JOURNAL BEARING DESIGN, LUBRICATION AND OPERATION

FOR ENHANCED PERFORMANCE

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Cover figure: Journal bearing in the old Porjus power station on the Lule älv.

Title page figure: Inside the large journal bearing on the cover showing texture and oil rings.

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Preface

The work of this thesis has been conducted at the Division of Machine Elements at Luleå University of Technology.

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Abstract

The increasing introduction of intermittent power sources combined with the de-regulation of electricity markets has led to increased instability in the electrical grid. This has led to increased start-up and shutdown of regulating power sources such as hydro-electric power plants and operation at non-ideal operating states both of which increase the wear and tear on machines. Likewise, the push for a less environmentally intrusive society has raised the importance of utilizing equipment with reduced impact on the natural surroundings.

These challenges lead to a need to improve the robustness of existing and new equipment to guarantee their usefulness in a future with increased operational instability. As a part of this improvement process, this work is focused on the guide/journal bearings which support the rotating portion of power generating machines. These bearings are studied using using a multi scaled approach covering small and large scale laboratory experiments as well as investigations of a full scale machine.

A journal bearing test machine was constructed to investigate a number of new synthetic lubricants and polymer bearing materials. These tests found that a significant reduction in power loss could be accomplished without significantly affecting the bearing’s minimum film thickness by changing from a traditional mineral oil to a high viscosity index oil of much lower base viscosity grade. The high viscosity index lubricants were then improved to reduce start-up friction as well. Further studies were conducted in small scale to determine the optimum lubricant characteristics for the startup problem. This knowledge was used to develop new lubricants to test in the journal bearing test machine which showed large reductions in power loss in the bearing and pumping system as well as greatly reduced bearing operating temperature.

Further experimental work led to the development of practical guidance for power plant operators contemplating a lubricant change. This technique focuses on the importance of maintaining equivalent viscosity in the minimum film thickness region after a lubricant change. Efficiency improvements can then be calculated by comparing the viscosity in the bulk of the bearing to that with the original lubricant.

Experimental work with polymer bearing facing materials demonstrated the dramatic reductions in break away friction that these materials can provide. A number of polymer composite materials were investigated for their friction characteristics at the moment of the start of sliding, finding that PTFE based materials were far superior to traditional white metal. This work with polymer faced bearings was extended to testing in a tilting pad journal bearing test rig
which allowed for identification of the dynamic characteristics resulting from changes in bearing pad material and configuration.

Investigations in the full scale with the Porjus U9 unit provided valuable insight into the dynamics of a full scale machine as well as needed data for the improvement of multi-physics models of bearings. Insights from the Porjus U9 experiments clarify many of the design challenges for large journal bearings in hydro power machines such as the thermal transients during startup and the dynamic effects during load changes.

The results from this work demonstrate that significant performance improvement of journal bearings is possible through the use of new lubricants, materials, and adjustments in operational methods.
Appended Papers

[A] G. Simmons, S. Glavatskih.
Synthetic Lubricants in Hydrodynamic Journal Bearings: Experimental Results.
Presented at World Tribology Congress, Kyoto, Japan, September 6-11, 2009
All bearing experimental work and the writing of paper A were accomplished by the author. Lubricants and their characteristics (viscosity, etc) were provided through collaboration with Statoil and Evonik.

Powerplant Lubricant Selection for Improved Efficiency and Environmental Impact Reduction.
All bearing experimental work and the writing of paper B were accomplished by the author. Modeling work was accomplished by Evgeny Kuznetsov with the author’s input.

[C] A. Golchin, G. Simmons and S. Glavatskih.
Break-away Friction of PTFE Materials in Lubricated Conditions.
Design of the experimental work and writing of paper C were accomplished by the author together with A. Golchin. The bulk of the experimental work was accomplished by A. Golchin. SEM work and analysis of the results was jointly done by the author and A. Golchin.

[D] G. Simmons, S. Glavatskih, M. Müller, Å. Byheden, and B. Prakash
Extending performance limits of turbine oils.
Accepted for publication in Tribology International
Lubricants studied in this work were formulated by Å. Byheden of Statoil Lubricants and R. Schweder and F. Lauterwasser of Evonik with input from all authors. Experimental work and writing of paper D were accomplished by the the author.
[E] G. Simmons, A. Cerda, I. Santos, S. Glavatskih and B. Prakash
Dynamic characteristics of polymer faced tilting pad journal bearings.
Manuscript to be submitted.
Bearings used in this work were designed by the author. Experimental work was conducted at DTU in collaboration with A. Cerda and I. Santos. Writing of paper E was accomplished by the author with input from all authors.

[F] G. Simmons, J.O. Aidanpää, M.J. Cervantes and S. Glavatskih
Operational transients in the guide bearings of a 10 MW Kaplan turbine
Accepted for publication in International Journal of Hydropower and Dams.
Experiments and data analysis were carried out by the author. Preparation of the machine for the experiments was conducted by a number of project participants. Writing of paper G was accomplished by the author with input from all authors.

Steady state and transient characteristics for guide bearings of a hydro-electric unit
Manuscript to be submitted.
Experiments and data analysis were carried out by the author while modeling was carried out by M. Cha. Writing of paper G was accomplished by the author and M. Cha with input from all authors.

Additional papers not included in thesis

G. Simmons, M. Mofidi and B. Prakash
Friction evaluation of elastomers in lubricated contact

A. Golchin, G. Simmons, S. Glavatskih and B. Prakash
Tribological behavior of polymeric materials in water lubricated contacts

M. Cha, G. Simmons and S. Glavatskih
Compliant pad bearing design considerations for hydropower units
Proceedings of HydroVision 2013, Denver, USA.
Chapter 1

Introduction

What are hydrodynamic bearings and what connection do they have to the field of tribology and the hydroelectric power industry?

Use of rotating machinery to convert raw power into useful work is a fundamental block of modern industrialized civilization. This rotating machinery is involved in almost every facet of life, be it the crank shaft in the engine of a tractor used to harvest soy beans, the hub on a bicycle, or the turbine/generator unit in a hydro-power station delivering electricity to the electrical grid. Regardless of their scale and use, rotating machines have some similarities: they all have at least one rotating part and that rotating part must be separated from the stationary portion of the machine by some sort of bearing. This bearing provides an easily sheared layer that allows the surface of the rotating part (shaft) to slide relative to the stationary part. This interface and the surfaces which bound it are central to the study of Tribology, the study of surfaces in relative motion including friction, lubrication and wear.

In the case of large rotating machines such as steam turbines, hydro-electric turbines and other electrical power generators the machine’s shaft is most commonly isolated from the stationary equipment through the use of journal bearings. To maintain low friction and minimize wear of the surfaces, these journal bearings are filled with lubricant.

Tribology and Journal Bearings

Tribology spans several scientific fields including mechanical engineering, materials science, physics and chemistry but its importance to everyday life is
CHAPTER 1. INTRODUCTION

Figure 1.1: The Stribeck curve showing the transition from boundary to mixed and finally to hydrodynamic lubrication regimes with changing speed, viscosity or load on the contact.

Often overlooked. To study the interaction between surfaces a few basic questions must be answered about the contact:

- Is the contact lubricated?
- In which lubrication regime does the contact operate?

This work focuses on contacts in which lubricant is always present. Lubricated contacts can be divided into three distinct zones according to the Stribeck curve, Fig. 1.1. A lubricated contact at rest is most often in the boundary or mixed lubrication regime in which the surface contours (asperities) of the two surfaces are well interconnected resulting in high friction. At the onset of relative motion between the two surfaces, pressure begins to build in the lubricant. Building lubricant pressure causes the load to shift from the asperities to the lubricant. The lubricant pressure continues to build with increased speed, decreased load, or increased lubricant viscosity which results in greater separation between the surfaces. As these factors are further increased, the lubricant pressure reaches a point where the surfaces’ asperities no longer make contact and the load shifts completely to the lubricant. The lubricant provides an easily sheared layer that leads to dramatic reduction of the friction in the contact as the load shifts from the asperities to the lubricant. Further increasing speed
Figure 1.2: Hydro-dynamic pressure buildup in a journal bearing. Lubricant is normally supplied to these bearings from the sides or top when the bearing operates with a horizontal shaft. With a vertical turbine, the bearing is often partly immersed in an oil bath and the pressure field rotates around the bearing with the shaft’s motion.

leads to an increase in shear rate which leads to somewhat increased friction in bearing applications. Wear of surfaces generally decreases from the boundary regime through the mixed regime and ideally does not occur after the contact has entered the hydrodynamic lubrication regime.

Journal bearings have been used in rotating equipment since the invention of the wheel. In its simplest form, a journal bearing consists of a cylinder built around an axle with a small clearance separating the two. To reduce wear of the surfaces, the clearance is filled with a lubricant which allows the contact to operate in the mixed or hydrodynamic regime. When the bearing operates in the hydrodynamic regime, a fluid pressure builds up in the bearing by a converging geometry effect in the lower portion of the bearing, shown in Fig. 1.2. Following the minimum clearance portion of the bearing, the clearance increases (the surfaces diverge). This diversion results in a pressure drop in the lubricant that leads to cavitation (this portion of the bearing is often referred to as the ‘cavitation zone’). Because of the pressure imbalance between the inlet side and the outlet side of the bearing, the shaft shifts towards the outlet side
which allows the pressure profile to balance with a resulting total force in the vertical direction.

The vast range of demands that different machine configurations place on their journal bearings has led to a wide variety of journal bearing designs, each suited to a specific application. This work focuses on large full film journal bearings that operate under hydrodynamic lubrication almost all of the time. The size of these bearings can range from a few centimeters to meters in diameter with the primary commonality being that the asperities on their surfaces are at least one order of magnitude smaller than the operating oil film thickness. Modern large journal bearings generally can be classified into two groups based on their geometry which is either fixed or adjustable.

A common adjustable geometry journal bearing is the tilting pad journal bearing (TPJB) as shown in Fig. 1.3. This type of bearing is composed of individual bearing pads (as in Fig. 5.12) which pivot to develop a pressure field in the oil film to support the load. The number of pads can be varied depending on the application and expected bearing loads. Bearings in large hydropower machines can have up to 10 or more pads while bearings in smaller horizontal machines may have only three pads.

In horizontal configurations, the literature often discusses 'load on pad' and 'load between pad' meaning that the gravitational load of the shaft is located over the bottom pad or between the bottom pads respectively. How-
ever, this is irrelevant in vertical configurations such as those in hydropower machines because the dynamic portion of the load rotates around the bearing while the load due to pad pre-load remains fixed over each bearing pad.

When bearings are lightly loaded (as in vertical machines), bearing pads are pre-loaded to provide a stabilizing load to the shaft and minimize dynamic motion. The pre-load is defined in Eq. (1.1).

\[
Preload = \frac{R_{pad} - R_{bearing}}{R_{pad} - R_{shaft}}
\] (1.1)

Lubricant in a TPJB can be provided using several different methods. The most simple is to fill the bearing with oil which is then circulated to a cooling system. More advanced lubrication methods involve spraying oil at the shaft between the bearing pads or injecting through the pad via a groove near the bearing pad’s inlet.

The bearing pad’s pivot generally makes a spherical contact or a line contact with the support structure. A spherical contact allows the bearing to tilt in both the axial and tangential direction which allows the bearing to self align to the shaft surface. A roller pivot which forms a line contact with the backing structure can also be used but alignment errors due to machining tolerances can potentially be an issue for larger machines.

**Hydro-Electric Power**

Hydropower harnesses the power of falling water. The two basic requirements to extract useful power from water are an elevation drop, which provides gravitational potential energy, and a flow which allows for the potential energy to be transformed into useful work. For example, a cup of water held in the air has potential energy which cannot be harnessed until the water is poured out or the cup is lowered. In most hydropower applications, potential energy is concentrated using a dam, Fig. 1.4, to abruptly change the elevation of a river’s water surface. The natural process of evaporation of sea water to form clouds which then drift over land and produce precipitation provides a supply of water which then flows into and along a river’s channel. Water is then channeled into the penstock and into a spiral casing where the flow is steered into the turbine which converts the potential energy of the water into kinetic (rotating) energy, Fig. 1.5. The rotating turbine’s energy can be tapped to accomplish many tasks, but in general it is used to turn an electric generator which in turn drives electrical current to electrical transmission lines and ultimately to end users.
Figure 1.4: Vittjärv hydro-electric power plant on the Lule River undergoing flood safety improvements autumn 2006.

Figure 1.5: Schematic of a hydro-electric power plant’s mechanical equipment with a Kaplan turbine. Water flows into the turbine through the inlet tube and spiral casing and returns to the river via the draft tube. The small figure on the right represents a person to provide a perspective of scale.
Like a cup of water poured onto the floor, a flowing river in its natural state releases almost all of its potential energy through friction, turbulence and noise. This lost power is apparent when standing near to any large waterfall or rapid. However, to the hydropower engineer, friction, turbulence and noise are forms of power losses that should be minimized. Impressively, modern large hydropower plants often operate at over 90% efficiency. This leaves little margin for further improvements. By comparison, the best thermal power plants are limited by their Carnot efficiency to approximately 60%. The remaining 40% is released to the environment in the form of heat. Wind turbines, solar cells and internal combustion engines harvest an even smaller fraction of their input energy. When the energy requirements for construction and operation are taken into account, hydroelectric power has been demonstrated to be the most effective power source available today [56].

In a modern hydroelectric power plant (see Fig. 1.5), the losses are generally attributed to turbulent and frictional losses in the waterways and turbine, resistive and fan losses in the generator and, most importantly for this work, losses due to friction in the bearings and seals. While technology and design of turbines and generators has progressed in the last century, basic design and operation of the bearings has changed very little since Kingsbury and Michell independently patented the tilting pad bearing at the beginning of the 20th century [80]. However, according to the insurance industry, wholly 40% of operational economic losses of hydropower plants are the result of bearing failures [3]. Whether the bearing itself fails or the machine is operated in such a way that leads to bearing failure is uncertain from the study. However, considering the micrometer tolerances in the bearing surfaces in comparison to the meter scale dimensions of those same bearings and the grand scale of the concrete structures they are installed in, not to mention the operational demands placed on them, it is a wonder that fluid film bearings function as well as they do.

**Materials**

Hydrodynamic bearings utilize a protective layer to guard the shaft against damage at machine start up and shutdown. This protective layer has traditionally consisted of a thick coating of bronze or tin alloy known as Babbitt or white metal. Various forms of white metal have been widely used in sliding bearings since their invention in the 19th century. While this material has been tried and proven for over a century in sliding bearings, its use is not without drawbacks. Because white metal is especially susceptible to adhesion at break
away, hydrostatic jacking systems are commonly used to transfer the load from the white metal layer’s surface to lubricant under high pressure during machine startup. These extra systems add complexity to the bearing system and start-up cycle which can reduce total machine reliability. Furthermore, because of white metal’s relatively low melting point, white metal bearings are susceptible to softening at higher temperatures. This directly limits the maximum mean pressure during operation to between 2 and 3 MPa depending on the specific geometry.

Progression of material science has allowed for development of new materials and processing techniques not earlier imaginable. Polytetrafluoroethylene (PTFE), with its low friction characteristics, has been considered to be one of the better available materials for bearings. However, its wear rate is higher than desirable [66] and PTFE can also have a tendency to creep under higher temperature loading and continuous use. Furthermore, when used in bearings, the insulating characteristics of most polymer materials can result in higher lubricant temperatures than for materials which more readily conduct heat away from the contact. This can result in differences in thermal expansion between the shaft and bearing which may lead to reduction of the bearing’s clearance. Other polymer materials such as ultra high molecular weight polyethylene (UHMWPE) provide excellent friction and wearing characteristics [67], but are severely limited by their maximum operating temperatures. Additionally, most polymer materials for low friction applications are developed with the goal of self lubricated operation often with solid lubricant particles embedded into the material. These solid lubricant particles are not necessary in high speed machinery because liquid lubricant is always present in the contact for cooling and to allow for operation in the hydrodynamic lubrication regime where the lowest friction and highest surface speeds are possible.

Countless new materials as well as blends of PTFE have been developed which maintain the same low friction offered by PTFE while providing added strength and wear resistance. These materials are based on composites of PTFE or other polymers such as PEEK [18]. Applying these more advanced polymer materials to sliding bearing applications could potentially eliminate the issues faced by current materials while allowing for significant improvements in machine performance.

Studies of a two-axial-groove journal bearing demonstrated that changes in length and clearance in combination with changes in bearing material and lubricant could provide significant reductions in power loss in bearings while improving their dynamic performance [24] [86].
Figure 1.6: Variation in dynamic viscosity vs. temperature for fluids with different viscosity indices. All fluids have equal viscosity at 100°C but viscosity at 40°C varies greatly.

**Lubricants**

As with material science, lubricant technology has progressed dramatically since the original development of mineral oil based lubricants in the 19th century. In the face of new developments such as custom synthesized lubricants and additives, many industries continue to operate with lubricants made to old specifications. New understanding of chemical processes provides the potential to tailor make lubricants for optimum performance in specific applications thus improving efficiency and machine safety. Equally important to developing and selecting the optimum lubricant for a given application is adapting the operation of the machine to best match the lubricant’s characteristics. Opportunities to dramatically change journal bearing performance by changing lubricant have not been previously available thus a firm understanding of the differences that can be expected from application of new lubricants needs to be developed.

Typical turbine lubricants available and used today are based either on classic high quality petroleum base stocks or some form of synthetic base stock such as ester, poly-alkylene-glycol (PAG), or poly-alpha-olefin (PAO). Additionally, any number of additives are blended into the base oil to improve specific characteristics such as anti-oxidation additives which extend lubricant
life, extreme pressure additives to protect surfaces in high pressure contacts or anti-foaming additives to reduce the lubricant’s tendency to trap air bubbles. Most important to this work are additives which improve the viscosity index (VI) of the lubricant.

A lubricant’s VI describes how the viscosity of a lubricant changes with variations in temperature. Generally, as temperature increases, the viscosity of a fluid decreases following a logarithmic relationship as shown in Fig. 1.6. The five lubricants have equal viscosity at 100°C but very different viscosities at 40°C due to the differences in their VI. Because fluid flow losses in a system are directly coupled to the viscosity of the fluid and the force needed to shear it, Eq. (1.2), any reduction in viscosity can potentially result in direct power savings due to reduced flow losses.

\[ F = \mu \cdot \frac{Area \cdot Velocity}{FilmThickness} \]  

Development of new lubricant additives has shown potential to increase VI to levels not before seen. These new additives have been observed to be able to provide decreased friction as well through the formation of polar bonds with metal surfaces [139]. Further development of functionalized poly-alkylmethakrylate (dPAMA) led to a description of the most desirable characteristics of dPAMA molecules for film forming [105]. Namely, the molecules should be large, have functional groups, and the functional groups should be concentrated in the central area of the molecule as opposed to being out on the fringes. Use of lubricants containing such additives could potentially allow for greatly improved performance in normal operation with the advantage of reduced friction at machine startup.

Lubricants also play an important role in the ongoing industry shift toward more environmentally and socially acceptable operations. Proper selection of base oils and feed stocks allows for production of lubricants with bio-degradability and non-toxicity. Following an in depth study of lubricant usage in aquatic environments, [16] the European Union established an eco label standard for lubricants [2] with specific requirements for bio-degradability and renewability of lubricants mostly in total loss applications such as chainsaws and outboard boat motors. Additionally a number of national standards regarding lubricant biodegradability have been developed such as SS 155434 [138] which include turbine lubricants.

Hydropower plants can potentially release significant quantities of lubricant into waterways. The greatest risk for leakage lies in the turbine bulbs of many older Kaplan turbines which are filled with hydraulic oil to lubricate the
turbine blade bearings and regulate the turbine blades. While these machines are slowly being upgraded to operate with 'oil-free' or water filled bulbs to eliminate a large potential leakage point, elimination of all leakage is not feasible in many cases and so using lubricant with some degree of bio-degradability provides an alternative. Other potential leakage risk points are the turbine guide bearing which sits immediately above the turbine in the power plant and the guide vane control hydraulics which are only separated from the turbine chamber by a few centimeters. The remainder of the bearings in the system are generally situated a safe distance from any potential leakage path to the river. However, the use of less hazardous materials in industrial applications offers additional, less tangible, benefits making it worthwhile to change lubricant to an EAL regardless of the specific application. Some of these benefits include the potential to reduce the health and safety risks and extra costs related to handling, storage and general use of lubricants.

**Dynamics**

Rotating machines, by their nature, are dynamically active. In most analyses, focus is made on excitations resulting from unbalances in the machine’s rotating parts. In the case of electrical generators, slight changes in frequency of the electrical grid can add further excitation to the machine. Hydropower machines are subjected to still more excitation generated by the water flow through the turbine and the turbine blade interactions with the guide vanes [77]. These excitations result in rotor motion that must be damped to allow the machine to operate safely and reliably. However, hydro power machines have very little damping built into their systems which, ideally, are built into a stiff concrete and bedrock foundation. The primary damping forces in a hydro power plant must thus come from the electrical grid connection, the water flow through the turbine, and the bearings. Traditionally, the bearings have been focused upon as they provide one of the greatest sources of direct stiffness and damping in rotating machines [29].

Further complicating the hydropower machine dynamics is their commonly vertical configuration. Unlike horizontal machines which operate with a generally constant eccentricity and attitude angle in their bearings due to the mass of their shafts, vertical machines are free to orbit within their bearings at an eccentricity governed by the pre-load of the individual bearing pads.

As with any rotating machine, it is especially true for hydro power turbines that small changes in bearing characteristics can have profound effects
on the machine’s performance. Nässelqvist [108] demonstrated that the increased damping provided by increasing bearing clearance kept a hydro power machine from dangerous dynamic excitation at startup when it was excited by a neighboring machine that shared a common draft tube. Similar effects on journal bearing dynamics and orbits were demonstrated when bearing pre-load was adjusted in any one of three bearings supporting a hydro power turbine [47].

More drastic changes in bearing characteristics may be possible by replacing the white metal surface with a compliant layer. Model results have shown that compliancy of the bearing’s surface can allow for more effective pressure development and higher pressure in the bearing. Additionally, because of the visco-elastic nature of polymer materials, bearings manufactured from polymers can potentially provide an extra degree of damping to the machine [23].
Chapter 2

State of the science

What have recent studies on journal bearings found, how have they been conducted, by whom, and what developments have been made to the technology?

A tremendous amount of research effort has been given to the subject of hydrodynamic journal bearings over the course of the second half of the 20th century and the beginning of the 21st century. Since Lund’s groundbreaking work [89] in the understanding of journal bearing dynamics, understanding of the function of hydrodynamics and journal bearings has been steadily developed with at least a couple research groups around the world producing a steady flow of research results. The advent of improved computers has given much to simulations of journal bearings however with ever improving simulation software, the need for tangible experimental results becomes essential.

Unfortunately, unlike much tribological testing, test equipment for journal bearings is neither standardized nor versatile. Further antagonizing this is the complexity of thermal and dynamic effects taking place in hydrodynamic journal bearings under operation, which extend over several length scales. This necessitates full scale or near full scale test equipment, which is in-turn large and expensive. The challenges associated with testing of journal bearings has led to the research being focused at several research centers around the world as opposed to broadly spread as is the case with some other fields of tribological research.

As in most research fields, reviews of the research have been accomplished with the most recent being that of Swanson [143], and his following doctoral thesis, [142] which well document experimental data for fixed geometry hydrodynamic journal bearings. The full range of bearings in terms of dynamics
were covered more recently by Tiwari et al [154] while methods and experimental data for extracting journal bearing dynamic coefficients were reviewed by Dimond et al [30] including a detailed analysis of bearing test machines with dynamic testing capabilities.

**Research groups**

A number of research groups contributed to the experimental dataset for hydrodynamic journal bearings during the previous 15 years. The following is a synopsis of their varying focuses and the work they have contributed during this period.

**University of Virginia**

One of these groups led by Ron Flack during the 90’s at the University of Virginia has determined characteristics of many different bearing geometries using a test rig in which the bearing housing was shaken against a rigidly mounted shaft [50], [83]. Of note in this series of works is that the error between the theoretical and experimental values is generally more reasonable than the majority of research work within this field. These works provided valuable bearing data regarding a number of bearing configurations. Investigations on pad pivot friction in tilting pad bearings [160], [161] found that rocker pivoting pads caused the shaft loci to take a straight line path to its equilibrium location under increased loading while in the ball and socket case, the shaft took a curved path. Similarly in the case of cross-coupled dynamic coefficients, the rocker pivoted bearing’s coefficients were equal to zero while the ball and socket pivoted bearing’s coefficient’s were not equal to zero, and were significant. These differences in performance of two otherwise similar bearings was attributed to the vast differences in friction produced in the ball and socket and rocker pivots.

When highly preloaded three-lobe journal bearings were studied [113], it was found that the thermal boundary conditions changed very little as loading direction was changed and additionally that the stiffness and damping were greatest perpendicular to the load and smallest in the load direction when the loading was centered on the pad. Testing of a range of excitation frequencies [114] demonstrated that there was little change in the stiffness and damping coefficients due to changes in excitation, and further that they remained linear through a wide range of loading (up to 30% eccentricity). Several different
preloads were experimented with on tilting pad journal bearings in [162], finding that the Sommerfeld number had little correlation with the thermal characteristics of the pad and that maximum precision was required in the lubricant inlet conditions to observe the slight differences in the bearing temperatures and to control the expansion of the bearing and shaft. This study was continued into the dynamic regime in [163] finding that negatively preloaded pads had a finite point of instability while positively preloaded pads displayed infinite stability. Unfortunately, this study was not able to produce confident values for bearing coefficient due to large uncertainties. Plans are currently in progress to extend this work into determination of dynamic characteristics in the turbulent regime [28] with a test rig utilizing magnetic bearings to support a 127 mm diameter shaft with surface speeds up to 150 m/s.

Texas A&M University

The other main group of researchers in the field of bearing dynamics during the last 15 years is led by Dara Childs and Luis San Andres at Texas A&M University. They conducted very similar work to Flack et al. using a host of similar test rigs generally with a stationary shaft and a bearing or squeeze-film damper excited by hydraulic shakers. This work has been centered on studying the characteristics of various bearing configurations.

Squeeze-film dampers were analyzed in [127] with significant difference (2x difference) between theory and measured results which is partially explained by air entrapment in the damper. This air entrapment issue was further confirmed in [126] which found that the squeeze film damper’s damping characteristics, while decreasing with increased air flow, showed little difference between those for dampers fed with pure oil and those fed with a 50% oil 50% air mixture. However, this difference also could have been caused by oil transfer from the support bearings to the squeeze film dampers. The squeeze film dampers were improved in [129] to include elastomer seals on their ends, thus trapping lubricant inside the damper. This significantly reduced the air entrapment and the damper’s characteristics were found to be quite independent of frequency while the seal provided greater damping at slower speeds than higher speeds.

The group has also investigated flexible pad bearings integrated into a squeeze film damper, [125], finding that the new arrangement greatly improved the system’s damping coefficients compared to operation with a tilting pad journal bearing only. Dynamic characteristics of flexible pad bearings were investigated in the load on pad configuration, [26], further helping the research
group to develop bearing models. Of note with this work is the high speeds (16,000 RPM) that the bearing was operated to and that the support bearings caused the most significant limitations. These tests were also accomplished with the bearing arranged in the load between pad configuration in [5] finding that the current models generally predict the bearing parameters. This set of tests was then accomplished again at higher loading (up to 2.2 MPa) in [6], [74] and [121] in both load-on pad and load-between pad configurations and with varying clearance finding that at these loading conditions, the bearing could be well modeled as frequency independent when added mass terms were added to the model formulation. In terms of static characteristics, temperatures reached 90° C with between 50 and 60 l/min of VG32 lubricant flow.

Hybrid bearings that function both in hydro-dynamic and hydro-static states were earlier studied when lubricated with water in a series of works beginning with a description of the test rig and initial test results in [84]. Good agreement was obtained between theoretical and experimental results, but most useful from this first study is the detailed descriptions of the challenges related to experimentation. The operating characteristics of high load, high speed hybrid bearings with water led to intense cavitation and material erosion with the initial material selection of brass. Hardened stainless steel was later used and generally solved this issue combined with a much finer filtering system.

This test equipment was then modified to include dynamic testing capabilities using an arrangement of springs and hydraulic shakers, [25], with a detailed analysis of the experimental uncertainties and parameter determination methods. Uncertainties determined in upstart testing ranged from under 5% for the stiffness and direct damping terms to much higher (up to 60%) for the added mass and cross-coupled damping terms. However, the higher uncertainties appeared to be more a result of the small values of the coefficients than large values of uncertainty. In [52] the rig was used to compare the performance of several different hybrid bearing geometries. This study found that while each of the four geometries tested had specific advantages, the use of an inlet angled against the flow of rotation provided the most beneficial characteristics. The uncertainty and a full comparison of the hybrid bearing test rig with theoretical models was detailed in, [53]. More investigation of water-lubricated hybrid bearings was conducted in [124] using a further refined model and finding that tangential injection of the lubricant provided clear advantages in terms of dynamic performance. However, it was also noted that the advantages offered by tangential lubricant injection were found to diminish as the speed increased and the bearing began to leave the hydrostatic regime and instead operate in the hydro-dynamic regime. Surface texturing on the bearing
pads was investigated in [46] on bearings that had earlier been tested without any modifications and it was found that the roughness significantly improved the bearing’s stiffness and damping characteristics, allowing for much higher speeds without the potential of rotor instability caused by the bearing.

This series of bearing research turned towards bearing characteristics in real applications in [87] with testing conducted on a hybrid bearing with a worn down surface and a plugged supply port. Results of the tests show that the bearing characteristics are dramatically influenced by the presence of a plugged inlet at higher eccentricities but little influence was observed at lower eccentricities. However, the bearing continued to function which was believed to be the result of an increase in hydrodynamic effects. In the case of the worn bearing, the changes in dynamic characteristics were less dramatic, demonstrating that the bearing could function reasonably well with a worn surface.

Pressure dam bearings were the topic of [7] finding that this form of bearing provided a large degree of stability and operated at generally higher eccentricities than similar tilting pad bearings. Of note is that the coefficients found in this study had generally good correlation to theoretically calculated values except in a few cases. Dynamic coefficients for a 5-pad, load-between-pad tilting pad bearing were investigated in [20] with exceptionally low error in both stiffness and damping coefficients. A random waveform excitation method was used and the stiffness and damping coefficients appeared to be independent of the excitation frequency.

Case Western Reserve and Cleveland State Universities

In [132] several methods are compared for extracting journal bearing dynamic coefficients from experimental data and several useful observations are made regarding test rig design and construction. This work was continued in [133], using a test rig with a controllable counter-rotating shaft initially reported in [131], with the conduction of experimental work that produced bearing dynamic coefficients within 30% of theoretically predicted values.

University of Wollongong

During this same period, Qiu and Tieu accomplished a number of works in which they determined dynamic coefficients of small (30 mm diameter) and large (200 mm diameter) hydrodynamic journal bearings. They first used rotor imbalances giving small displacements to the rotor [152] with quite good agreement between theory and measured values. A theoretical study is con-
Conducted in [117] investigating misalignment affects on bearing dynamic coefficients and operational characteristics, with results of note that the misaligned bearing demonstrated more stability than the properly aligned bearing under similar loading. The thermal effects within the journal were investigated [153] under misalignment and with varying loads, finding that temperature in the bearing was in the expected range with the highest temperatures occurring in or immediately after the region of lowest film thickness. Further work included harmonic excitation of the shaft via a journal bearing [118] with a detailed analysis of the calculation method and uncertainty in the measured values of the bearings’ dynamic coefficients.

Another method which involves impacting a small rotating shaft supported by two journal bearings was utilized in [119] to determine bearing dynamic characteristics with quite reasonable correlation to theoretically calculated values, however it was found that a significant portion of the error could be attributed to the coupling between the motor and shaft. To reduce this effect, the authors conducted several repeat tests at varying shaft frequencies.

**University of Poitiers**

Michel Fillon at the University of Poitiers has led a group conducting testing on bearings in realistic operating conditions to help better understand what happens within the bearing in non-ideal operating situations. Monmousseau began with rapid start-up temperature tests [102] finding that thermal effects in the fluid film and thermal effects within the pad had a significant effect on the performance of the bearing. Further it was concluded that any future models should take these effects into account to obtain more realistic analysis of tilting pad bearings.

Tests were continued in [104] to determine the bearing response to static and dynamic loading conditions caused by failure of a component within a rotating machine. A theoretical study was accomplished in which several different models were compared finding that for accurate modeling, all thermal characteristics should be included and, additionally that the response of a machine to transient dynamic loading is to a large degree dependent on the ratio of the dynamic loading to the static loading. Sudden shaft excitation produces a mechanical transient period on the order of a few revolutions of the shaft while the thermal transient period is much longer, with the specific lengths of these transient periods being governed by the ratio of dynamic to static loading.

Rapid start-up and the case of a bearing seizure was investigated in [103] finding that seizure is caused by a combination of coupled mechanical and
thermal effects with the mechanical effects having a shorter response time than the thermal effects. Additionally, it was found that the primary cause of film thickness reduction, ultimately leading to total loss of clearance and seizure, is shaft expansion as opposed to pad expansion. Dynamic and static characteristics are investigated in [100] at high speeds and under loading, finding that the bearing clearance changes by as much as 20μm from 3000 to 15,000 RPM. Of special note was that their rotor exhibited a response as well as their bearing demonstrating that rotors in bearing test rigs need to be extremely stiff.

Characteristics of mis-aligned tilting pad bearings [99] with pressure, viscosity, and temperature curves were analyzed as well as minimum film thicknesses for a host of different operating states. Finally a numerical study of the safe operating regimes for tilting pad journal bearings, [101], found that the chance of bearing seizure could be greatly reduced by accelerating the system slowly or by increasing the inlet lubricant temperature. This same study found that as seizure occurs in the bearing, temperatures and pressures rapidly spike at the same time as minimum film thickness reaches a minimum.

The mis-alignment theme was continued in [115] helping to develop a 3-dimensional approach to THD analysis and finding in terms of experimental results that the misalignment direction was not detectable in the center of the bearing although the misalignment itself caused an increase in the temperature and a decrease in pressure in the center of the bearing. Maximum temperatures in the case of the misaligned bearing, predictably, occurred at the side of the bearing where the misalignment was applied. Effects of pad fluttering were investigated in case study format in [73] finding that bearing characteristics, including the elimination of flutter, were significantly improved when the leading edges of the bearing pads were machined at a slight angle, 11° and a depth of approximately 250 μm. With this modification, not only did the bearing pads stop fluttering, but the bearing itself began operating at a much lower temperature.

**Korea Heavy Industries**

An industry researcher, Hyun Cheon Ha of Korea Heavy Industries and Construction, conducted valuable applications focused work first regarding the effects of inlet pressure on the thermal performance of tilting pad journal bearings, [71], finding that greater flow was present at higher inlet pressures, and vice versa. From comparison of the results to a theoretical model, it was concluded that any thermal models must include the inlet pressure in order to accurately determine the bearing’s thermal characteristics.
This same practical applications based approach was continued in [72] in which the original 300 mm diameter test rig was improved to include dynamic excitation. The effects of excitation frequency on dynamic coefficients were investigated, finding that the stiffness and damping vary only slightly with excitation frequency with the greatest variation occurring at low speed, high load combinations. Of note in these tests is the rather low bearing pressures used of between 110 kPa and 220 kPa.

Full scale testing of bearing pre-loads in a 300 MW hydropower journal bearing with 1450 mm diameter were accomplished in [70] to determine the thermal effects on the pads and vibration frequencies of the main shaft caused by changes in the bearing pad pre-load. In this study, one of very few involving full scale machines, it was found that increasing the pre-load on the pads caused a reduction in pad temperature and an increase in bearing stiffness. The increase in bearing stiffness resulted in a dramatic drop in the vibration of the shaft and machine. The author, however points out that this increase in stiffness could lead to greater risk of damage in machines operating outside of optimal operation ranges.

National Research Council of Canada

Another group more focused on industrial applications, the National Research Council of Canada, also conducted research on large bearings during this time. Although they did not focus heavily on dynamics and dynamic coefficients, they were focused on developing better understanding of bearings for the hydropower industry through investigations of thermal characteristics. In [32] a tilting pad journal bearing with a leading edge groove was evaluated in comparison to a standard bearing, finding superior performance for the bearing with the leading edge groove (LEG). Temperatures were found to be equal at the leading edge of the pad with large reductions in temperature in the case of the LEG bearing further into the pads. In some cases this temperature reduction was near 20°C and it was concluded that the difference was primarily caused by the hot oil carryover from one pad to the next. The geometry of the LEG bearing prevented hot oil from the exit of a pad to be carried over to the inlet of the following pad.

Following the trend of improving tilting pad journal bearing design, the performance differences between offset pivot and centered pivot bearings were investigated in [14] using both load-on-pad and load-between-pad configurations. This work found that the offset pivot bearing operated with considerably lower pad temperatures than the centered pivot bearing and further that the
load between pad configuration yielded lower temperatures than the load-on-pad. The temperature difference between load-on-pad and load-between-pad configurations became quite large at higher speeds but was nominal at lower speeds.

Other studies included operation of an offset pivot bearing in reverse [27], finding that the bearing seemed to function properly albeit with much higher temperatures. Comparison of general bearing performance with varying viscosity grade of oil was conducted as well [15], finding that ISO VG68 led to higher power loss, higher temperature, and a thicker oil film than ISO VG32 under the same operational conditions.

**Luleå University of Technology**

On the lubricant side of the journal bearing research field, work has of late been focused on improving models and understanding of the effects of newer synthetic lubricants in journal bearings. Work conducted by McCarthy investigated the more fundamental aspects of improving bearing performance, including use of synthetic lubricants, [97], and use of PTFE based materials was examined in [96]. These studies found improved bearing performance with the synthetic lubricants including reduced power loss, while the studies on materials highlighted the potential improvements in startup friction provided by PTFE composites.

**Other individuals**

**El-Butch**

Modeling work in regards to the effect of mis-alignment in bearings has been accomplished relatively recently by El-Butch [42]. This study found that misalignment led to an increase in the maximum pressures and that bearing materials with some degree of elasticity aided to absorb bearing misalignment. In an earlier study the same author [40] performed similar modeling on a bearing with various levels of air bubble content in the lubricant, finding that the pressure peak in the bearing was both increased and delayed as bubble content increased.

El-Butch et-al also conducted an initial study, [41], in which tilting-pad journal bearings with rubber and steel pads were compared, finding that rubber pads improved bearing performance without sacrificing minimum film thickness. However, thermal effects were not included in the rubber model due to
uncertainties in the rubber’s thermal characteristics, and experimental data was not included, or mentioned, for comparison and model validation.

Ma

Ma and Taylor conducted work on journal bearings as well [90]. This work primarily focussed on the differences between elliptical and plain journal bearings under higher pressures (such as those found in internal combustion engines). Notable results were that the temperature fade and oil flow rate in the elliptical bearings were significantly greater than in the circular bearings. This follows theory that more lubricant can leak out in the inlet and outlet area of an elliptical bearing than in a similar circular bearing due to the larger clearances in the elliptical bearing.

Syverud

At the Norwegian University of Science and Technology, Syverud conducted studies of some unique effects in journal bearings [146]. It was found that the temperature of the shaft plays a major role in the bearing surface temperature and further that thermal expansion should be taken into account so far as proper measurement and calibration of the eccentricity is concerned. It was later theorized through the use of a vacuum [145] that the high level of cooling observed in the cavitation zone is the result of an endothermic evaporation process initiated by the exceptionally low pressures in the cavitation zone.

Experimental approaches

At the simplest level, testing of journal bearings requires a bearing, a shaft and sensing equipment. Assuming the bearing is to be tested under loaded conditions, a method of producing a load between the bearing and shaft is needed which then necessitates supportive structure to counter-act or develop the shaft loading.

In the vast majority of rotating machines, the shaft floats freely in the bearing while loads are produced by a combination of gravitational forces and un-balances in the rotor’s loading. Full scale tests of machines have been conducted in industrial applications with actual loading, [70], [107]. Similar work is conducted when rotating machinery is balanced prior to delivery. The primary drawback with this form of testing, especially on vertical machines, is
that the loading parameters are nearly impossible to measure directly. Furthermore, controlling and calibrating the loads placed on the bearings in most cases is not possible or un-realistically expensive with an installed machine. Unfortunately, this arrangement is rather difficult to reproduce in the laboratory environment as the mass of an entire machine is needed to produce the proper loads.

Therefore, the vast majority of bearing test equipment utilizes a shaft held stationary by support bearings and loaded by the test bearing – the opposite of actual machine arrangement. Primary loading of the bearing is accomplished using hydraulic cylinders connected through a coupling with multiple degrees of freedom [97], or by the use of an air bellows [32]. More complex arrangements have also been considered [50]. Dynamic testing is accomplished by shaking the test bearing using electro-magnetic or hydraulic shakers.

Other solutions have been successful as well including placing the test bearing at the end of a shaft and then loading the shaft mechanically or with a magnetic bearing as in [82], however this method can produce other challenges dealing with rotor-dynamics as in the case of [142]. Another potential drawback to this method is that by mounting the bearing at the end of the shaft, mis-alignment caused by the fact that only one end of the shaft is shaken is possible.

Using a stationary test bearing allows for the most accurate measurement of the bearing characteristics but unfortunately as noted above it can be difficult to isolate and control the shaft’s motion. While using traditional roller bearings is possible, the added dynamic effects caused by the roller bearings can drown out the characteristics of the test bearing itself. Thus the ultimate solution to this problem is to carry and load the shaft using magnetic bearings which have the advantage of not contacting the shaft and so not hindering the shaft’s motion. However, the costs both monetary and time of developing such a test rig and its control system are daunting.

Table of Papers

A compilation of the geometry, load, speed, and collected data for the recent experimental work regarding hydrodynamic journal bearings is detailed in Table 2.1. The various data collected is abbreviated as described in Table 2.2.

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Table 2.1: Test-Rig Operating Characteristics
<table>
<thead>
<tr>
<th>Reference</th>
<th>Bearing Type Type</th>
<th>Diameter (mm)</th>
<th>L/D</th>
<th>Speed (m/s)</th>
<th>Load (MPa)</th>
<th>Measurements</th>
</tr>
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<tbody>
<tr>
<td>[83]</td>
<td>3-lobe</td>
<td>70.0</td>
<td>0.75</td>
<td>9.20</td>
<td>1.30</td>
<td>D, Fi, Pi, T</td>
</tr>
<tr>
<td>[50]</td>
<td>2-axial groove</td>
<td>70.0</td>
<td>0.75</td>
<td>8.24</td>
<td>0.73</td>
<td>D, Fi, Pi, T</td>
</tr>
<tr>
<td>[161]</td>
<td>TPJB (pivot types)</td>
<td>70.0</td>
<td>0.75</td>
<td>8.24</td>
<td>0.73</td>
<td>D, F, U</td>
</tr>
<tr>
<td>[160]</td>
<td>TPJB (pivot types)</td>
<td>70.0</td>
<td>0.75</td>
<td>8.24</td>
<td>0.73</td>
<td>D, F, U</td>
</tr>
<tr>
<td>[113]</td>
<td>three-lobe</td>
<td>70.0</td>
<td>0.75</td>
<td>3.30</td>
<td>0.12</td>
<td>D, Fi, T, U</td>
</tr>
<tr>
<td>[114]</td>
<td>three-lobe</td>
<td>70.0</td>
<td>0.75</td>
<td>3.30</td>
<td>0.13</td>
<td>D, Fi, U</td>
</tr>
<tr>
<td>[162]</td>
<td>tilt pad (preload)</td>
<td>70.0</td>
<td>0.75</td>
<td>8.24</td>
<td>0.73</td>
<td>Fi, T, U</td>
</tr>
<tr>
<td>[163]</td>
<td>tilt pad (preload)</td>
<td>70.0</td>
<td>0.75</td>
<td>8.24</td>
<td>0.73</td>
<td>D, F, U</td>
</tr>
<tr>
<td>[125]</td>
<td>squeeze film damper</td>
<td>76.2</td>
<td>0.30</td>
<td>35.89</td>
<td>NA</td>
<td>D, U</td>
</tr>
<tr>
<td>[127]</td>
<td>squeeze film damper</td>
<td>132.0</td>
<td>0.24</td>
<td>24.87</td>
<td>0.04</td>
<td>D, F, Pi</td>
</tr>
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<td>[126]</td>
<td>squeeze film damper</td>
<td>96.3</td>
<td>0.24</td>
<td>0.00</td>
<td>NA</td>
<td>D, U</td>
</tr>
<tr>
<td>[129]</td>
<td>squeeze film damper</td>
<td>127.0</td>
<td>0.20</td>
<td>0.00</td>
<td>0.14</td>
<td>D</td>
</tr>
<tr>
<td>[26]</td>
<td>flex. pivot TPJB LOP</td>
<td>117.0</td>
<td>0.65</td>
<td>97.97</td>
<td>1.03</td>
<td>D, T, F, U</td>
</tr>
<tr>
<td>[5]</td>
<td>flex. pivot TPJB LBP</td>
<td>117.0</td>
<td>0.65</td>
<td>73.48</td>
<td>1.03</td>
<td>D, T, F, U</td>
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<tr>
<td>[6]</td>
<td>flex. pivot TPJB LOP</td>
<td>117.0</td>
<td>0.65</td>
<td>73.48</td>
<td>1.03</td>
<td>D, T, F, U</td>
</tr>
<tr>
<td>[74]</td>
<td>flex. pivot TPJB LOP</td>
<td>117.0</td>
<td>0.65</td>
<td>73.48</td>
<td>2.21</td>
<td>D, T, F, U</td>
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<td>flex. pivot TPJB LOP</td>
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<td>D, U</td>
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<tr>
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<td>1.00</td>
<td>99.70</td>
<td>1.40</td>
<td>P, F</td>
</tr>
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<td>[25]</td>
<td>hybrid water</td>
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<td>1.00</td>
<td>99.70</td>
<td>1.40</td>
<td>D, P, T, F, U</td>
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<tr>
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<td>hybrid water</td>
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<td>1.00</td>
<td>113.65</td>
<td>2.93</td>
<td>D, P, F, U</td>
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<td>hybrid water</td>
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<td>99.70</td>
<td>1.40</td>
<td>D, P, F, U</td>
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<tr>
<td>[124]</td>
<td>hybrid water</td>
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<td>1.00</td>
<td>99.70</td>
<td>1.40</td>
<td>D, P, F, U</td>
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<tr>
<td>[46]</td>
<td>hybrid water</td>
<td>76.2</td>
<td>1.00</td>
<td>99.70</td>
<td>1.40</td>
<td>D, P, F, U</td>
</tr>
<tr>
<td>[87]</td>
<td>hybrid water</td>
<td>76.2</td>
<td>1.00</td>
<td>99.70</td>
<td>1.40</td>
<td>D, P, F, U</td>
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<td>[7]</td>
<td>pressure dam</td>
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<td>1.03</td>
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<td>69.39</td>
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<td>[133]</td>
<td>hydrostatic 4 pad</td>
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<td>0.47</td>
<td>12.04</td>
<td>D, F, U</td>
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<td>[131]</td>
<td>hydrostatic 4 pad</td>
<td>115.0</td>
<td>0.47</td>
<td>12.04</td>
<td>D, F, U</td>
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<td>D, F</td>
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<td>25.4</td>
<td>1.00</td>
<td>4.65</td>
<td>D, F</td>
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<td>[153]</td>
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<td>D, T, F, Pi, F, U</td>
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<td>41.87</td>
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<td>[119]</td>
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<td>1.00</td>
<td>4.65</td>
<td>D, F</td>
<td></td>
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<td>[102]</td>
<td>4-pad TPJB</td>
<td>100.0</td>
<td>0.70</td>
<td>8.37</td>
<td>1.43</td>
<td>F, T</td>
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<tr>
<td>[104]</td>
<td>4-pad TPJB</td>
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<td>0.70</td>
<td>8.37</td>
<td>1.43</td>
<td>F, T</td>
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<tr>
<td>[103]</td>
<td>4-pad TPJB</td>
<td>100.0</td>
<td>0.70</td>
<td>8.37</td>
<td>1.43</td>
<td>F, T</td>
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<td>[100]</td>
<td>4-pad TPJB</td>
<td>100.0</td>
<td>0.70</td>
<td>8.37</td>
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<td>[99]</td>
<td>4-pad TPJB</td>
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<td>0.70</td>
<td>8.37</td>
<td>1.43</td>
<td>F, T</td>
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<td>[101]</td>
<td>4-pad TPJB</td>
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<td>0.70</td>
<td>8.37</td>
<td>1.43</td>
<td>F, T</td>
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<td>0.70</td>
<td>20.93</td>
<td>1.29</td>
<td>F, P, T</td>
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<td>0.32</td>
<td>D</td>
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<td>66.16</td>
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<td>Fi, Pi, T</td>
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<td>[72]</td>
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</tr>
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<td>[70]</td>
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<td>0.28</td>
<td>34.15</td>
<td>NA</td>
<td>D, F</td>
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Table 2.2: Measurements and analyzed bearing characteristics abbreviations

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<tr>
<th>Measurement / Analysis</th>
<th>Type</th>
<th>Abbreviation</th>
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<tr>
<td>Dynamic Coefficients</td>
<td>Dynamic</td>
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<td>Film Thickness</td>
<td>Static</td>
<td>F</td>
</tr>
<tr>
<td>Film Thickness</td>
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<td>Fi</td>
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<tr>
<td>Film Pressure</td>
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<td>P</td>
</tr>
<tr>
<td>Film Pressure</td>
<td>Continuous</td>
<td>Pi</td>
</tr>
<tr>
<td>Bearing Temperature</td>
<td>Static</td>
<td>T</td>
</tr>
<tr>
<td>Uncertainty</td>
<td>Calculated</td>
<td>U</td>
</tr>
</tbody>
</table>

**Technological development**

A historical analysis of the previous years of hydrodynamic journal bearing research shows a steady improvement in technology and the understanding of the way in which geometrical parameters affect performance characteristics. Many of these trends were continued to test the limits of modeling software and determine whether new operational arrangements were acceptable. Development of the leading edge groove (LEG) bearings was allowed through more complete understanding of the stability effects of TBJJ’s. Likewise, testing of the location and type of pivot used in journal bearings helped to better understand the mechanisms which occur in TPJB’s.
Some work regarding use of new lubricants has been conducted, finding both advantages and disadvantages to new synthetic lubricants but very little deeper investigation has been undertaken. A similar case can be seen in the materials side of journal bearing research. Work regarding bearings with surfaces other than white metal is extremely rare considering the effort spent maintaining white metal faced bearing pads and the time and money lost due to their failure [3], [19]. Some work has been accomplished regarding PTFE lined bearings using a copper mesh backing [39], [58], [98], [137]. This work validates the usefulness of PTFE in hydrodynamic bearings but does not lead to further development of the technology. Models have been developed to further understand the effects of compliant materials in journal bearings, but to date, very little laboratory or industry work has been made on this topic. Without this understanding, the status quo will continue to be used in new design.

**Future needs and research gaps**

With ever increasing demands placed on electrical power networks through the large scale introduction of variable power sources (wind power, solar power, etc.) demands placed on electrical generators is only expected to increase. Further, these variable power sources by their nature of variability place undesirable demands on the journal bearings that support them. A few of these demands can be higher loading, increased starting and stopping, wider operating ranges, faster acceleration and thermal cycling. Thus, better understanding of the support bearings is essential to allow for the new uses of power generation machines. While current dynamic characteristics of bearings are normally calculated to within 50%, better accuracy is highly desired. Finally, the role that new materials and lubricants play on operating characteristics is generally unknown and must be well understood for their wider acceptance and adoption.
Chapter 3

Objectives

What are the goals of this research work?

The objective of this work is 'Determination of dynamic characteristics of hydrodynamic journal bearings' to improve their operation in large machines by reducing power losses and environmental impact while at the same time increasing machine reliability.

Specifically this work aims to:

- Investigate the performance of and further develop new environmentally adapted lubricants for use in large sliding bearing applications.
- Investigate the potential of new materials to improve transient and dynamic operation of journal bearings.
- Develop practical guidelines for the application of new sliding bearing and lubricant technology in industry.
Chapter 4

Methods

How is this research carried out and how do the individual pieces fit together?

This thesis provides a modern investigation of journal bearing design, testing, and application. Countless simulations have shown the potential of changes in bearing technology, but for any change in design or operation to be adopted by a greater audience, strong support and proof of viability based in experiments are necessary. Because full scale experiments are expensive and inflexible, experiments should be focused at the small scale. However as pointed out by Taylor [148], scaling effects of experiments in tribology can lead to uncertain results. This necessitates the inclusion of full scale experiments in any development campaign.

Conducting experiments over such a range of scales requires the use of a variety of test conditions and equipment. Descriptions of the testing equipment used throughout this work to study the journal bearing contact are provided in Chapter 5.

The primary issue to be solved for today’s machines is increased starting and stopping. Because the bearing and shaft are in contact (and presumably in boundary or mixed lubrication) before start and after stop, the case can be well studied using reciprocating small scale test equipment, see Chapter 6. Testing in the small scale is further advantageous for studying wear as it allows for the study of a large number of materials and lubricants. Statistical accuracy can be guaranteed through the use of a large number of tests.

The weakness of small scale testing is the inability to investigate geometry and thermal effects such as those that govern the function of hydrodynamic journal bearings. Likewise, for industry acceptance of new materials and tech-
niques testing at as near to full scale as possible must be accomplished. This large scale testing requires the use of a full scale test rig with a large journal bearing used in actual equipment. The construction, development and test results of this rig are detailed in Chapter 7.

Safe operation of rotating machinery requires control of a machine’s dynamic characteristics. For new developments in journal bearings to be acceptable, their effects on the dynamic characteristics of machines must be understood or at a minimum benchmarked to existing solutions. Dynamic testing of new bearing solutions was originally planned to be accomplished using a modified version of the full scale test rig. Unfortunately, a combination of financial, design, and material availability issues made it not feasible to complete the modifications. Dynamic testing was therefore accomplished using equipment at the Department of Mechanical Engineering of the Technical University of Denmark. This equipment and the development of these tests are described in Chapter 8.

Finally, the opportunity to benchmark computer models and smaller scale experimental results with truly full scale equipment was taken using the U9 machine at the Porjus hydro-electric power station on the Lule River. This heavily instrumented machine allowed for the measurement of a host of characteristics of three separate and different tilting pad journal bearings under a variety of operating conditions. The details of this work are provided in Chapter 9.
Chapter 5

Experimental equipment

What experimental equipment was used?

Several testing arrangements were utilized to study hydrodynamic journal bearings from laboratory scale tribo-meters to a full scale machine. Small scale experiments were primarily conducted using a reciprocating block on plate arrangement while a block on disk arrangement was used for complimentary testing and investigation of stick slip at very slow sliding speeds. A journal bearing test rig was designed and built to perform steady state testing while dynamic testing was conducted using a two pad tilting pad journal bearing test rig at the Technical University of Denmark. Experiments and measurements in the full scale were conducted on the Porjus U9 machine in Porjus, Sweden. The range of scale over these very different pieces of test equipment is best illustrated by each machine’s rated power: the block on plate and block on disk have a rating in the 100 W range, the motor on the tilting pad journal bearing test rig is rated at 3 kW while the motor powering the larger journal bearing test rig is 43 kW and the Porjus U9 machine is rated at 10 MW.

Laboratory Tribometers

The reciprocating test arrangement used in this work was adapted to represent a small area of a sliding bearing. Rectangular flat bearing specimens are held in contact with a stationary flat steel counter-surface in an oil bath as shown in Fig. 5.1. The bearing specimens for testing at 1 MPa were 5 mm x 16 mm while they were 5 mm x 8 mm for all other tests. These different sizes were necessary to keep test loads within the optimum range for the reciprocating ma-
Figure 5.1: The small scale reciprocating test arrangement.

Figure 5.2: Example of measurement result from breakaway friction testing for Oil B at 8 MPa mean load. Breakaway friction is determined from the maximum peak at the start of each sliding cycle. Approximately 30 seconds of testing is displayed.
machine. The oil bath temperature was measured using a thermo-couple which was connected in a feedback loop to guarantee constant oil temperature. Sliding speed and frequency were set at the test machine’s lower limit to allow for isolation of the point at which break-away occurred in each start-stop cycle. Using a slow cycle speed also helped to stabilize any dynamic effects in the machine and additionally allowed for a maximum number of data points per stroke. A typical example of several cycles of data from break-away testing is shown in Fig. 5.2.

Both a 10 mm and 15 mm stroke length were tested through the course of the experiments. In the end, the shorter stroke length was selected as the primary configuration as it minimized the total sliding distance and therefore the wear of the specimens. The shorter stroke length also provided a slightly slower turn around at the end of each stroke which, in turn, allowed for more accurate data sampling.

The maximum friction value (at break-away) was recorded for each cycle over the entire one hour test period while the shape of the break-away curve was recorded at the end of the test period. Analysis of the data showed that not only did the break-away friction vary significantly for different materials and lubricants, but the variation in break-away over the course of each test also varied significantly for some materials and lubricants while it remained essentially constant for others. The reciprocating arrangement was used with all lubricants and materials that were tested in the small scale and provided clear, repeatable results.

To gain deeper insight into the friction effects at the moment of break-
away, a block on disk arrangement was developed, Fig. 5.3. This arrangement utilized the same block specimens as the reciprocating test equipment with the exception that the plate was replaced with a disk which rotated. The block specimen was first engaged with the plate and the contact was loaded to the desired contact pressure, then the plate was very slowly set into motion. This resulted in a dynamic loading of the block holder and stick slip at the system’s natural frequency. The block on disk arrangement was used for only a couple lubricants and materials because the time consuming and ungainly nature of the test machine and its software made extended test campaigns and analysis of their results unrealistic.

**Steady State Journal Bearing Test Rig**

Inclusion of the thermal and hydrodynamic effects present in journal bearings is absolutely necessary for any development of journal bearing technology. The best way to include these effects is through the use of actual journal bearing geometry loaded under equivalent conditions to those in a full scale machine. This requirement led to the development of the steady state journal bearing test rig which utilized a modified journal bearing from an actual machine.

The basic arrangement of the test rig is illustrated in Fig. 5.4. A motor
Figure 5.5: Journal bearing test rig. Black hoses to the right and left return lubricant to the storage tank. The white hose in the center provides pressurized air to the air cushion loading system.

(red) is connected, via a flexible coupling (yellow), to a shaft (teal). The shaft is supported by two roller bearings (purple) and the test bearing is mounted between these two bearings. A slip ring (gray) is used on the end of the shaft opposite the motor to connect on-board sensors to the data collection system. The test bearing housing (blue and green) is supported by 4 load cells (green) which are bolted to a steel block (black). Loading is provided by an air cushion (light blue) placed under the steel block. The foundation is filled with sand to stabilize the machine with extra mass as well as improve the damping of high frequency structural resonance.

One of the key goals in design of this test machine was modularity. The primary elements of the test rig are designed to accommodate industry standard bearing designs while specialized bearings can be installed using custom manufactured holders which are easily swapped. The majority of electrical couplings to the shaft and bearing are standard quick connectors which greatly reduce time for repairs and modifications. The use of standard components was stressed throughout the design and construction of the rig to ensure that as many components as possible could be later re-used for other laboratory test equipment. Furthermore, the design allows for relatively simple assembly and disassembly requiring 1 person and a crane (complete removal and disassembly of the shaft requires less than 2 hours).
Figure 5.6: Journal bearing test rig shaft showing installation of sensor plugs on shaft outer diameter. The hollow portion of the shaft allows for cable routing and ends where it intersects the through bore at the sensor plugs.

Figure 5.7: Sensor plug from test rig shaft showing arrangement of pressure and displacement sensors as well as temperature sensor locations.
The initial design of the test rig had pressure sensors and inductive displacement sensors installed within the shaft to provide pressure and displacement profiles of each revolution. These sensors were mounted in two opposing caps held onto the shaft with custom formed socket head cap screws (Fig. 5.6). One cap held 2 piezo-electric pressure sensors while the other cap was equipped with 1 piezo-electric pressure sensor and 2 inductive displacement sensors (Fig. 5.7). A description of the areas covered by the final sensor arrangement is provided in Fig. 5.8.

During initial testing, it was observed that the inductive sensors operated very well, providing film thickness profiles with good accuracy. Unfortunately, during the first round of testing with synthetic lubricants, the sensor elements released from their housings and ceased to function properly. This problem was investigated and a solution was developed that mechanically held the sensing elements in their housings. However, solution of this issue brought another problem into light. Having released from their housings, the sensor elements now compressed and decompressed with each revolution due to the high oil film pressure in the bearing. This led to erratic readings where the oil film became suddenly extremely thick at the point of highest film pressure. An attempt was made to solve this issue, but in the end, the inductive sensors were removed and replaced with filler screws.

Without displacement sensors inside the shaft, determination of shaft eccentricity became a challenge which led to the installation of eight displacement sensors outside of the shaft, four on each side of the bearing housing, with an initial 90° offset. These sensors allowed for measurement of the static eccentricity and attitude angle of the shaft during operation. Because the loading system relies on a pressure load which allows the bearing to float around the shaft, determining when the shaft is centered in the bearing poses yet another challenge.

The challenge of finding the zero point of stationary displacement sensors with absolute certainty is well documented in the literature. The most common solution is to operate the bearing with no load at the test rig’s maximum speed as in [14], [15]. This type of operation causes the shaft to converge as near to zero eccentricity as possible and ideally provides a measurement of the location at which the shaft and bearing centers coincide. Alternatively, the shaft center can be found by placing the bearing at a relative eccentricity of 1 in several locations and calculating the intersection of these locations. Both methods were used in this work with the second method (loading and unloading) being observed to provide the most repeatable zero location. Because the bearing center was found with the system ‘cold’, a temperature sensor was
placed inside the shaft at the displacement measurement location to monitor the potential for thermal expansion. During testing, shaft temperature at the measurement point was observed to be generally constant regardless of the specific lubricant used. It is believed that the shaft temperature at the measurement point is more affected by heat from the roller bearings than by the heat generated in the test bearing.

Similar challenges were experienced with the pressure measurements, with the difference being that the pressure sensors were designed and able to withstand the journal bearing environment. The greatest challenge with the pressure sensors was isolating them from the test fluid without affecting their measurements. Because of the cavitation zone in the journal bearing the volume in front of the pressure sensor is filled with and emptied of oil every revolution. This filling and emptying can result in a damping of the measurement output from the sensor. Further, due to the high rotation speed, inertial effects tend to throw the oil out of the sensing hole when the sensor rotates through the cavitation zone. To solve this issue, the cavity between the sensor and the oil film was filled with flexible silicone which, ideally, remains in the cavity through the entire revolution. This eliminates the need to fill the cavity with oil every revolution but adds further complication to the equipment assembly.
and preparation as any air bubbles or sticking of the silicone to the cavity walls can affect the shape of the pressure curve.

To verify the pressure sensors on the shaft, two pressure sensors were installed in the bearing housing and sealed to the bearing sleeve with a pressed o-ring coupling. The pressure measured in these sensors was lower than that measured in the shaft. This difference was believed to be caused by leakage at the o-ring seal between the sleeve and the bearing’s housing. However, the pressure sensors in the housing generally verified the function of the sensors on the shaft.

Temperature sensing throughout the system is accomplished using both thermocouples and inexpensive medical grade NTC thermistors. Both sensing methods have advantages and disadvantages and these are taken into account based on their application location. Thermocouples are used throughout the test rig’s monitoring and control system to monitor temperatures of oil and coolant water because of their robustness and the simplicity of mounting through pressure fittings. Thermistors are used in the shaft because several thermistors can be coupled to a common ground which reduces the number of channels through the slip ring. Also supporting this choice is that thermistors are not as sensitive to signal noise as thermocouples due to their much higher voltage level (0 to 10 V as opposed to μV for thermocouples). The same reasons justified the use of thermistors in the bearing sleeve where reducing the number and bulk of wires was crucial for proper assembly. Furthermore, their low cost eliminated the need of re-using sensors which simplified assembly of the test bearing sleeves.

Additional challenges were met in development of the software used to operate and collect data from the test rig. The large number of sensors operating at both high and low speeds required the development of a complex LabView program for data collection. Processing of all data was accomplished using a number of codes written using MatLab.

Dynamic Journal Bearing Test Rig

Experiments on the dynamic characteristics of polymer faced journal bearings were conducted at the Technical University of Denmark’s Solid Mechanics Laboratory. The test equipment used utilizes two counter-acting tilting pads to simplify the tilting pad journal bearing system. However, because these experiments were conducted with a static load which was much larger than the dynamic loading only one of the pads (the lower pad) carried the load. This
allowed for removal of the upper bearing pad and operation of the test rig with only the lower pad which simplified the experimental procedure and eliminated any potential issue that could be caused by variations in bearing clearance.

Unlike many dynamic testing machines which excite the bearing, this particular machine excites the shaft directly and thereby provides a more realistic excitation of the journal bearing. The bearing pads are outfitted with thermocouples to measure film temperature and static load is provided by a block and tackle to the beam which allows for loading up to 6 kN in the bearing or about 1.5 MPa mean pressure for the full sized bearing pad.

The bearing is fully flooded with a constant supply of oil at approximately 3 l/min. A suction line is used to return oil and leakage oil to the 40 liter tank. A diagram of the test rig is provided in Fig. 5.9. The dynamic system is characterized using two load cells to determine the static and dynamic load on the pad as well as a displacement sensor to determine the shaft’s motion. Dynamic loading was provided using a shaker which allowed for excitation loads of up to 300 N.

Dynamic characterization of the bearing pads is accomplished using a shaker which excites the shaft with a frequency range from 0 to 300 Hz. This frequency range allowed for analysis of both low frequency excitations common in large hydropower machines as well as higher frequency excitations caused by the electrical network. The load and displacement signals collected during the frequency burst are then analyzed as described in Paper E using the signal power spectral density. The stiffness and damping are easiest to compare using this method near to a system natural frequency where system
motion is largest, which in this case is the natural frequency of the beam.

**Full Scale Hydropower Machine**

Following construction of the new Porjus power station, the old station, Fig. 5.10, was mostly decommissioned from active use and turned into a museum, education, and research facility. The research facility in the power station consists of the U9 machine (Fig. 5.11) a 10 MW Kaplan turbine coupled to a Powerformer generator. This generator is unique in that it provides electricity directly to the electrical network at high voltage without the need for a transformer. The machine was installed as a testbed for research in the many facets of hydropower, including everything from the fluid dynamics in the penstock and around the turbine to the coupling between the electrical grid and the machine’s rotor dynamics. Most interesting for this work are the three different guide bearings which stabilize the machine’s vertical shaft and how these bearings affect and are affected by the machine’s dynamic motions during operation.

To investigate this coupling between bearing and rotor, a large number of sensors were installed in and around each of the guide bearings. Each of the bearing pads has 2 inductive displacement sensors and 2 thermocouples as well as a load cell in place of the tilting pin as displayed in Fig. 5.12. Four displacement sensors are mounted in each bearing housing to determine the shaft motion relative to the bearing housing. Four additional sensors are mounted...
from the concrete foundation at each bearing to determine the relative motion between the shaft and the surrounding structure.

All of these sensors are coupled to a data collection network which in turn is steered by a computer in the machine’s control room. A common trigger signal is used to synchronize the data sets collected for the entire machine. The large volume of data produced during experiments is saved in bulk and processed later using MatLab.
Figure 5.12: Sensor arrangement for a typical bearing tilting pad in the Porjus U9 machine.
Chapter 6

Experiments using tribo-meters

How can start and stop performance be improved?

Most of the phenomena involved in hydrodynamic journal bearings such as the buildup of pressure and thermal gradients are highly dependent on the bearing’s geometry and therefore require the full scale to properly test. However, prior to the buildup of a hydrodynamic film during startup, the shaft and bearing surfaces are in contact with each other. By using a small portion of this contact geometry with full scale shaft and bearing surface textures and contact pressure, this pre-start contact can be scaled. This is fortunate because the transition from stationary to sliding is also the operation range that most commonly leads to expensive bearing damage.

A reciprocating test rig, was initially used to simulate startup in sliding bearings while a block on disk uni-directional sliding arrangement was used for detailed study as described in Chapter 5. Two studies were conducted to investigate the effects of changes in bearing materials and lubricants.

Bearing materials

A host of materials are now available that can potentially eliminate one of the greatest weaknesses of current journal bearing technology: startup and shutdown. When a machine is started from rest or brought to a stop, the hydrodynamic film which supports the machine during operation is not present and the bearing and shaft surfaces contact each other. Traditionally, white metal material has been used on the sliding surface of bearings both to decrease friction at startup and to provide a sacrificial layer which can wear without damaging
the shaft. However, because the friction of white metal against steel is actually quite high, hydraulic jacking systems are used in most large machines to provide oil film pressure and separate the shaft surface from the bearing until surface speeds are high enough to produce adequate hydrodynamic film or during extended periods of slow speed operation. New materials have the potential to greatly reduce the break-away friction and possibly even eliminate the need for complex hydraulic jacking systems.

To determine the material characteristics most important at break-away in hydrodynamic bearings, several different composites of PTFE were tested in a small scale test. Performance of the PTFE materials varied greatly but all of them were superior to white metal material. The specific fillers used in the materials had significant impact on both the break-away friction shown in Fig. 6.1 and the variation of the break-away friction over the course of successive starts and stops displayed by Fig. 6.2. Generally, the glass filled PTFE composites had higher break-away friction and greater variation over the course of the tests than the other PTFE composites and virgin PTFE.

The reasons for these observations became clear when specimens were examined in a scanning electron microscope (SEM). PTFE composite materials with hard particles such as black glass and glass fiber were found to have iron deposits on their surfaces signifying that they had been wearing on the steel counter surface. SEM investigation (Fig. 6.3) showed that this wearing most
Figure 6.2: Variation in breakaway friction vs. contact pressure at 25°C. Error bars represent one standard deviation of three repeated tests for each point.

Figure 6.3: SEM image of black glass filled PTFE tested at 8 MPa and 25°C. White areas correspond to iron, glass particles are light gray, and the bulk material is dark gray.
likely occurred through a process of abrasion as piles of iron appear to have collected in front of and behind the sharp glass fibers. This abrasive effect is explained by the high hardness of the glass fibers which form sharp edges when they are broken or sheared. These sharp edges act as a cutting tool against the steel counter-surface preventing the buildup of an effective transfer film or the wearing down of the PTFE material to a smooth surface.

The other materials including bronze filled PTFE, carbon filled PTFE and pure PTFE provided very smooth worn surfaces with no sign of counter surface material. Of note is that bronze filled PTFE was the only material which had decreasing variability in its break-away friction levels with increasing temperature. This was most likely due to a softening of the bronze with increasing temperatures allowing for a smoother sliding surface to more rapidly build.

Further testing with a 72 hour extended stop found that the PTFE based materials had very little change in their break-away friction while Babbitt showed a large increase in break-away friction following the stop. This provided a clear demonstration of the improvements that polymer surfaces on bearings can provide during startup in lubricated conditions.

**Lubricants**

To develop the optimum lubricant for startup conditions, an initial round of three lubricants were investigated (G, O, and V) in both the reciprocating and uni-directional sliding test rigs. These initial studies highlighted the desirable lubricant characteristics at startup, namely that the lubricant should provide a low static friction followed by a steady sliding friction.

The dramatic differences between the three different lubricants is clear in Fig. 6.4, Fig. 6.5 and Fig. 6.6. In this case, the test started by pressing the bearing surface into the steel counter surface. Pressure was held for 1 minute after which the disk was accelerated to an extremely slow speed. The specimen holder was then loaded until it released, resulting in stick slip at the holder’s natural frequency. In the case of both lubricants ‘O’ and ‘V’, the release was followed by significant stick-slip. Lubricant ‘G’ provided a much lower stick friction followed by a consistent sliding friction and no observable stick-slip.

This initial work clarified the effect of lubricant formulation on break-away characteristics. To further develop the initial results, small batches of a number of lubricant formulations were prepared with varying concentrations of a variety of VI improving additives. The characteristics of these lubricants are provided in Table 6.1.
Figure 6.4: One cycle showing a start from stop, acceleration to steady speed and deceleration to stop. Test materials are white metal block sliding against a steel disk, lubricated with lubricant ‘V’.

Figure 6.5: One cycle showing a start from stop, acceleration to steady speed and deceleration to stop. Test materials are white metal block sliding against a steel disk, lubricated with lubricant ‘O’.
Figure 6.6: One cycle showing a start from stop, acceleration to steady speed and deceleration to stop. Test materials are white metal block sliding against a steel disk, lubricated with lubricant 'G'.

Table 6.1: Characteristics of tested lubricants

<table>
<thead>
<tr>
<th>Lubricant</th>
<th>Base Oil</th>
<th>Viscosity (mm²/s)</th>
<th>VI</th>
<th>Additive Type</th>
</tr>
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<tbody>
<tr>
<td></td>
<td></td>
<td>40 °C</td>
<td>100 °C</td>
<td></td>
</tr>
<tr>
<td>A</td>
<td>White Oil</td>
<td>14.9</td>
<td>3.86</td>
<td>163</td>
</tr>
<tr>
<td>B</td>
<td>White Oil</td>
<td>15.0</td>
<td>4.73</td>
<td>274</td>
</tr>
<tr>
<td>C</td>
<td>White Oil</td>
<td>14.7</td>
<td>5.35</td>
<td>364</td>
</tr>
<tr>
<td>D</td>
<td>White Oil</td>
<td>14.9</td>
<td>5.08</td>
<td>323</td>
</tr>
<tr>
<td>E</td>
<td>White Oil</td>
<td>14.9</td>
<td>5.67</td>
<td>394</td>
</tr>
<tr>
<td>F</td>
<td>White Oil</td>
<td>14.9</td>
<td>6.47</td>
<td>475</td>
</tr>
<tr>
<td>G</td>
<td>White Oil</td>
<td>15.3</td>
<td>7.87</td>
<td>581</td>
</tr>
<tr>
<td>O (SE15)</td>
<td>Synthetic Ester</td>
<td>15.5</td>
<td>4.46</td>
<td>215</td>
</tr>
<tr>
<td>V</td>
<td>Mineral Oil</td>
<td>14.5</td>
<td>3.26</td>
<td>83</td>
</tr>
<tr>
<td>Z</td>
<td>White Oil</td>
<td>14.5</td>
<td>8.16</td>
<td>640</td>
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</table>
A full test was conducted with all of the lubricants with the aim of determining which lubricant (if any) would be interesting to formulate in a larger batch for use in the journal bearing test rig. These tests were conducted using the reciprocating test rig which allowed for precise control of the temperature in the contact as well as a steady and repeatable result. An additional reason for selecting the reciprocating test rig was that it required very little time to change conditions during testing. This led to a fully randomized test at two contact pressures with 5 repeats. The results of the full study are shown in Fig. 6.7.

From the results of the lubricant study in the reciprocating test rig, it was clear that certain lubricant formulations provided advantageous performance for the journal bearing application. As in the initial work with the unidirectional sliding rig clear trends were observed regarding the lubricants’ friction characteristics. The lubricants either provided a low break-away friction which remained constant during the test, provided a slightly higher break-away friction which increased initially then remained constant, or, as in the case of lubricant 'V', the lubricant provided a high break-away friction that increased during the course of the test. Clearly a low and steady break-away friction is most desirable for sliding bearing applications.

While 'G' and 'A' performed by far the best of all of the formulations, they
also had drawbacks. The additive in lubricant ‘A’ did not provide the high VI desired for the full film lubrication regime while the stability of lubricant ‘G’ was uncertain. It was therefore decided to blend two more lubricants using similar additives to those in ‘A’ and ‘G’ except in a formulation which was more stable. These lubricants, ‘H’ and ‘Z’, were found to have better friction characteristics than the majority of the lubricants, albeit not as good as ‘A’ and ‘G’. Due to the advantageous break-away friction characteristics and its exceptionally high VI, lubricant ‘Z’ was selected for further testing in the journal bearing test rig.
Chapter 7

Journal bearing experiments

What can be done to improve steady state performance?

Work with the full scale journal bearing test rig covered primarily lubricant development. While the test rig is constructed to accommodate a number of different bearing types and geometries, full utilization of this potential was not realized for a number of reasons. However, the extensive lubricant research program pushed to the limits of lubricant technology and provided answers to many questions about the practicality and application of extremely high VI lubricants. Results from these studies are provided below. Results from the less extensive polymer faced bearing study can be found in [135].

New Synthetic Lubricants

Initial studies were conducted with a number of new synthetic lubricants with high viscosity index (VI) finding significant performance differences between standard VI and high VI lubricants. The high VI lubricants provided generally lower power loss compared to the standard lubricants. Reduced power loss resulted in significantly lower operating temperatures in the bearing and lubrication system. Characteristics of the new lubricants together with two standard mineral oil based turbine oils used for comparison are detailed in Table 7.1.

Study of the lubricants accomplished in Paper A provided justification for improving lubricants with high doses of VI improvers. Studies of the breakaway characteristics of the lubricants opened up potential for even higher levels of VI improvers in lubricants. It was thus decided to determine the limits of lubricant formulations combining low viscosity with extremely high VI. This
Table 7.1: Characteristics of the tested lubricants

<table>
<thead>
<tr>
<th>Grade</th>
<th>Abbreviation</th>
<th>Base Oil</th>
<th>Viscosity, (40, ^\circ\text{C})</th>
<th>Viscosity, (100, ^\circ\text{C})</th>
<th>VI</th>
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</thead>
<tbody>
<tr>
<td>ISO VG68</td>
<td>VG68</td>
<td>Mineral</td>
<td>67.3</td>
<td>8.79</td>
<td>103</td>
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<tr>
<td>ISO VG32</td>
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<td>Mineral</td>
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<td>SE32</td>
<td>Synthetic Ester</td>
<td>32.1</td>
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<td>ISO VG22</td>
<td>SE22</td>
<td>Synthetic Ester</td>
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<td>ISO VG22</td>
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<td>SE15</td>
<td>Synthetic Ester</td>
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<tr>
<td>ISO VG15</td>
<td>HV15</td>
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</tr>
<tr>
<td>ISO VG28</td>
<td>HV28</td>
<td>Synthetic Ester</td>
<td>28.0</td>
<td>13.5</td>
<td>519</td>
</tr>
</tbody>
</table>

led to the development of additive and base oil combinations which allowed for the blending of HV15 and HV28. These two extreme lubricants are very unique in that their VI is at the limit of the VI scale and the specific VI improving additives used to formulate them were demonstrated to provide lower friction at break-away. Such a high VI was attained by combining a very low viscosity base oil with a high concentration of VI improving additives, more detail of the additives is given in [140, 141].

The synthetic lubricants generally demonstrated equivalent film thickness to mineral based lubricants with double the base viscosity grade. For example, SE32 provided similar film thickness to VG68 in all but the slowest speed and highest load points. Likewise, SE15 would make an acceptable replacement for VG32.

While HV15 did not provide the thick oil films that the other lubricants provided, it did provide adequate film thickness (see Fig. 7.1 and Fig. 7.2) with up to a 40% reduction in power losses (see Fig. 7.3 and Fig. 7.4). This effect is well described by the minimum viscosity measured in the lubricants where it is seen that the viscosity of the extreme VI lubricants in the highest temperature area of the bearing remained much lower than that with VG68 at low loads, Fig. 7.6.

HV28 provided matching eccentricity to VG68 at all loads and speeds with an admirable 5% to 15% reduction in power loss. At high loads, HV15 and HV28 provide a more viscous lubricant film than VG68 (when calculated from temperature), however it seems that in the case of HV15 significant temporary
Figure 7.1: Bearing eccentricity at 1 MPa.

Figure 7.2: Bearing eccentricity at 2 MPa.
Figure 7.3: Power loss at 1 MPa mean load as a percentage of VG68.

Figure 7.4: Power loss at 2 MPa mean load as a percentage of VG68.
Figure 7.5: Minimum viscosity in the bearing at 1 MPa.

Figure 7.6: Minimum viscosity in the bearing at 2 MPa.
shear thinning occurs which in reality leads to a lower viscosity than calculated at high speed.

Due to changes in the test rig, it was not possible to compare HV15 to VG32, however from the comparison of HV28 and VG68 it is clear that the extreme VI lubricants could outperform traditional lubricants of at least two times their viscosity grade. Considering the performance comparison of HV15 with VG68, it can be inferred that HV15 would be a good candidate to replace VG32 or even VG46 in some cases.

Machine Operation

The study of new synthetic lubricants highlighted the operational differences between high VI lubricants and standard lubricants. However, the methodology for application of the new lubricants in industry remained uncertain. Clearly, new lubricants offer potential for improved machine performance, but how should one select the best lubricant for a specific application? Further, how should a machine’s operation be changed to take advantage of the new lubricant? These two questions were explored by performing tests in which the goal was to match performance of two lubricants. Minimum viscosity of all lubricants was calculated from the maximum temperature data. The specific temperature required for VG68 mineral oil was then calculated at the specified viscosity. The test machine was consequently operated at equivalent load and speed while varying the temperature of the lubricant to yield equivalent minimum viscosity to that of the other lubricants.

Eccentricity and power loss differences for this test program are provided in Paper B. The power loss differences between the lubricants were generally smaller than expected. However, this discrepancy can be explained by the decrease in eccentricity observed at higher speeds. Equivalent eccentricity should have led to greater savings in power loss for the synthetic lubricants compared to the mineral oil. When taken in their entirety, the results demonstrate the applicability and the limits of simply matching viscosity at the maximum temperature location in the bearing.

Following the initial experiments conducted by heating up VG68, another set of experiments was conducted during which the test lubricant was instead cooled in an effort to match eccentricity with VG68. The power loss and eccentricity results are provided in Fig. 7.4 and Fig. 7.2 (‘SE32 to match VG68’ and ‘SV22 to match VG68’). While it was possible to match VG68 in almost all of the test cases with SE32, the real weakness for both SE32 and SV22 was
at low speed and high load. Even by cooling the bearing inlet oil to a frigid 15°C with 8°C cooling water, the minimum viscosity was still not adequately high enough to provide an equivalent oil film to that provided by VG68.

From the theory of matching viscosity and maintaining equivalent machine safety, a simple formula for prediction of changes in power loss resulting from changing the VI or specific heat, $\rho C_p$, of a lubricant was developed. The method uses operating temperatures of the original lubricant to predict operating temperatures with a different lubricant. The temperature rise between the oil bath and the maximum temperature is assumed to be equivalent for the new and the original lubricant and so the oil bath temperature for the new lubricant can thus be determined. The relative change in power loss is then found from a simple comparison of the oil bath lubricant viscosity using Eq. 7.1, where $A$ is a description of the percent area of the bearing which does not carry load.

$$\Delta \text{losses} = A \times \left[ 1 - \frac{\mu_{\text{new}}}{\mu_{\text{old}}} \right]$$

(7.1)

Most lubricant suppliers do not include information on the thermal characteristics of their lubricants, however if $\rho C_p$ is known, a more accurate power loss calculation can be made. By assuming that the power used to develop load carrying capacity in the bearing is constant regardless of the lubricant and that this power usage results only in heating of the lubricant, Eq. 7.2 can be used to provide a prediction of the new lubricant’s temperature rise in the contact. Using this method allows for a higher oil bath temperature which in turn allows for lower viscosity in the oil bath and thereby further decreased power losses.

$$\Delta T_{\text{new}} = \frac{\rho_{\text{old}} C_p \Delta T_{\text{old}}}{\rho_{\text{new}} C_p}$$

(7.2)

Changes in power loss can readily be given a monetary value that can provide tangible justification for or against lubricant changes in specific applications where it is difficult to define or place a value on the benefits of reduced environmental risk.
Chapter 8

Dynamic journal bearing experiments

How do polymer faced bearing pads perform in terms of dynamics?

An understanding of the dynamic response of polymer faced journal bearings is necessary to 'close the loop' in their development. After a number of false starts, it was decided to conduct this dynamic testing at the Technical University of Denmark in Lyngby, Denmark. A number of test pads were prepared for these tests (detailed in Paper E) and a test plan was developed. The test bearing pads were designed to allow for evaluation of the effects of varying material characteristics of the polymer layer, varying the contact area and thereby mean pressure in the bearing, and varying the material characteristics of the pivot. Further, their geometry and the method of using screws to connect the polymer faces to the steel backing was designed so that the parts could be interchangeable and easily modeled.

In operation at the heaviest load and highest speed, maximum pad temperatures were slightly higher for the smaller surface area pads than for the full surface area pads. Likewise, as shown in Table E.4 the pads entirely made of PEEK had slightly higher temperatures than those with a steel backing. The increased thickness of insulating material (PEEK) in the entirely PEEK pads is believed to cause the entirely PEEK pads to operate with higher temperatures than the steel backed pads. The steel backed PEEK and PTFE pads appeared to behave similarly to each other with PTFE being slightly warmer than PEEK at its trailing edge. This difference is believed to be caused by the differences in pad geometries and deformation under load.
Investigation of the stiffness for the varying pad configurations at low frequency (0 to 50 Hz) provided an understanding of how changes in journal bearing pads could affect hydropower machines while testing to higher frequency (0 to 300 Hz) allowed for investigation of the pads’ damping characteristics. At low frequency, the pads constructed entirely of PEEK provided lower stiffness than those with a PEEK facing and steel backing as seen in Fig. 8.1. The PTFE faced pad with a steel backing performed similarly to the entirely PEEK pad at low excitation frequencies. The large error at 42 Hz in Fig. 8.1 is due to the interference from the operating frequency of the rotor.

Testing using a higher frequency excitation range allowed for characterization of both the stiffness and damping characteristics. Trends at higher frequency were similar to those at lower frequency for stiffness as seen in Fig. 8.2 with the PTFE pad being slightly stiffer than the entirely PEEK pad. However, pads made entirely from PEEK provided equivalent to greater damping than the steel backed pads (Fig. 8.3). This increased damping is believed to be the result of the combination of the PEEK material’s internal damping and deformation in the Hertzian contact at the pad’s pivot. Pivot deformation for the steel backed pads was much smaller due to the differences in hardness between
Figure 8.2: Stiffness for high frequency range at 2500 rpm and 2400 N load showing the mean values plus and minus one standard deviation from 3 tests.

Figure 8.3: Damping for high frequency range at 2500 rpm and 2400 N load showing the mean values plus and minus one standard deviation from 3 tests.
PEEK and steel.

Lower stiffness with higher damping could be useful in applications such as hydropower where loads are generally light but highly dynamic. However, application of these types of polymer faced pads requires a reliable method of attaching the polymer facing to the backing. Using machine screws to attach the polymer facing to the steel backing as in these experiments functioned admirably in a laboratory environment but it is uncertain how reliable this method would be in an industrial setting. Further, machining soft polymer such as PTFE to the tight tolerances necessary in journal bearing applications poses many practical challenges in and of itself. The use of bearing pads entirely constructed of a harder polymer such as PEEK can be a viable alternative that eliminates both the need of a reliable bonding method and the manufacturing issues.
Chapter 9

Full scale experiments

How does the full scale machine behave?

Experimental work with the Porjus U9 machine provided a large amount of experimental data on the dynamic motions which a hydropower machine undergoes during both steady state and transient operation. These experiments also highlighted a number of effects that are in many ways unique to vertical machines. While these phenomena are discussed and hypothesized in the literature and by professionals, very few experimental results from vertical machines have made it into published media.

During analysis, the Porjus U9 results were divided into three groups to provide a complete picture of the machine via the bearings. In steady state conditions, the orbits and loads in the bearings were analyzed to compare the bearing response with multi-physics models while transient conditions were investigated to examine the extreme cases of machine operation and to better understand just how machines respond to extreme load scenarios. Finally the dynamic characteristics of the bearings and their surrounding structure were analyzed to provide insight into the bearings’ response to transient conditions.

While the majority of the sensors in the machine functioned properly during all measurements there were a few sensors which became problematic. Three of the load cells in the turbine bearing appear to have developed a loose contact and so they sometimes provided false signals. Other sensors such as those between the foundation and the brake disc came into contact with the brake disc and succumbed to tribological processes.
Figure 9.1: Orbit of the shaft in Bearing 2's housing. Dimensions are in mm and 40 complete orbits are shown as well as the average orbit and the locations of the bearing pads.

**Steady State**

Measurements during steady state operation were conducted at a range of loads from 40% to 95% with both 'cool' and 'warm' lubricating oil. Measurements with 'cool' lubricant were conducted while the machine was warming up and so bearing inlet oil did not have constant temperature for all load states. Measurements with 'warm' lubricant were conducted after the machine had been in operation and temperatures in the bearing oil baths had reached equilibrium. Having measurements at two different lubricant bath temperatures allowed for the comparison of the bearing’s characteristics with different oil viscosities.

Orbits, loads and temperatures from the bearings at steady state are shown in Fig. 9.1, Fig. 9.2 and Table 9.1.

Results in the steady state demonstrate the strong correlation between the temperature of the individual bearing pads and the static load carried by each pad after start (see Fig. 9.3). The pads with greater load tend to have significantly higher temperature. The dynamic load resulting from imbalance of the rotor is quite low compared to the static load as seen in Fig. 9.2.
Figure 9.2: Dynamic portion of the bearing load in Bearing 2. Dimensions are in Newtons and 40 complete rotations are shown as well as an average rotation and the locations of the bearing pads.

Table 9.1: Bearing and oil bath temperature in Bearing 2.

<table>
<thead>
<tr>
<th>Location</th>
<th>Outlet °C</th>
<th>Inlet °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pad 1</td>
<td>54.0</td>
<td>40.1</td>
</tr>
<tr>
<td>Pad 2</td>
<td>51.4</td>
<td>39.3</td>
</tr>
<tr>
<td>Pad 3</td>
<td>59.8</td>
<td>48.9</td>
</tr>
<tr>
<td>Pad 4</td>
<td>48.8</td>
<td>36.6</td>
</tr>
<tr>
<td>Pad 5</td>
<td>54.1</td>
<td>40.2</td>
</tr>
<tr>
<td>Pad 6</td>
<td>57.4</td>
<td>42.0</td>
</tr>
<tr>
<td>Oil bath</td>
<td>38.2</td>
<td>28.4</td>
</tr>
</tbody>
</table>
Figure 9.3: Static load on each pad during start-up for Bearing 2.

Start

Startup of the machine demonstrated the fraction of the total bearing load that is due to preload. Because the machine accelerates through a wide range of speeds during startup, the bearing static loads resulting from pad preload could be clearly observed as seen in Fig. 9.3. Prior to start, the bearing pads have very little load, while the load steadily increases with increasing rotational speed. There is then a shift that occurs when the rotor magnetizes and the machine is connected to the grid. Load continues to shift slightly while the output of the machine increases up to the best operating point at 80% power. While startups were conducted both with ‘cold’ bearings and ‘warm’ bearings, very little difference was observed in the bearing behaviour for the different start conditions.

Stop

While machine startup was limited to a specific procedure, stopping the machine was a different procedure entirely. The machine was stopped via a ‘standard’ stop procedure as well as a ‘quick/emergency’ stop procedure from several different operating points. Furthermore, a number of unplanned stops
occurred during testing when the machine’s built in alarm system detected potentially dangerous levels of vibration over an extended period of time. This allowed for a number of cases to study. In general however there was very little difference between quick stop and a slow stop in terms of the response measured in the bearings.

**Load Change**

The change in load on the bearings over the course of a machine load change can be seen in Fig. 9.4. Interestingly, there is very little effect on the bearing static load as the static load is primarily the result of bearing preload and the associated hydrodynamic pressure. However, there is a slight change in the dynamic load on the bearing pads through the load change which is most likely caused by the increased force in the generator or flow in the turbine. This effect can clearly be seen in the case of load change from 40% to 80% power in (Fig. 9.5).
Figure 9.5: Dynamic load on each pad in bearing 1 during load change from 40% to 80%.

**Grid Disconnect**

Allowing the machine to accelerate freely by disconnecting the generator from the electrical grid while leaving the guide vanes open was the most extreme test accomplished on the Porjus U9 machine. The machine was allowed to accelerate for 1 second from 25% and 50% load in these experiments. As the machine began accelerating, the bearing orbit decreased due to the increased bearing pad load. While the machine appeared to be only slightly affected by disconnection of the generator, the effect of closing the guide vanes led to a very large low frequency orbit and load in both the lower generator and turbine bearing as seen in Fig. 9.6 and Fig. 9.7. The upper generator bearing appeared nearly unaffected in comparison to the lower two bearings. The low frequency caused by the closing of the guide vanes was only slight from 25% load. However, this difference is reasonable because following grid disconnection, the machine ran in 'safe mode' at 17% load. The difference in load between 50% and safe mode being much larger than that from 25% to safe mode accounts for the large difference in the results.
Figure 9.6: Eccentric displacement of the shaft in all bearings during grid disconnect.

Figure 9.7: Total bearing load in all bearings during grid disconnect.
Dynamic Characteristics

Estimation of the dynamic coefficients for each of the three bearings was accomplished using the direct stiffness and damping method described by Näs selqvist et al. [109]. This method assumes that the cross-coupled terms are very small compared to the direct stiffness and damping and that the fluid inertia terms are also very small. The equations of motion for the system are converted into polar coordinates to account for each pad’s contribution to the total load. The derivative of the displacement is then determined for each data point directly using a finite difference scheme.

The displacements, derivatives, and forces are then assembled in matrix form:

\[
\begin{bmatrix}
    f_x \\
    f_y
\end{bmatrix}
= \begin{bmatrix}
    k & 0 \\
    0 & k
\end{bmatrix}
\begin{bmatrix} u_x \cos(\alpha) \\
    u_y \sin(\alpha) \end{bmatrix}
+ \begin{bmatrix} c & 0 \\
    0 & c
\end{bmatrix}
\begin{bmatrix} \dot{u_x} \cos(\alpha - \theta) \\
    \dot{u_y} \sin(\alpha - \theta) \end{bmatrix}
\tag{9.1}
\]

Where \( \alpha \) is the angle of each individual pad from the x-axis and \( \theta \) is the phase shift between the shaft displacement and force response. This matrix is then converted to polar coordinates such that:

\[
\begin{align*}
    f &= \sqrt{f_x^2 + f_y^2} \\
    u &= \sqrt{u_x^2 + u_y^2} \\
    \dot{u} &= \sqrt{\dot{u}_x^2 + \dot{u}_y^2}
\end{align*}
\tag{9.2}
\]

Which leads to the simplification of Eq. (9.1) to:

\[
f^2 = u^2 k^2 + \dot{u}^2 c^2 + 2 u \dot{u} k c \cos(\theta)
\tag{9.3}
\]

The above equation can then be assembled as a matrix composed of the data from three successive revolutions to provide a running estimate of \( k \) and \( c \) around the bearing:

\[
\begin{bmatrix}
    f_p^2 \\
    f_{p+1}^2 \\
    f_{p+2}^2
\end{bmatrix}
= \begin{bmatrix}
    u_p^2 & \dot{u}_p^2 & u_p \dot{u}_p \cos(\theta_p) \\
    u_{p+1}^2 & \dot{u}_{p+1}^2 & u_{p+1} \dot{u}_{p+1} \cos(\theta_{p+1}) \\
    u_{p+2}^2 & \dot{u}_{p+2}^2 & u_{p+2} \dot{u}_{p+2} \cos(\theta_{p+2})
\end{bmatrix}
+ \begin{bmatrix} k^2 \\
    c^2 \\
    2kc
\end{bmatrix}
\tag{9.4}
\]

Solution of Eq. (9.4) for \( k \) and \( c \) gives the bearing stiffness and damping at each measurement point around the bearing. In the case of these experiments, 250 data samples were taken for each revolution, providing an angular
Table 9.2: Mean stiffness and damping of the bearings and structure before warming up.

<table>
<thead>
<tr>
<th></th>
<th>Oil Film</th>
<th>Bearing</th>
<th>Foundation</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Stiffness</strong></td>
<td>(10^6) N/m</td>
<td>(10^6) N/m</td>
<td>(10^6) N/m</td>
</tr>
<tr>
<td>Bearing 1</td>
<td>660 ±100</td>
<td>48 ±7</td>
<td>15 ±1</td>
</tr>
<tr>
<td>Bearing 2</td>
<td>630 ±60</td>
<td>163 ±11</td>
<td>5.9 ±0.2</td>
</tr>
<tr>
<td>Bearing 3</td>
<td>189 ±14</td>
<td>–</td>
<td>12 ±2</td>
</tr>
<tr>
<td><strong>Damping</strong></td>
<td>(10^6) Ns/m</td>
<td>(10^6) Ns/m</td>
<td>(10^6) Ns/m</td>
</tr>
<tr>
<td>Bearing 1</td>
<td>7.7 ±1.4</td>
<td>6.8 ±1.3</td>
<td>0.17 ±0.02</td>
</tr>
<tr>
<td>Bearing 2</td>
<td>8.5 ±1.0</td>
<td>3.3 ±0.3</td>
<td>0.67 ±0.03</td>
</tr>
<tr>
<td>Bearing 3</td>
<td>3.2 ±0.2</td>
<td>–</td>
<td>2.0 ±0.4</td>
</tr>
</tbody>
</table>

resolution of approximately 1.5° when the machine is running at its standard operating speed.

Although this method required significant smoothing of the data to eliminate noise, it was still possible to observe changes in the stiffness of the bearing and structure as the machine warmed up to steady state operation. Stiffness and damping was analyzed across the oil film, across the bearing and across the bearing structure (from the concrete foundation) both when the machine was ‘cold’ and after the machine had reached operating temperature. Temperature in the bearing was approximately 3°C colder in the inlets and 1°C colder in the outlets of the bearing pads for the ‘cold’ case. These small temperature differences seem to mask the real difference in temperature and thermal expansion of the different structural components because the two cases had differences in stiffness and damping of up to 20% for their components.

Interestingly, the bearings seem to react differently to the machine’s warming up cycle. There is nearly no net change in the stiffness and damping for all of the bearing systems when the entire bearing system including the structure is included. However, the stiffness and damping of the individual components change significantly over the warm up cycle. In the case of the lower generator bearing and turbine bearing, the stiffness and damping provided by the oil film are approximately 20% higher when the machine is cold compared to after it is warmed up. No significant difference was noted for the upper generator bearing.

These stiffness calculations highlight the issue that the oil film is stiffer
Table 9.3: Mean stiffness and damping of the bearings and structure after warming up.

<table>
<thead>
<tr>
<th></th>
<th>Oil Film</th>
<th>Bearing</th>
<th>Foundation</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Stiffness</strong></td>
<td>$10^6$ N/m</td>
<td>$10^6$ N/m</td>
<td>$10^6$ N/m</td>
</tr>
<tr>
<td>Bearing 1</td>
<td>$670 \pm 85$</td>
<td>$59 \pm 17$</td>
<td>$14.6 \pm 1$</td>
</tr>
<tr>
<td>Bearing 2</td>
<td>$530 \pm 30$</td>
<td>$425 \pm 90$</td>
<td>$6.3 \pm 0.3$</td>
</tr>
<tr>
<td>Bearing 3</td>
<td>$161 \pm 12$</td>
<td>–</td>
<td>$11 \pm 2$</td>
</tr>
<tr>
<td><strong>Damping</strong></td>
<td>$10^6$ Ns/m</td>
<td>$10^6$ Ns/m</td>
<td>$10^6$ Ns/m</td>
</tr>
<tr>
<td>Bearing 1</td>
<td>$8.1 \pm 1.3$</td>
<td>$7.2 \pm 2.6$</td>
<td>$0.15 \pm 0.01$</td>
</tr>
<tr>
<td>Bearing 2</td>
<td>$7.0 \pm 0.6$</td>
<td>$3.6 \pm 0.8$</td>
<td>$0.77 \pm 0.1$</td>
</tr>
<tr>
<td>Bearing 3</td>
<td>$2.6 \pm 0.2$</td>
<td>–</td>
<td>$1.6 \pm 0.4$</td>
</tr>
</tbody>
</table>

than the bearing in some cases and softer than the bearing in other cases. The bearing and the film in all cases provided greater stiffness and damping than their foundations. The oil film also seems to provide greater damping than the other components in some cases and less damping in other bearings. Regardless, the bearings appear to orbit within the foundation at least as much as the shaft orbits within the bearings. More effective damping of these orbits may be possible by increasing the stiffness in the foundations while increasing the assembly clearance in the bearing to reduce the bearing’s motion within the foundation.
Chapter 10

Conclusions

What are the most important findings of this research?

The work of this thesis began with study of a hydrodynamic journal bearing in a test rig. These test results then led to development of the small scale test arrangement using the block on plate and block on disk setups to study characteristics of the hydrodynamic journal bearing contact at startup. Results from the small scale studies then led back to the journal bearing test rig both in the laboratory at Luleå University of Technology and the laboratory at the Technical University of Denmark. Lessons learned in the laboratory allowed for an efficient and targeted study of the full scale machine in the Porjus power station. Finally, the results of study with the full scale machine provided valuable feedback for the interpretation and application of the earlier work.

This holistic approach has led to the following conclusions:

- High viscosity index synthetic lubricants can greatly reduce power loss in journal bearings while maintaining safe machine operation.
- Select additives can both increase viscosity index to extremely high levels and reduce friction at startup.
- Startup friction can further be reduced through the use of select polymer bearing materials.
- Fabricating bearing tilting pads entirely of a polymer such as PEEK can lead to an increase in damping with a decrease in stiffness and an increase in oil temperature.
- Full scale machine measurements demonstrate that the longer term transients in hydropower machines can significantly impact bearing performance.
Taken as a whole, these conclusions provide a basis for application of new materials, bearing pad designs and new lubricants to enhance performance of large journal bearings. Several potential synergies between these conclusions are also worth highlighting.

Changing to a high VI lubricant with low base viscosity leads to decreased oil temperatures due to decreased heat production (in the form of power loss). Using an entirely polymer bearing leads to increased oil temperatures due to pad insulation. These two effects may cancel out each other, leading to a net decrease in heat and power loss for the bearing with little change in oil film temperatures.

Lower heat production in the bearings can decrease the magnitude of thermal transient effects in the machine resulting in less variation and greater predictability of machine geometry and bearing characteristics through start and stop cycles.

Reducing startup friction and using more wear resistant materials such as polymer faced pads can reduce the risks associated with machine startup and the thermal transients in the machine, thereby improving reliability.

Improved reliability and robustness will allow large journal bearings and electrical power generating machines to continue to operate effectively regardless of society’s changing demands.
Chapter 11

Looking forward

What questions remain and what paths can future research follow?

In many ways this work has succeeded in the tasks it set out to accomplish, however the real work of applying the concepts presented herein lies ahead.

In terms of machine operation there is a lot of low hanging fruit that could be easily picked by making small adjustments to the way machines are operated. These small adjustments such as regulating lubricant temperature in a more thoughtful way or allowing higher maximum bearing temperatures could result in meaningful power savings.

Further improvement in machine performance can be provided through the application of new lubricants in real machines. Such a change could decrease power loss and improve machine safety at startup. However, fluid formulation and chemical compatibility must be checked prior to usage to avoid any negative consequences for packings and seals. Before these tests can be accomplished, a final lubricant product must be developed.

On the topic of lubricants, the long term reliability of new extremely high VI lubricants must be investigated for them to be applied to hydropower machines. Additionally, a better method for describing a lubricant’s temperature-viscosity relationship is needed as the current viscosity index (VI) scale (which was designed as a scale of 0 to 100) loses its relevance and becomes unstable at high VI levels. This new VI scale should have physical units as opposed to the current system with non-dimensional values based on the comparison between a ‘bad’ and ‘good’ oil.

Changes to bearing design are not as straightforward. Here, the primary limiting factor seems to be the designers’ knowledge of the potential of modern materials and design tools. Bearing design is all too often left as an af-
terthought to fulfill the rotor dynamicist’s requirements. Lack of understanding of the bearing’s function and optimization then leads to overly conservative designs based on rudimentary calculations. While this design method has been shown to work acceptably in most cases, it does not allow for engineering optimization. It is here that new manufacturing techniques and new materials in combination with modern design and simulation tools have the potential to provide the greatest improvements to hydrodynamic journal bearings. For example, a bearing pad could be designed using advanced models with varying material characteristics across its surface. Manufacturing of such a bearing could then be accomplished by blending materials [17] using 3D printing techniques. Such a method could potentially provide bearing pads that provide an optimum balance of load carrying capacity and dynamic characteristics with reduced power loss.

And of course, these designs must be validated first with laboratory experiments and then tested in the field...
Paper A

Synthetic lubricants in hydrodynamic journal bearings: experimental results
A.1 Abstract

Synthetic lubricants and additives have seen many major improvements in recent years. However, very little is known about the performance peculiarities of these new lubricants in actual machines. To fill this gap, a new full-scale hydrodynamic journal bearing test rig has been constructed to evaluate the behavior of conventional and new bearing designs, synthetic lubricants and variations in operating parameters. This test rig’s bearing has diameter 180 mm with measuring capabilities including continuous film thickness and film pressure as well as temperature. The new machine was used to compare a number of synthetic lubricants to mineral based lubricants, finding that performance of the synthetic lubricants was superior to their mineral-based counterparts of much higher viscosity grade. These tests showed that high viscosity index (VI) synthetic lubricants had higher viscosity in the region of maximum pressure and lower viscosity elsewhere in the bearing than similar mineral based lubricants. This reduction in viscosity in low pressure zones was found to produce a measurable reduction in friction and power loss in the bearing system. This paper provides comparative performance results of several formulations of current and future turbine oils from measurements of losses, oil film thickness and temperature under a range of operating parameters. Lubricants tested include ISO VG68 and VG32 mineral based turbine oils (VG68 and VG32), ISO VG32 synthetic ester based oil (SE32), two ISO VG22 synthetic ester based oils (SE22 and SV22) and ISO VG15 synthetic ester based
It was found that SE32 and VG68 provided similar performance at lower speeds while SV22 provided similar performance to VG68 at the highest speed. Likewise, SE22 and SV22 provided similar performance to VG32 at low speeds while SE15 provided similar performance to VG32 at medium to higher speeds. Generally the results demonstrate the potential for replacing mineral based lubricants with high performance synthetic lubricants of significantly lower viscosity grade without sacrificing bearing safety.

A.2 Introduction

Ever increasing demands on power producers to meet new, stiffer, environmental regulations, regarding hazardous materials, and rapidly changing operational requirements, caused by the recent buildup of renewable and variable power sources have begun to take their toll on, the generally old, equipment in power plants. Many of these plants are or soon will be going through major overhauls. Furthermore, while power plant efficiency is already exceptional, Gagnon [56], very little has been done to further improve this already high level of efficiency. In this regard, the bearings stand out as one source of losses in the plant with potential for significant improvements from a relatively small investment. One proposed way to reduce these losses while maintaining operating characteristics is to change lubricating oil to a higher viscosity index (VI) lubricant with a lower viscosity base oil. Further, many newer lubricants are environmentally adapted (EAL), thus allowing power plants to decrease their environmental footprint while potentially increasing power output. This topic was explored with several different lubricants in the early 90’s by Boehringar and Neff [11] finding significant improvement in machine performance after changing to a di-ester based lubricant. Later, Swanson et al. [144] found that the thermal performance of mineral-based oils could be improved through the use of VI improvers to match that of synthetic lubricants. A more focused study by Dmochowski and Webster [34] investigated the case of poly-alpha-olefin based lubricants finding that lubricant with high VI provided performance equivalent to higher viscosity lubricants with lower VI ratings. A similar study was performed using by Vijayaraghavan and Brewe [158] finding that changes in journal bearing performance could be realized by adjusting viscosity-temperature-pressure relationships.

Comparison between mineral based lubricants of varying grade accomplished by Ferguson et al. [48] for large tilting pad thrust bearings, Glavatskikh and Decamillo[61] for smaller tilting pad thrust bearings, and Brockwell et al.
A.2. INTRODUCTION

[15] for tilting pad journal bearings found that ISO VG32 lubricants resulted in reduced temperatures and power loss as well as reduced oil film thickness when compared to higher viscosity grade lubricants such as ISO VG46 and ISO VG68.

Comparisons of synthetic lubricants to mineral-based lubricants have shown that a lower viscosity synthetic ester can provide performance improvements over standard mineral based lubricants. In the early 1980’s work with synthetic lubricants by New and Schmaus [110] found significant reduction in power losses for thrust bearings when mineral oil (VG32) was replaced with a much lower viscosity grade di-ester (VG10) lubricant. Glavatskih and Larson [62] compared performance of ISO VG68 mineral-base and ISO VG46 synthetic-base lubricants and Dmochowski and Webster [34] compared ISO VG46 mineral-base and ISO VG32 synthetic-base lubricants. These later studies found similar results to those of New and Schmaus but the tested lubricants had smaller differences in their base viscosity grades than those earlier studied.

Results from a more recent study by McCarthy et al. [97] using synthetic ester-based EALs and a Babbitt faced plain journal bearing are in line with and expand upon the earlier work, demonstrating the improved performance provided by synthetic lubricants as well as the potential effects of VI improvers. These tests showed that EALs, because of their inherent higher VI, provide higher viscosity in the region of lowest film thickness and lower viscosity in the rest of the bearing than standard mineral-based lubricants. This reduction in viscosity throughout the bearing was found to lead to a measurable reduction in power losses in the bearing system while maintaining desirable film thickness.

The advent of new forms of polymer-based viscosity index improvers, Stohr et al. [140], has allowed for production of lubricants with exceptionally high viscosity index and acceptable shear stability. A number of potential new lubricants were studied by Kuznetsov and Glavatskih [85] using a thermoelastohydrodynamic numerical model during the development of an improved journal bearing solution. This study found that maximizing VI with reductions in lubricant base viscosity led to the greatest improvements in bearing characteristics. While guidance from [85] was used in the development process for the lubricants used in the present study, many uncertainties exist regarding the function of the VI improvers in combination with the specific lubricant chemistries in actual applications.

The current study expands on the earlier efforts by examining synthetic ester based lubricants with high VI for potential replacement of much higher ISO grade mineral oil based lubricants. Specifically, an ISO VG32 synthetic
ester is compared to an ISO VG68 mineral oil, and an ISO VG32 mineral oil is compared to two ISO VG22 synthetic esters and an ISO VG15 synthetic ester.

**A.3 Equipment**

Testing carried out in this study utilizes the full scale journal bearing test machine shown in Figure A.1. This test equipment features a 180 mm diameter shaft coupled to a motor capable of providing journal speeds up to 6000 RPM (56 m/s surface speed). The maximum test bearing load is 140 kN equating to 10 MPa for a 0.4 L/D bearing. Loading is accomplished through the use of an air cushion and manual pressure regulator. Load measurement is accomplished with four load cells.

Instrumentation installed onboard the shaft allows for measurement of the film thickness with improved accuracy at thinner films. Additionally, onboard pressure transducers provide continuous film pressure at bearing centerline, 10 mm from the centerline, and 20 mm from the centerline. Due to geometrical constraints, film pressure sensors are arranged with two sensors on one side of the shaft and one mounted on the opposite side, alongside the two displacement sensors. Fifty NTC (negative temperature coefficient) thermistors and a number of thermocouples have been strategically located both in the bearing and throughout the oil feed and cooling systems, providing temperature profiles of the oil film to help validate recently developed numerical bearing models. The arrangement of these sensors is displayed in Fig. A.2. Power loss can be determined from calculations of thermal characteristics of the oil and cooling systems as well as by measuring the motor power demands, however...
in this study, power loss is determined from the motor power demand as some of the oil characteristics (eg. specific heat vs. temperature) were uncertain. The test bearing installed during this study is a 130 mm long two axial groove bearing with a radial clearance of 160 μm.

A.4 Lubricants

The lubricants used in this study include standard mineral-oil based turbine oils (VG68 and VG32), as well as synthetic ester based oils with viscosity index improvers (SE32, SE22, SV22, and SE15) as detailed in Tab. A.1.

A.5 Testing parameters

Testing was accomplished so as to ensure that bearing inlet conditions were the same for each of the lubricants. Inlet temperature was controlled in the inlet channel immediately before entrance into the test bearing itself and held at 40±0.5°C in all tests, while oil flow was held constant at 2 l/min and increased to 3 l/min at speeds of 28 and 33 m/s (3000 and 3500 RPM). This flow rate was confirmed to be adequate using calculations proposed by Martin [94]. Speed
Table A.1: Characteristics of tested lubricants

<table>
<thead>
<tr>
<th>Grade</th>
<th>Abbreviation</th>
<th>Base Oil</th>
<th>Viscosity, 40 °C (mm²/s)</th>
<th>Viscosity, 100 °C (mm²/s)</th>
<th>VI</th>
</tr>
</thead>
<tbody>
<tr>
<td>ISO VG68</td>
<td>VG68</td>
<td>Mineral</td>
<td>67.3</td>
<td>8.79</td>
<td>103</td>
</tr>
<tr>
<td>ISO VG32</td>
<td>VG32</td>
<td>Mineral</td>
<td>33.7</td>
<td>5.62</td>
<td>105</td>
</tr>
<tr>
<td>ISO VG32</td>
<td>SE32</td>
<td>Synthetic Ester</td>
<td>32.1</td>
<td>8.46</td>
<td>259</td>
</tr>
<tr>
<td>ISO VG22</td>
<td>SE22</td>
<td>Synthetic Ester</td>
<td>21.4</td>
<td>5.87</td>
<td>245</td>
</tr>
<tr>
<td>ISO VG22</td>
<td>SV22</td>
<td>Synthetic Ester</td>
<td>20.4</td>
<td>6.81</td>
<td>340</td>
</tr>
<tr>
<td>ISO VG15</td>
<td>SE15</td>
<td>Synthetic Ester</td>
<td>15.5</td>
<td>4.46</td>
<td>226</td>
</tr>
</tbody>
</table>

was varied through a range from 9 to 24 m/s (1000 to 2500 RPM) for SE32 and SE22 and from 9 to 33 m/s (1000 to 3500 RPM) for VG68, VG32, SV22, and SE15. Specific bearing load was set at 1, 1.5 and 2 MPa (approx. 23 to 46 kN). These speeds and loads were chosen to keep testing parameters within the operational range of bearings in hydroelectric power turbine applications. Upon operational state change, results were recorded once steady state was achieved (normally 15 to 20 minutes after parameter change). Due to the relatively low thermal mass of the test assembly, changes from one operating state to another led to a much more rapid change in temperature than reported in earlier work with a similar arrangement [97], thus allowing for time savings in the experimental process. Average temperature readings over 10 seconds were taken to reduce noise due to the electronics. Film thickness and pressure were determined from the average signal recorded over 2 seconds at 10 kHz sampling speed. Thus the lower resolution at higher speeds was offset by an increase in the number of revolutions and vice versa at lower speeds. Oil changes were accompanied by a system drain and cleaning of all exposed and accessible surfaces.

A.6 Uncertainty

Due to the complexity of the bearing test system, a certain degree of uncertainty is associated with each of the measurements as described in Tab. A.2. Sources of uncertainty are primarily associated with the measurement equipment with the exception of the film thickness measurements which are a combination of measurement equipment error and mathematical error developed
Table A.2: Measurement uncertainty

<table>
<thead>
<tr>
<th>Source</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermocouple (type K)</td>
<td>±1.0° C</td>
</tr>
<tr>
<td>Film thickness</td>
<td>±3μm</td>
</tr>
<tr>
<td>Thermistor</td>
<td>±1.0° C</td>
</tr>
<tr>
<td>Rotational speed</td>
<td>±0.4%</td>
</tr>
<tr>
<td>Friction torque</td>
<td>±0.2%</td>
</tr>
<tr>
<td>Flow rate</td>
<td>±3.0%</td>
</tr>
<tr>
<td>Bearing load</td>
<td>±0.25%</td>
</tr>
<tr>
<td>Film pressure</td>
<td>±0.2 MPa</td>
</tr>
</tbody>
</table>

through the averaging process of minimum film thicknesses. Additional uncertainty with the on-shaft film thickness measurements results from the fact that the amplifiers for these sensors were mounted inside of the shaft and therefore were not held at a constant temperature as the manufacturer recommends. The shaft temperature in the area of the amplifiers was affected by the roller support bearings which varied greatly in temperature with speed. Temperature also varied in these bearings with loading, but to a very small degree compared with the variation from speed changes. This allowed for comparison of film thickness between different lubricants at individual speed settings, but not between different speed settings. Testing for repeatability of the test procedure was conducted with the original lubricant (VG68) following completion of the full series of experiments. These results were equivalent to the initial results and demonstrated that the test rig and procedure provided a repeatable accuracy through the study’s duration.

A.7 Results and discussion

The results of the testing program highlighted the changes in machine performance that were realized through use of varying lubricant and VI combinations. The tested lubricants are thus compared in two groups: higher viscosity grade (VG68 and SE32) and lower viscosity grade (VG32, SV22, SE22, and SE15).

From the film pressure profiles it was clear that the bearing was operating in the fully hydrodynamic lubrication regime with no starvation. Examples of film pressure profiles are displayed in Fig. A.3.
A.7.1 VG68 and SE32

The comparison between VG68 and SE32 provides the clearest difference in oil performance of this study. System power loss differences between the two lubricants range from nearly 20% at low speeds to just under 10% at 24 m/s as shown in Fig. A.4. The same trend was seen at a lower load setting, shown in Fig. A.5. Minimum oil film thickness was found to be nearly equal in all tests and potentially thicker in the case of SE32 at some speeds as shown in Fig. A.6. To some extent, this could be expected from calculation of the viscosity in the minimum film thickness region, Fig. A.7. The viscosity at the region of minimum film (approximately 90° to 130°) is found to be significantly higher for SE32 than VG68. However, elsewhere in the bearing, the viscosity is much lower for SE32 than VG68 leading to the measured net decrease in power losses, presumably through viscous losses, seen in Fig. A.4 and A.5. It should be noted that temperatures around the top half of the bearing were generally in the range of the lowest temperatures measured in the lower half of the bearing.

In the case of hydroelectric power applications, which primarily operate at slower bearing surface speeds than other turbo-machinery, the primary lubricant in use currently is ISO VG68 mineral based turbine oil such as that included in this study. Previous studies demonstrated that losses can be reduced by changing to a higher performance lubricant of lower viscosity grade. Comparing VG68 to SE32 shows the potential for greater reductions in losses through a shift to an even lower viscosity grade lubricant without a loss in bearing safety. Equivalent film thickness is achieved with the SE32. It is believed...
Figure A.4: Power loss for system at 2 MPa loading

Figure A.5: Power loss for system at 1 MPa loading
Figure A.6: Difference in minimum film thickness between 'X' lubricant and VG68 at 2 MPa load.

Figure A.7: Dynamic viscosity 2 MPa, 24 m/s (2500 RPM)
that further savings could be realized in thrust bearings as well as pumping and control systems due to the much lower viscosity of the synthetic lubricant at system temperatures. Qualitative evidence of this was observed in the test rig’s oil pump which operated at lower loading with SE32 than VG68 throughout the entire test, however the current study has not taken into account these differences in losses in any calculations or measurements.

A.7.2 VG32, SE22, SV22 and SE15

Lower viscosity grades such as ISO VG32 are the most common grades used in gas and steam turbines operating at higher speed with higher temperatures than in hydro-electric power applications. Comparison between standard mineral based turbine oil, VG32, and synthetic lubricants becomes slightly more complex due to the reduced variation in viscosity through the bearing. Thus several different combinations of base lubricant and viscosity index improver have been tested to determine the combination which provides the greatest reductions in power loss while maintaining bearing reliability.

Investigating the power losses produced by each lubricant shows that in some cases the synthetic lubricants out-perform the standard ISO VG32 turbine oil while in other cases, they produce greater power loss as seen in Fig. A.4 and A.5. However, in cases having greater power loss with the synthetic lubricants, the minimum film thickness is also greater. SE15 provides significant power loss reductions under all test conditions when compared to VG32
with film thickness that is nearly equal. Of note is that SE15 has the lowest VI of the synthetic lubricants tested and it is believed that, given future improvement in additive and lubricant chemistry, this lubricant could provide the same stark improvement that was seen in the case of SE32 vs. VG68. Furthermore, the lower operating temperatures provided by SE15 could potentially greatly reduce oil oxidation when operated under high speeds and loads, thus extending lubricant life and machine reliability.

When tested at higher speeds, it was found that SE15 provided equivalent film thickness and lower power loss than VG32 while SV22 yielded higher power loss than VG32, and SE22 yielded equivalent power loss to VG32. The case of SV22 at higher speeds (and higher temperatures) highlights the nature of high VI lubricants. Because of the high VI, the viscosity of the lubricant is higher through much more of the bearing than with a corresponding low VI lubricant. While this leads to increased power losses, it also leads to increased film thickness, as the difference in base oil viscosity is counteracted by the high VI provided by the additives.

The maximum temperature in the center of the bearing is displayed in Fig. A.8 for all lubricants at 2 MPa load.

Examination of the minimum lubricant viscosity (calculated from the maximum temperature) over the range of testing at 2 MPa load (Fig. A.9) clarifies the difference between the VI of the lubricants. At lower speeds, the synthetics provide much lower minimum viscosity than the mineral oils as seen in

Figure A.9: Dynamic viscosity 2 MPa load and varying speed
A.8 Conclusion

This work has investigated the performance of a range of synthetic ester based lubricants with viscosity index improvers in comparison to industry standard mineral oil based turbine oils in journal bearing applications. The study results generally demonstrate that synthetic lubricants in combination with VI improvers provide improved bearing operation. The following conclusions can be drawn:

- High VI ISO VG32 synthetic ester lubricant offers an acceptable general
replacement to ISO VG68 mineral oil, providing reductions in power loss and maintaining or improving journal bearing safety through equal or greater film thickness.

- High VI ISO VG15 synthetic ester lubricant offers an alternative to ISO VG32 mineral oil at higher speeds such as for gas and steam turbines, providing reductions in power loss with equivalent lubricant film thickness. With further improvement in viscosity index improvers and base oil development, similar lubricants could provide better performance than the mineral based lubricants at both high and low speeds.

- High VI ISO VG22 synthetic ester lubricant offers an alternative to ISO VG32 mineral oil in low speed, high load applications such as hydroelectric power plants or shipping, providing reductions in power loss and equivalent film thickness. However, at higher speeds, this lubricant results in greater power loss due to the thicker film it provides.

- Very high VI ISO VG22 and high VI ISO VG22 synthetic esters demonstrate the potential to develop custom lubricants for optimal operation in applications where much higher viscosity grade lubricants, such as ISO VG68, are operated at high temperatures such as in gas or steam turbines.

A.9 Acknowledgments

The research presented in this paper has been carried out as part of a project in conjunction with the Swedish Hydropower Center - SVC. SVC has been established by the Swedish Energy Agency, Elforsk and Svenska Kraftnät in partnership with academic institutions. Additional acknowledgment is given to Statoil Lubricants and Evonik RohMax Additives GmbH for material support.
Paper B

Powerplant lubricant selection for improved efficiency and environmental impact reduction
Powerplant lubricant selection for improved efficiency and environmental impact reduction

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Luleå, Sweden

B.1 Abstract

Computational models were used to optimize bearing performance by adjusting a number of lubricant properties. This computational optimization showed that the most beneficial characteristics to hydrodynamic bearing operation were high viscosity index (VI) and high specific heat capacity. Four environmentally adapted synthetic lubricants were developed to provide these characteristics including: ISO VG32 with 259 VI, ISO VG22 with 245 VI, ISO VG22 with 336 VI, and ISO VG15 with 226 VI. A full scale bearing test machine was then operated with these lubricants in addition to mineral based turbine oils, ISO VG68 with 103 VI and ISO VG32 with 105 VI, to determine the effect on bearing performance and to validate the models. The new lubricants reduced bearing power loss by up to 20% and significantly reduced bearing temperatures with somewhat reduced film thickness. The machine was then operated to provide equivalent minimum viscosity with the new lubricants by varying inlet temperature, finding that changes in power loss were less substantial with equivalent minimum viscosity.

Comparison of simulated and experimental results led to development of a simple, practical method to estimate benefits and operational parameters for lubricants based on viscosity grade, viscosity index and a simplified description of the machine’s bearings. Other, less tangible, factors considered are bio-degradeability and impact of power loss reduction.
B.2 Introduction

The increasing focus on renewable energy sources including wind, solar and hydro-electric power has lead to the installation of industrial machines in natural environments. To minimize the hazardous environmental impact that these machines can cause, many owners are evaluating and switching to new, environmentally adapted lubricants (EALs). However, lubricant change intervals can exceed 30 years and changing lubricant in large rotating machines is a major undertaking often requiring replacement of packings and extensive cleaning. With significant purchase price differences between mineral and synthetic lubricants, a balance between initial costs and operational benefits must be clear.

Meanwhile, developments in the synthetic lubricants which make up EALs have made it possible to not only reduce environmental impact but at the same time significantly improve machine performance. This has turned lubricants from necessary maintenance products into performance enhancing investments.

With a host of different synthetic lubricants in production, all with unique base oils and additives, it can be quite confusing for the end user to decide which lubricant to use, and to determine how a specific lubricant can affect a specific machine’s performance.

B.2.1 Experimental Studies

A number of experimental studies have been conducted on fluid film bearings, both thrust and journal, in laboratories and in the field which have helped to develop a description of the effects that can be expected following a lubricant change. While the most commonly reported result is a decrease in maximum temperature of the bearing, the specific results and conditions of the experiments have some variation.

In laboratory tests, power loss reductions and equivalent film thickness were found to result from changing to a thinner synthetic lubricant in thrust bearings by New and Schmaus [110] and later by Glavatskiih and Larsson [62]. Boehringer and Neff found significant improvements in machine performance upon changing to a di-ester based lubricant [11] in a full scale hydropower thrust bearing. Similar improvements in a full scale hydropower machine were found by Glavatskiih [60] upon changing to an ester based lubricant in a combined thrust journal bearing. Ferguson et al. [48] investigated ISO VG68, VG46 and VG32 in a large thrust bearing test rig and proposed using a special
numerical modeling software package to predict maximum bearing temperatures from oil bath temperatures and thereby predict bearing performance characteristics. Calculated results agreed well with experiments on a large thrust bearing. Lower power losses were found for lower viscosity grade (VG) and higher oil bath temperatures. Significant decreases in film thickness were also observed but it was argued that the lower film thickness was still adequate to maintain machine safety.

Investigations with journal bearings by Swanson et al [144] in work with VI improvers found that the thermal performance of mineral-based oils could be improved to match that of synthetic lubricants. Separately, Dmochowski [34] and McCarthy et al [97] found similar effects in their studies on the performance of journal bearings with high VI polyalphaolefin and ester based lubricants. However, power loss reductions in journal bearing studies were generally lower magnitude, percentage wise, than those found for thrust bearings and laboratory results have shown less reduction than field experiences.

The majority of earlier experimental work has kept inlet or oil bath temperature constant so as to keep initial operating conditions equivalent for each of the lubricants. Power loss, film thickness and temperature were then found to compare performance of the varying lubricants. Temperatures throughout the bearing were used to calculate lubricant viscosity which was then compared, finding that in the case of the higher VI lubricants, viscosity was equivalent in the region of highest temperature (lowest viscosity) [34, 97]. The findings from McCarthy et al showed that the film thickness was lower for the higher VI lubricants until the speed and load had increased to a point at which the minimum viscosities of the lubricants became equivalent and it was concluded that the reduced power losses provided by the higher VI lubricants were the result of lower bulk lubricant viscosity in the bearing.

B.2.2 Numerical Studies

The earliest models of fluid film bearings handled viscosity as a constant through the bearing rather than as a parameter locally affected by changes in temperature, however it was soon found that variation of lubricant viscosity plays an essential role in the Reynolds equation. This effect of including viscosity effects was shown as early as 1933 by Kingsbury [81] to reduce power loss significantly, up to 60% in some cases. Later, the generalized Reynolds equation with the variable viscosity term was derived by Dowson [35] which significantly improved the accuracy of the numerical models. The next step was done in the area of thermal boundary conditions. A heat flux based bound-
ary was proposed by Dowson and March [36] and an adiabatic boundary by Pinkus and Bupara [116]. Around the same time, the effect of thermoelastic deformation has been taken into account Rohde and Kong Ping [122].

Through the 1980’s and ’90’s, models for fluid film bearings progressed in accuracy with the growth of computing power. Recently, Tanaka and Hatanaka [147] presented a very accurate numerical model including detailed description of the boundary conditions and solution scheme.

While numerical and experimental studies have provided much broader understanding of bearing function and methods for improving bearing performance, very few studies have provided useful guidance for the practicing engineer. The present study seeks to provide practical guidance for the end user regarding methodology to analyze changes in lubricant.

**B.3 Experimental setup**

Lubricants were tested in a 180 mm diameter, two-axial groove, Babbitt coated journal bearing with L/D ratio of 0.7. The test equipment, Fig. B.1, has a shaft equipped with two inductive displacement sensors and four thermistors on the shaft to continually measure film thickness and shaft temperature. Temperature along the bearing surface is measured using 46 thermistors arranged to determine circumferential and axial film temperature profiles. Lubricant supply is controlled using an in-line flow meter while lubricant temperatures are monitored using thermo-couples and temperature is controlled using a heat exchanger. Two inductive displacement sensors are located outside the housing for redundancy in film thickness and eccentricity measurements. The shaft is
Table B.1: Measurement uncertainty.

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Type</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Displacement (Internal)</td>
<td>Inductive</td>
<td>± 5 μm</td>
</tr>
<tr>
<td>Displacement (External)</td>
<td>Inductive</td>
<td>± 5 μm</td>
</tr>
<tr>
<td>Temperature</td>
<td>Thermistor</td>
<td>± 0.5 °C</td>
</tr>
<tr>
<td>Temperature (Thermocouple)</td>
<td>(K)</td>
<td>± 1 °C</td>
</tr>
<tr>
<td>Speed</td>
<td>Motor Electronics</td>
<td>0.4%</td>
</tr>
<tr>
<td>Power Loss</td>
<td>Motor Electronics</td>
<td>0.6%</td>
</tr>
</tbody>
</table>

Table B.2: Lubricants used in experimental studies.

<table>
<thead>
<tr>
<th>Grade</th>
<th>Abbreviation</th>
<th>Viscosity, ( (\text{mm}^2/\text{s}) )</th>
<th>VI</th>
</tr>
</thead>
<tbody>
<tr>
<td>ISO VG68</td>
<td>VG68</td>
<td>67.3 8.79 103</td>
<td></td>
</tr>
<tr>
<td>ISO VG32</td>
<td>VG32</td>
<td>33.7 5.62 105</td>
<td></td>
</tr>
<tr>
<td>ISO VG32</td>
<td>SE32</td>
<td>32.1 8.46 259</td>
<td></td>
</tr>
<tr>
<td>ISO VG22</td>
<td>SE22</td>
<td>21.4 5.87 245</td>
<td></td>
</tr>
<tr>
<td>ISO VG22</td>
<td>SV22</td>
<td>20.4 6.81 340</td>
<td></td>
</tr>
<tr>
<td>ISO VG15</td>
<td>SE15</td>
<td>15.5 4.46 226</td>
<td></td>
</tr>
</tbody>
</table>

held in place by two identical roller bearings and is powered by a 43 kW motor which provides power and speed measurements through its controller. Loading is applied to the bearing housing using a pneumatic cushion equipped with four 50 kN load cells. Uncertainty for the various measurements is provided in Table B.1.

The lubricants investigated in this work consist of industry standard mineral-based ISO VG68 and ISO VG32 turbine oil as well as four environmentally adapted synthetic lubricants (EALs) including an ISO VG32, two ISO VG22 and an ISO VG15. Characteristics of these lubricants are detailed in Table B.2 as well as the abbreviations used for convenience.

Initial testing of the lubricants was accomplished in the same manner as the earlier mentioned experimental studies and is detailed in Table B.3. Inlet temperature was kept constant while load and speed were varied with measure-
Table B.3: Testing program for experimental studies.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Speed</td>
<td>RPM</td>
<td>1000, 1500, 2000, 2500</td>
</tr>
<tr>
<td>Mean Pressure</td>
<td>MPa</td>
<td>1.0, 1.5, 2.0</td>
</tr>
<tr>
<td>Inlet Temp.</td>
<td>°C</td>
<td>40 ± 1.0</td>
</tr>
<tr>
<td>Oil Flow</td>
<td>l/minute</td>
<td>2.0 ± 0.05</td>
</tr>
</tbody>
</table>

ments taken upon reaching steady state at each testing point. After completion of the initial round of tests, the maximum temperatures in the bearing at each state were compared and calculations of minimum viscosity were made.

Further testing with VG68 was then accomplished by setting the maximum film temperature to these calculated values by changing the inlet temperature to compare lubricant performance under equivalent film viscosity. In these further tests, effort was made to achieve steady state in all cases, however, due to the high temperatures involved and efficient system cooling through radiation, some tests could not be performed at steady state while temperatures required for comparing VG68 and SE15 were only achievable in a few cases. This caused power loss comparisons for the lower VG lubricants to be somewhat less reliable than those of the higher VG lubricants.

**B.4 Numerical model**

**B.4.1 Model Description**

Numerical experiments were conducted using a thermo-hydro-dynamic (THD) model described fully by Kuznetsov et al [86]. This model includes thermal effects and flow (oil film pressure) analysis. Pressure distribution is found by solution of the simplified the Reynolds equation. The finite difference technique is used with zero boundary conditions for discretization of Reynolds equation. Cavitation effects are considered by introduction of a switch function. Analysis of thermal effects is performed by solving the energy and heat transfer equations while an iterative successive-over-relaxation (SOR) method is then used to solve the final system of equations.

Boundary conditions are set to give zero heat flux for the shaft and conservation of the heat flux for oil-bearing and bearing-air interfaces. It is assumed
that the cold oil flow is split equally between two supply grooves. The oil mixing prediction is based on a predefined mixing coefficient, Tanaka and Hatak- enaka [147]. Both equations are then discretized using the finite difference method and solved simultaneously with the iterative SOR method.

The solution process begins from a uniform viscosity guess, from which the Reynolds equation is solved. Next, the obtained solution is used to predict the temperatures within the oil film and the bearing. The influence of thermal expansion of the shaft and bearing are then accounted for and a new viscosity field is computed using the previously obtained temperatures. The process is then iterated using the newly calculated viscosity as the new guess. This process is repeated in a loop until no changes in viscosity are visible, that is the difference in lubricant temperatures between the two preceding iterations satisfies the convergence criterion.

### B.4.2 Numerical Experiment Setup

Geometry for the numerically modeled bearing was identical to that of the actual test bearing. Discretization was performed on a mesh grid of size 256 points in the circumferential, 33 points in the axial and 21 points in the cross-film directions. Convergence criteria are, pointwise, $10^{-6}$ for the pressure and $10^{-7}$ for the temperature of the oil fluid film.

The goal of the numerical experiment was to achieve the same load and minimum film thickness for oils VG32, VG68, SE32, SE22 and SV22 and compare the variation of the power loss and oil maximum temperature versus rotational speed. To accomplish this, eccentricity was fixed at 0.80 and the model was run with a range of supply temperatures from 5°C to 80°C. Then an interpolation of load, maximum oil temperature and power loss versus supply oil temperature was performed to determine values for the given load. This yielded equivalent performance at 52 kN load (2.2 MPa mean bearing pressure), slightly higher than the 2.0 MPa maximum mean pressure used in the experiments.

### B.5 Results

The experimental and numerical investigations both demonstrated that lubricants of widely varied viscosity grade and viscosity index can provide film forming function in a journal bearing. The results further highlight the sensitivity of power losses in journal bearings to small changes in viscosity in
the bulk of the bearing. An examination of the temperature through the bearing pointed out viscosity zones that occur when lubricants are replaced while comparing the power loss and change in eccentricity allowed for further comparison of the lubricants.

B.5.1 Temperature and Viscosity Zones

During the initial investigation with constant inlet temperature of $40^\circ$ it was found that film thickness in the bearing was equivalent for both VG68 and SE32 in cases where the lubricant viscosity was equal or greater for SE32 in the area of highest temperature, Fig. B.2. In this particular case, SE32 yielded approximately 10% power reduction compared to VG68. To describe where the difference in losses come from, the bearing was divided into the three zones shown in Fig. B.2. Zone 1 is the region on the loaded half of the bearing where the lubricant has relatively high viscosity which can add to losses, Zone 2 can be considered the primary load bearing area where pressure and temperature are greatest, while Zone 3 is the unloaded region of the bearing where greater viscosity adds to losses only. From this division it can be determined that having the proper viscosity in the most heavily loaded region allows for the desired minimum film to form. Likewise, decreased viscosity through the rest of the bearing allows for reduction in power losses from lubricant shearing. This hypothesis led to the development of the viscosity matching experiments as well as the numerical investigation with fixed eccentricity and load.

Figure B.2: Temperature and Viscosity at 2 MPa load and 24 m/s.
The results of the numerical investigation reinforced the relationship between maintaining eccentricity and viscosity in Zone 2. As shown in Fig. B.3, the minimum viscosity in the bearing was very similar for all lubricants at any given speed. The slightly higher viscosity seen for the high VI lubricants reflects the earlier finding that matching eccentricity occurs when the high VI lubricant has slightly higher viscosity in Zone 2 than the low VI lubricants. The maximum viscosity found in Zone 1 and 3 was lower for the high VI lubricants in the model as shown in Fig. B.4. The low viscosity in the unloaded regions shows the potential loss reduction allowed by the high VI lubricants.
B.5.2 Power Loss

Unlike the initial investigations, changes in power loss were much smaller in the viscosity matching experiments. However, comparison between VG68 and the high VI lubricants showed that the high VI lubricants tend to have lower power loss with increasing speed. Comparison between all lubricants and VG68 are shown in Fig. B.5 and Fig. B.6 for 1 MPa and 2 MPa mean pressure respectively. Very little change in power loss was observed for lower speeds, and in most cases, the power loss was seen to increase with the thinner lubricants at the lowest speed. No trend was observed for VG32 which appeared to produce a slightly higher power loss at all speeds compared to VG68. This agrees well with Ferguson et al. [48] who found that VG32 and VG68 behaved similarly at equivalent viscosity. Similar trends were observed in the model results except that the differences in power loss were within the error of the interpolation scheme.

B.5.3 Eccentricity Change

Eccentricity behaved similarly to power loss in the experimental investigation when VG68 was compared to the high VI lubricants as shown in Fig. B.7 and B.8 for 1 MPa and 2 MPa mean bearing pressure respectively. At lower speeds, the synthetic lubricants were unable to carry the same film thickness as their thicker mineral oil counterparts. However, as speed increases, the eccentricity provided by the synthetic lubricants became approximately equal to that of the mineral lubricant until at the highest speeds, the synthetics pro-
vided lower eccentricity. The eccentricity provided by VG32 was similar to that of the synthetic lubricants, with a higher eccentricity at lower speeds and lower eccentricity in relation to VG68 at higher speeds. This behavior is supported by the minimum viscosity results from the model, Fig. B.3 in which the high VI lubricants have slightly higher minimum viscosity at all speeds for equivalent eccentricity and load. In the experiment, minimum viscosity was computed from the maximum measured temperature which was slightly lower than the actual maximum film temperature which led to minor discrepancies in matching VG68 with the other, much thinner lubricants.

This effect is believed to be a result of the location of the temperature measurement as the maximum temperature measured in the bearing is somewhat lower than the actual maximum temperature in the oil film. Because this difference in temperature increases with speed and because the temperature matching was done at the lower temperature, the synthetic lubricants actually had higher viscosity in the maximum temperature region than the VG68 lubricant. This phenomenon further helps explain why the lower viscosity synthetics provided lower eccentricity than SE32 as well as the slight difference in power loss of the SE22 and SV22 compared to SE32.

**B.6 Method for lubricant selection**

With a host of new lubricants available a simple method of evaluating potential lubricants should be followed to allow for selection of the best alternative. Any evaluation should include a determination of the value of biodegradability, cost
of disposal and lubricant lifetime. Following this, a determination of which lubricants can be used practically, and an evaluation of how they should be operated should be carried out. Finally the various options can be compared to determine what performance changes can be expected through replacement of the existing lubricant.

### B.6.1 Environmental Factors

Long service life and environmental friendliness often work against each other in the case of bio-lubricants. Bio-degradable lubricants, such as rapeseed oil have been found to perform well in short tests, McCarthy et al. [97], however...
ever as noted by Schneider [134] many plant-based lubricants oxidize much more rapidly than mineral oil based lubricants. Some ester based lubricants have been found to have oxidation stability equivalent to or better than mineral oils. However, ester based lubricants have been found to affect sealing materials which can turn a lubricant change into a complicated process including replacing incompatible sealing materials with compatible materials. A different view of the issue from an economist, Mann [92], reported that changing industries to bio-based lubricants is more successful when governments institute the change rather than industries take the initiative themselves. In spite of this, many industries have changed lubricants and been successful as reported by Broekhuizen [16] so the coupling between economic and environmental advantages clearly depends on the specific application.

An often overlooked aspect of environmental impact reduction is that improved efficiency of existing equipment reduces the need for new construction. Furthermore, while most renewable energy sources are variable (wind, solar, tidal, etc), most existing power sources (fossil fuel, nuclear, and hydro power) can be regulated and thus are more valuable to the electricity grid than variable sources. These regulating power machines are generally much larger than variable power sources, thus small improvements in efficiency can result in significant real increases in power output.

B.6.2 Viscosity Matching

Earlier recommendations for changing lubricant in machines involved using specially developed computer models to predict performance changes from an oil change. The aim of the current study has been to develop a simpler method through a clearer understanding of the functional differences in lubricants. It has been shown that film thickness is maintained in the bearing as long as viscosity of the new lubricant matches or exceeds that of the old in the loaded zone of the bearing. Thus when changing lubricant from a low VI to a high VI, viscosity should be calculated in the loaded zone for the old lubricant, then from that viscosity, the required maximum temperature of the new lubricant can be calculated.

The bearings of most large machines already have temperature sensors in the oil bath and in the pads that can be used to determine the bath and 'maximum' temperature. Because of inconsistency in how temperature sensors are mounted in the pads in industry, the difference between the pad temperature and the maximum temperature in the oil film is not usually clear. However, given that these sensors should always give a temperature colder than the ac-
tual maximum, calculating with them guarantees that the minimum viscosity in the film will be greater for the new lubricant than the old. While this assumption predicts less savings through changing lubricant it trades savings for improved machine safety in the form of greater film thickness. The bearing should then be operated using the calculated maximum temperature of the new lubricant to maintain equivalent minimum viscosity.

B.6.3 Estimating Performance Changes

As the load carrying function of a fluid film bearing is provided by only a portion of the lubricated surface, it can be assumed that losses in the load carrying portion of the bearing should be nearly equivalent regardless of the lubricant if equivalent function is to be maintained. However, by allowing for viscosity reduction through the unloaded part of the bearing, churning losses can be significantly decreased with the use of lower viscosity grade but higher VI lubricants. Because churning losses are directly associated to viscosity, it follows that an estimate of the percentage change in power losses can be described by Eq. B.1.

\[ \Delta \text{losses} = A \ast [1 - \frac{\mu_{\text{new}}}{\mu_{\text{old}}}] \]  

(B.1)

Where \( A \) is the percentage of the bearing’s area which contributes to churning losses only (e.g. non-load bearing area) and \( \mu_{\text{new}} \) and \( \mu_{\text{old}} \) are the viscosity of the oil bath for the new and old lubricants. \( A \) can vary greatly depending on the type of bearing in question. For example, a tilting pad thrust bearing in a power plant has considerably greater churning area than a plain journal bearing due to gaps between pads and vertical surfaces of the shaft. The temperature increase, \( \Delta T \), between the bath and the maximum temperature is assumed to be the same for the lubricants as losses are assumed to be equal in the loaded portion of the bearing. This is not entirely correct as in the experimental work, it was found that synthetic lubricants had a smaller \( \Delta T \) which was partly due to the difference in specific heat capacity, \( \rho C_p \), of the lubricants; synthetic esters have slightly higher \( \rho C_p \) than mineral oils, Pettersson [112]. Given equivalent losses developed in the oil film, the lubricant with higher \( \rho C_p \) experiences lower \( \Delta T \).

In the case that accurate measurements of \( \rho C_p \) are known, \( \Delta T \) can be reasonably estimated using Eq. B.2.

\[ \Delta T_{\text{new}} = \frac{\rho_{\text{old}} C_{\text{old}} \Delta T_{\text{old}}}{\rho_{\text{new}} C_{\text{new}}} \]  

(B.2)
Including $\rho C_p$ in the calculations allows for more accurate estimation of bath temperatures which, in the case of increased $\rho C_p$, allows for a greater decrease in losses due to lower bath viscosity.

The total change in power loss can finally be calculated using estimates of the bearing losses in the system. For hydro-electric power generators the bearing losses are approximately 0.2% [48] of the machine’s output but this varies with machine speed.

This value appears low at first glance, however with an output of 100 MW from the machine, the bearing losses are on the order of 200 kW which is quite significant. Unlike in other industries where new lubricants are compared in terms of the savings from less frequent oil changes and changes in maintenance practices, Johnson [76], the value and the power savings in a power plant can also be directly applied as increased power output. This increased power output can then be valued at sales prices for electric power and the operation schedule of the machine.

B.6.4 Practicalities

It should be noted that the approach of matching viscosity presents practical limits in terms of the relationship between viscosity grade (VG) and VI. The trend for optimum performance is to reduce VG and increase VI, however this poses a challenge. The additives used to increase VI also increase VG which leads to the selection of ever thinner base fluids. Additionally, to provide matching viscosity, the required viscosity occurs at an increasingly lower temperature leading to an even lower temperature in the oil bath. An example to exemplify this being that to replace VG68 with SE15 in a hydropower turbine bearing with maximum temperature around 75°C would require the oil bath to be less than 5°C which is clearly unreasonable. In the same application, substitution with SE32 would leave the oil bath at around 30°C, a low but much more realistic temperature in a hydropower station.

The argument has previously been made that oil film thickness can be safely reduced to some degree [48, 15] by operating with a higher maximum temperature thus allowing for further loss reduction. However, this decision should be taken separately from a change of lubricant.

B.6.5 Practical Example

As part of a research program, the lubricant in one 10 MW turbine at the Porjus power station on the Lule River in northern Sweden was changed from
ISO VG68 mineral oil to an ester based ISO VG46 lubricant with VI 150. The original maximum temperature was 95°C and the original oil bath temperature was 30°C. Matching viscosity between the old lubricant and the new lubricant results in that the maximum temperature of the new lubricant should be 91°C which leads to a bath temperature of 30°C. This calculation includes a small change in $\rho C_p$ from 1.8 MJ/cm$^3$ to 1.9 MJ/cm$^3$ incorporated using Eq. 7.2. Assuming an area of losses in the large, fully immersed combined thrust/journal bearing of 50% results in a prediction for power savings in the bearing of 16%. After the actual oil change, the maximum temperature decreased to 84°C because coolant flow was unchanged from prior to the oil change. This resulted in the oil bath being a cool 24°C during the winter months. As could be expected from such a drop in bearing temperature, the oil film thickness increased by 20%. During the summer months, the temperature in the bath was allowed to increase to 35°C while the maximum temperature increased to 88°C. An 18.5% reduction in bearing power loss was measured by the temperature change in the coolant flow compared to the original lubricant while the film thickness remained thicker than prior to the oil change. The reduction in bearing power loss allowed for an increase of production of 30 kW which, while small, is not insignificant considering the relatively small size of the machine.

This example highlights the validity of the simplistic viscosity matching method. After changing lubricant, temperatures and savings predicted by the methodology were reasonably close to the actual values observed. While the operators in this example could have yielded greater reductions in power loss by allowing the new lubricant to operate warmer, they were satisfied with increased safety in terms of increased film thickness.

### B.7 Conclusion

This work has provided comparison between two standard mineral based turbine oils and four synthetic based environmentally adapted turbine oils. The comparison of lubricant performance led to the development of a simple method for the practicing engineer to determine the economic value of changing lubricant in a machine in terms of power savings and increased output potential as well as the new operating parameters for the bearings. This allows the following conclusions to be drawn:

- Equivalent machine performance (similar losses and film thickness) is provided when a high viscosity grade lubricant is replaced with a lower
viscosity grade lubricant operated at lower maximum and inlet temperatures.

- Improved machine performance (equivalent eccentricity and decreased losses) can be provided when high viscosity grade lubricants are replaced with lower viscosity grade and higher viscosity index lubricants operated at lower temperatures. This decrease in losses occurs due to a decrease in losses in the non-load bearing portions of the bearing.

- The optimum operating conditions for the new lubricant and changes in bearing performance can be predicted using a simple comparison of the minimum viscosity in the maximum temperature zone of the bearing. Matching viscosity in the maximum temperature region allows for reduced viscosity in the rest of the bearing which in turn allows for reductions in power loss in the bearing.

- Reductions in power loss in addition to the improved environmental characteristics of the new lubricants can make up for the initially higher cost of changing lubricant.

B.8 Acknowledgements

The research presented in this paper has been carried out as part of a project in conjunction with the Swedish Hydropower Center - SVC. SVC has been established by the Swedish Energy Agency, Elforsk and Svenska Kraftnät in partnership with academic institutions. Additional acknowledgment is given to Statoil Lubricants and Evonik RohMax Additives GmbH for material support.
Paper C

Break-away Friction of PTFE Materials in Lubricated Conditions
C.1 Abstract

This study investigates the tribological characteristics at initiation of sliding (break-away friction) of several polytetrafluoroethylene based materials. Four PTFE composites, pure PTFE, and white-metal were tested in a reciprocating tribo-meter with the block on plate configuration against a steel counter surface. Apparent contact pressure and oil temperature were varied from 1-8 MPa and 25-85°C respectively. SEM investigations revealed wear patterns of the PTFE materials and the abrasive nature of hard fillers.

Bronze-filled, carbon-filled, and pure PTFE were found to provide lower break-away friction and less variation over the course of testing and generally superior properties.

C.2 Introduction

Sliding bearings in use in most large power generation machines are designed to operate in the fully hydrodynamic regime with a thick oil film separating the sliding and stationary surfaces. In the full film regime, these bearings provide low friction and extremely long service life. White metal (Babbitt) materials have traditionally provided acceptable performance. However, changes in electricity markets and the introduction of variable power sources have resulted in more frequent starts and stops of power generating machines. Because the Babbitt material currently in use is not optimum for these conditions due to its
potential for being damaged by seizure at start-up, hydraulic jacking systems are often used to flood the bearing pads and lift the machine prior to start-up.

Since its invention in the 1930's, countless studies have found PTFE to provide low friction in dry sliding, Biswas and Vijayan [10]. However, PTFE is also associated with some of the highest wear rates among crystalline polymers in dry contacts, Friedrich et al [54]. This deficiency has led to the use of fillers to improve the mechanical and wear properties of the PTFE matrix and has been widely documented by Bahadur and Tabor [8], Briscoe et al [13] and Xue et al [164]. A review of work related to polymers, including PTFE, with nano-particle fillers is provided by Friedrich et al [55]. Gong et al [68] found that the choice of the counter-surface material's chemical properties had no effect on the wear rate of PTFE. Presumably, wear of the PTFE is primarily caused by the roughness of the counter surface as mentioned by Akagaki et al [4]. A comparison of tribo-testing techniques for polymer materials was accomplished by Samyn et al [123] finding that the selection of test arrangement significantly affected the friction and wear rates in dry conditions.

Little work has been accomplished in regards to PTFE’s characteristics in lubricated conditions. Zhang et al [166, 167, 165] found that the friction and wear of PTFE were greatly reduced by adding lubrication. This result can probably be explained by the high sliding speeds used during the tests, 1.5 - 4 m/s, which could have allowed for hydrodynamic lubrication. A recent study on wear rates in lubricated conditions, McCarthy and Glavatskih [96], of PTFE composites found a marked decrease in wear rate of PTFE materials filled with MoS$_2$ and fiberglass, black glass, bronze, and carbon. However, Akagaki et al [4] and Zhang et al [166] found that PTFE-based composites exhibit different tribological behavior compared to pure PTFE material with varying pressures and temperatures in lubricated conditions.

Journal bearing configurations made of a variety of polymers were tested in dry conditions by Ünlü et al [156] finding that PTFE provided the lowest coefficient of friction and some of the highest wear rates of the materials tested. Further journal bearing tests with both carbon composite and bronze composite PTFE in dry conditions were conducted by Tevrüz [149, 150] finding in both cases that the friction coefficient decreased with increasing contact pressure while the friction coefficient increased with increasing temperature.

Work has been accomplished to investigate PTFE-based composites for hydrodynamic journal bearing applications, Iwai et al [75], finding that PTFE-based composites performed better than Babbitt in lubricated conditions. Pure PTFE has long been used as a bearing surface material in large hydrodynamic bearings, Glavatskih et al [64].
C.3. EXPERIMENTAL STUDY

Investigating start-up in sliding bearings, it was found that journal bearings rapidly become hydrodynamic following an initial moment of break-away friction, Bouyer et al [12]. While wear of the bearing was observed after these tests, it was deemed that the bearing continued to function acceptably. On the other hand, industrial experience of bearing failures due to wiping or failure of the bearing’s Babbitt surface has led to the widespread use of hydrostatic jacking systems during machine start-up. Reduction of the break-away friction by changing to low friction and gently wearing bearing materials has potential to simplify machine start-up while also controlling the risk of bearing failure.

This work seeks to address the issue of break-away friction in hydrodynamic bearings through focused testing of bearing materials in a simplified configuration. This configuration is designed to isolate the transition from static to dynamic friction at machine start-up by eliminating hydrodynamic effects which become more dominant as speed increases.

C.3 Experimental study

The break-away friction characteristics of some commercially available PTFE-based composite materials sliding against steel plates under lubricated conditions are studied using a reciprocating block on plate configuration. These tests are conducted over a wide range of contact pressures and temperatures. Sliding speeds in all tests are kept extremely slow to eliminate the possibility of hydrodynamic effects. The results are compared with those of pure PTFE and Babbitt material.

C.3.1 Materials

Testing was performed using four commercially available PTFE-based composites together with pure PTFE and Babbitt material. Details of the PTFE-based composites are given in Table C.1. Compression modulus was determined on-site for comparison of the materials to each other. The materials have been chosen due to their availability and to develop upon earlier study of PTFE-based composites by McCarthy and Glavatskih [96]. Babbitt specimens were removed from a journal bearing sleeve and consist of a steel backing with a minimum of 0.5 mm of aluminium-tin (AlSn40Cu1) coating on the bearing surface.

The counter surface in all testing was low carbon steel plate ground and polished to a surface roughness of 0.4 μm Ra with structure parallel to the sliding direction which is a typical surface roughness and direction for counter
Table C.1: PTFE composite material characteristics.

<table>
<thead>
<tr>
<th>Filler Material</th>
<th>Compression Modulus [GPa]</th>
<th>Specific density</th>
</tr>
</thead>
<tbody>
<tr>
<td>40% Bronze</td>
<td>0.99</td>
<td>3.96</td>
</tr>
<tr>
<td>25% Carbon</td>
<td>0.85</td>
<td>2.15</td>
</tr>
<tr>
<td>25% Black glass</td>
<td>0.66</td>
<td>2.19</td>
</tr>
<tr>
<td>20% Glass fiber and 5% MoS2</td>
<td>0.78</td>
<td>2.28</td>
</tr>
<tr>
<td>Pure virgin PTFE</td>
<td>0.46</td>
<td>2.16</td>
</tr>
</tbody>
</table>

Table C.2: Test lubricant characteristics.

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Density at 40°C</td>
<td>0.92 kg/l</td>
</tr>
<tr>
<td>Density at 100°C</td>
<td>0.86 kg/l</td>
</tr>
<tr>
<td>Kinematic viscosity at 40°C</td>
<td>31 mm²/s</td>
</tr>
<tr>
<td>Kinematic viscosity at 100°C</td>
<td>6.2 mm²/s</td>
</tr>
</tbody>
</table>

surfaces in hydrodynamic bearings. Lubricant used in all testing was commercially available synthetic ester based turbine oil as detailed in Table C.2. This lubricant was chosen as it is representative of the lubricants being used in new and renovated hydrodynamic bearings in large machines.

C.3.2 Experimental setup

Experiments were carried out using a reciprocating tribo-meter under lubricated conditions in the block on plate configuration. This configuration was chosen to eliminate a converging gap which could lead to hydrodynamic lubrication and to ensure that contact pressure remained constant with varying wear rates of the materials. Providing a constant contact is a common challenge in block on ring studies of soft materials against hard materials. The conditions are nearly identical to those in hydrodynamic thrust bearings at start-up, while because of the small scale the conditions are similar to those in large hydrodynamic journal bearings at start-up. The steel counter-surfaces were fixed in a temperature controlled oil bath while the block specimens were fastened to the
C.3. EXPERIMENTAL STUDY

Load Plate Test block

Figure C.1: Diagram of block on plate test arrangement.

Reciprocating motion was provided by an eccentric transmission driven by an electric motor which was geared down through a gearbox. Loading was accomplished via a spring connected to the reciprocating arm and was varied to provide specified contact pressures. Friction force measurements were taken continuously using a piezoelectric load cell. A full description of the experimental arrangement is detailed in Table C.3 and a diagram showing the test configuration is provided in Fig. C.1. Tests were conducted three times each to ensure repeatability and minimize uncertainty.

Testing for the dependence of break-away friction on contact pressure was conducted at 25°C and tests for the dependence of break-away friction on temperature were conducted at 2 MPa contact pressure. Additional tests were conducted at the maximum temperature and load conditions, 85°C and 8 MPa, to determine whether, or not, temperature effects and contact pressure effects acted independently of each other.

Further tests were conducted to determine how break-away friction was affected by extended periods of loading without operation in the presence of lubricant. For these tests, specimens were run-in to a steady state condition (10 minutes). They were then shifted to the start-stop point at the end of the stroke and left stationary in that position under load for 72 hours. The test was then re-started and the first several start-stop cycles were recorded to determine the break-away friction for each cycle.
Table C.3: Experimental conditions for short term tests with pressure and temperature variation.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Load</td>
<td>80 to 320 N</td>
</tr>
<tr>
<td>Contact Pressure</td>
<td>1 to 8 MPa</td>
</tr>
<tr>
<td>Oil Bath Temperature</td>
<td>25, 45, 65, 85 °C</td>
</tr>
<tr>
<td>Stroke Length</td>
<td>5 mm</td>
</tr>
<tr>
<td>Stroke Duration</td>
<td>5 seconds</td>
</tr>
<tr>
<td>Test Duration</td>
<td>3 hours</td>
</tr>
<tr>
<td>Total Sliding Distance</td>
<td>10.8 m</td>
</tr>
<tr>
<td>Steel Surface Roughness Ra</td>
<td>0.4 μm</td>
</tr>
</tbody>
</table>

C.3.3 Break-away friction

Break-away friction is determined from the friction load data stream provided by the test rig. The value of break-away friction is taken from the maximum value of each stroke occurring when the reciprocating test specimen goes through the stop and start at the end of each stroke. Figure C.2 gives an example of the raw data stream from a typical test of carbon-filled PTFE showing a number of starts and stops with steady break away friction. An average of these peak values over the length of the test is used to determine the average break-away friction for the materials. The lack of symmetry over each stroke of the friction curve results from the flexing of the materials and lubricant presence in the contact. During the stop, the surfaces settle against each other as lubricant is squeezed out of the contact, leading to the dramatic break-away peak. However, because lubricant is fed into the contact all the way until the stop, there is no friction peak at the end of the stroke. This effect is increased by the flexing that occurs in materials with lower elastic modulus, such as PTFE. Babbitt displayed much more symmetry over the course of each stroke with a lower peak at the end of the stroke. The slight difference in the shape of the positive and negative portions of the curve is the result of the geometry of the test rig arrangement. The load, which is provided by a spring and beam, varies slightly over one test cycle by approximately 1 to 2% of the total load value.

Running-in effects are included in the average friction results and are taken into account in terms of the variation in the break-away friction through the duration of the test. This method was used because some of the materials did not provide a steady state friction after running-in. Further supporting this decision
C.3. EXPERIMENTAL STUDY

is the application of this test program to hydrodynamic sliding bearings which only experience break-away friction at start-up of the machine. With the exception of pumped storage machines, these bearings typically experience less than one start per day over the course of their several year life-cycle. Thus any variation of break-away friction, including changes during running-in, can have significant consequences for machine operation.

C.3.4 Material preparation

The polymer blocks were prepared in two different dimensions to achieve desired contact pressures with the practical load range of the test rig which is in the range of 80N to 400N. The dimensions of the contact surface of block specimens for 1MPa tests were 5mm parallel to and 16 mm perpendicular to the sliding direction, while for the rest of the loading conditions the contact surface dimensions were 5mm parallel to and 8mm perpendicular to the sliding direction. This made it possible to achieve apparent contact pressures of 1 MPa to 8 MPa. The block specimens were cleaned in an ultrasonic bath using industrial petroleum (heptane) followed by ethyl alcohol and dried in air prior to each test.
C.4 Results and discussion

The experimental setup was found to produce highly reproducible results, allowing for clear comparisons to be made between the performance of the various materials. In general, each of the test materials followed one of three different trends. The first of these trends was that of Babbitt which provided the highest levels of break-away friction. This high friction phenomenon reflects typical industrial applications of Babbitt bearings which generally limit mean contact pressures to 2.5 MPa without the use of hydrostatic lifting systems at startup. Another trend was displayed by the materials with glass and fiberglass fillers. These materials tended to wear the counter-surfaces, leading to decreased roughness and higher levels of friction, but not as high as Babbitt. The final response that was observed was that of the bronze- and carbon-filled PTFE, as well as pure PTFE, which did not appear to wear the counter-surface and provided low friction with little variation due to temperature or loading.

C.4.1 Contact pressure effects

Results of break-away friction in relation to variation in contact pressure generally showed a slightly decreasing trend in friction coefficient with increasing contact pressure. This is consistent with empirical formulas proposed by Zhang et al’s [166] findings in regards to PTFE composites in lubricated conditions. The exception to this was fiberglass and MoS2 material which did not show a consistent trend with varied loading.

Results for the average break-away friction over the entire test period of 3 hours are shown in Fig. C.3 and clearly indicate that pure PTFE consistently provided the lowest levels of friction with bronze-filled PTFE also providing low friction. Carbon filled PTFE was found to provide lower friction at higher loads than at lower loads while black-glass filled PTFE and fiberglass filled PTFE provided highest break-away friction of the PTFE composites. All PTFE based materials provided significantly lower friction than Babbitt and these friction levels were generally 25-50% those of Babbitt.

Analysis of the variation in break-away friction over the course of the test cycle provides insight into a material’s behavior in an actual application. In the case of the materials tested in this study, a stark contrast could be seen in the response of the materials to variations in pressure. Figure C.4 shows the difference between the minimum and maximum break-away friction observed for materials during the course of each test. This demonstrates the stability or instability of the materials’ break-away friction when subjected to increased
loading. In the case of carbon-filled and pure PTFE, relatively little difference was observed between the minimum and maximum break-away friction. It was quite a different case for the remainder of the materials which had large levels of variation in break-away friction with changes in loading and no consistent trends in the variation observed during the tests.

### C.4.2 Temperature effects

Results of break-away friction in relation to changes in temperature in the oil bath are displayed in Fig. C.5. It was found that pure PTFE and bronze-filled PTFE produced the lowest friction as temperature increased with carbon-filled PTFE also providing low friction. As in testing with pressure variation, fiberglass filled PTFE yielded higher coefficients of friction than the other PTFE composites while Babbitt provided much higher break-away friction than all of the other materials. Black glass filled PTFE approached carbon filled PTFE at higher temperatures but had equivalent friction to fiberglass filled PTFE at lower temperatures. The fiberglass filled PTFE demonstrated an increasing trend in break-away friction with increasing temperature which was most likely a result of the decreasing lubricant viscosity which could have allowed for the hard fibers to come more into contact with the counter surface. A similar effect is believed to be the cause of the increase in friction observed for
Figure C.4: Variation in break-away friction vs. contact pressure at 25°C. Error bars represent one standard deviation of three repeated tests for each point.

Babbitt with increased temperature. It is believed that the lower viscosity lubricant more readily squeezed from the contact, allowing the Babbitt to have greater area in contact with the counter-surface. The other PTFE composites appeared to have very little change in their average break-away friction values with increased temperature.

When variation in friction was analyzed with increasing temperature, significant trends were observed, Fig. C.6. The fiberglass filled PTFE had inconsistent variation in friction coefficient while bronze-filled PTFE demonstrated the opposite trend with a decrease in variation in break-away friction at increasing temperatures coupled with a decrease in the average friction. Black glass filled PTFE had slightly decreasing variation in friction coefficient with increasing temperature. Variation in break-away friction for carbon filled and pure PTFE at increased temperature was marginal as both materials provided very stable break-away friction levels through the entirety of the test range.

C.4.3 Pressure and temperature interaction

Testing for the interaction between temperature and pressure effects were conducted for all materials except black-glass and fiberglass-filled PTFE at 85°C.
Figure C.5: Break-away friction vs. oil bath temperature at 2 MPa. Error bars represent one standard deviation of three repeated tests for each point.

Figure C.6: Variation in break-away friction vs. oil bath temperature at 2 MPa. Error bars represent one standard deviation of three repeated tests for each point.
and 8 MPa loading. Analysis of variance of these results confirmed the influences of temperature and pressure on the break-away friction for all materials. However, no interaction was found between the effects of changes in temperature and changes in pressure meaning that changes in temperature can be considered to act independently of changes in contact pressure and vice versa.

C.4.4 Prolonged stop and restart

Testing of the effects of an extended stop under load, such as in the practical case of a machine stop, were conducted with Babbitt, carbon-filled, bronze-filled, and pure PTFE. The poor performance of black-glass- and fiberglass-filled PTFE in the earlier testing disqualified these types of fillers from further testing and so they were not included in the time consuming extended stop tests. Results of this series of testing are displayed in Fig. C.7. The value at the '0' cycle is the average break-away friction from the one minute prior to the stop while '1', '2', etc. are the first, second and subsequent cycles after the extended stop.

One of the most notable features of these results is that the maximum friction returns to values measured prior to the prolonged stop after just one stroke, regardless of the material. However, in regards to the materials, it was observed that Babbitt produced a much greater break-away friction after a prolonged stop than the PTFE based materials. The break-away friction observed in the case of Babbitt was also much greater in proportion to the steady state friction measured prior to the prolonged stop. It is felt that the large increase in break-away friction for Babbitt was caused by the Babbitt material conforming to the steel counter-surface and ‘squeezing-out’ the lubricant from the contact. In the case of PTFE materials, reduction of lubricant in the contact is believed to allow PTFE to operate in the ranges of low friction that it is known for in dry contacts while the lubricant that was not ‘squeezed-out’ of the contact could have been sealed under pressure within the contact similar to the mechanism proposed by Persson et al [111] for the case of vehicle tires on asphalt.

C.4.5 Counter-surface wear

The effects the materials had on the steel counter-surface were analyzed using an optical profilometer (WYKO NT1100). Roughness average (Ra) was measured at the same points in the center of the wear track both before and after testing to determine the degree of polishing or roughening caused by the test. These results are displayed in Fig. C.8 showing that only the glass-filled PTFE
based materials had a significant effect on the counter-surface at high load. Error bars in the figure represent the accuracy of the measurement method and error associated with comparison of samples before and after testing. Given the significant surface polishing provided by these materials at the 6 MPa load level it is felt that the trend observed at 1 MPa loading also represents polishing by the glass-based materials even though the difference is very near to the uncertainty of the measurement method. Observations by Iwai et al [75] under constant sliding conditions with similar fiberglass-filled material also found wearing of the counter-surface with the exception that counter-surface roughness increased. The difference in these observations from the current study can be explained by the testing configurations. The reciprocating action of the current tests likely trapped wear particles in the contact region, breaking them into a media which polished both surfaces instead of removing them from the contact as in the block on ring configuration.

This wearing effect made the glass- and fiberglass-filled composites unsuitable for the final application in hydrodynamic bearings and so they were not included in the, time consuming, long-term breakaway testing.

Figure C.7: Break-away friction before and after 72 hour pause of testing under loading at 2 MPa and 25°C. Error bars represent one standard deviation of two repeated tests for each point.
C.4.6 SEM investigation

Further investigations utilizing a scanning electron microscope revealed a number of unique tribological characteristics of the materials under break-away sliding. Overall however, SEM examination further confirmed the experimental findings regarding suitability of materials for the application. The sliding direction in all SEM images is approximately in the vertical direction.

Babbitt

SEM evaluation of the worn Babbitt surface revealed a significant wear scar with grooves cut into the material by the steel counter-surface. A representative image of the worn Babbitt surface is displayed in Fig. C.9. Counter-surface material was not detected on the worn surface signifying that wearing occurred on the Babbitt material only. Similar wear scars were seen on the Babbitt specimens for both high and low loads regardless of temperature.

Black-glass filled PTFE

Investigation of the black-glass filled PTFE revealed the reason for the change in surface roughness observed on the counter-surfaces. The fibers of the black glass had in many locations become sharpened cutting surfaces with collections of iron particles piled up in the direction of sliding. This effect is clearly...
Figure C.9: SEM image of Babbitt tested at 6 MPa and 25°C.

C.4. RESULTS AND DISCUSSION

demonstrated in Fig. C.10 in which the darkest areas are PTFE, the gray areas
are fibers and the white areas were found to be rich in iron. Given that the
PTFE material did not initially contain iron, it can only be assumed that the
iron presence was the result of wearing on the counter surface. In this case,
it appears that the PTFE did very little to reduce the friction in the contact as
large portions of the load appear to have ridden on the harder glass fibers.

Fiberglass and MoS$_2$ filled PTFE

Analysis of the fiberglass and MoS$_2$ filled PTFE revealed large areas of
blended MoS$_2$ and iron as well as sharp fibers. As in the case of black glass,
these sharp fibers tended to collect the other softer materials in the direction of
sliding and produced significant polishing on the counter-surface. This effect
is shown in Fig. C.11. While the MoS$_2$ appeared to have smoothed out the
PTFE surface, it does not appear to have reduced friction in lubricated condi-
tions as the friction levels were generally the same or greater for MoS$_2$ and
fiberglass filled PTFE than black glass filled PTFE.
Figure C.10: SEM image of black glass filled PTFE tested at 8 MPa and 25°C.

Figure C.11: SEM image of MoS$_2$ and fiberglass filled PTFE tested at 8 MPa and 25°C.
C.4. RESULTS AND DISCUSSION

Bronze filled PTFE

The wearing characteristics of bronze filled PTFE were very consistent between high and low pressures and high and low temperatures. In all cases, the bronze particles had scratches in parallel with the sliding motion. However, depending on the load, the bronze particles and the PTFE surrounding them appeared to have worn at different rates. This difference is highlighted by comparison of Fig. C.12 and Fig. C.13 showing bronze filled material tested at 2 and 8 MPa respectively. At 2 MPa, it appears that the surfaces of the worn bronze particles (white regions) and the bulk material (gray) are approximately level with smooth transitions between filler and bulk material. This is not the case at 8 MPa where gaps (black areas) can be distinguished around the edges of the filler particles. Additionally, the bronze particles occupy a much larger area of the contact region than at lower pressure loading. It is thought that the bronze particles are more wear resistant than the PTFE material and so they collect at the surface and are pushed into the bulk material as it is steadily worn away. At some point equilibrium may be reached when the bronze and PTFE wear at equal rates. This equilibrium appears to be pressure dependent as it was observed to occur at 2 MPa and not at 8 MPa.

Figure C.12: SEM image of bronze filled PTFE tested at 2 MPa and 25°C.
Carbon filled PTFE

Carbon filled PTFE had consistent performance at low and high pressures and low and high temperatures. In all cases, the carbon particles wore evenly with the PTFE bulk material as shown in Fig. C.14 in which the darker patches are carbon and the lighter gray areas are PTFE. The primary difference between heavier and lighter loads was observed in the smoothness of the worn surface with higher loads yielding a markedly smoother surface than lighter loads as shown in Fig. C.15. This smoothing at higher loads could have caused the trend of reducing friction with increasing load that was observed with variations in pressure.

Pure PTFE

Pure PTFE showed very similar wear characteristics to carbon filled PTFE with an even smoothing of the surface in general without the complex characteristics seen for the composite materials. As could be expected, counter-surface particles were not observed as in the case of the glass filled materials. The smooth surface seen for pure PTFE could have helped to yield the very low friction coefficient observed for all pressures with PTFE. Figure C.16 shows
Figure C.14: SEM image of carbon filled PTFE tested at 2 MPa and 85°C.

Figure C.15: SEM image of carbon filled PTFE tested at 8 MPa and 25°C.
the smoothing of the PTFE surface at high load and temperature.

C.5 Conclusions

Break-away friction was investigated in reciprocating, sliding, block-on-plate contact in the presence of lubricant for a number of PTFE based materials as well as Babbitt material. The counter surface was mild steel. Sliding speed was kept as low as possible and mean contact pressure was varied from 1 to 8 MPa while oil bath temperature was varied from 25 to 85°C. A summary of the major findings is as follows:

- Highest break-away friction levels were provided by Babbitt and these were considerably higher than those of the PTFE-based materials. Pure PTFE provided the lowest friction levels while carbon- and bronze-filled PTFE also produced low break-away friction.

- Increased loading generally resulted in a slight decrease of break-away friction, except in the case of fiberglass filled PTFE which had no consistent trend.
C.5. CONCLUSIONS

- Pure PTFE and carbon-filled PTFE provided very low variation in friction through changes in both temperature and pressure.

- The break-away friction coefficient with increasing temperature was found to increase for Babbitt and fiberglass-filled PTFE materials while it decreased slightly in the cases of, black-glass-filled and bronze-filled PTFE. Break-away friction was not affected by temperature in the cases of carbon-filled and pure PTFE.

- Contact pressure and temperature were found to influence the break-away friction independently of each other for bronze-filled, carbon-filled, and pure PTFE as well as Babbitt, i.e. there was no significant interaction between temperature and pressure effects.

- Significant polishing of the steel counter-surface was observed in the cases of both black-glass-filled and fiberglass and MoS₂ filled PTFE materials at all pressures. This was clarified by the presence of iron on the PTFE surfaces. Measurable polishing was not observed on the counter-surfaces for the other materials.

- Break-away friction for Babbitt increased greatly following an extended stop before returning to the pre-stop value. Bronze-filled, carbon-filled, and pure PTFE showed a much smaller increase in break-away friction following an extended stop before returning to pre-stop levels.
Paper D

Extending performance limits of turbine oils
D.1 Abstract

New turbine oils providing both extremely high viscosity index (VI) and improved boundary/mixed lubrication performance are investigated. Comparisons are made in both laboratory scale testing using typical journal bearing sliding surfaces (steel and white metal) and full scale testing using a hydrodynamic journal bearing test machine. The results from these studies demonstrate the effectiveness of new, high VI, turbine oils for reducing friction at machine startup and improving performance during full film operation.
D.2 Introduction

Increasing instability in the electrical power supply due to the rapid introduction of new energy sources such as wind and solar has led to an increase in the start and stop frequency of regulating power machines (commonly gas-fired turbines and hydro-electric power plants). These regulating machines are more often being forced to operate in non-optimal ranges. This can lead to higher bearing loads and increased oil temperature which shortens the lifetime of the lubricant and can harm machinery. Further, due to the high startup friction provided by traditional oils and bearing materials, hydraulic jacking systems are often used to provide a lubricating film in thrust and radial bearings during start-up, however this extra system increases the complexity of the machine and the start-up procedure. Lubricants for these machines have traditionally been based on mineral oil and are generally considered hazardous to the environment which can be a significant drawback in renewable energy applications such as hydropower in which machines are placed in direct contact with sensitive ecosystems. However, developments in lubricant technology can potentially solve these issues and at the same time significantly improve machine performance.

Earlier work has investigated reduction of the start-up (break-away) friction through the use of polymer materials [66] finding that significant reductions in the friction at startup could be realized with the use of polymer faced bearings. Texturing the bearing surface has been proposed to reduce sliding friction in the full film [63]. Texturing also helps to retain lubricant in the contact after shutdown and thus can reduce startup friction [93]. Reduction of this start-up friction through lubricant design presents a further method of improving startup characteristics. This could be especially useful for existing machines for which a lubricant change may be more realistic than replacement of the bearings.

While the viscosity index improving characteristics of poly-alkylmethacrylates (PAMA and dPAMA) have long been known [157] a number of recent investigations have found improved lubrication properties using functionalized PAMA lubricant additives which attach to surfaces through polar bonds. Spikes [139] provides a discussion of the effect various forms of boundary friction additives can have on both conformal and non-conformal contacts. Glovnea et al [65] investigated the topic further to include surfaces with realistic roughness, finding that functionalized additives reduced the rolling-sliding friction in contacts with rougher surfaces. Further development of this topic by Müller et al [105] led to a description of molecular characteristics to provide the thick-
D.2. INTRODUCTION

est possible films, namely that the molecules should be large, have functional
groups, and that the functional groups need to be concentrated in the molecule.
Studies were continued with a full range of functionalized additives to further
optimize the additives’ molecular architecture by Fan et al [45] finding that
functionalized block co-polymers provided the thickest boundary films.

Muraki and Nakamura [106] investigated the effects of PAMA additives at
the transition from thin film to sliding friction, observing thicker films at low
speeds with significant shear thinning of the additives as speed increased.

An investigation of the dynamic characteristics of the bonds between fric-
tion modifying additives and surfaces was conducted by Mazuyer et al [95]
finding that the boundary layer deformed elastically and that this deformation
was reversible.

Polymer additives were used by Quinchia et al [120] to successfully ad-
just the base viscosity of high-oleic sunflower oil. Viscosity at 40°C and
100°C were dramatically impacted. However it seems that the additives de-
graded the viscosity/temperature characteristics of the base oil as the viscosity
index of the formulated lubricant decreased with increased additive concen-
trations. Similar investigations were made by Biresaw et al. [9] with bio-
degradable vegetable oils finding that while the oils had very high VI ini-
tially, their viscosity-temperature characteristics degraded with treatment for
increased oxidation stability.

In laboratory tests, power loss reductions and equivalent film thickness
were found to result from changing to a thinner synthetic lubricant in thrust
bearings by New and Schmaus [110] and later by Glavatskih and Larsson [62].
Boehringer and Neff found significant improvements in machine performance
upon changing to a di-ester based lubricant [11] in a full scale hydropower
thrust bearing. Similar improvements in a full scale hydropower machine were
found by Glavatskih [60] upon changing to an ester based lubricant in a com-
bined thrust and journal bearing. Ferguson et al. [48] investigated ISO VG68,
VG46 and VG32 in a large thrust bearing test rig and proposed using a nu-
merical modeling software package to predict maximum bearing temperatures
from oil bath temperatures and thereby predict bearing performance charac-
teristics. Calculated results agreed well with experiments on a large thrust
bearing. Lower power losses were found for lower viscosity grade (VG) and
higher oil bath temperatures. Significant decreases in film thickness were also
observed but it was argued that the lower film thickness was still adequate to
maintain machine safety.

Investigations with journal bearings by Swanson et al [144] in work with
VI improvers found that the thermal performance of mineral-based oils could
be improved to match that of synthetic lubricants. The effects of changing viscosity index was also investigated by Ma and Taylor [91] who found that increasing VI tended to decrease power losses, bearing temperature and film thickness. Separately, Dmochowski [34] and McCarthy et al [97] found similar effects in their studies on the performance of journal bearings with high VI polyalphaolefin and ester based lubricants. However, power loss reductions in journal bearing studies have generally lower magnitude, percentage wise, than those found for thrust bearings and laboratory results have shown lower power savings than field experiences.

In the majority of earlier experimental work inlet or oil bath temperature has been kept constant in order to keep initial operating conditions equivalent for each of the lubricants. Power loss, film thickness and temperature were then found to compare performance of the varying lubricants. Temperatures throughout the bearing were used to calculate lubricant viscosity which was then compared, finding that in the case of the higher VI lubricants, viscosity was equivalent in the region of highest temperature (lowest viscosity) [34, 97]. The findings from McCarthy et al [97] showed that the film thickness was lower for the higher VI lubricants until the speed and load had increased to a point at which the minimum viscosities of the lubricants became equivalent. It was concluded that the reduced power losses provided by the higher VI lubricants were the result of lower bulk lubricant viscosity in the bearing.

High viscosity index lubricants were also investigated in a low speed journal bearing by Kasai et al. [78] finding that high VI lubricants led to reduced bearing friction in some cases and higher maximum oil film pressure when compared to a polyalphaolefin base oil. New forms of VI improvers such as comb polymers developed by Stohr et al. [141, 140] have the potential to provide even greater performance improvements.

While the bearing performance of environmentally adapted lubricants with high concentrations of VI improvers was earlier documented [136], further improvements in additive technology have allowed for the formulation of lubricants with functionalized VI improvers. This allows for a lubricant with even higher VI and the benefits of functionalized PAMA in boundary lubrication. The current study begins by investigating the characteristics of a number of VI improvers for their potential to reduce friction at machine startup. Lubricants are also formulated using selected VI improvers and their performance is evaluated in a full scale journal bearing test rig. Testing in the full scale is accomplished to determine the operational limits of the new lubricants as compared to a standard ISO VG68 mineral oil.
D.3 Experimental study

Characterization of the startup characteristics of very high VI lubricants was conducted using two laboratory scale experimental setups. A reciprocating block on plate arrangement was used to examine the effects of a large number of starts and stops at low frequency. This reciprocating test rig was chosen for the extended portion of the study because of its highly stable and repeatable motion and results. Detailed studies on the stick slip effect at break-away were conducted using a block on disk arrangement to allow for more precise control of the acceleration and pre-loading of the contact. Testing of the lubricants’ performance in full film hydrodynamic conditions was carried out using a full scale journal bearing test rig.

D.3.1 Lubricants

This study began by investigating a variety of lubricant additives for applicability both in the hydrodynamic lubrication regime and at break-away in the boundary regime. The characteristics of the optimum base oil/additive blends were then determined to develop a new generation of lubricants. The lubricants used throughout the study include white oil with high concentrations of VI improvers, mineral oil without VI additives, and synthetic ester with high concentrations of VI improvers.

Studies in break-away were begun with lubricants O, V, and G detailed in the upper portion of Table D.1. From these initial studies, a number of lubricants were formulated with the same white oil to allow for comparison of the additive performance without effects of any viscosity or base oil differences between the lubricants and a further break-away study was conducted using all of the lubricants detailed in Table D.1. PAMA refers to polyalkyl methakrylate, dPAMA refers to dispersant PAMA, and ‘comb polymer’ is described in [141, 140].

Initial studies in the full film regime [136] provided the basis for the aim of reducing power loss in hydrodynamic contacts by increasing lubricant VI. Inclusion of SV22 and SE32, which were formulated with more traditional VI improvers, in the current study allows for comparison of lubricants over a range of VI. HV15 and HV28 were formulated based on the results from the break-away friction testing and to provide desirable viscosity characteristics to improve performance in full film conditions. All of these lubricants are detailed in the lower portion of Table D.1. The synthetic ester (SE) base oils and PAO/synthetic ester blend (PA) used for a number of the lubricants have
Table D.1: Characteristics of tested lubricants. MO = Mineral Oil, SE = Synthetic Ester, PA = 60% PAO and 25% Synthetic Ester blend, WO = White Oil, CP = Comb Polymer, HP = Hydraulic Package.

<table>
<thead>
<tr>
<th>Lubricant</th>
<th>Base Oil</th>
<th>Viscosity (mm²/s)</th>
<th>VI Type</th>
<th>Concentration</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>WO</td>
<td>14.9 3.86 163</td>
<td>dPAMA</td>
<td>15</td>
</tr>
<tr>
<td>B</td>
<td>WO</td>
<td>15.0 4.73 274</td>
<td>dPAMA</td>
<td>19</td>
</tr>
<tr>
<td>C</td>
<td>WO</td>
<td>14.7 5.35 364</td>
<td>dPAMA</td>
<td>9</td>
</tr>
<tr>
<td>D</td>
<td>WO</td>
<td>14.9 5.08 323</td>
<td>dPAMA</td>
<td>14</td>
</tr>
<tr>
<td>E</td>
<td>WO</td>
<td>14.9 5.67 394</td>
<td>PAMA</td>
<td>12</td>
</tr>
<tr>
<td>F</td>
<td>WO</td>
<td>14.9 6.47 475</td>
<td>CP</td>
<td>21</td>
</tr>
<tr>
<td>G</td>
<td>WO</td>
<td>15.3 7.87 581</td>
<td>HP</td>
<td>21</td>
</tr>
<tr>
<td>H</td>
<td>WO</td>
<td>14.9 4.11 195</td>
<td>dPAMA</td>
<td>15</td>
</tr>
<tr>
<td>O</td>
<td>SE</td>
<td>15.5 4.46 226</td>
<td>PAMA</td>
<td>4</td>
</tr>
<tr>
<td>V</td>
<td>MO</td>
<td>14.5 3.26 83</td>
<td>None</td>
<td>–</td>
</tr>
<tr>
<td>Z</td>
<td>WO</td>
<td>14.5 8.16 640</td>
<td>CP</td>
<td>20</td>
</tr>
<tr>
<td>VG68</td>
<td>MO</td>
<td>67.3 8.79 105</td>
<td>None</td>
<td>–</td>
</tr>
<tr>
<td>SV22</td>
<td>PA</td>
<td>20.4 6.81 340</td>
<td>PAMA</td>
<td>15</td>
</tr>
<tr>
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<td>SE</td>
<td>32.1 8.46 259</td>
<td>PAMA</td>
<td>16</td>
</tr>
<tr>
<td>HV15</td>
<td>WO</td>
<td>14.8 8.29 637</td>
<td>CP</td>
<td>20</td>
</tr>
<tr>
<td>HV28</td>
<td>SE</td>
<td>26.8 13.5 519</td>
<td>CP</td>
<td>30</td>
</tr>
</tbody>
</table>
Table D.2: Experimental conditions for reciprocating tests.

<table>
<thead>
<tr>
<th>Condition</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Load</td>
<td>80 and 320 N</td>
</tr>
<tr>
<td>Contact Pressure</td>
<td>2 and 8 MPa</td>
</tr>
<tr>
<td>Oil Bath Temperature</td>
<td>38.2 °C</td>
</tr>
<tr>
<td>Stroke Length</td>
<td>5 mm and 15 mm</td>
</tr>
<tr>
<td>Stroke Duration</td>
<td>5 seconds</td>
</tr>
<tr>
<td>Test Duration</td>
<td>1 hour</td>
</tr>
<tr>
<td>Total Sliding Distance</td>
<td>10.8 and 21.6 m</td>
</tr>
<tr>
<td>Steel Surface Roughness</td>
<td>Ra 0.4</td>
</tr>
<tr>
<td>Block material</td>
<td>AlSn_{40}Cu_{1}</td>
</tr>
</tbody>
</table>

Figure D.1: Test arrangement for reciprocating tests.

over 60% biodegradability in accordance with the OECD 301B test with up to 5% additive, however biodegradability of the formulated lubricants with high concentrations of additives was not investigated. VG68 is a typical ISO VG68 mineral oil included for reference.

D.3.2 Break-away friction

Reciprocating setup

Experiments carried out in the block on plate configuration were accomplished using a reciprocating tribo-meter with the contact submerged in an oil bath as in the earlier work of Golchin et al. [66]. Arrangement of this test is shown
in Fig. D.1. This configuration was chosen to eliminate a converging gap which could lead to hydrodynamic lubrication and to ensure that contact pressure remained constant over the course of each test. The conditions are nearly identical to those in hydrodynamic thrust bearings at start-up with contacting surfaces at about 2 MPa mean load. Likewise, the setup represents the most extreme portion of the conformal cylindrical contact in a journal bearing where local contact pressure can peak over 8 MPa.

The steel counter-surface was fixed in a temperature controlled oil bath while the block specimen was fastened to the reciprocating arm. Blocks were cut from a journal bearing sleeve and consisted of a steel backing with an aluminium-tin (AlSn40Cu1) sliding surface having minimum thickness 0.5 mm. Reciprocating motion was provided by an eccentric transmission driven by an electric motor which was geared down through a gearbox. Loading was accomplished via a spring connected to the reciprocating arm and was varied to provide specified contact pressures. Friction force measurements were taken continuously using a piezoelectric load cell. A full description of the experimental conditions is detailed in Table D.2. Testing was fully randomized and tests were repeated at least four times each to ensure repeatability and minimize statistical uncertainty.

Break-away friction in reciprocating tests was determined from the friction load data stream provided by the test rig. The value of break-away friction is taken from the maximum value when the reciprocating test specimen stops
and changes sliding direction at the end/beginning of each stroke. Figure D.2 gives an example of the raw data stream from typical tests of three of the tested lubricants showing a start and stop cycle with break away at the beginning and stopping friction at the end of the stroke.

Temperature was held at 38.2°C to mimic conditions in the oil baths of large bearings as well as to keep the lubricants’ viscosity as equal as possible, thus eliminating the effects variations in lubricant viscosity may have had on the contact. A full set of tests was conducted for all oils using the short stroke length (5 mm). Only lubricants G, V, and O were additionally tested using a long stroke length (15 mm) as they represent the three extremes of the lubricant and additive combinations used in this study.

**Block on disk setup**

To further understand the friction effect at break-away a second test arrangement was developed using a pin on disk test setup as shown in Fig. D.3. This arrangement utilized the same geometry as the earlier reciprocating tests with the primary difference being that the flat plate was replaced with a disk whose rotation was controlled using a servo motor coupled to a planetary gearbox. Rotating the disk at extremely low speeds allowed for examination of the dynamic effects which occur at break-away and further clarified the performance differences between the lubricants.

The block on disk arrangement also allowed for a somewhat more realistic breakaway than the reciprocating test arrangement by allowing full control of
the extent of the stop and acceleration times. A testing cycle consisted of the block being engaged and loaded against the disk to the desired load then left idle for 1 minute before rotation was started. The speed of rotation was then increased to 5 RPM (0.02 m/s sliding speed) before being slowed down to a stop. After stopping, the block was disengaged from the disk, the disk was rotated to flush the contact with lubricant and the cycle was started again. To determine the lubricant response to long periods under load such as in the case of a machine stop, the idle time was extended from 1 minute to 3 hours. Loads during testing were varied between 49 and 392 N (approximately 1.2 to 9.8 MPa mean contact pressure). All testing in the block on disk configuration was conducted at room temperature (approximately 23°C).

D.3.3 Hydrodynamic Lubrication

Testing carried out in the hydrodynamic regime aimed to simulate the conditions in journal bearings used for electrical power generating machinery such as gas, steam, and hydropower turbines with high sliding speeds and constant loads. This work was conducted using the full scale journal bearing test machine described in detail in [136]. This test equipment features a 180 mm diameter shaft coupled to a motor capable of providing journal speeds up to 6000 RPM (56 m/s surface speed). The maximum test bearing load is 140 kN equating to 10 MPa for a 0.4 L/D bearing. For safety, maximum bearing surface temperatures are limited to approximately 120°C which limits the maximum speed and mean load in the current study to 3500 RPM (33 m/s) and
2 MPa. Speed is incremented in steps of 500 RPM from 1000 RPM to 3500 RPM. Loading is accomplished through the use of an air cushion and electronic pressure regulator and is applied at three levels providing 1 MPa, 1.5 MPa and 2 MPa mean pressure on the bearing. Load measurement is accomplished with four load cells. Eight inductive displacement sensors are mounted on the bearing housing (four per side) to determine relative eccentricity.

Instrumentation installed onboard the shaft includes pressure transducers to provide continuous film pressure at the bearing centerline, 10 mm from the centerline, and 20 mm from the centerline. Due to geometrical constraints, film pressure sensors are arranged with two sensors on one side of the shaft and one mounted on the opposite side, NTC (negative temperature coefficient) thermistors are installed near to the pressure sensors to measure the shaft temperature. NTC thermistors are also installed in the bearing shell (in contact with the oil film) to measure oil film temperature. A number of type K thermocouples are used to measure temperature throughout the oil supply system and cooling system. The arrangement of these sensors is displayed in Fig. D.4. Power losses are determined from the motor’s power demands. While power loss could also be determined through the thermal characteristics of the cooling system it is not done so in this work as some of the oil characteristics (e.g. specific heat vs. temperature) as well as the thermal characteristics of the shaft and support bearing system were uncertain. The test bearing installed during this study is a 130 mm long two axial groove bearing with a radial clearance of 160 μm.

Oil inlet temperature was initially maintained at 40°C with a flow rate of 2 liters/minute for speeds up to 24 m/s (2500 RPM) and 3 liters/minute for speeds over 24 m/s. Oil inlet temperature was later adjusted to control the maximum oil temperature and thereby the minimum oil viscosity in the bearing to control bearing eccentricity. When controlling inlet temperature the coldest achievable inlet oil temperature was 14°C.

D.4 Results

D.4.1 Break-away studies

The results of the testing program revealed the dramatically varying effects that new lubricants can have on the break-away friction. The effects of contact pressure, acceleration rate and sliding speed on break-away and slow speed sliding performance of the lubricants were also investigated. Additionally, the effect of extended idle periods under loading was examined.
Figure D.5: Comparison of breakaway friction for short and long stroke length

Initial work with oils G, O, and V showed that break-away friction was 30% lower with G than for O and V which provided approximately equivalent friction as shown in Fig. D.5. The additives used in G seemed to have a major effect on the break-away friction not seen in the lubricant without additives (V). The synthetic ester with high VI (O) generally behaved more like the reference oil (V) than G, but was less sensitive to changes in acceleration rate. As O also has a very high concentration of VI improvers, it can be concluded that the specific type of VI improvers affects the break-away friction in this type of contact.

Acceleration rate was not found to have an effect on the break-away friction for oil G, but seemed to affect the friction in the case of O and V. Changes in acceleration increased the dynamic loading at breakaway because the faster acceleration rate provided less time to build up the contact stress needed to cause a break-away. This increased dynamic load seems to have primarily affected the base oil (V) with the other lubricants not showing a significant response to change in stroke length. The dramatic change in friction for V at high load is thought to be the result of a combination of V’s tendency to stick-slip at break-away and the dynamics of the testing apparatus. This uncertainty led to the further studies using the block on disc configuration.

Testing break-away using the block on disc arrangement allowed for the use of extremely slow acceleration rates as well as the ability to maintain a constant, yet very slow, sliding speed to initiate stick-slip behavior. V and O not only provided much higher friction at break-away, but they also led to significant stick slip as shown in Fig. D.6 and D.7. A small degree of stick slip was observed with G as can be seen following the initial break-away in Fig. D.8. With increasing sliding speed G provides a constant sliding friction while
Figure D.6: Stick slip behavior of V (mineral base oil) showing one cycle of acceleration to 0.02 m/s.

Figure D.7: Stick slip behavior of O (synthetic ester with VI improvers) showing one cycle of acceleration to 0.02 m/s.
Figure D.8: Stick slip behavior of G (white oil with VI improvers) showing one cycle of acceleration to 0.02 m/s.

V and O continue to have significant stick-slip.

Significant qualitative differences between the lubricants associated with their stick-slip behavior were also observed. O produced a large amount of noise and vibration through the tests while V produced slightly less noise and the contact with G was silent following each loading cycle’s initial break-away with no noticeable vibration. A dry contact produced some noise at a higher frequency than the lubricated tests and all tests produced a 'popping noise' during the initial release of the contact caused by the inherent dynamic loading of the test apparatus.

Figure D.9: Breakaway friction for all lubricants tested.
Based on the initial results, several further formulations were developed to match the characteristics of the lubricants from the initial tests with a final additive and lubricant combination. The results from the complete study are shown in Fig. D.9. Many of the additives provided very consistent friction compared to lubricants V and O, but in only some cases was the friction reduced significantly. Of note are lubricants C, D and F which while providing very little variability in friction did not effectively reduce friction in the contact. Lubricant A led to dramatically reduced friction in the contact but had low viscosity index compared to the other lubricants because the additives used to increase the VI in A are normally used as friction modifiers. Lubricant G also had very low friction in addition to an extremely high VI, however it was uncertain whether it was stable enough to safely operate in the turbine application where temperatures can be well over 100° C.

Lubricants H and Z were blended to try to combine VI boosting and low friction characteristics. They successfully reduced the break-away friction, albeit not as dramatically as in the case of A or G. For lubricant Z, the VI was also increased to an extreme level. The large difference in VI between H and Z had little effect on the friction which highlights the importance of the specific structure of the additives in reducing friction.

D.4.2 Extended stopping

The break-away friction results for extended idle time are displayed in Fig. D.10. The benefits provided by G in earlier testing were not observed at the highest loads after 3 hours. However at lower loads, G continued to provide a lower breakaway friction than O and V following an idle period. Lubricant G appeared to provide a tribofilm that remained in the contact at much higher pressure than the other lubricants. The base oil (V) and the synthetic ester (O) on the other hand do not generate the same low friction tribofilm in the contact resulting in friction levels close to those seen in dry contacts.

D.4.3 Hydrodynamic lubrication

Studies in the hydrodynamic lubrication regime highlighted the large differences in performance provided by the viscosity index improved lubricants as well as the limits of high VI / low base viscosity lubricants. Power losses were reduced for nearly all cases of synthetic lubricants compared to VG68. These power loss savings resulted in lower heat generation within the bearing and thereby lower temperatures within the oil film. The slight step that can be ob-
served in most of the figures between 2500 and 3000 RPM is due to the change in lubricant flow rate between these two speeds. The increased lubricant flow rate was necessary to keep the bearing from becoming starved, but led to increased cooling. The datasets are slightly shifted from each other along the speed axis in all figures regarding hydrodynamic studies to better differentiate the datasets and their error bars.

Power loss of the lubricants as a percentage of that for VG68 is displayed in Fig. D.11 and D.12. It can clearly be observed that SE32 and SV22 have much lower power loss than VG68 at all speeds with the greatest percentage difference at low speeds and loads. HV15 provided dramatically lower power loss compared to that with VG68, however this is to be expected given its very low viscosity at lower temperatures. HV28 provided much lower power loss than VG68 at lower speeds with slightly lower power loss at higher speeds. The differences in power loss can to some degree be explained by the corresponding differences in maximum measured lubricant viscosity (calculated from minimum bearing temperature) in the bearing as shown in Fig. D.13 and D.14. Shearing a thinner lubricant in the bulk of the bearing should result in lower power loss than shearing a thicker lubricant through the same geometry.

Shaft eccentricity for all cases is displayed in Fig. D.15 and D.16. Values for eccentricity cannot be considered as absolute due to uncertainty in the effect of the combination of shaft and housing thermal expansion. However comparisons at individual speeds and loads can be made as the difference in temperature between the shaft and housing varied very little between lubricants at any specific speed and load. SE32 generally provides lower eccentricity than the thinnest lubricants at 1 MPa, but does not provide the thick films seen with the extremely high VI lubricants. At all loads and speeds, SE32 had slightly
higher eccentricity than VG68, and at the highest speed, SE32 had slightly higher eccentricity than SV22. SV22 generally provided slightly higher eccentricity than SE32 at low speeds and loads and slightly lower eccentricity than SE32 at higher speeds and loads. In comparison to VG68, SV22 was not able to provide equivalent eccentricity.

Examining the extremely high VI lubricants, HV28 provided equivalent eccentricity to that with VG68 in all cases with 1 MPa mean load. For the heaviest load, 2 MPa, HV28 provided slightly higher eccentricity than VG68. HV15 always had the highest eccentricity. This is to be expected from its generally lower minimum viscosity at all testing speeds.

The differences in eccentricity are to some degree explained by a comparison of the viscosity in the most heavily loaded part of the bearing at low speeds such that lubricants with higher minimum viscosity should provide lower eccentricity. However, at higher speeds the minimum viscosity and eccentricity do not correlate as well. Minimum viscosity for the lubricants calculated from the maximum temperature is provided in Fig. D.17 and D.18. The minimum viscosity in all cases is calculated from the maximum measured temperature in the bearing shown in Fig. D.19 and D.20.
Figure D.12: Power loss for lubricants at 2 MPa.

Figure D.13: Maximum viscosity in the bearing at 1 MPa.
Figure D.14: Maximum viscosity in the bearing at 2 MPa.

Figure D.15: Shaft eccentricity at 1 MPa
Figure D.16: Shaft eccentricity at 2 MPa

Figure D.17: Minimum viscosity in the bearing at 1 MPa.
D.4. RESULTS

Figure D.18: Minimum viscosity in the bearing at 2 MPa.

Figure D.19: Maximum temperature in the journal bearing at 1 MPa mean load.
D.5 Discussion

While it is clear that newer VI improving additives can provide reduced friction in boundary lubricated contacts, the application of high concentrations of these same VI improvers to lubricants used primarily in the hydrodynamic regime leads to a slightly more complex situation.

Upon first examination, a lubricant with maximum VI and minimum base oil viscosity would seem to provide optimum performance. However, such high VI stiffens the lubricant’s thermal response which is beneficial for hydraulic applications requiring consistent viscosity over a wide temperature range. Power generating machinery, in contrast, has a much narrower operational temperature window and requires a specific film thickness at a controlled temperature.

At low speeds the high VI lubricants are, in most cases, simply not thick enough to provide equal film thickness to VG68. In Fig. D.17 it is observed that the minimum viscosity for VG68 is double that of HV15 at 1000 RPM. The maximum viscosity for VG68 is nearly triple that of HV15 for the same conditions in Fig. D.13. These stark differences in viscosity are reflected in
both the differences in eccentricity (Fig. D.15) and power loss (Fig. D.11) where the thick film provided by VG68 is paid for with a 40% difference in power loss.

The higher viscosity at higher temperatures of high VI lubricants as exemplified by HV28 in Fig. D.17 is somewhat misleading. From the minimum viscosity of 14 mm$^2$/s one would expect a much thicker oil film than for VG68 which has a viscosity below 8 mm$^2$/s in the same conditions. Instead, VG68 and HV28 provide equal eccentricity with only a small difference in power loss. This could be partly due to viscosity variation within the hydrodynamic pressure zone caused by the temperature gradients. Since the shaft eccentricity is affected by the three dimensional viscosity field, an equivalent viscosity value should be used to compare oil performance. Another factor which appears to be contributing to the higher than expected eccentricity of HV28 and HV15 at high speeds is temporary shear thinning of the additive formulations as described by Muraki and Nakamura [106].

Thus increasing a lubricant’s VI narrows the optimum speed and load range for which it can provide acceptable film thickness and reduced power loss. Outside of this range the lubricant is either unable to provide the same film thickness due to its low viscosity or yields greater power loss, due to the thicker film that it develops compared to traditional lubricants. Fortunately most machines used for electrical power generation operate at constant speed with relatively constant bearing load, which helps to eliminate the need for a great range of flexibility in the lubricant.

If necessary, some flexibility in the application ranges of the lubricants can be provided by adjustment of the oil inlet temperature. This flexibility however becomes more limited as the lubricant’s VI is increased. Additionally, the limitations of oil inlet temperature are governed by the machine’s location and access to effective cooling. Even so, in the case of SV22, the ingoing oil to the bearing housing was cooled to 14°C using water at 8°C to provide matching viscosity to VG68 at the lowest speed and highest loads. Matching the minimum viscosity of HV15 and VG68 at low speed was not possible. Clearly such cooling capabilities are not practical for many machines thus choosing a lubricant with higher base viscosity such as SE32 or HV28 would be necessary.

However, it can be argued that a slightly thinner oil film may be worth trading for the potential 15% reduction in power loss that could be achieved by operating with SE32 or 40% reduction with HV15. Furthermore, the boundary friction reducing properties of the additives in both HV15 and HV28 can help to protect the bearing and shaft even with a very thin oil film.
D.6 Conclusion

This work has investigated the application of high VI lubricants to hydrodynamic journal bearings to improve performance through the entire range of operation from startup to the full film regime. From these experimental studies, the following conclusions can be drawn:

- The new high VI lubricants significantly reduce friction at break-away compared to more traditional lubricants.
- The new lubricants provide a significant reduction in stick slip during low speed sliding.
- In the full film hydrodynamic regime, the new high VI lubricants deliver equivalent performance to traditional mineral oil based lubricants with the benefit of reduced power loss.
- Increasing the lubricant VI narrows the breadth of the optimum bearing sliding speed and load for a lubricant compared with traditional mineral oil based lubricants in journal bearing applications.
- Correct application of new lubricant technology provides significantly improved performance in journal bearings with base oils that are much less harmful to the natural environment than those in traditional lubricants.

D.7 Acknowledgments

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Paper E

Dynamic characteristics of polymer faced tilting pad journal bearings
Dynamic characteristics of polymer faced tilting pad journal bearings

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E.1 Abstract

Dynamic characteristics of polymer faced tilting pad journal bearings are presented. Investigations are conducted using a single pad, load on pad configuration over a range of shaft speeds and loads. Two polyether ether ketone (PEEK) faced pads, one polytetrafluoroethylene (PTFE) faced pad and two entirely PEEK pads are investigated to determine the effect of varying mean bearing pressure and pivot characteristics as well as different material properties of the polymer layer. Experimental results indicate increased damping and decreased stiffness and slightly increased oil film temperature for entirely PEEK pads compared to pads with a PEEK lining and steel backing. Similar effects were observed by using a softer (PTFE) pad liner with a steel backing.
E.2 Introduction

Changing demands on the electrical network and growing supply of un-regulated power sources such as wind and solar have resulted in a change of roles for large power generating machines. Machines in hydro-electric and thermal power plants which were originally designed to provide a steady base load are now being operated to fulfill grid regulation and peak power needs. This has resulted in a dramatic increase in the number of starts and stops which machines undergo as well as an increase in operation at non-ideal operating states. Both starting/stopping and non-ideal operation contribute significantly to the wear and tear of machines with components that come into contact with each other such as bearings and seals taking the brunt of the wear [43]. Furthermore, with bearing failures estimated to cause 40% of a hydropower plant’s operating losses [3], it is clear that the current technology is inadequate for the markets demands.

Current bearing technology consists primarily of white metal faced sliding bearings, technology initially developed at the beginning of the 20th century but little changed since the 1970’s. Polymer faced journal bearings have been investigated for use primarily in slowly rotating or oscillating applications by Ukonsaari [155] and Gawarkiewicz and Wasilczuk [57]. While these studies included temperature measurements, the contacts were almost always in boundary lubrication, far from the hydrodynamic lubrication regime.

Studies conducted by Fleszar [51] investigated a number of bearing liner materials for use in sliding applications finding little performance difference between woven and molded liners made of phenol resins with PTFE additives. Melting temperatures for all materials tested were over 240°C. Phenol resin based materials with asbestos and carbon fiber reinforcement were investigated by Kim, Lee and Park [79] and [88] for use in marine propeller shaft bearings. These studies found that carbon fiber provided less wear than asbestos and additionally proposed a method of manufacturing a polymer sleeve bearing with a 4 mm sleeve thickness. Attachment of the sleeve to the backing ring was not discussed.

Hydrodynamic sliding bearings utilizing virgin PTFE bonded using a copper mesh have been used in Russia and China for some time, Glavatskih [58], but detailed laboratory studies have only been conducted on thrust bearings utilizing this technology. Reports from industrial installation of these types of bearings have found improved machine operating characteristics as well as potential for reductions of bearing size, Glavatskih et al [64]. Simulations of such bearings have predicted slightly higher lubricant temperatures, Kuznetsov et al
E.3 EXPERIMENTAL SETUP

[86], as could be expected by the lower thermal conductivity of polymers in comparison to metals.

Recent reports of installed PTFE faced bearings, Simmons et al [137], Et- tles et al [44] and Dwyer-Joyce et al [39] report good performance of plain PTFE faced thrust bearings however very little has been reported regarding PTFE faced journal bearings in hydrodynamic applications. However, modeling of compliant lined bearings was accomplished by Thomsen [151].

The dynamics of tilting pad journal bearings are generally well understood with a large body of work covering bearing design and modeling as well as experimental results. The body of this work is summarized by Dimond et al. [30]. Other recent studies of horizontal journal bearings have focused on dynamics of the tilting pads [31] and controlling the pads’ dynamic response [130]. Recent numerical studies of tilting pad journal bearings by Fillon, Dmochowski and Dadouche have focused on the effects that manufacturing and assembly tolerances can have on the bearing performance [49, 33].

The vast majority of research work and experimental results covering bearing dynamics is focused on bearings with white metal or other bronze based surface material. Research on polymer faced bearings, which have been used successfully in both Russia and China for several decades, has not been prioritized by the major bearing research groups and therefore the bearing characteristics needed for designing safe systems are also somewhat unknown. Thus the aim of this work is to investigate the dynamic characteristics of polymer faced and entirely polymer tilting pad journal bearings.

E.3 Experimental setup

The test equipment consists of a stationary two pad bearing housing with a shaft which can be loaded vertically using a block and tackle system as shown in Fig. E.1. Dynamic excitation is accomplished using an electromagnetic shaker in the vertical direction. A full description of the testing equipment is provided in [21]. A range of base and dynamic loads were utilized to fully characterize the polymer faced pads. Testing was conducted over a range of speeds from 500 to 2500 rpm in 500 rpm increments. Two loads were used during dynamic testing, 3000 N and 4800 N which generated mean pressure of 0.5 and 0.9 MPa for the complete pads and 1.1 and 1.8 MPa for the pads with smaller surface area. Dynamic excitation was accomplished using two different frequency sweeps. A sweep from 0 to 50 Hz was used to identify the stiffness at lower frequency excitation and a 0 to 300 Hz sweep was used to
identify the damping characteristics at the system’s natural frequency.

The test rig is equipped with one load cell to measure the static and dynamic load on the bearing and one load cell to measure the dynamic impulse provided to the bearing by the shaker. The displacement of the shaft is measured using an inductive displacement sensor while temperature in the pads is monitored at 4 points per pad using type K thermocouples. Shaft speed is set using the motor controller.

Three different pad material configurations were tested with two different surface geometries. This made for a total of six sets of polymer face pads. The basic geometry of the pads is shown in Fig. E.2. Following testing of the initial set of pads (with 100% of their surface area) a second full set of pads with approximately 40% of the surface area of the original set were also tested. Due to geometric constraints, bearing pads with 40% surface area had
E.4. DYNAMIC CHARACTERIZATION

Figure E.2: Temperature sensor arrangement and layout of bearing pads.

only 2 temperature sensors per pad. The three material configurations used were 100% PEEK, a layer of PEEK with a steel backing, and a layer of PTFE with a steel backing. To simplify characterisation and modeling of the pads, compliant liners are attached to their steel backings using machine screws. These three combinations allowed for the study of a range of pad compliancy. Further detail of the bearing geometry is provided in Table E.1.

E.4 Dynamic characterization

Defining an experimental identification procedure for the dynamic coefficients of the tilting pad bearing begins by setting up the equation of motion for the system. The test rig can be represented by an equivalent mechanical system depicted in Fig. E.3. It consists of a rotating journal supported in a rigid lever arm, with a pivot at one end. The journal is normally supported by a two pad, load on pad, tilting pad journal bearing arrangement, however to simplify the system, only the lower pad is used in this work. A harmonic excitation force \( f(t) \) with frequency \( \omega \) is applied at the free end of the arm to perform the identification procedure. The oil film forces generated in the bearing are linearized around the static equilibrium position and represented by reduced equivalent stiffness \( K \) and damping \( C \) coefficients. These in general are a function of the excitation frequency \( \omega \). The dynamic behavior of this equivalent mechanical system can be described by a single degree of freedom \( y(t) \), describing the vertical displacement of the free end of the lever arm, measured from the static equilibrium position. The corresponding displacement of the journal inside the tilting pad bearing arrangement can then be related linearly to the displacement
Reduction of the original system to the equivalent single degree of freedom system described above entails the introduction of some simplification assumptions. Firstly, in order to assume $y(t)$ as the only coordinate required to describe the position of the equivalent system, the lever arm must behave as a rigid body. This assumption is fulfilled if none of the arm flexible modes are excited by the excitation force $f(t)$. Thus, this assumption will be valid within a certain frequency range of the excitation force. Secondly, in order to directly relate the displacements at the arm end with the journal vertical displacement, the rotation of the lever arm around its pivot point must be small. This condition is fulfilled considering that the maximum measured motion at the end of the 500mm long lever arm during any of the loading cycles was less than 60 μm. Thirdly, the representation of the bearing oil film forces using equivalent stiffness and damping coefficient entails neglecting any contribution from the oil film to the inertia of the system. This condition is respected if the bearing operates within the laminar regime. Finally, linearization of the oil film forces using the equivalent dynamic coefficients is valid if the journal vertical displacements are small enough compared to the bearing clearance. Typically, the literature suggests keeping the journal displacements below 20% of the bearing radial clearance in order to keep the oil film force linearization a valid approach. The original two pad design was designed for a 300 μm clearance which allowed for a maximum motion of 60 μm to maintain linearity. This limit of 60 μm was met in testing with a single pad.

Considering the above assumptions, the equation of motion for the reduced mechanical system can be given by:

![Figure E.3: Test arrangement model system.](image)
E.4. DYNAMIC CHARACTERIZATION

\[ \hat{M} \ddot{y} + \hat{C} \dot{y} + \hat{K} y = f(t) \]

\[ \hat{M} = \frac{I}{l_2^2} \]

\[ \hat{C} = C(\omega) \left( \frac{l_1}{l_2} \right)^2 \]

\[ \hat{K} = K(\omega) \left( \frac{l_1}{l_2} \right)^2 + K_{\text{static}} \]

(E.1)

In Eq. (E.1), \( I \) is the combined inertia of the arm and journal, related to its rotation around the arm lever pivot point, and \( l_1, l_2 \) is the position of the journal center and free arm end, measured with respect to the lever arm pivot point. The forcing term \( f(t) = F_0 e^{i \omega t} \) corresponds to a harmonic excitation, and because linear behavior is being assumed for the system, the system response can be expressed as \( y(t) = Y_0 e^{i \omega t} \). Thus, Eq. (E.1) can be written in the frequency domain as follows:

\[ \frac{Y_0}{F_0} = \frac{1}{\hat{H}(\omega) - \hat{M} \omega^2} \]

\[ \hat{H}(\omega) = \hat{K}(\omega) + i \omega \hat{C}(\omega) \]

(E.2)

In Eq. (E.2), the function \( \frac{Y_0}{F_0} \) must be determined experimentally. Since such relationship corresponds to the frequency response function of the system, it can be determined as follows:

\[ \frac{Y_0}{F_0} = FRF(\omega) = \frac{S_{yy}(\omega)}{S_{yf}(\omega)} \]

(E.3)

In Eq. (E.3), \( S_{yy} \) corresponds to the power spectral density of the measured displacement \( y(t) \), whereas \( S_{yf} \) corresponds to the cross-power spectral density of the applied force \( f(t) \) relative to the displacement \( y(t) \). Consequently, the bearing reduced dynamic coefficients can be calculated from:
E.5 Uncertainty

Uncertainty in the experimental results comes from the measurement uncertainty of the equipment as well as the accuracy of the geometry of the machined parts. Uncertainty of the measurements and sensors is detailed in Table E.2. To further quantify the error in the experiments and calculations of stiffness and damping, 3 rounds of testing were conducted with each pad. For each test round the pad was re-mounted in the test rig to include any variation resulting from alignment and the assembly procedure.

The other source of uncertainty in the experimental determination of the bearing pad dynamic characteristics is related to the machining of the bearing pads. All efforts were made to produce different pad configurations with as near to exactly the same dimensions as possible. However, due to the different material characteristics of the pad configurations and the varying response of PEEK and PTFE to machining, some variation in geometry was inevitable. This variation was most obvious for the PTFE surfaced pads due to the difficulty of accurately machining PTFE. Measurements of the diameter of curvature of the pad sliding surfaces and shaft are provided in Table E.3 together with the measured tolerance of the curvature. The tolerance of the diameter varies for the different pads due to settling and deformation of the material both during and after machining.
Table E.2: Measurement uncertainty.

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
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</thead>
<tbody>
<tr>
<td>Temperature</td>
<td>±0.5 °C</td>
</tr>
<tr>
<td>Static Load</td>
<td>±50 N</td>
</tr>
<tr>
<td>Dynamic Load</td>
<td>±5 N</td>
</tr>
<tr>
<td>Displacement</td>
<td>±3 μm</td>
</tr>
</tbody>
</table>

Table E.3: Diameter of pad sliding surface.

<table>
<thead>
<tr>
<th>Pad</th>
<th>Curvature Diameter (mm)</th>
<th>Tolerance Axial (mm)</th>
<th>Tolerance Roundness (mm)</th>
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</thead>
<tbody>
<tr>
<td>PEEK/Steel</td>
<td>100.047</td>
<td>±0.063</td>
<td>±0.005</td>
</tr>
<tr>
<td>Small PEEK/Steel</td>
<td>100.020</td>
<td>±0.08</td>
<td>±0.001</td>
</tr>
<tr>
<td>PTFE/Steel</td>
<td>100.111</td>
<td>±0.184</td>
<td>±0.0301</td>
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<tr>
<td>PEEK</td>
<td>99.885</td>
<td>±0.019</td>
<td>±0.001</td>
</tr>
<tr>
<td>Small PEEK</td>
<td>99.880</td>
<td>±0.075</td>
<td>±0.001</td>
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</tbody>
</table>
Table E.4: Oil temperature along pad at 2500 RPM and 2.4 kN load.

<table>
<thead>
<tr>
<th>Pad</th>
<th>5°</th>
<th>25°</th>
<th>45°</th>
<th>65°</th>
</tr>
</thead>
<tbody>
<tr>
<td>PEEK/Steel</td>
<td>41.6</td>
<td>43.8</td>
<td>46.1</td>
<td>45.1</td>
</tr>
<tr>
<td>Small PEEK/Steel</td>
<td>38.8</td>
<td>44.4</td>
<td>47.8</td>
<td>38.5</td>
</tr>
<tr>
<td>PTFE/Steel</td>
<td>42.0</td>
<td>44.3</td>
<td>46.7</td>
<td>47.2</td>
</tr>
<tr>
<td>PEEK</td>
<td>43.8</td>
<td>46.1</td>
<td>48.8</td>
<td>45.4</td>
</tr>
<tr>
<td>Small PEEK</td>
<td>32.1</td>
<td>46.6</td>
<td>50.6</td>
<td>26.8</td>
</tr>
</tbody>
</table>

### E.6 Results

Experimental work on the bearing pads in both steady state conditions and with a dynamic load provided insight into the behavior of the various geometries and pad configurations. To clarify the uncertainty in stiffness and damping figures, all figures with results detail the mean of three tests plus and minus one standard deviation.

#### E.6.1 Steady state

Oil film temperature varied slightly between the different bearing pads. However, the general trend shows higher temperatures for the pads with smaller surface area and entirely made of polymer. Interesting to note is the decrease in temperature at the outlet of the fully PEEK pad. It is believed that this pad deformed around its pivot point. Also of note is that the temperatures given for the inlet and outlet of the small PEEK and small PEEK/Steel pad are outside of the pad’s load bearing surface. They are included as they give a good indication of the conditions before the inlet and after the outlet for these cases.

#### E.6.2 Dynamics

Investigation of the dynamic characteristics of the pads was begun by checking the coherence of the force and displacement of the pad. Coherence for the two different frequency sweeps is provided in Fig. E.4 and Fig. E.5. Coherence for the low frequency sweep (Fig. E.4) was much better for 500 rpm tests than the higher speed tests at 1500 and 2500 rpm, mostly as a result of the loss of coherence at the rotating frequency and the overtones of the rotating frequency. This loss of coherence is clearly observed in Fig. E.5 where the 2500 rpm test
Figure E.4: Coherence function of $F$ and $y$ for low frequency range with PEEK/Steel pad and 1500 N load.

Figure E.5: Coherence function of $F$ and $y$ for high frequency range with PEEK/Steel pad and 2400 N load.
Figure E.6: Magnitude and phase of response function for high frequency range with PEEK/Steel pad and 2400 N load.

(approximately 42 Hz) has poor coherence at the first three overtones within the sweep range (84 Hz, 126 Hz, and 168 Hz). The same effect occurs for in the 500 rpm tests where poor coherence occurred at approximately 8 Hz, 16 Hz, and 33 Hz.

Examples of the system response for the lowest and highest speed are provided in Fig. E.6. Coherence is provided in Fig. E.5. The system natural frequency appears to be in the range of 225 Hz when the phase of the response shifts. As the frequency approaches the maximum excitation frequency of 300 Hz, the coherence begins to decrease.

The frequency sweep to 50 Hz helped to provide higher resolution at the low frequencies which most hydropower machines operate under. Comparing the responses of the two pad configurations with small surface area (small PEEK and small PEEK/Steel) demonstrates the large difference in stiffness provided by the steel backing compared with the PEEK backing. Using a steel backing greatly increases the stiffness of the pad at lower excitation frequencies with no significant difference between low speed operation (Fig. E.7) and high speed operation (Fig. E.8).

This difference in stiffness, while present, was not as dramatic for the pads
Figure E.7: Stiffness for low frequency range at 500 rpm and 2400 N load.

Figure E.8: Stiffness for low frequency range at 2500 rpm and 2400 N load.
Figure E.9: Stiffness for low frequency range at 500 rpm and 2400 N load.

with the full surface area at low frequencies as shown in Fig. E.9 and Fig. E.10. The stiffest configuration, PEEK/Steel, is only somewhat stiffer than the other two configurations which provide equivalent stiffness. Of note in the low frequency sweep is the unavoidable loss of coherence near to the rotating frequency.

Examining the results for excitation from 0 to 300 Hz, the stiffness generally increased with increasing frequency. The difference between small PEEK and small PEEK/Steel was nearly constant through the frequency sweep at both low speed (Fig. E.11) and high speed (Fig. E.12). Similar trends were observed for the large pads with PEEK/Steel being the stiffest and PTFE/Steel and PEEK having lower stiffness both at low speed (Fig. E.15) and high speed (Fig. E.16). In the higher frequency sweep the PEEK pad may be slightly softer than the PTFE/Steel pad, but the difference in this case lies within the uncertainty of the experiment.

Damping characteristics of the different pads were obtained at the natural frequency of the system at around 240 Hz. Of note is the shift in natural frequency provided by the pads of varying stiffness. The small pads (Fig. E.13 and Fig. E.14) both provided damping that was generally equivalent to that provided by the larger pads. Notably, the small PEEK pad provided slightly
E.7 Discussion

Many theoretical investigations have been made into the use of polymer faced sliding bearings in the full film hydrodynamic regime, however, few experimental studies have been performed. This is especially the case regarding dynamic response of journal bearings. From this initial study a number of important points regarding polymer faced bearings should be brought to light.

The deformation and compression of the bearing’s surface and pivot seem to affect both the stiffness and damping characteristics of the bearing. Furthermore, manufacturing compliant materials such as PTFE with acceptably fine tolerances and correct critical dimensions poses many challenges in and of itself. These challenges are magnified at smaller scale as the tolerances become greater damping than the small PEEK/Steel pad. A similar trend was observed for the large pads wherein PEEK provided slightly greater damping than PEEK/Steel. PTFE/Steel and PEEK/Steel were indistinguishable from each other in terms of damping. In the case of the large pads, the PEEK pad led to a significant reduction in the system natural frequency while the PTFE/Steel pad yielded a slight reduction in the system natural frequency.

Figure E.10: Stiffness for low frequency range at 2500 rpm and 2400 N load.

\[ \text{Frequency, Hz} \times 10^6 \]

\[ \text{Stiffness, N/m} \]

- **PEEK/Steel**
- **PTFE/Steel**
- **PEEK**

\[ \text{Frequency, Hz} \]
Figure E.11: Stiffness for high frequency range at 500 rpm and 2400 N load.

Figure E.12: Stiffness for high frequency range at 2500 rpm and 2400 N load.
Figure E.13: Damping for high frequency range at 500 rpm and 2400 N load.

Figure E.14: Damping for high frequency range at 2500 rpm and 2400 N load.
Figure E.15: Stiffness for high frequency range at 500 rpm and 2400 N load.

Figure E.16: Stiffness for high frequency range at 2500 rpm and 2400 N load.
Figure E.17: Damping for high frequency range at 500 rpm and 2400 N load.

Figure E.18: Damping for high frequency range at 2500 rpm and 2400 N load.
larger in relation to the overall dimensions.

Furthermore, a method of providing accurate and reliable bonding between the surface and backing materials is not straightforward. This is especially true for laboratory developments to support computer models. The method of using screws in this study was successful for the PEEK surface due to its higher stiffness. The attachment method worked for the PTFE surface in lab conditions, but would not be recommended for industrial applications due to its sensitivity to screw tension and the difficulty of machining threads in PTFE.

The practical challenges associated with attachment of a compliant surface to a hard backing are a known issue with relatively few solutions for tilting pad bearings. Construction of the entire pad from a harder but somewhat compliant material such as PEEK or other engineering plastic can provide a way of eliminating the issue while still providing the advantages of compliant lined pads. Exemplifying this is the entirely PEEK pad in this work which provided very similar stiffness to the PTFE/Steel pad with slightly greater damping.

A clear effect of constructing the entire pad in a polymer material was observed in the case of the small pads. The PEEK pad provided lower stiffness and higher damping than the PEEK/Steel pad. This is believed to be the result of a combination of the deformation of the pivot contact and the internal damping of the polymer material.

E.8 Conclusions

Experiments have been conducted to determine the dynamic characteristics of polymer faced tilting pad journal bearings. From the tests of three different bearing configurations, the following conclusions can be drawn:

- Pads manufactured entirely of PEEK were slightly warmer than the pads with a steel backing.
- PEEK and PTFE surfaces seem to provide equivalent thermal characteristics when backed by steel.
- Increasing mean contact pressure on the entirely PEEK pad (by reducing surface area) led to lower stiffness and higher damping than equivalent pads with lower mean contact area.
- Replacing the PEEK liner with PTFE led to slightly reduced stiffness with equivalent damping.
E.9 Acknowledgments

The research presented in this paper was carried out as part of a project in conjunction with the Swedish Hydropower Center - SVC. SVC was established by the Swedish Energy Agency, Elforsk and Svenska Kraftnät in partnership with academic institutions. The experimental work of this study was carried out at the laboratories of the Solid Mechanics research group of the Technical University of Denmark and was in part funded by the Swedish Research School in Tribology.
Paper F

Operational transients in the guide bearings of a 10 MW Kaplan turbine
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Operational transients in the guide bearings of a 10 MW Kaplan turbine

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F.1 Abstract

Experiments are conducted using a 10 MW Kaplan hydropower machine with a PowerFormer™ generator, the Porjus U9 research machine. This machine is outfitted with an extensive array of sensors to determine oil film thickness, pad tilting, pad load and oil temperature in all three guide bearings as well as motion of the shaft in relation to both the bearing housings and the concrete foundation. Test results for all three guide bearings are examined during both steady state and transient operation.

F.2 Introduction

The increasing build-up of intermittent power sources such as wind and solar has resulted in changes in the way that hydro-electric power plants operate.
Traditionally, large hydropower plants provided a steady power supply to the grid with machines operating at or near maximum efficiency with minimal adjustments and few starts and stops. However, with increasing supply of intermittent power, hydro-electric power machines are more often called upon to provide regulating power to maintain grid frequency. This leads to operation at non-ideal operating points as well as an increase in starts and stops, all of which increase wear and tear on the machine in general and especially on the hydrodynamically lubricated bearings supporting the machine.

The dynamics of journal/guide bearings are generally well understood with a large body of work covering bearing design and modeling as well as experimental results. The body of this work is summarized by Dimond et al. [30].

The vast majority of research work and experimental results focus on bearings supporting horizontal shafts in which gravitational force plays a key role in stabilizing the shaft at a specific attitude angle and eccentricity. Meanwhile, most hydropower machines have vertical shafts with essentially no stabilizing load which results in large bearing orbits and a greater sensitivity to other excitations.

Work more relevant to hydro-electric power machines was carried out by Feng and Chu [47] who predicted the orbits of a vertically configured pump/turbine with varied pre-load between the pads. Experiments undertaken by Gustavsson et al. [69] and Nässelqvist et al. [107] allowed the authors to determine the stiffness and damping of guide bearings experimentally using strain gages mounted on the structure and in the support pins of tilting pad bearings. These results were then compared to models for horizontal machines finding that the calculations for horizontal machine eccentricity adequately predicted the orbit magnitude of the vertical machine. Aside from the work by Gustavsson, Nässelqvist, and Aidanpää, very few experimental studies have focused hydropower machine guide bearings.

The increasingly dynamic operation of hydropower machines brings with it many questions regarding the safety of different operations such as startups and shutdowns to more extreme situations such as complete loss of connection to the electrical grid. This study aims to investigate these particular transient cases in terms of their effect on the machine’s guide bearings and guide bearing foundations.
F.3 Experimental setup

The experimental work of this study was carried out on turbine/generator U9 at the old Porjus power station on the Lule river in the far north of Sweden. Generator unit U9 Fig. F.1 has a rated power of 10 MW and is unique in that it has a PowerFormer™ generator which is directly connected to the electrical grid without transformers. The machine’s Kaplan turbine has six runner blades, 20 guide vanes and 18 stay vanes and it rotates operates at 600 RPM. Porjus U9 is used occasionally in production but it is primarily used for research to improve and better understand hydropower machines and their operation. To support the research goals it is outfitted with an extensive sensor arrangement described in [22].

Crucial for this work is the measurement system for the guide bearings. Each housing of each of the machine’s three guide bearings has four displacement sensors to measure shaft motion relative to the bearing. Four additional displacement sensors at each bearing are solidly mounted in the concrete foundation. Inside the bearing, each of the tilting pads has two displacement sensors to measure oil film thickness at the inlet and outlet of each pad and two thermo-couples to measure oil temperature. Furthermore, the pivot pin of each pad has been replaced with a load cell to directly measure pad load. The sen-
Figure F.2: Front (left) and back (right) sides of a bearing pad with installed sensors for oil film thickness (A), pad load (B) and film temperature (C).

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature</td>
<td>±1°C</td>
</tr>
<tr>
<td>Housing to shaft</td>
<td>±2 μm</td>
</tr>
<tr>
<td>Foundation to shaft</td>
<td>±4 μm</td>
</tr>
<tr>
<td>Oil film thickness</td>
<td>±2 μm</td>
</tr>
</tbody>
</table>

The layout of the three bearings in this study is provided in Fig. F.3, Fig. F.4, and Fig. F.5. The machine is configured with two bearings, each with six pads, supporting the generator and a third eight pad bearing which supports the turbine. While the upper two bearings are of similar design, Bearing 3 is more similar to a cylindrical bearing without any open space between the pads. Bearing sizes are detailed in Table F.2.

The study began by starting the machine from cold conditions after being out of operation for several days. Shaft speed over the course of both start and shutdown are provided in Fig. F.6. A number of start and stop cycles were conducted over the course of testing with the bearings both cold and at steady state temperature. No significant variation in the acceleration/deceleration rates of the machine was observed for starting and stopping with varying bearing lubricant temperatures.

Investigation of the transient effects on the bearing resulting from changes in the machine’s operation state were investigated by performing several different state changes. The studies began with state changes of 10% in output power from 40% to 95%, then were continued with larger state change steps
Figure F.3: The arrangement of the pads and coordinate system in Bearing 1 of the U9 machine.

Figure F.4: The arrangement of the pads and coordinate system in the middle guide bearing of the U9 machine.
Figure F.5: The arrangement of the pads and coordinate system in the turbine guide bearing of the U9 machine.

Figure F.6: Rotational speed of the machine during normal start and normal stop cycle.
### Table F.2: Bearing characteristics.

<table>
<thead>
<tr>
<th>Bearing Type</th>
<th>Bearing</th>
<th>Lubricant supply</th>
<th>Bearing Diameter</th>
<th>Radial Clearance</th>
<th>Pad Length</th>
<th>Pivot Offset</th>
<th>Pad Height</th>
</tr>
</thead>
<tbody>
<tr>
<td>Upper Generator Bearing</td>
<td>1</td>
<td>Oil bath, leading edge injection</td>
<td>956 mm</td>
<td>150 μm</td>
<td>28°</td>
<td>64.3%</td>
<td>150 mm</td>
</tr>
<tr>
<td>Radial clearance</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pad length</td>
<td></td>
<td></td>
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<td></td>
<td></td>
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<tr>
<td>Pivot offset</td>
<td></td>
<td></td>
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<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pad height</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Lower Generator Bearing</td>
<td>2</td>
<td>Oil bath, leading edge injection</td>
<td>651 mm</td>
<td>150 μm</td>
<td>35°</td>
<td>61.4%</td>
<td>150 mm</td>
</tr>
<tr>
<td>Radial clearance</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pad length</td>
<td></td>
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<td></td>
<td></td>
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</tr>
<tr>
<td>Pivot offset</td>
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</tr>
<tr>
<td>Pad height</td>
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<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Turbine Bearing</td>
<td>3</td>
<td>Oil bath, leading edge injection</td>
<td>449.4 mm</td>
<td>150 μm</td>
<td>44°</td>
<td>61%</td>
<td>150 mm</td>
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<tr>
<td>Radial clearance</td>
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<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pad length</td>
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<tr>
<td>Pivot offset</td>
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<td>Pad height</td>
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</tr>
</tbody>
</table>
of 20%. These state changes were conducted both with the machine warmed up to operating temperature and immediately following a cold start with the machine’s systems in a ‘cold’ state below normal operating temperature.

Higher risk testing was then conducted by disconnecting the machine from the electrical network while running at 25% and 50% loads. During these tests the automatic shutdown system was modified to allow the machine to run without load for approximately 1 second prior to closure of the turbine guide vanes. This procedure allowed for de-coupling of the effects of the generator and the water turbine and also because the machine accelerated freely, it was possible to collect bearing data at higher speeds than under normal operation. Shaft speed for the grid disconnect testing is provided in Fig. F.7.

F.4 Results

Results from the steady state measurements highlighted the unique characteristics of large vertical bearings. The transient studies helped to quantify the risks associated with transient operation and describe dynamic responses that can be expected from machines going through transients.
Table F.3: Bearing and oil bath temperature in Bearing 1.

<table>
<thead>
<tr>
<th>Location</th>
<th>Outlet °C</th>
<th>Inlet °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pad 1</td>
<td>58.3</td>
<td>50.8</td>
</tr>
<tr>
<td>Pad 2</td>
<td>58.0</td>
<td>50.7</td>
</tr>
<tr>
<td>Pad 3</td>
<td>60.9</td>
<td>51.0</td>
</tr>
<tr>
<td>Pad 4</td>
<td>56.8</td>
<td>49.3</td>
</tr>
<tr>
<td>Pad 5</td>
<td>56.2</td>
<td>49.7</td>
</tr>
<tr>
<td>Pad 6</td>
<td>58.6</td>
<td>51.3</td>
</tr>
<tr>
<td>Oil bath</td>
<td>32.6</td>
<td>25.3</td>
</tr>
</tbody>
</table>

Table F.4: Bearing and oil bath temperature in Bearing 2.

<table>
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<tr>
<th>Location</th>
<th>Outlet °C</th>
<th>Inlet °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pad 1</td>
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<tr>
<td>Pad 2</td>
<td>51.4</td>
<td>39.3</td>
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<tr>
<td>Pad 3</td>
<td>59.8</td>
<td>48.9</td>
</tr>
<tr>
<td>Pad 4</td>
<td>48.8</td>
<td>36.6</td>
</tr>
<tr>
<td>Pad 5</td>
<td>54.1</td>
<td>40.2</td>
</tr>
<tr>
<td>Pad 6</td>
<td>57.4</td>
<td>42.0</td>
</tr>
<tr>
<td>Oil bath</td>
<td>38.2</td>
<td>28.4</td>
</tr>
</tbody>
</table>

Table F.5: Bearing and oil bath temperature in Bearing 3.

<table>
<thead>
<tr>
<th>Location</th>
<th>Inlet °C</th>
<th>Outlet °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pad 1</td>
<td>53.5</td>
<td>57.2</td>
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<td>Pad 2</td>
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<td>Pad 3</td>
<td>—</td>
<td>55.4</td>
</tr>
<tr>
<td>Pad 4</td>
<td>54.0</td>
<td>56.7</td>
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<tr>
<td>Pad 5</td>
<td>54.6</td>
<td>56.7</td>
</tr>
<tr>
<td>Pad 6</td>
<td>—</td>
<td>56.6</td>
</tr>
<tr>
<td>Pad 7</td>
<td>—</td>
<td>53.9</td>
</tr>
<tr>
<td>Pad 8</td>
<td>53.1</td>
<td>55.4</td>
</tr>
<tr>
<td>Oil bath</td>
<td>42.9</td>
<td>36.9</td>
</tr>
</tbody>
</table>
Figure F.8: Load on each bearing pad through one revolution for Bearing 2.

Figure F.9: Orbit of the shaft in Bearing 2’s housing. Dimensions are in mm and 40 complete orbits are shown as well as the average orbit and the locations of the bearing pads.
Figure F.10: Dynamic portion of the bearing load in Bearing 2. Dimensions are in Newtons and 40 complete rotations are shown as well as an average rotation and the locations of the bearing pads.

Figure F.11: Static load on each pad during start-up for Bearing 1.
Figure F.12: Static load on each pad during start-up for Bearing 2.

Figure F.13: Static load on each pad during stop for Bearing 1.
Figure F.14: Static load on each pad during stop for Bearing 2.

Figure F.15: Dynamic load on each pad in bearing 1 during load change from 40% to 80%.
Figure F.16: Static load on each pad in bearing 1 during load change from 40% to 80%.

F.4.1 Steady State

The bearing temperatures measured in the guide bearings were somewhat lower than expected. However, in light of the load and rotational speed, they seem to be reasonable as shown in Table F.3, Table F.4 and Table F.5 when compared to other experimental journal bearings with similar load [136]. Of note is that the temperature of several pairs of opposing pads (such as pad 3 and 6 in bearing 2) are quite a bit higher than the other pads. The temperatures in all of the pads correlate strongly with the load on each pad such that pads sitting opposite each other have very similar temperatures.

Studies of the machine’s bearings showed lower loads and smaller orbits than initially expected. The load on each pad over the course of one revolution is shown in Fig. F.8 for Bearing 2 and it highlights the difference between journal bearings in vertical and horizontal machines. While the load in a horizontal bearing is quite easily predicted, knowing the mass of the machine and the bearing’s geometry, the load in a vertical machine is heavily affected by the mis-alignments in the generator and any dynamic effects in the turbine as well as the preload of the individual bearing pads. In this study, the machine’s orbit in the foundation is almost entirely governed by the unbalance force in
F.4. RESULTS

the generator and excitations from interaction of the turbine and water flow. This leads to a maximum mean pressure on any individual bearing pad to be approximately 0.4 MPa.

The effect of the shaft’s motion and misalignment of the machine correlate directly with the orbit of the shaft within the oil film. Taking into account the loads from Fig. F.8 and the orbit from the same moment shown in Fig. F.9 it can be observed that the most heavily loaded pads, six and three, which sit opposite each other, effectively force the orbit into an oval shape within Bearing 2.

While the static portion of the bearing load varied greatly for each set of opposing bearing pads, the dynamic portion of the pad load was quite constant. A summation of the total dynamic load is provided in Fig. F.10. While some variation occurs through each revolution, the dynamic load appears to remain in the range of 1600 N for Bearing 2.

F.4.2 Start

Startup of the machine was conducted both from a ‘cold’ state after several days out of operation and from a ‘warm’ state immediately following a stop. In the case of cold startup, pad temperature rapidly increased to within 15°C of steady state operating temperature. During warm startup, pad temperature returned to operating temperature almost immediately due to the large thermal mass of the system.

Bearing pad mean load during startup rapidly increases as the turbine accelerates to operating speed as shown in Fig. F.11 and Fig. F.12. Load is spread unevenly over the pads due to mis-alignment or mis-forming of the bearing housing. Once the machine has reached full speed, the generator is phased into the electrical grid. The slight mis-alignment of the generator can be observed during phasing in (at approximately 65 seconds on Fig. F.11 and Fig. F.12) by the shift in load from bearing pads 5 and 6 to pads 2 and 3 in bearing 1 and from pads 3, 4 and 5 to pads 1, 2 and 6 in bearing 2.

Following startup the machine goes through a period of warming up over several hours of operation. Due to the geometric layout of the machine, warming up leads to varying effects in the bearings and support structures. Bearing 1 is mounted on a quite flexible structure which acts as the ceiling for the generator space. As such, heat from the generator can warm up the bearing structure. Additionally, Bearing 1 sits in the same housing as the thrust bearing which supports the weight of the entire machine. This bearing produces a significant amount of heat which leads to heating of the surrounding structure. The shaft is also heated by the thrust bearing and because this bearing is very near the
top of the shaft, the cold water in the turbine provides very little cooling. The result of this is that the shaft in Bearing 1 expands more than the surrounding structure and bearing housing which leads to smaller clearance in the bearing after warming up.

Bearing 2 sits alone under the generator. While the bearing is mounted in the generator space, the bearing’s underside is the ceiling for the lower level of the machine where the close proximity of the cold river water and bedrock structure leads to colder temperatures. The structure for Bearing 2 is approximately equivalent in stiffness to that of Bearing 1. This allows Bearing 2’s housing to expand during warm up. While the shaft also expands, it appears to expand less than the bearing, most likely due to the cooling from the turbine. The sum of these expansions is a slight increase of the clearance in the bearing.

Bearing 3 is surrounded by a very stiff structure which is cooled by the water flowing through the turbine. The shaft is also cooled by the turbine flow. Even though the bearing is quite warm during operation and the shaft appears to expand, this expansion appears to be matched by the bearing expansion and results in a slight decrease in the bearing clearance.

F.4.3 Stop

The machine was shut down using both the normal and emergency stop cycles. Through these two cycles, the machine was noted to accelerate over operational speed immediately following de-magnetization while the guide vanes were in the process of closing. While this acceleration did not seem to result in any dangerous excitations of the machine, it did lead to a slight increase in bearing pad load. The increase in bearing pad load (from increased hydrodynamic pressure) led to a slight increase in temperature, demonstrating the
F.4. RESULTS

Significant heating of the lubricant due to shearing.

Otherwise, during shutdown, the bearings cool rapidly to the oil bath temperature. This cooling directly follows the decrease in rotational speed. Heating of the oil and bearing and load on the bearing pads appears to be primarily due to oil shear as opposed to loads from dynamic motion.

In regards to thermal expansion, following shutdown, Bearing 1 and 2 appear to cool slowly. Bearing 3 on the other hand rapidly returns to pre-operation geometry due to the strong cooling effect of the water in the turbine.

F.4.4 State changes

In general, the effects on the bearings from load changes on the machine were small. In most cases, the only effect from the load state change was a change in the dynamic load on each of the bearing pads. The change in the dynamic portion of the load on each pad in bearing 1 is displayed in Fig. F.15. The static load on each pad remained nearly constant through even the largest load changes from 40% to 80% load.

No discernible change in pad temperature was observed from changes in load state. Through even the largest load state change from 40% to 80%, the temperature on all pads remained constant as seen in Fig. F.16. This makes it clear that the primary source of heating and power loss in the bearing pads is the result of the oil shearing and hydrodynamic pressure buildup.

F.4.5 Disconnection from grid

Disconnecting the machine from the electrical grid led to rapid acceleration of the machine as shown in Fig. F.7. From 50% load, the machine accelerates...
from 600 to 700 RPM before the turbine’s guide vanes closed fully, cutting off the inlet water flow to the machine. The case of 25% load is much milder but the machine still is able to accelerate from 600 RPM to nearly 640 RPM in only a few seconds. This rapid acceleration and its inherent danger for the machine limited the maximum load prior to disconnecting the machine from the grid to 50% of full capacity. Even from 50% power, disconnection of the electrical grid and de-magnetization as well as closure of the guide vanes provided significant instantaneous excitation to the generator and rapid closure of the guide vanes heavily excited the turbine. These extreme excitations are displayed in Fig. F.17 and Fig. F.18.

Immediately following disconnection of the electrical grid, it can be observed that the orbit in Bearing 3 begins to shrink and stabilize as the turbine flow becomes the primary source of damping in the machine. The opposite occurs in the generator bearings where the orbit begins to grow slightly due to the loss of the stabilizing effect of the magnetized generator. Increased bearing static load due to the increased rotational speed also seems to add to the stabilization effect immediately following grid disconnection. Bearing temperatures rise slightly during the disconnection cycle, but it is uncertain whether this is caused by increased shaft speed or the extra dynamic motion resulting from closure of the guide vanes. Disconnection of the generator from the grid provides much less excitation than closure of the guide vanes. The flow in the turbine that resulted from guide vane closure excited the entire machine at a very low frequency of approximately 1.3 Hz as is seen in Fig. F.18. The corresponding dynamic bearing loads generated by these excitations were the largest observed during the experimental test series as displayed in Fig. F.17. However, the water in the turbine seems to provide significant damping of low frequency motion as the motion is damped out and nearly undetectable after only a few cycles.

**F.5 Discussion**

The body of knowledge and experimental data available for horizontal turbines is quite extensive. The same cannot be said for vertical turbines. The experimental results presented herein provide a basis for the development of bearing models which in turn can lead to further development of hydropower machines.

Critical analysis of the data brings to light a number of interesting points in the design of hydropower machines. The load on any bearing pad in even the
most extreme case of grid disconnection from 50% load reached a maximum momentary contact pressure of 0.4 MPa in Bearing 1, 0.4 MPa in Bearing 2 and 0.3 MPa in Bearing 3. Under normal operation at 'best efficiency point' the bearing pad loads are slightly lower than the maximums observed during grid disconnect, albeit with a much smaller dynamic component.

Considering that the pad load appears to be primarily the result of bearing preload as opposed to actual dynamic loading, there is potential to improve the machine’s operation. A slight reduction of bearing preload on the most heavily preloaded pads could reduce the oil temperature at those pads by several degrees. Even a slight reduction of bearing pad maximum temperature by 5°C could lead to power savings of several kW. Considering the high efficiency of hydropower machines, a very small percent savings of the total power output can result in a substantial reduction of the total losses.

Further reduction of power losses could be achieved by increasing the load on each individual pad through reduction of the number of pads or reduction of the size of the pads. The load on each bearing pad in Bearing 3 could be doubled and still lie below the normal bounds for journal bearings used in other, horizontal, turbines. The power savings associated with such a change would be significant.

F.6 Conclusion

The transient nature of journal bearing operation has been studied on the three unique journal bearings of a full scale hydropower machine. Dynamic loads were found to be on the order of static loads in the most extreme loading cases, but in general the pad loading was quite low and primarily a result of bearing preload. Analysis of the temperature and heating of the bearing pads demonstrated the potential for significant power savings in the machine.

F.7 Acknowledgments

The research presented in this paper was carried out as part of a project in conjunction with the Swedish Hydropower Center - SVC. SVC was established by the Swedish Energy Agency, Elforsk and Svenska Kraftnät in partnership with academic institutions. Additional acknowledgment to the Porjus Stiftelsen with representatives from Vattenfall, Alstom and Andritz for use of the experiment facilities in Porjus.
Paper G

Steady state and dynamic characteristics for guide bearings of a hydro-electric unit
G.1. ABSTRACT

Submitted.

Steady state and dynamic characteristics for guide bearings of a hydro-electric unit

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G.1 Abstract

Experiments are conducted using a 10 MW Kaplan hydropower machine which is outfitted with an extensive array of sensors to determine oil film thickness, pad load and oil temperature in all three guide bearings as well as motion of the shaft in relation to both the bearing housings and the concrete foundation. Test results for all journal bearings are compared to a commercial rotor dynamics model and results for the central journal bearing are compared to a multi-physics model to provide insight into the machine’s steady state and dynamic characteristics and their variations during normal operation.
G.2 Introduction

De-regulation of electricity markets and the build-up of intermittent power sources such as wind and solar have led to changes in the way that hydro-electric power plants operate. Traditionally, large hydropower plants have provided steady power supply to the grid with machines operating at or near maximum efficiency with minimal adjustments and few starts and stops. However, because of ongoing changes in the electrical power network, hydro-electric machines are more often used to provide regulating power to maintain grid frequency. This leads to operation outside of ideal operating ranges as well as an increase in starts and stops which can lead to larger dynamic loads and increased wear and tear on the machine. Thus a clear understanding of the machine’s dynamic characteristics is essential for safe operation.

The dynamics of journal/guide bearings are generally well understood with a large body of work covering bearing design and modeling as well as experimental results. The body of this work is summarized by Dimond et al. [30]. Nearly all of the work on journal bearings has focused on horizontal machines in which gravity provides a key stabilizing force.

On the contrary, most hydropower machines have vertical shafts with essentially no stabilizing load which results in potentially large shaft orbits and a greater sensitivity to other excitations. Study of a vertical rotor and machine by White et al. [159] found that dramatically decreasing the radial clearance in a pump’s journal bearings moved a critical speed outside of the machine’s operating range. Another study of vertical bearings by San Andres and De Santiago [128] found that the characteristics of a vertical plain bearing under high load could generally be predicted by modeling it as a horizontal configuration but that the fluid inertia terms were significantly greater in the experiments than predicted by models.

Investigation of a hydro-electric power machine was carried out by Feng and Chu [47] who predicted the orbits of a vertically configured pump/turbine with unevenly pre-loaded pads. Experimental work with hydropower machines was conducted by Gustavsson et al. [69] and Nässelqvist et al. [107] in which the authors determined the stiffness and damping of guide bearings experimentally using strain gages mounted on the structure and in the support pins of tilting pad bearings. These results were then compared to models for horizontal machines finding that the calculations for horizontal shaft eccentricity adequately predicted the vertical shaft’s orbit magnitude. Aside from the work by Gustavsson, Nässelqvist, and Aidanpää, very few experimental studies have focused on hydropower machines.
Figure G.1: Arrangement of Porjus turbine/generator unit U9.

G.3 Equipment

The limited number of experimental studies conducted on vertical rotating machines do not provide the accurate data needed to develop robust machine models to aid in development of new machines and upgrades and refurbishments of existing hydropower machines. Thus using the unique test facility at Porjus and a well developed model, this study aims to begin filling in the gaps in experimental data and model studies regarding the guide bearings of large vertical rotating machines.

G.3 Equipment

The experimental work of this study was carried out on turbine-generator U9 at the old Porjus power station on the Lule river in the far north of Sweden. Turbine unit U9 Fig. G.1 is unique in that its generator is directly connected to the electrical grid without transformers. The machine’s Kaplan turbine has six runner blades, 20 guide vanes and 18 stay vanes and it operates at 600 rpm. Its three guide bearings are described in Table G.1. While Porjus U9 is used occasionally in production it is primarily used for research and development to improve and better understand hydropower facilities and their operation. To support the research goals it is outfitted with an extensive sensor arrangement described in [22].

Relevant to the current work is that the housing of each of the machine’s three guide bearings have four displacement sensors to measure shaft motion relative to the bearing. Four additional displacement sensors at each bearing
Figure G.2: Sensor layout along the centerline for the pads of Bearing 1 (top), Bearing 2 (middle) and Bearing 3 (bottom). Sliding direction is from left to right.

are solidly mounted in the concrete foundation. Inside the bearing, each of the spherical pivot tilting pads has two displacement sensors to measure oil film thickness at the inlet and outlet of each pad and two type K thermo-couples to measure oil temperature. Thermo-couples are installed 6 mm below the pad surface using the technique developed in [59] which allows oil from the contact to continually leak past the thermo-couple providing a faster response to oil film temperature changes. Furthermore, the pivot pin of each pad has been replaced with a load cell to directly measure pad load. The sensor arrangement for the pads is displayed in Fig. G.2.

Lubricant in the bearings for all testing was ISO VG68 turbine oil and is provided via a combination of oil bath and leading edge injection. The shaft rotates in the clockwise direction.

G.4 Uncertainty

The uncertainty associated with each of the measurements is presented in Table G.2. Sources of uncertainty are primarily associated with the accuracy of the measurement equipment. In the case of film thickness sensors, zero
Table G.1: Bearing characteristics.

<table>
<thead>
<tr>
<th>Bearing Type</th>
<th>(Bearing 1)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of pads</td>
<td>6</td>
</tr>
<tr>
<td>Bearing diameter</td>
<td>956 mm</td>
</tr>
<tr>
<td>Shaft diameter</td>
<td>955.57</td>
</tr>
<tr>
<td>Radial clearance</td>
<td>150 μm</td>
</tr>
<tr>
<td>Pad length</td>
<td>28°</td>
</tr>
<tr>
<td>Pivot offset</td>
<td>64.3%</td>
</tr>
<tr>
<td>Pad height</td>
<td>150 mm</td>
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</table>

<table>
<thead>
<tr>
<th>Bearing Type</th>
<th>(Bearing 2)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of pads</td>
<td>6</td>
</tr>
<tr>
<td>Bearing diameter</td>
<td>651 mm</td>
</tr>
<tr>
<td>Shaft diameter</td>
<td>650.57</td>
</tr>
<tr>
<td>Radial clearance</td>
<td>150 μm</td>
</tr>
<tr>
<td>Pad length</td>
<td>35°</td>
</tr>
<tr>
<td>Pivot offset</td>
<td>61.4%</td>
</tr>
<tr>
<td>Pad height</td>
<td>150 mm</td>
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</tbody>
</table>

<table>
<thead>
<tr>
<th>Bearing Type</th>
<th>(Bearing 3)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of pads</td>
<td>8</td>
</tr>
<tr>
<td>Bearing diameter</td>
<td>449.4 mm</td>
</tr>
<tr>
<td>Shaft diameter</td>
<td>449.85</td>
</tr>
<tr>
<td>Radial clearance</td>
<td>150 μm</td>
</tr>
<tr>
<td>Pad length</td>
<td>44°</td>
</tr>
<tr>
<td>Pivot offset</td>
<td>61%</td>
</tr>
<tr>
<td>Pad height</td>
<td>150 mm</td>
</tr>
</tbody>
</table>
values for the sensors were found prior to installation but it was not possible to calibrate these values after assembly. Measurements of film thickness are therefore presented relative to the mean measurement from each sensor. Likewise, shaft orbit is determined from the center of the measured orbit. While the tolerances for roundness and form of the shaft in the bearing are small, the tolerances are not as small outside the bearing housing where the sensors used to determine shaft orbit are located. The error generated by these tolerances and roughness can be observed as a slight noise in the orbit plots and is included in the uncertainty of these measurements.

G.5 Model description

To better understand the experimental results, a multi-physics model was developed. This model is a further development of that described in [23] and [24] to account for the geometry of the test bearings and the vertical bearing arrangement. The model incorporates the Reynolds equation, structural deformations and a simplified energy equation as well as assumptions of an incompressible, viscous, Newtonian fluid with laminar flow.

Study using the model focuses on Bearing 2 because it has clearly defined boundary conditions. Unlike Bearing 3, Bearing 2 is not affected by the powerful cooling provided by the water in the turbine. Furthermore, the free air flow around the bearing housing and shaft provides a stable thermal environment. Because Bearing 2’s housing contains a guide bearing only the heat generation in the bearing can be directly related to the fluid shearing and load on the bearing pads. This is not the case in Bearing 1 where the thrust bearing contributes significantly to the heating and dynamic motion in the guide bearing.

Rotor dynamics modeling of the bearings was accomplished using a commercial rotor dynamics code [1].

Table G.2: Measurement uncertainty.

<table>
<thead>
<tr>
<th></th>
<th>±</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature</td>
<td>1°C</td>
</tr>
<tr>
<td>Housing to shaft</td>
<td>3 μm</td>
</tr>
<tr>
<td>Foundation to shaft</td>
<td>4 μm</td>
</tr>
<tr>
<td>Film thickness</td>
<td>2 μm</td>
</tr>
</tbody>
</table>
Table G.3: Bearing and oil bath temperature in Bearing 1.

<table>
<thead>
<tr>
<th>Location</th>
<th>Outlet °C</th>
<th>Inlet °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pad 1</td>
<td>58.3</td>
<td>50.8</td>
</tr>
<tr>
<td>Pad 2</td>
<td>58.0</td>
<td>50.7</td>
</tr>
<tr>
<td>Pad 3</td>
<td>60.9</td>
<td>51.0</td>
</tr>
<tr>
<td>Pad 4</td>
<td>56.8</td>
<td>49.3</td>
</tr>
<tr>
<td>Pad 5</td>
<td>56.2</td>
<td>49.7</td>
</tr>
<tr>
<td>Pad 6</td>
<td>58.6</td>
<td>51.3</td>
</tr>
<tr>
<td>Oil bath</td>
<td>32.6</td>
<td>25.3</td>
</tr>
</tbody>
</table>

G.6 Results

Experimental results from the tests generally follow expectations from journal bearing theory. However, because most bearing models and practical experience are focused on horizontal machines, the presented experimental results have a number of important deviations from initial expectations.

G.6.1 Experimental Results

The bearing temperatures measured in the guide bearings were highly dependent on the bearing’s geometry but generally seem to be reasonable for hydrodynamic bearings [136] as shown in Table G.3, Table G.4 and Table G.5. However, the pads of the most lightly loaded bearing, Bearing 3 appear to be significantly warmer than equally loaded pads in the other bearings. This extra heating is believed to be due to carryover of warm lubricant from one pad to the next which retains heat in the bearing and prevents cooling. Also of note is that the temperature of several pairs of opposing pads (such as pad 3 and 6 in Bearing 2) are quite a bit higher than the other pads. The temperatures in all of the pads correlate strongly with the load on each pad such that pads sitting opposite each other have similar temperatures.

Studies of the machine’s bearings showed generally light loads and small orbits compared to loads in horizontal machines for electrical power generation. Shaft orbits within the upper bearings are provided in Fig. G.3 and Fig. G.4. The load on each pad over the course of one revolution is shown in Fig. G.5, Fig. G.6 and Fig. G.7 for Bearings 1, 2 and 3 respectively which
Table G.4: Bearing and oil bath temperature in Bearing 2 for experiment and maximum pad temperature from multi-physics model.

<table>
<thead>
<tr>
<th>Location</th>
<th>Outlet °C</th>
<th>Inlet °C</th>
<th>Model °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pad 1</td>
<td>54.0</td>
<td>40.1</td>
<td>47.0</td>
</tr>
<tr>
<td>Pad 2</td>
<td>51.4</td>
<td>39.3</td>
<td>47.2</td>
</tr>
<tr>
<td>Pad 3</td>
<td>59.8</td>
<td>48.9</td>
<td>47.0</td>
</tr>
<tr>
<td>Pad 4</td>
<td>48.8</td>
<td>36.6</td>
<td>47.0</td>
</tr>
<tr>
<td>Pad 5</td>
<td>54.1</td>
<td>40.2</td>
<td>47.2</td>
</tr>
<tr>
<td>Pad 6</td>
<td>57.4</td>
<td>42.0</td>
<td>47.0</td>
</tr>
<tr>
<td>Oil bath</td>
<td>38.2</td>
<td>28.4</td>
<td>40.0</td>
</tr>
</tbody>
</table>

Table G.5: Bearing and oil bath temperature in Bearing 3.

<table>
<thead>
<tr>
<th>Location</th>
<th>Outlet °C</th>
<th>Inlet °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pad 1</td>
<td>57.2</td>
<td>53.5</td>
</tr>
<tr>
<td>Pad 2</td>
<td>54.6</td>
<td>51.5</td>
</tr>
<tr>
<td>Pad 3</td>
<td>55.4</td>
<td>—</td>
</tr>
<tr>
<td>Pad 4</td>
<td>56.7</td>
<td>54.0</td>
</tr>
<tr>
<td>Pad 5</td>
<td>56.7</td>
<td>54.6</td>
</tr>
<tr>
<td>Pad 6</td>
<td>56.6</td>
<td>—</td>
</tr>
<tr>
<td>Pad 7</td>
<td>53.9</td>
<td>—</td>
</tr>
<tr>
<td>Pad 8</td>
<td>55.4</td>
<td>53.1</td>
</tr>
<tr>
<td>Oil bath</td>
<td>42.9</td>
<td>36.9</td>
</tr>
</tbody>
</table>
Figure G.3: Orbit of the shaft in Bearing 1’s housing. Dimensions are in mm and four complete orbits are shown.

Figure G.4: Orbit of the shaft in Bearing 2’s housing from experiment (thin blue line) and simulation (thick red line). Dimensions are in mm and four complete orbits are shown. Pad locations are shown for reference.
highlight the difference between journal bearings in vertical and horizontal machines. While the static load in a horizontal bearing is quite easily predicted, knowing the mass of the rotor and the bearing’s geometry, the static load in a vertical bearing of a hydropower turbine is primarily affected by uneven pull by the generator, fluid effects in the turbine and the pre-load of the individual bearing pads. The effect of this can be observed in Fig. G.8 and Fig. G.9 in which the bearing pad loads increase with rotation speed (Fig. G.10) and then shifts (at approximately 60 seconds) when the generator is connected to the electrical network. This leads to a maximum mean pressure on any individual bearing pad of approximately 0.4 MPa.

The evolution of the oil film temperature through the startup process is provided in Fig. G.11 for Bearing 1 and Fig. G.12 for Bearing 2. This temperature evolution further highlights the effects of the variation in pre-load of the individual bearing pads. In Bearing 1, the pads have a much smaller spread in temperatures than in Bearing 2 reflecting the more even pre-loading in Bearing 1 compared to Bearing 2.

The effect of the misalignment of the bearings correlates directly with the orbit of the shaft within the oil film. Taking into account the loads from Fig. G.6 and the orbit from the same instant shown in Fig. G.4 it can be observed that the most heavily loaded pads, Pad 6 and Pad 3, effectively force the orbit into an oval shape within Bearing 2.

While the static portion of the bearing load varied greatly for each set of opposing bearing pads, the dynamic portion of the pad load remained generally constant. A summation of the total dynamic load in Bearing 1 and Bearing 2 are provided in Fig. G.13 and Fig. G.14 respectively. While some variation occurs through each revolution, the dynamic load appears to remain in the range of 2800 N in Bearing 1 and 1600 N in Bearing 2.

Bearing 2 had the greatest variation in film thickness and load between its pads. In Fig. G.6 it is observed that pads with the lightest load (pads 2 and 5) also had the greatest range of motion. Likewise, the most heavily loaded pads (pads 3 and 6) had the smallest range of motion. The pads of Bearing 3 are generally lightly loaded and the film thickness and tilt angle vary significantly over the course of one revolution as observed in Fig. G.7. Load for pads 7 and 8 is not displayed as these two load cells were not operational.

G.6.2 Model results

The pre-load of each pad in the model was initially tuned to provide an identical orbit and static pad load to those observed in the experiments, however
Figure G.5: Bearing 1 dynamic portion of film thickness for the leading and trailing edges of all pads as well as the load on the pad over the course of one revolution. Dashed line is leading edge, dotted line is trailing edge, solid is the load on the pad.
Figure G.6: Bearing 2 dynamic portion of film thickness for the leading and trailing edges of all pads as well as the load on the pad over the course of one revolution. Dashed line is leading edge, dotted line is trailing edge, solid is the load on the pad.
Figure G.7: Bearing 3 dynamic portion of film thickness for the leading and trailing edges of all pads as well as the load on the pad over the course of one revolution. Dashed line is leading edge, dotted line is trailing edge, solid is the load on the pad.
Figure G.8: Static load on each pad during startup for Bearing 1. Rotor is connected to electrical grid at approximately 70 seconds.

Figure G.9: Static load on each pad during startup for Bearing 2. Rotor is connected to electrical grid at approximately 70 seconds.
Figure G.10: Machine speed during startup.

Figure G.11: Temperature at the outlet side of each pad during startup for Bearing 1. Rotor is connected to electrical grid at approximately 70 seconds.
Figure G.12: Temperature at the outlet side of each pad during startup for Bearing 2. Rotor is connected to electrical grid at approximately 70 seconds.

Figure G.13: Dynamic portion of the bearing load in Bearing 1. Dimensions are in Newtons and four complete orbits are shown.
Figure G.14: Dynamic portion of the bearing load in Bearing 2 from experiment (thin blue line) and simulation (thick red line). Dimensions are in Newtons and four complete orbits are shown. Pad locations are shown for reference.
when the orbit matched that of the experiments, the dynamic load on the individual pads was much lower than that observed. Matching the dynamic load and the static load led to the larger orbit shown in Fig. G.4. The discrepancy in orbit in this case is most likely due to the fact that the bearing’s structure in the model is infinitely stiff. Calculation of the dynamic characteristics (Table G.6) show that this is not the case.

The pad temperatures in the model are lower due to a combination of factors. Firstly, the model’s method of handling the energy losses in the bearing is important to mention. To simplify the model and decrease computational time a simplified energy equation is used which is not able to take into account all the energy used to shear the lubricant. The lower energy losses in turn lead to less heating of the lubricant and a thicker lubricant film for a given pad load. More importantly, the actual inlet temperature to the bearing pads has large uncertainty. While the experiment gives a value for the oil at the surface of each pad, the oil at the shaft surface may be much warmer due to lubricant carryover from the previous pad.

G.7 Discussion

Experimental datasets gathered from full scale machines are generally sparse in the academic literature. This is especially true for vertical hydropower turbines which until recent years have been almost entirely overlooked by the rotor dynamics and tribology research communities. From these initial results it can be determined that while the machine generally behaves as would be expected from theory, the vertical configuration provides many unique effects. The inherent instability of the vertical rotor results in an orbit at a generally constant eccentricity as would be expected from a very lightly loaded horizontal machine. However, the light loading makes the bearing’s performance more sensitive to unavoidable variation in assembly tolerances as described by Fillon, Dmochowski and Dadouche [49, 33]. This sensitivity is very clear in the test results for the individual pads as exemplified by Bearing 2.

The experimental results allowed for calibration of the multi-physics model. As suggested by Feng and Chu [47], by matching the orbit in the model with that measured in the experiment, it was possible to determine the degree of misalignment in the real bearing. In some cases, such as in Fig. G.4 the model was able to clarify the shaft motion with a more round orbit than that which was measured. It is believed that the squarish shape of the orbit in this case results from a combination of the construction of the machine with three bearings
Table G.6: Mean stiffness in the three bearings obtained from experiment (Exp) for both steady state (warm) and cold conditions, rotor dynamics model (RD), and multi-physics model (MP).

<table>
<thead>
<tr>
<th>Location</th>
<th>Stiffness (N/m)</th>
<th></th>
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</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Warm</td>
<td>Cold</td>
<td></td>
</tr>
<tr>
<td>Bearing 1</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Oil Film (Exp)</td>
<td>6.7x10^8</td>
<td>6.6x10^8</td>
<td></td>
</tr>
<tr>
<td>Oil Film (RD)</td>
<td>6.3x10^8</td>
<td></td>
<td>–</td>
</tr>
<tr>
<td>Housing (Exp)</td>
<td>5.9x10^7</td>
<td>4.8x10^7</td>
<td></td>
</tr>
<tr>
<td>Foundation (Exp)</td>
<td>1.5x10^7</td>
<td>1.5x10^7</td>
<td></td>
</tr>
<tr>
<td>Bearing 2</td>
<td></td>
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<tr>
<td>Oil Film</td>
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<td>Oil Film (MP)</td>
<td>3.5x10^8</td>
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<td>–</td>
</tr>
<tr>
<td>Oil Film (RD)</td>
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<td>–</td>
</tr>
<tr>
<td>Housing (Exp)</td>
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<td>1.6x10^8</td>
<td></td>
</tr>
<tr>
<td>Foundation (Exp)</td>
<td>6.3x10^6</td>
<td>5.9x10^6</td>
<td></td>
</tr>
<tr>
<td>Bearing 3</td>
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</tr>
<tr>
<td>Oil Film (Exp)</td>
<td>1.6x10^8</td>
<td>1.9x10^8</td>
<td></td>
</tr>
<tr>
<td>Oil Film (RD)</td>
<td>3x10^8</td>
<td></td>
<td>–</td>
</tr>
<tr>
<td>Foundation (Exp)</td>
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<td>1.2x10^7</td>
<td></td>
</tr>
</tbody>
</table>
arranged on three different axes as well as possible slight misalignment of the displacement sensors. Such misalignments are unavoidable in large complex sensor installations when sensor installation must be adapted to real geometric and operational constraints.

The dynamic load and orbit in each of the bearings can be further used to characterize the stiffness and damping of the lubricant film, bearing and the foundation structure around the bearing using the methods of Gustavsson, Nässelqvist and Aidanpää [69, 107]. The mean stiffness of the three bearings calculated using this method is provided in Table G.6 for the components of the bearings both immediately following startup (cold) and at steady state after several hours of operation (warm). Due to the complexity of the calculations, very large uncertainty (on the order of 20%) should be associated with the stiffness values listed. This is especially important in Bearing 3 where data for two faulty load cells (on pads 7 and 8) was substituted with load measured from the opposing pads (pads 3 and 4) which was then shifted 180 degrees. Housing stiffness is not calculated for Bearing 3 as the displacement sensors there provided unreliable data.

The stiffness provided by each of the three bearings is closely related to the static load on the bearing pads. This is accentuated by Bearing 3 which has much lower stiffness than the other bearings due to the light static loading on its bearing pads.

Calculation of the stiffness from the multi-physics model results was accomplished using the same method as with the experimental results. The exception being that several rotations of the shaft were simulated by perturbing the model results with random noise. The multi-physics model predicts a lower stiffness than the experimentally determined value as is clear by the larger shaft orbit in the multi-physics model. A commercial rotor dynamics code (for horizontal machines) was also used to validate the calculations using the mean bearing pad load as the bearing load. All of these calculations are provided in Table G.6.

From these stiffness calculations it can be observed that the stiffness provided by the structure around the bearing is considerably lower than that of the oil film in the bearings. In order to allow for convergence in the rotor dynamics calculations and multi-physics model, this structural stiffness is assumed to be much higher than the oil film stiffness. The experimentally calculated stiffnesses demonstrate that this assumption is clearly incorrect and that the structure may be softer than the bearing oil film. This highlights that in design of hydropower plants, care should be taken to properly determine the characteristics of the structure surrounding the bearings to guarantee that the models
used to develop the bearing and rotor design provide accurate results.

Further, warming up of the machine seems to affect the different bearings differently. Bearing 1 and Bearing 2 become stiffer, but the oil film in Bearing 2 becomes softer through following warm up. The stiffness of Bearing 3 on the other hand decreases slightly through the warm up process. Increasing the regularity of startup and shutdown increases the thermal cycling of the machine leading to constantly changing bearing characteristics. The long term effects of such a situation are uncertain but presumably could lead to less stability of the machine.

G.8 Conclusion

Experiments have been conducted in the journal bearings of a full scale hydroelectric power machine to determine their characteristics. These experiments were then compared to a multi-physics model of one of the bearings and a rotor dynamics model including all bearings.

Variation in load, oil film thickness, pad tilting and temperature of the individual bearing pads was highly dependent on the alignment and individual pad pre-load from assembly.

A multi-physics model was able to capture many bearing characteristics, but the accuracy of the model was highly dependent on the boundary conditions including the accuracy of the bearing’s geometry, the inherent misalignment that occurs at assembly and the response of the surrounding structure.

In hydropower machines, stiffness of the supporting structure may be more important to machine performance than the stiffness of the bearing alone and the stiffness can vary greatly through a startup process due to the machine’s thermal transients.

G.9 Acknowledgments

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