular or parallel to the pressure gradient, i.e., with permeability equal to $k_{\perp}$ or $k_{\parallel}$ respectively.

It can be seen here that for the perpendicular arrangement, the permeability is smallest and goes to zero at $f = \pi/4$, which was defined as the maximum fiber-volume fraction. For the parallel case, no such limit appears in the equation. This means that even at the maximum fiber-volume fraction, a flow of oil is still possible.

Consider the first case, as described in Eq. [8], where the fibers stay in the orthogonal arrangement. Figure 8 shows that the permeability is lower than for the uniaxial parallel fiber arrangement but follows the same trend.

Interesting is the second case, where the fibers that are initially parallel, tilt during oil bleed and finally become perpendicular according to Eq. [15]. Then the permeability decreases to values similar to the uniaxial perpendicular case. The permeability finally goes to zero, which means that the oil bleeding stops while the microstructure still contains oil.

The permeability derived for fibers perpendicular to the oil flow, Eq. [9], has to be used with care because this equation was originally derived for fiber-volume fractions $f > 0.35$. The grease used in the current study has an initial fiber-volume fraction of

TABLE 3—GREASE PROPERTIES

Grease properties			
ρ_{grease}	930	kg·m ⁻³	grease density
ρ_{oil}	870	kg·m ⁻³	base oil density
η_{40}	98.5	Pa·s	base oil viscosity at 40°C
η_{100}	10.5	Pa·s	base oil viscosity at 100°C
b	200	nm	fiber diameter
f_{mo}	0.26	-	fiber mass fraction

Fig. 8—Permeability models for fluid flow through the fibrous soap microstructure.

0.24, but this can be lower for other greases. However, in the initial orthogonal structure also, parallel fibers are present such that the model is not only depending on Eq. [9]. In addition, during oil bleed, higher fiber-volume fractions are reached where Eq. [9] can be used without problems.

The simplest model is considered where the grease microstructure is assumed to be isotropic such that Eq. [8] applies and the oil bleed will decrease in time due to the decrease in permeability. Here this decrease only comes from the increasing fiber-volume fraction. Figure 9 a and Fig. 9 b show the percentages of oil loss respectively at different temperatures and at different speeds. The symbols represent the experiments, and the lines represent the numerical model. The model shows trends similar to those of the experiment but predicts higher values for the oil loss and oil-bleeding rate. The isotropic model, using Eq. [8] for the total permeability, overestimates the experimental results. This means that the predicted permeability is too high and has to decrease faster with oil loss. Finally, when the maximum fiber-volume fraction is reached, the oil bleed has to stop. This, however, does not happen here since the permeability does not become zero at high fiber-volume fractions as was shown in Fig. 8.

The stopping of oil bleeding is fulfilled in the anisotropic model where the fibers that are initially parallel with the pressure gradient slowly tilt perpendicular during oil bleed. Here the oil-bleed rate will reduce faster, and there is a limit for the maximum percentage of oil loss when the maximum fiber-volume fraction is reached.

Figure 10 a shows that the results obtained with the anisotropic model using Eq. [15] predicts the oil loss with better accuracy. Here the oil loss at 120°C is still overestimated in the beginning but finally approaches the measured values. At lower temperatures, 40°C and 60°C, the first hours are described very well but the model somewhat overestimates the oil loss at later times. Also at lower speeds of 1000 rpm, as shown in Fig. 10 b, the anisotropic model describes experimental results reasonably well. Overall, the model seems to capture the main effect of speed and temperature on the oil-bleeding rate.

Fig. 9—Oil loss due to oil bleed for an isotropic grease microstructure. The symbols represent the experiments, and the lines represent the model.

Figure 11 shows an extrapolation of the anisotropic model up to 1000 h and includes more temperatures and rotational speeds. All curves finally reach the same maximum percentage of oil loss and give a good indication of the time it takes to reach this maximum. At increased temperatures or rotational speeds, the maximum percentage of oil loss is reached earlier. This could be correlated to relubrication intervals of bearings, which also become shorter when temperatures or speeds are high.

It is anticipated that the oil-bleed model can be used for all greases that have microstructures similar to that of the lithium complex grease studied here. This means that also lithium hydroxystearate greases and calcium greases can be modeled in the same way. However, the grease properties presented in Table 3 will be different and have to be measured for each individual grease.

It must be recognized that the model presented here is not yet complete apart from conditions based on uniform fibers and the homogenous microstructure. Other forces might become rel-

Fig. 10—Oil loss due to oil bleed of an anisotropic grease microstructure where fibers that are initially parallel and then tilt to become perpendicular to the pressure gradient or oil flow. The symbols represent the experiments, and the lines represent the model.

evant at conditions where the centripetal force, which is the driving force in the current model, becomes small. Here one can think of the capillary force acting at the grease-air interface. Including capillary forces into the model results in a reduced oil-bleed rate and in a threshold force below which there is no oil bleed. Indications for capillary effects are found in static grease-bleeding experiments where the temperature was lowered after some time and it was observed that the oil that did bleed out of the grease was sucked back into the grease again. Also, the reduction of the oil-bleed rate found in the current experiment at reduced filling rates of the cup, i.e., grease mass, seems to indicate the existence of capillary forces. It is expected that the dependence on filling rate of the cup will become visible in the model when additional forces like the capillary forces, fiber interactions, and nonhomogeneous microstructures are included. The model also does not include other secondary effects such as the force needed to deform the soap network during bleeding; oxidation of the oil,

Fig. 11—Oil loss due to oil bleed over a long time span as an indication of relative grease life.

which results in growth of fiber diameter (Salomonsson et al. (6)); and the buoyant force due to the difference in mass density between the thickener fibers and the oil. Including this last effect might lead to an explanation for the oil layer sometimes found on top of the grease after long storage times.

Nevertheless, the current model gives a good indication of the oil-bleed rate when including the anisotropic deformation of the soap microstructure.

CONCLUSION

The microstructure of a lithium complex grease has been visualized using AFM. The images show that the grease consists of solid fibers embedded in oil. These fibers have a twisted rope-like shape with a diameter of $\sim 0.2 \mu\text{m}$.

The oil bleeding of grease has been modeled as viscous flow through a porous soap microstructure subjected to a centripetal body load. The permeability of the soap microstructure has been assumed to be homogeneous and has been derived for an isotropic and anisotropic fiber arrangement. The model results show that the soap microstructure must become anisotropic during oil bleed, i.e., the fibers all tilt perpendicular to the pressure gradient and oil flow, to predict the experimental results and make the oil bleed stop at a certain fiber-volume fraction.

The model has been validated with experiments which have been run over a period of 24 h and can be used with good confidence to simulate the bleeding process over longer periods of time at various speeds and temperatures. The oil-bleed model can be used as input for a replenishment model for grease-lubricated contacts.

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Paper F

Film thickness model for grease lubricated bearing seals with an axial contacting lip.

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Abstract

A theoretical model is presented to predict the oil film thickness in an axial sealing contact based on grease properties and operating conditions. It is assumed that a small amount of grease will form an oil reservoir on the rotating part and slowly supplies oil to the sealing contact. The oil bleed model from a previous study is implemented and in addition oil loss due to centripetal forces and the seal pumping action are taken into account. The results show that, depending on the operating conditions, an oil film is present in the sealing contact for a certain period of time. The oil film thickness decreases in time due to the decreasing oil supply from the grease reservoir and oil loss from the contact due to centrifugal forces. Seal pumping has only little effect and the seal material and geometry are therefore not important for the predicted time until the mixed lubrication regime is reached. This time depends on the oil viscosity, rotational speed, and seal contact radius and scales with the $\eta/(n^2 \cdot d_s)$ parameter. Here, also the volumetric size of the grease reservoir has a large impact on maintaining the film thickness in time.

Keywords

Bearing seal, grease lubrication, film thickness.

INTRODUCTION

Bearing seals with multiple contacting seal lips are lubricated with grease in order to reduce friction and wear, and improve sealing performance. The grease provides oil to the sealing contact and an oil film separates the sliding surfaces. The replenishment mechanisms of lubricating greases are very different from those of oils due to the grease semi-solid behaviour. Unlike oil, grease may not freely flow to the sealing contact, limiting the amount of lubricant supply for lubrication. Consequently, the lubricant film thickness in the contact decreases, resulting in higher contact temperatures and increased seal wear (Dürnegger and Haas [1]). This limited contact replenishment is very similar to what happens in rolling bearings, as described in Lugt [2], where grease is initially pushed to the sides, away from the moving contact surfaces, and subsequently forms a grease reservoir on the seal or counter surface. The supply of oil to the contact is limited by the grease oil bleeding characteristics. Baker [3] found that the oil

bleed rate increases with temperature and strongly depends on the grease type. Baart et al. [4] presented a one dimensional physical model for predicting the oil bleed from the grease.

In the current study this oil bleed model is applied to bearing seals with an axial contacting lip. The supply of oil to the contact and loss from the contact are modeled to predict the film thickness as a function of time.

Figure 1 Bearing seal with a) an axial sealing lip contact with details in Figure 2, and b) a grease reservoir with details in Figure 3.

NOMENCLATURE

A_r	Cross section grease reservoir	m^2
b	Seal contact width	m
C_1	Constant	-
C_2	Constant	-
D_f	Soap fiber diameter	m
d_s	Sealing contact diameter	m
F_{body}	Specific body force	N/m^3
F_{lip}	Specific lip force	N/m
f_o	Initial soap mass fraction	-
G	Duty parameter	-
H_o	Initial height grease reservoir	m
h	Film thickness	m
h_{max}	Maximum film thickness	m
k	Permeability	m^2
n	Rotational speed	min^{-1}
∇p	Pressure gradient	Pa/m
Q_{body}	Flow rate oil loss - body force	m^3/s
Q_{feed}	Flow rate oil feed	m^3/s
Q_{loss}	Flow rate oil loss	m^3/s
Q_{pump}	Flow rate oil loss - pumping	m^3/s
R_c	Radius axial contact	m
R_o	Radius grease reservoir	m
r	Radius	m
t	Time	s
u_r	Radial velocity	m/s
V_o	Initial volume oil in contact	m^3
V_{oil}	Volume oil in contact	m^3
W	Width grease reservoir	m
η	Oil viscosity	$Pa \cdot s$
η_{40}	Base oil viscosity at 40°C	$Pa \cdot s$
η_{100}	Base oil viscosity at 100°C	$Pa \cdot s$
ρ	Grease density	kg/m^3
ω	Angular velocity	rad/s

THEORY

In grease lubricated seals the lubrication condition, or film thickness, is assumed to be determined by the availability of lubricant near the sealing contact. Grease may form an oil reservoir on the seal and/or counter surface depending on which part is moving. Assuming that the counter surface moves, most grease will be thrown onto the stationary seal and a relatively small amount of grease will remain on the counter surface, as indicated in Figure 1, which will then slowly release oil for lubrication of the sealing contact. This oil flow is driven by the large centrifugal forces that result from the rotational movement. The stationary grease on the seal will also release oil but on a much longer timescale due to the absence of these

forces. The formation of the grease reservoir depends on the amount of grease that is applied and the initial grease distribution in the volume between the seal and counter surface. In practice, system dynamics like vibrations, centrifugal forces, and shock loads exert forces on the grease reservoirs and may redistribute the grease. Additionally, grease may creep and grease which is in contact with a moving surface may degrade, as explained in e.g. Mérieux et al. [6], to provide some replenishment of the sealing contact as well. For the model presented in this work it is assumed that after a churning phase, the grease is distributed through the volume between the seal and counter surface and small permanent grease reservoirs are formed on the rotating part. No creep flow, dynamics, and grease degradation are assumed to be present afterwards. After the churning phase a maximum film thickness in the sealing contact is assumed, which is determined by the operating conditions and seal lip force. Subsequently, the film thickness may decrease depending on oil loss, due to seal pumping and centrifugal effects while there will be replenishment from the grease reservoir.

Model

The oil feed to and oil loss from the sealing contact is simulated to predict the film thickness inside the sealing contact. The *oil loss* from the contact takes place due to the natural pumping action of the seal and centrifugal forces acting on the oil film in the gap as indicated in Figure 2. Figure 3 schematically presents the *oil feed* due to oil bleed from a grease reservoir on the rotating counter surface. A balance between the oil feed and oil loss mechanisms determines the amount of oil in the sealing contact as

$$V_{oil} = \int_0^t (Q_{feed} - Q_{loss}) dt + V_o \quad (1)$$

where V_{oil} is the volume of oil in the contact as a function of time t and Q_{feed} and Q_{loss} are the flow rates of the oil feed and oil loss respectively. The initial volume of oil in the contact is V_o and results from the churning phase. Assuming a uniform oil film in the contact, the oil film thickness is defined as

$$h = \frac{\int_0^t (Q_{feed} - Q_{loss}) dt + V_o}{2\pi R_c b} \quad (2)$$

where R_c is the radial position of the contact and b is the width in the contact that contains the oil as defined in Figure 2.

Figure 2 Dimensions of the axial sealing contact from Figure 1 with a) oil loss from seal pumping and b) ingested meniscus and oil loss from centrifugal body load.

Oil feed

The specific oil feed from the grease reservoir is based on the oil bleed model presented in Baart et al. [4]. In this model the grease is considered as a porous medium where the grease thickener forms a solid microstructure that holds the base oil inside. Due to external body forces the oil will flow through the microstructure and bleeds out at the outer radius. This approach has been based on Darcy's law [7] which gives the fluid velocity in its general form as

$$u = -\frac{k}{\eta} \nabla p \quad (3)$$

where k is the permeability, η the base oil viscosity, and ∇p the pressure gradient. Baart et al. [4] show how the permeability, or ability for the oil to flow through the thickener microstructure, is a function of the oil volume fraction in the grease and the orientation of the microstructure. The pressure gradient ∇p in Darcy's law can be replaced by a body force, e.g. gravity or centrifugal force. Consequently the oil flow rate or oil loss is a function of the grease microstructure, temperature, and rotational speed and in its most simple form reads

$$Q_{bleed} = -2\pi R_o W \frac{k(f)}{\eta(T)} \rho \omega^2 r, \quad (4)$$

where R_o the outer radius of the grease reservoir, W is the width of the grease reservoir, f is the thickener volume fraction in the grease, η is the oil viscosity from the Walther equation [8], and the body force on

the oil given by the oil density ρ , the angular velocity ω , and the radius r . Characteristic dimensions of a simplified grease reservoir are given in Figure 3. The oil bleeding, or oil supply to the sealing contact, has a maximum at the beginning and reduces as a function of time. The reader is referred to Baart et al. [4] for more details on the model.

Figure 3 Dimensions of the grease reservoir from Figure 1 located on the rotating part after the churning phase.

Oil loss

The oil loss from the sealing contact results from body forces on the oil film in the contact and the natural pumping action of the seal such that

$$Q_{loss} = Q_{pump} + Q_{body} \quad (5)$$

where Q_{pump} is the oil loss from the natural pumping of the seal and Q_{body} is the oil loss from body forces.

Oil loss from natural pumping of the seal is predicted using empirical equations from Horve [5]. These equations have been derived from extensive experimental work on oil lubricated radial shaft seals of which both sides are fully flooded with oil. It is assumed that the equations can also be applied to axial contacting seals. Since the seal geometry, i.e. lip angles, and seal material are not included in this model, it is assumed that the axial sealing contact in Figure 2 is equal to the seal geometry and material that Horve used. From the equations Horve [5] presents, an equation for the seal pump rate is derived assuming fully flooded conditions

$$Q_{pump} = 1.04 \cdot 10^{-8} R_c^3 \frac{n}{60} G^{-1/3}, \quad (6)$$

where R_c is the radial position of the contact, n the shaft speed in rpm, and G the duty parameter given as

$$G = 2\pi b \frac{n}{60} \frac{\eta}{F_{lip}}, \quad (7)$$

where F_{lip} is the specific lip force. The film thickness can also be calculated from equations presented in [5]. This is the film thickness at fully flooded conditions and consequently the maximum film thickness possible as:

$$h_{max} = 0.011 R_c G^{5/9}. \quad (8)$$

The natural pumping equation and maximum film thickness equation from Horve are only valid when the seal is fully flooded. Stakenborg [9] and Salant [10] showed that as soon as all the oil is pumped from the air side to the oil side of the seal, the oil meniscus will ingest into the sealing contact as indicated in Figure 2b. When this happens, pumping due to the asymmetric pressure distribution as described by Kammüller [11] will no longer take place. This hypothesis was proven through numerical simulation by Salant [10] who showed that even multiple equilibrium positions for the meniscus exist. Consequently the seal will not continue pumping oil. The volume of oil that is subsequently present in the sealing contact is given as

$$V_{max} = 2\pi R_c b h_{max}, \quad (9)$$

where b is the width of the oil volume when the meniscus is ingested and h_{max} is the maximum film thickness as in fully flooded conditions. This volume V_{max} is considered as the maximum volume of oil in the contact below which the seal no longer pumps oil according to Eq. (6). Oil loss from seal pumping is dominating when the oil volume in the contact is $V_{oil} > V_{max}$. When $V_{oil} < V_{max}$ then seal pumping is no longer present and oil loss only takes place due to the centrifugal forces on the oil film in the sealing contact. Figure 2 graphically defines these lubrication conditions and corresponding contact dimensions.

Assuming a linear velocity profile over the film thickness in circumferential direction, the centrifugal forces on the oil film vary from zero on the stationary seal surface, where $z = 0$, to the maximum value on the rotating flinger surface, where $z = h$. The specific body force then becomes

$$F_{body} = \rho \left(\omega \frac{z}{h} \right)^2 r, \quad (10)$$

where ρ the oil density, ω the angular shaft velocity and r the radial position. Considering a thin layer, the body force induces shear flow defined by

$$\eta \frac{d^2 u_r}{dz^2} = -\rho \left(\omega \frac{z}{h} \right)^2 r, \quad (11)$$

where u_r is the flow velocity in radial direction. Because the width b is small compared to the contact radius R_c , it can be assumed that the radial flow velocity is constant through the contact width such that r in Eq. (11) can be replaced by R_c . Eq. (11) can be integrated twice and after applying the boundary conditions where $u_r = 0$ at the walls, the radial velocity u_r is then found as

$$u_r(z) = -\frac{\rho \omega^2 z R_c}{12 \eta h^2} (h^3 - z^3) \quad (12)$$

Integrating Eq. (12) over the gap height and multiplying with the contact circumference gives the oil loss from body forces as

$$Q_{body} = 2\pi R_c \frac{\rho \omega^2 R_c}{40 \eta} h^3 \quad (13)$$

The oil loss Q_{body} will result in a reduction of the lubricant film thickness in the seal contact keeping the width b constant.

To summarize the theoretical model three phases of operating conditions can be identified.

- Phase 1: In the churning phase the grease is being churned and the grease reservoirs are formed. The film thickness in the contact is equal to the maximum film thickness h_{max} .
- Phase 2: Oil bleed from the grease reservoir supplies oil to the sealing contact. At the same time oil is lost from the contact due to centrifugal forces and seal pumping as long as fully flooded conditions remain as in Figure 2a.
- Phase 3: The maximum film thickness can not be maintained due to insufficient replenishment of the contact. The oil meniscus is ingested into the sealing contact and seal pumping stops as in Figure 2b. Oil loss from the contact is continued due the body forces and the film thickness decreases slowly in time.

To predict the film thickness in time, Eq. (2) is numerically integrated according to the calculation

scheme in Figure 4 where the different phases are included. Here the two conditions as described in phase 2 and phase 3, and shown in Figure 2, can be identified.

Figure 4 Calculation scheme for film thickness calculation.

RESULTS AND DISCUSSION

A prediction of the oil film thickness in the axial sealing contact of a specific bearing seal and grease type at different operating conditions is made using the parameters presented in Table 1. Parameters related to the grease properties are taken from Baart et al. [4] which apply for the lithium complex grease with mineral base oil as used in their study. Other parameters are related to the seal design in Figure 1 and Figure 2 and seal pumping parameters from Horve [5].

A_r	$0.5 \cdot 10^{-6}$	m^2
b	$0.4 \cdot 10^{-3}$	m
D_f	$0.2 \cdot 10^{-6}$	m
F_{Lip}	20	N/m
f_o	0.26	-
H_o/W	2	-
R_c	$44 \cdot 10^{-3}$	m
R_o	$42 \cdot 10^{-3}$	m
η_{40}	0.10	Pa s
η_{100}	0.011	Pa s
ρ	900	Kg/m^3

Table 1 Parameter values for the grease and seal design.

To evaluate the influence of the operating temperature, rotational speed and grease reservoir, the model is used to calculate the time until a critical film thickness $h_{crit} = 0.25 \mu\text{m}$, equal to the counterface surface roughness, is reached. The critical film thickness may

be used as an indication for the transition from the full film lubrication to the mixed lubrication regime where some direct contact between the seal and counterface may occur.

Figure 5 shows the predicted film thickness as a function of time and operating conditions including and excluding a grease reservoir. The maximum film thickness depends on the rotational speed and temperature, i.e. the base oil viscosity. In the case that a grease reservoir is present, i.e. $A_r = 0.5 \text{ mm}^2$, for a limited amount of time the maximum film thickness remains constant before it decreases due to the limited replenishment of the sealing contact. The curves for the two speeds at 70°C show that at lower speeds the maximum film thickness is present for a much longer time since at high speed the oil loss from the sealing contact is much higher. Figure 5 also shows the predicted film thickness without the presence of a grease reservoir. Here the film thickness starts to decrease instantaneously and it takes significantly shorter time to reach the critical film thickness. For example at 70°C and 2000 rpm it takes with the grease reservoir about 80 h to reach the critical film thickness of $0.25 \mu\text{m}$ and without the grease reservoir only 0.5 h.

Figure 5 Oil film thickness in the sealing contact: starting from fully flooded conditions.

In Figure 5 the cases which include a grease reservoir all start with the same grease reservoir size. In reality the formation of the grease reservoir during the churning phase will also depend on the operating conditions and it is likely that a smaller grease reservoir is formed at higher temperatures and higher speeds. Consequently, the difference between the two curves at a constant temperature of 70°C will be even more pronounced. The initial grease fill and the distribution of the grease in the sealing system are

therefore of crucial importance for the formation of the grease reservoir and the total amount of oil that can be available for lubrication. In the model, a temperature increase only results in a reduction of the base oil viscosity. Consequently, equal model results, i.e. equal film thickness, can be obtained when at higher temperature grease with higher viscosity base oil is used such that the effective base oil viscosity is unchanged.

Figure 6 shows the time until the critical film thickness is reached for several operating conditions. Here the symbols represent the model results from Eq. (2). The results show how the critical time is a function of temperature for different rotational speeds and a fixed grease reservoir size.

Figure 7 shows that this critical time can be significantly extended when a larger initial grease reservoir A_r is present, provided that $H_o/W = 2$, at fixed rotational speed.

Figure 6 Operating time until critical film thickness is reached, $A_r = 0.5 \text{ mm}^2$. Points are model calculation results.

Figure 7 Operating time until critical film thickness is reached, $n = 1000 \text{ rpm}$. Points are numerical calculations.

The lines in Figure 6 and Figure 7 represent a fit to the results of the physical model in Eq. (2). This fit can be used as a simple engineering model which is based on some characteristic parameters. In both the oil bleeding model in Eq. (4) and the oil loss model in Eq. (13) the influence of the oil viscosity, rotational speed and seal contact radius on the flow rate is given as $(\omega \cdot R_c)^2 / \eta$ provided that $R_o \sim R_c$. Here the $\omega \cdot R_c$ is very similar to the $n \cdot d_m$ term, where n is the rotational speed in rpm and d_m is the mean diameter as commonly used in the modelling of rolling bearings. For the axial seal lip a characteristic term $\eta / (n^2 \cdot d_s^2)$ can be defined where $d_s = 2 \cdot R_c$ is the diameter of the sealing contact. According to the film thickness equation in Eq. (2) this term has to be divided by the contact radius giving $\eta / (n^2 \cdot d_s)$ and is used as de basis for the engineering model. Here also the influence of the size of the grease reservoir is included which has been presented in Figure 7. Including two fitting constants the engineering model reads

$$t_c = [C_1 A_r + C_2] \left[\frac{\eta}{(n^2 \cdot d_s)} \right], \quad (14)$$

where for this particular grease type and case $C_1 = 2 \cdot 10^{15} \text{ (kg} \cdot \text{m)}^{-1}$, $C_2 = 6 \cdot 10^6 \text{ kg}^{-1} \cdot \text{m}$, and $A_r = WH_o$ provided that $H_o/W = 2$. The seal geometry is included in d_s , the base oil viscosity and temperature in η and the rotational speed in n . The results of Eq. (14) are plotted as continuous lines in Figure 6 and Figure 7. Figure 8 shows how the results from the physical model in Figure 6 fall on one line when scaled with the characteristic term $\eta / (n^2 \cdot d_s)$.

Figure 8 Operating time until critical film thickness is reached based on the $\eta / (n^2 \cdot d_s)$ parameter for $A_r = 0.5 \text{ mm}^2$.

A physical model to predict the lubricating conditions and film thickness in an axial sealing contact has been presented. Based on this model a simplified

engineering model has been developed that can be used to predict the time until the mixed lubrication regime is reached. It is possible to include a more advanced pumping model including the actual seal lip geometry. However, when the pumping phase (phase 2) is short compared to the phase where the film thickness decays (phase 3) and the maximum film thickness is at least a few times higher than the critical film thickness, the pumping rate has little effect on the predicted time to mixed lubrication. Under these conditions the present engineering model is also valid for other axial seal materials and geometries. In addition, the model can be extended with e.g. a prediction of the grease reservoir formation and percolation theory for calculations below the critical film thickness, i.e. in the mixed lubrication regime.

CONCLUSION

A model has been presented to predict the lubricant film thickness in the sealing contact taking into consideration replenishment of the contact from grease reservoirs. Here the grease reservoir can only supply oil for lubrication of the sealing contact for a limited time period and subsequently the film thickness decreases due to centrifugal forces acting on the oil layer. It is shown how the film thickness decreases with increasing rotational speed, increasing operating temperature and decreasing grease reservoir size. It is concluded that the presence and dimension of a grease reservoir has a very large impact on the lubricating conditions in the sealing contact. The model is used to derive a simplified engineering model to calculate the time until the mixed lubrication is reached.

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