Tribological Design of Nano/Magnetorheological Fluid Journal Bearings

Ph.D. Dissertation of Dimitrios A. Bompos

Dipl.-Ing. Mechanical & Aeronautics

Patras 2015
Πρακτικό συνεδρίασης
Επταμελώς Εξεταστικής Επιτροπής για την κρίση της διδακτορικής διατριβής
tου υποψηφίου διδάκτορα κ. Δημητρίου Μπόμπου

Η Τριμελής Συμβουλευτική Επιτροπή για την Διδακτορική Διατριβή του υποψήφιου Διδάκτορα κ. Δημητρίου Μπόμπου, Διπλωματούχου Μηχανολόγου Μηχανικός του Τμήματος Μηχανολόγων και Αεροναυτικών Μηχανικών του Πανεπιστημίου Πατρών (ΠΠ), με τίτλο

«Τριβολογικός Σχεδιασμός Εκράνων Ολισθήσεως με Νάνο-Μαγνητορεολογικά Ρευστά»,

Η Τριμελής Συμβουλευτική Επιτροπή συγκροτήθηκε με απόφαση του Τμήματος Μηχανολόγων και Αεροναυτικών Μηχανικών του Πανεπιστημίου Πατρών, στη Γενική Συνέλευση με Ειδική Σύνθεση Αρ. 3 την 20.03.2012 και αποτύχησαν από τους:

1) Παντελή Γ. Νικολακόπουλο, Επιβλέποντα (σύμφωνα με την απόφαση της Γενικής Συνέλευσης με Ειδική Σύνθεση Αρ. 5 της 24/02/2015), Επίκεφαλη Καθηγήτρια του Τμήματος Μηχανολόγων & Αεροναυτικών Μηχανικών ΠΠ, με γνωστικό αντικείμενο "ΣΤΟΙΧΕΙΑ ΜΗΧΑΝΩΝ - ΤΡΙΒΟΛΟΓΙΑ"

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3) Νικόλαο Αντωνιάνη, Καθηγήτρια Τμήματος Μηχανολόγων & Αεροναυτικών Μηχανικών ΠΠ, με γνωστικό αντικείμενο "ΑΝΑΛΥΣΗ ΚΑΙ ΣΧΕΔΙΑΣΜΟΣ ΚΑΤΑΣΚΕΥΩΝ ΚΑΙ ΜΗΧΑΝΩΝ ΜΕ ΕΜΦΑΣΗ ΣΤΙΣ ΥΠΟΛΟΓΙΣΤΙΚΕΣ ΜΕΘΟΔΟΥΣ"

και συμπληρώθηκε κατόπιν αποφάσεως της 1ης Γενικής Συνέλευσης με Ειδική Σύνθεση της 6.10.2015 με τους κκ.:

- Michel Fillon, Καθηγήτριας Τμήματος Mechanics, Structures and Complex Systems, του Ινστιτούτου Πρίμε του Πανεπιστημίου του Ποιτιέ της Γαλλίας με γνωστικό αντικείμενο: "ΥΔΡΟΔΥΝΑΜΙΚΗ ΚΥΛΙΝΔΡΙΚΩΝ ΚΑΙ ΩΣΤΙΚΩΝ ΕΔΡΑΝΩΝ ΟΛΙΣΘΗΣΗΣ".

- Αθανάσιος Μιχαήλη. Καθηγητής Τμήματος Μηχανολόγων Μηχανικών ΑΠΘ με γνωστικό αντικείμενο: "ΣΤΟΙΧΕΙΑ ΜΗΧΑΝΩΝ, ΚΑΤΑΣΚΕΥΑΣΤΙΚΗ ΑΝΑΛΥΣΗ ΚΑΙ ΣΥΝΘΕΣΗ ΜΗΧΑΝΩΝ, ΤΡΙΒΟΛΟΓΙΑ".

- Νικόλαο Ασηφάγκαθο, Καθηγητής Τμήματος Μηχανολόγων & Αεροναυτικών Μηχανικών ΠΠ, με γνωστικό αντικείμενο: "ΥΔΡΟΔΥΝΑΜΙΚΗ ΗΛΕΚΤΡΟΜΗΧΑΝΟΛΟΓΙΚΩΝ ΣΥΣΤΗΜΑΤΩΝ ΜΕ ΕΜΦΑΣΗ ΣΤΗ ΡΟΜΠΟΤΙΚΗ".

[1]
Η εξεταστική επιτροπή συνεδριάσει σήμερα Παρασκευή, 5.2.2016 και ώρα 17:00, στην αίθουσα συνεδριάσεων του Τμήματος. Κατά την συνεδρίαση ο υποψήφιος διδάκτορας κ. Δημήτριος Μπόμπους ανέπτυξε δημόσια και εκτενές την διατριβή του ενώπιον της εξεταστικής επιτροπής και αφού απάντησε στις υποβληθείσες ερωτήσεις περατώθηκε η εξέταση.

Εν συνεχεία, στην κατ’ ιδίαν συνεδρίαση η εξεταστική επιτροπή, κατόπιν συζήτησης, αποφάσισε ομόφωνα ότι η διατριβή αποτελεί πρωτότυπη και οισιαστική συμβολή στην προαγωγή της επιστήμης, και που συγκεκριμένα στον τριβολογικό σχεδιασμό εδράνων αλιάθησης λ υπαινομένων με μαγνητορεολογικά και νανομαγνητορεολογικά λαπαντικά. Μετά από ονομαστική ψηφοφορία, τα μέλη της εξεταστικής επιτροπής υψήσαν ομόφωνα υπέρ της απονομής του τίτλου του Διδάκτορα στον κ. Δημήτριο Μπόμπου, τη δε διατριβή βαθμολόγησαν με τον βαθμό «……………….».

Η αναγγέλθηση του υποψήφιου διδάκτορα θα γίνει από την Γενική Συνέλευση με Ειδική Σύνθεση του Τμήματος Μηχανολόγων & Αεροναυπηγών Μηχανικών του Πανεπιστημίου Πατρών, σύμφωνα με το άρθρο 13 του νόμου 2083/92.

Η επικοινωνή με εξεταστική επιτροπή

Επίκουρος Καθηγητής, Δρ. Παντελής Νικολακόπουλος

Καθηγητής, Δρ Χρήστος Α. Παπαδόπουλος

Καθηγητής, Δρ Νικόλαος Αναφαντής

Καθηγητής, Δρ Michel Fillon

Καθηγητής, Δρ Αθανάσιος Μιχαηλίδης

Καθηγητής, Δρ Ασπράγκαθος Νικόλαος

Καθηγητής, Δρ Παύλος Χατζηκωνσταντίνου
Meeting Record

Of the Seven Member Examination Committee for the judgement of the Doctoral Dissertation of
the candidate doctor Mr. Dimitrios Bompos

The Three Member Advisory Committee for the Doctoral Dissertation of the candidate Doctor Mr.
Dimitrios Bompos, Dipl. Mechanical Engineer of the Department of Mechanical Engineering and
Aeronautics of Patras University, entitled:

«Tribological Design of Journal Bearings using Nano-Magnetorheological Fluids»,

was composed with the decision of the Department of Mechanical Engineering and Aeronautics,
University of Patras, in the General Assembly with Specific Composition Nr. 3 of 20/03/2012 and
comprises of:

1) Pantelis G. Nikolakopoulos, Supervisor (according to the decision of the General Assembly with
Special Composition Nr. 5 of 24/02/2015), Assistant Professor of the Department of Mechanical
Engineering and Aeronautics, University of Patras, in the discipline of "Machine Elements -
Tribology",

2) Chris A. Papadopoulos, Professor of the Department of Mechanical Engineering and Aeronautics,
University of Patras, in the discipline of "Computer Aided Machine Design"

3) Nikolaos K. Anifantis, Professor of the Department of Mechanical Engineering and Aeronautics,
University of Patras, in the discipline of “Analysis and Design of Constructions and Machines with
emphasis on Calculation Methods”

And was supplemented with the decision of the 1st General Assembly with Special Composition,
06.10.2015 with the following members:

- Michel Fillon, Professor in the Mechanics, Structures and Complex Systems Department, of Institute
  Pprime of University of Poitiers, France in the discipline of: “Hydrodynamics of Journal and Thrust
  Bearings”.

- Athanasios Mihailidis, Professor in Mechanical Engineering Department of Aristotle University of
  Thessaloniki, Greece in the discipline of “Machine Elements Design, Structural Analysis and
  Synthesis of Machines, Tribology”.

- Nikolaos Aspragathos, Professor of the Department of Mechanical Engineering and Aeronautics of
  UoP, Greece in the discipline of “Dynamics of Electromechanical Systems with Emphasis in
  Robotics”.

[1]
Prof. Pavlos Hatzikonstantinou, of the Department of Mechanical Engineering and Aeronautics of UoP, Greece in the discipline of “Applied Mathematics with Emphasis on the Use of Mathematics and Computational Methods in Solving Physical and Mechanical Problems of the Broad Engineering Interests”.

The examination committee met today, Friday, 05.02.2016 at 17:00, in the Conference room of the Department of Mechanical Engineering and Aeronautics. During the meeting the candidate doctor Mr. Dimitrios Bompos developed publicly and in extent his doctoral dissertation in front of the examination committee and after he answered all the submitted questions the examination was terminated.

Subsequently, in the private session of the examination committee, after discussion, it was decided unanimously that the dissertation is an original and substantial contribution to the promotion of science in the field of the journal bearings and more specifically in the tribological design of journal bearings using magnetorheological and nanomagnetorheological fluids. After roll-call voting, the members of the examination committee voted unanimously in favor of awarding the title of Doctor to Mr. Dimitrios Bompos, while the dissertation was rated «Excellent».

The nomination of the candidate in doctor will take place in the General Assembly with Special Composition of the Department of Mechanical Engineering and Aeronautics, University of Patras, according to the Article 13 of the Greek Law 2083/92.

The Seven Member Examination Committee,

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<td>Professor, Dr. Pavlos Hatzikonstantinou</td>
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This thesis is dedicated to my parents Thanos and Helene, for their support
to my wife Evangelia, for her encouragement
my daughter Helene, for her smile
and my supervisor, for his weekends away from home.
Tribological Design of Nano/Magnetorheological Fluid Journal Bearings

Thesis title: *Tribological Design of Nano/Magnetorheological Fluid Journal Bearings*

Dimitrios A. Bompos

This thesis is submitted in partial fulfillment of the requirements for the degree of Doctor of Philosophy

Supervisor: Assistant Professor Pantelis G. Nikolakopoulos

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Acknowledgements

This work has started at mid-2010 when the financial crisis has already made its presence in Greece. During this period, this work has been funded by the Research Comitee of the University of Patras under the auspices of the Karatheodori 2009 program. The program enabled me and my advisor to conduct high-end research using resources of the University of Patras. Moreover, three journal papers and six presentations have been funded by the program, in the same time that our country faced severe austerity measures. And it the support didn't seized there. After three years of work under the "Karatheodoris 2009" C.923 program with title "Tribological Design of Nano/Magnetorheological Fluid Journal Bearings", the Research Committee of the University of Patras continued its support to our work and honored myself and my advisor with the "Karatheodoris 2013" E.039 program which enabled me to complete my PhD focused on what I had to do rather than worry about my welfare in a country in deep recession. This crucial support comes from the redistribution of resources from programs that the University was awarded mainly from the EU. So, it is thanks to the hard work of colleagues of these Laboratories and Researchers in our University that this work was accomplished, despite the economic hardship that Greece endures. I have to underscore my profound gratitude to the colleagues at the University of Patras that I will never know in person. Thanks to their efforts I was allowed to complete this Dissertation. Last but not least I would like to thank my advisor, Prof. Pantelis G. Nikolakopoulos, for his long hours in the lab, working with me even during weekends away from his family and his children.
This PhD Thesis was funded by the "Karatheodori 2009" C.923 and "Karatheodori 2013" E.039 programs.
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Abstract

The journal bearings are components of machines that support rotating shafts. The available damping capacity and their low friction are some of their advantages. Examples of use of the journal bearings extent from micro applications, such as hard disc drives, to large size applications, such as bearings of gas turbine shafts. With the evolution of "smart" materials, i.e. materials with controllable physical properties, the perspective of active adjustments of journal bearings' performance has been made possible. In this work the magnetorheological (MR) and the nanomagnetorheological (NMR) fluids have been examined as possible candidate fluids for use as lubricants in journal bearings.

MR fluids are solutions of ferromagnetic particles into a conventional mineral oil lubricant. A suitably orientated magnetic field aligns the particles into chains, which hinder the lubricant flow. Thus, the apparent viscosity of the MR fluid is increased. In its active state, the MR fluid tends to behave as a viscoplastic solid. Under a certain amount of shear stress, the particle chains break and the MR fluid flows as a liquid. This amount of shear stress is called yield stress. The magnetic field intensity has the ability to adjust the yield stress of the MR fluid, making the control of the rheological behavior of this type of fluid possible.

A subcategory of the MR fluids is the NMR fluids. The major difference between the two is the smaller particle size of the NMR fluids. In the case of MR fluids, the particles may size from 100 to 10 \( \mu \text{m} \). The major drawback of these fluids is the sedimentation of the particles. NMR fluids employ particles which size less than 10 \( \mu \text{m} \) and up to several nanometers. The main effect of the decreased particle size is a higher viscosity that may offer significant damping. Additionally, the smaller size of the particles prevents the formation of the intermolecular forces that cause sedimentation. Ultimately while the yield stress of the NMR fluids is rather low, i.e. lower of the MR fluids, the control of the properties of the NMR fluids may become easier.

In this PhD thesis, the simulation of a MR/NMR fluid journal bearing is presented. The effect of the field intensity on the performance of the journal bearing is discussed. The main performance parameters such as the relative eccentricity, the normalized friction coefficient, the lubricant flow and the dynamic coefficients are presented for a
range of Sommerfeld number values. The rotordynamic behavior of a shaft using a MR and NMR fluid journal bearing is presented. Moreover, the capacity of the MR fluid journal bearing to compensate for wear present on the bushing of the bearing is examined, using the standard wear pattern present in journal bearings, due to contact between the journal and the bushing.

The rheological behavior of the MR and NMR fluids was studied mostly using a macroscopic viscosity model. However, the influence of the particles on the flow of the lubricant was also examined and useful conclusions were drawn concerning the validity of the macroscopic approach.

Ultimately, the journal bearing performance using an in-situ prepared NMR fluid is examined experimentally. The dynamic coefficients are determined and the orbit of the shaft is presented with and without the presence of the magnetic field.

The most important conclusions drawn from the experiments included in this thesis is the ability of the nanomagnetorheological fluid journal bearing to reduce the extent of orbit, increase its damping capacity and activate rapidly. One major drawback that is matter of future research is the ability of the magnetorheological and nanomagnetorheological fluids to be used with some sort of filtering devices in the lubricant circuit. This would enhance the quality of the base fluid and would allow longer periods between maintenance inspections. All in all, despite some practical aspects that remain to be solved, the ability of control of the load carrying capacity, damping and friction coefficient is significant and certainly worth of further development.

Περίληψη
Τα έδρανα ολίσθησης είναι στοιχεία μηχανών τα οποία υποστηρίζουν περιστρεφόμενους άξονες. Η ικανότητα απόσβεσης κραδασμών και η χαμηλή τριβή συγκαταλέγονται στα πλεονεκτήματά τους. Παραδείγματα της χρήσης των εδράνων ολίσθησης βρίσκουμε τόσο σε εφαρμογές μικρής κλίμακας μεγέθους, όπως οι σκληροί δίσκοι, όσο και εφαρμογές μεγάλης κλίμακας μεγέθους τα έδρανα αξόνων αεριοστροβίλων. Με την εξέλιξη των "ξυπνών υλικών", δηλαδή υλικών με
Τα μαγνητορεολογικά ρευστά είναι διαλύματα φερομαγνητικών σωματιδίων σε ένα συμβατικό ορυκτέλαιο. Ένα κατάλληλα προσανατολισμένο μαγνητικό πεδίο ευθυγραμμίζει τα σωματίδια σε αλυσίδες, οι οποίες εμποδίζουν την ροή του λιπαντικού. Με αυτόν τον τρόπο, το φαινόμενο έξωδες του μαγνητορεολογικού ρευστού αυξάνεται. Στην ενεργό του κατάσταση, το μαγνητορεολογικό ρευστό τίνει να συμπεριφερθεί όπως ένα ιξωδοπλαστικό στερεό. Υπό την επίδραση διατμητικών τάσεως συγκεκριμένης έντασης, οι αλυσίδες διαρρηγνύονται και το μαγνητορεολογικό υλικό ρέει ως υγρό. Το ύψος της διατμητικής τάσης κάτω από την οποία προκύπτει η ροή, ονομάζεται τάση διαρροής. Η ένταση του μαγνητικού πεδίου μεταβάλει την τάση διαρροής του μαγνητορεολογικού ρευστού, επιτρέποντας τον έλεγχο της ρεολογικής συμπεριφοράς των ρευστών αυτού του τύπου.

Μια υποκατηγορία των μαγνητορεολογικών ρευστών είναι τα νανομαγνητορεολογικά ρευστά. Η μεγάλη διαφορά μεταξύ τους είναι το μικρότερο μέγεθος των σωματιδίων των νανομαγνητορεολογικών ρευστών. Στην περίπτωση των μαγνητορεολογικών ρευστών τα σωματίδια έχουν διάμετρο μεταξύ 10 εως και 100 μ.μ. Τα νανομαγνητορεολογικά ρευστά έχουν διάμετρο μικρότερο των 10 μικρούς και μερικά νανόμετρα. Το κύριο αποτέλεσμα του μικρότερου μέγεθους των σωματιδίων των μαγνητορεολογικών ρευστών είναι ότι μπορούν να προσφέρουν σημαντικά αυξημένη απόδοση. Επιπρόσθετα, το μικρότερο μέγεθος των σωματιδίων προλαμβάνει το σχηματισμό των διαμοριακών δυνάμεων που προκαλούν ιζηματοποίηση. Τέλος, ενώ η τάση διαρροής είναι χαμηλότερη στα νανομαγνητορεολογικά ρευστά, εντούτοις η ικανότητα ελέγχου των ιδιοτήτων τους είναι ευκολότερη.

Σε αυτή την Διδακτορική Διατριβή, παρουσιάζεται προσομοίωση των εδράνων με μαγνητορεολογικά και νανομαγνητορεολογικά ρευστά. Η επιρροή της έντασης του μαγνητικού πεδίου στις επιδόσεις του εδράνου αναλύεται. Οι κύριες παράμετροι των επιδόσεων του εδράνου, όπως η σχετική εκκεντρότητα, ο κανονικοποιημένος συντελεστής τριβής, η ροή του λιπαντικού και οι δυναμικοί συντελεστές
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Τα πιο σημαντικά συμπέρασματα που προέκυψαν από τα πειράματα που περιλαμβάνονταν σε αυτή τη διατριβή είναι η ικανότητα των εδράνων να μειώσουν την έκταση της τροχιάς του άξονα, να αυξήσουν την ικανότητα απόσβεσης και να ενεργοποιηθούν ταχέως. Ένα σημαντικό μειονέκτημα που αποτελεί αντικείμενο για περαιτέρω έρευνα είναι η αδυναμία των μαγνητορεολογικών και νανομαγνητορεολογικών ρευστών να χρησιμοποιηθούν σε συνδιασμό με κάποιο τύπο φίλτρου λιπαντικού. Αν αυτό το μειονέκτημα αίρονταν, η ποιότητα του λιπαντικού θα βελτιωνόταν και θα επιτρέπονταν μεγαλύτερος χρόνος μεταξύ επιθεωρήσεων συντήρησης. Εν τέλει, παρά κάποια πρακτικά ζητήματα που μένουν να επιλυθούν, η ικανότητα ελέγχου της παρεχόμενης στήριξης από το έδρανο, της απόσβεσης και του συντελεστή τριβής είναι σημαντική και αναδυκνύει την ανάγκη για περαιτέρω ανάπτυξη.

Dimitrios A. Bompos
1 Introduction

1.1 Journal bearings

Journal bearing performance is studied by a growing number of researchers, using both theoretical and experimental methods. Connors [1] studied the effect of lubricant supply rate in the case of a long bearing in a theoretical study. One of the most important papers concerning journal bearing design was the work of Raimondi and Boyd [2]. In this work the numerical solutions of Reynolds' equation are obtained for centrally-loaded bearings and various cases of full and partial arc journal bearings. Assumptions are based on constant viscosity and no film rupture. Gertzos et al. [3] presented a CFD analysis in which for Bingham lubricated journal-bearing performance characteristics, such as relative eccentricity, attitude angle, pressure distribution, friction coefficient, lubricant flow rate, and the angle of maximum pressure, are derived and presented for several length over diameter ($L/D$) bearing ratios and dimensionless shear numbers of the Bingham fluid. The above diagrams presented in the form of Raimondi and Boyd charts, and can easily be used in the design and analysis of journal bearings lubricated with Bingham fluids. Shenoy et al. [4] used ANSYS CFX in order to perform a hydroelastic analysis and stress calculation of a plain journal bearing. Li et al. [5] applied computational fluid dynamics coupled with rotordynamics simulation of a rotor.

Thomsen et al. propose a new technique of avoiding misalignment with bearing flexibility and compliant liners [12]. In [13] Jiugen et al. studied the effects of misalignment on thermal hydrodynamic lubrication in multi-pad journal bearing. Glienicke et al. [14] calculate the journal bearing dynamic coefficients both analytically and experimentally for various cases of journal bearing geometry including plain four lobe and lemon type bearing. A complete review of the available methods developed for the calculation of the stiffness and damping coefficients and identification of the dynamic properties of journal bearings is provided by Tiwari et al. in [15]. The use of surface modification in order to achieve performance improvement in journal bearings has been the subject of numerous works [10, 16]. The main issue with the application of surface texturing techniques in journal bearings is the fact that possible benefits are limited in a specific set of operating conditions. Lu and Khonsari [17] have shown that dimples may affect positively the journal bearing performance especially under boundary lubrication regime. On the other hand, the application of artificial surface texturing in thrust bearings and parallel sliders has shown higher potential benefits in comparison to journal bearings as shown in [18, 19]. Since the benefits from artificial texturing application occur under specific operating conditions [20], making the most out of this technique requires active control over the operating conditions themselves.

1.2 Wear in journal bearings

Wear is a type of surface damage that arises from the relative motion between interacting solid surfaces. It is a dynamic and complex process that incorporates surface and material properties, operating conditions, stresses, lubricants and geometry. Wear plays an important role in determining life span of machine elements. The lifetime of a component depends on wear. It is difficult to measure with precision wear due to its dynamic and complex nature. The onset and development of wear in plain hydrodynamic journal bearings under repeated stop/start cycles have been studied experimentally by Mokhtar [21]. Hashimoto et al [22] calculated the main performance characteristics of worn journal bearings using a model based on Reynolds equation which they validated with experimental data. Dufrane et al. [23] investigated wear in steam turbines and took measurements during overhaul periods to determine the extent and nature of the wear. They established two models of wear geometry for use in further analysis of the effect of wear on hydrodynamic
lubrication. Fillon et al. [24] performed a thermohydrodynamic analysis of a worn plain journal bearing, using the generalized Reynolds equations along with Elrod algorithm for cavitation. They were able to calculate the main performance characteristics of a worn journal bearing.

1.3 Rotordynamics

The simulation of rotating machinery is always an active item for research. Chouksey et al. [25] performed modal analysis taking into account the fluid film forces and the rotor material damping. In [26] L. San Andres discusses the role of inertia in the calculation of rotodynamic force coefficients and their effect on the stability of a rotor-bearing system. R.G. Kirk et.al [27], evaluated the stability and transient response of a high-speed automotive turbocharger. They used various computer models with different bearing designs and properties to obtain the linear stability threshold speeds and also the nonlinear transient response, in the high speed automotive turbochargers.

Using Navier Stokes equations in the calculation of fluid film pressure on a journal bearing is a demanding computational task. On the other hand, Reynolds equation, the most widespread method of calculating the lubricant pressure, ignores the inertial phenomena. The inclusion of the inertial phenomena in the Navier Stokes equations provides an accurate method of calculation of the journal bearing's dynamic charasterics and their stability [28]. Chasalevris et. al [29] solved the Reynolds equation and calculated the bearing dynamic characteristics in order to extract the dynamic response of a defective multi-bearing shaft system. In [30] Gounaris and Papadopoulos investigated the feasibility of crack identification in rotating shafts by coupled response measurements.

1.4 Smart fluids

Smart fluids are fluids whose physical properties may be controlled through the application of an external electric or magnetic field. Electrorheological and magnetorheological fluids consist the two major categories of smart fluids. Vance and Ying [31] developed and demonstrated the dynamic behavior of the rotor systems supported on the multi-disk ER fluid damper. Dimarogonas and Kollias [32, 33] studied stability of a rotor system supported by journal bearings with ER fluid theoretically and compared the capability of three kinds of ER fluid damper for
controlling the rotor vibration. Nikolajsen and Hoque [34] presented a multi-disk ER fluid damper operating in shear flow mode and studied the effectiveness of the multi-disk ER fluid damper in controlling the vibration of rotor systems when passing through the critical speeds. Nikolakopoulos and Papadopoulos [35] studied the dynamic characteristics of a controllable journal bearing lubricated with the ER fluid. Based on the Bingham fluid theory, Tichy [36], Dorier and Tichy [37] analyzed the dynamic characteristics of fluid film force and the existing conditions and manner of core in the ER fluid squeeze-film damper and journal bearings.

1.5 Magnetorehological fluids applications

Magnetorheological fluid (MRF) is a manageable fluid that exhibits drastic changes in rheological properties adjustable and interchangeable to the applied magnetic field strength. The fluid is potentially advantageous to be employed in many applications. MRF is a kind of controllable or smart fluids whose rheological properties can be dramatically and reversibly varied by the application of an external magnetic field in a very short period of time. The MRF has the property of a normal viscosity in the absence of an external magnetic field, but in the presence of a strong magnetic field immediately solidifies to a grease state.

The MRF’s are one of the most active “smart materials” of the current range. Most research in the application of the MRF’s is focused on structural vibration control and flow power system. The use of MR fluids in various applications, such as dampers [38, 39], brakes [40] and surface finishing [41] where control of friction and vibrations becomes a useful capability, stimulated research concerning the field of journal bearing lubrication [42, 43].

Stanway et al. [44] and Wang and Meng [45] made a survey study in the state of the MRF’s and the application of the MRF’s in several mechanical engineering systems. There are many papers dealing with the application of the magnetorheological fluids for controllable dampers [46-50] for seismic response control of frame structures [51] and vibration control of large structures [52]. The rapid, reversible and dramatic changes in its rheological properties provide a possibility of control in flow power systems [53, 54]. Salloom et al. [55] present a comparative simulation of various valve designs using MR fluids. Wiltsie et al. [56] present a robot which uses a MR fluid as an adhesive matter which permits the robot to climb.
1.6 Rheology of magnetorheological fluids

López-López et. al. [57] investigated the interaction between micron-sized magnetizable particles dispersed in a ferrofluid upon application of a magnetic field. NMR fluids contain small particles (d<=30nm) made of Fe₂O₃. This results in a weaker MR effect than the one present in conventional MR fluids. On the other hand the sedimentation stability of these magnetic fluids is higher [58]. The rheological behavior of MR and NMR fluids can be simulated using the Bingham plastic model. Tichy [59] proposed a differential method with an iterative algorithm to study Bingham plastic flow in a journal bearing.

Zhang et al [38] study the role of sedimentation after a long time of lack of motion on the effectiveness of a MR fluid damper for automotive applications. Mitsoulis [60], presented a review work for several problems of viscoplastic flows, such as entry and exit flows from dies, flows around a sphere and a cylinder, and squeeze flows. The high density of magnetorheological fluids is set to induce significant inertial effects for which Navier Stokes are better suited than Reynolds [61, 62]. Ghaednia et al. [63] study the performance of magnetorheological fluid journal bearings under the influence of temperature. A significant aspect of the rheology of the magnetorheological fluids is temperature. While in conventional lubricants the rise of temperature is known to be linked with drop of viscosity, in the case of MR fluids the relationship of temperature and viscosity is not linear. For instance in [64] Chen et al. suggest that the MR fluid viscosity may become insensitive to temperature or even increase in certain cases. At any rate, the existence of particles will always give a relative advantage to the MR fluid in comparison to its base oil.

1.7 Nanomagnetorheological fluids

The size of the paramagnetic particles in MR fluids ranges between 10µm and 100µm. While in many applications this size is acceptable, the case of the journal bearings is somewhat different. Typical values of radial clearance in journal bearings range 50 to 200µm.
It is evident that the size of the paramagnetic particles in this case may become an obstacle, some of the particles would be left out of the lubricant film of the journal bearing and the performance benefits expected from the application of the magnetic field would not ensue. Thus smaller size particles would be more suitable in the application of the MR fluid in journal bearings. On the other hand, while some rheological benefits are present in the case of Ferrofluids (FR fluids), which make use of particles with size less than 30nm, (e.g. lower sedimentation), their yield stress is extremely limited. This limitation is linked with the high porosity of iron based particles of this very low size. The intermediate category of NanoMagnetoRheological (NMR) fluids in which particles with size between several tens of nanometers and tens of micrometers, is better suited for use in the limited space of journal bearings, with limited sedimentation [65] and higher yield stress.

Kim et al. [66] have conducted a comparative study on the effect of particle size to the properties of a MR fluid. More specifically they compared the performance of a MR and a NMR fluid on apparent viscosity, yield stress and other parameters. Moving a step forward, NMR fluids have also been proposed for certain applications. Safarik et al [67] studied the applicability of magnetic nanoparticles in biomedical applications, while Fijalkowski [68] introduces a novel internal combustion engine, which uses a NMR mechatronic commutator as a replacement of the crankshaft and the connecting rod mechanisms. The particle size of the paramagnetic particles, inside the MR fluid volume, plays a significant role in the rheological behavior of the MR fluids and their physical properties. It is important to suggest how this parameter can contribute to the
performance of this kind of fluids. Chaudhuri et al. [69], presented the effects of substitution of micron size powder by nanometer size powder in MR fluids. They found that by mixing a reasonable percent of nano iron powder in the NMR fluid, a substantial change in the rheological characteristics is obtained. Vekas [65] examines the similarities and main differences between the MR and NMR fluids through various examples.

1.8 Magnetorheological fluid journal bearing and rotor dampers

Hesselbach and Abel-Keilhack [42] investigated the influence of the magnetic field on the bearing gap of hydrostatic bearings with MRF’s. They found that, in a closed loop control, a nearly infinite stiffness, only limited by the resolution of the measuring system, can be achieved. The results showed that the concept of a hydrostatic bearing with MRF’s can overcome the drawbacks (stiffness and response time) of conventional hydrostatic bearing.

Kim et al. [70], presented a controllable semi-active smart fluid damper (SFD) using magnetorheological fluids, focusing on its design and modeling. It offers a comprehensive design method and an innovative experimental identification and modeling technique for MR-SFDs. They constructed a prototype MR-SFD and investigated how some of the critical design parameters affect the performance of the MR-SFD. Additionally they characterized the damper’s dynamic behavior experimentally using a novel excitation method that adopts active magnetic bearing (AMB) units. In modeling the dynamic behavior of the MR-SFD, they employed the describing function method. The describing function analysis effectively captured the non-linear dynamic behavior of the MR-SFD. Carmignani et al. [71] presented an analytical, numerical and experimental study off a magnetorheological squeeze-film damper. Numerical simulations were carried out in order to evaluate the dynamic behaviour of the damped rotor as a function of the current supplied to the adjustable device. A linear model that depicts the main characteristics of the system has been developed as a useful tool in damper and control design. They tested different fluids, and an optimal fluid has been singled out. The tests conducted on the selected fluid shown that it is possible to have the optimum conditions for each steady rotational speed.
Urreta et al. [43] summarizes the work carried out in the development of hydrodynamic lubricated journal bearings with magnetic fluids. Two different fluids have been analyzed, one ferrofluid from FERROTEC APG s10n and one magnetorheological fluid from LORD Corp., MRF122-2ED. Theoretical analysis has been carried out with numerical solutions of Reynolds equation, based on apparent viscosity modulation for ferrofluid and Bingham model for magnetorheological fluid.

The authors in order to validate their model, designed, manufactured and set up a test bench, where their preliminary results shown that magnetic fluids can be used to develop active journal bearings. The design of a magnetorheological squeeze-film damper is presented and discussed by Forte et al. [72]. A numerical simulation has been carried out in order to evaluate the dynamic behavior of the damped rotor as a function of the magnetic field strength. The authors made a test rig of a slender shaft supported by two oilite bearings and an unbalanced disk. The damper was interfaced with the shaft through a rolling bearing and the electric coils generate the magnetic field whose field lines cross the magnetorheological film.

Kuzma in [73] presents an analysis of an infinite magnetohydrodynamic journal bearing, using a modified form of the Reynolds' bearing equation. Unlike the MR fluids, in this case the liquid is not a standard lubricant but rather a homogenous fluid which can be magnetized.

While the MR fluids offer high yield stress and thus controllability, certain limitations, such as sedimentation of the iron based particles of the MR fluids, have motivated some research towards the use of NMR fluids [74].

1.9 Objectives of this PhD thesis

Magnetorheological (MR) fluid technology is a relatively new addition in the market of smart lubricants which expands very rapidly. MR fluids technology has been successfully employed already in various low and high volume applications. Among other examples, the automotive industry paved the way for the commercial use of MR fluids in clutches, dampers and brakes. The use of MR fluids offers the ability of control, create the opportunity of increased functionality and lower maintenance costs, especially in these applications where intense vibrations are expected.
The overall objective of this research was to study the advantages and disadvantages of MR and Nano MR fluids in journal bearing applications from a tribological and rotor dynamics perspective. Although MR fluids are investigated thoroughly since they were first discovered, very little research has been performed on the tribological design of the journal bearings. More specifically, the objectives of this research were:

1. To simulate the magnetic field and to investigate the influence of MR fluids on the static performance of journal bearings. The influence of magnetic field intensity for a range of several bearing's slenderness ratios is investigated, in terms of bearing load capacity and friction force.

2. To investigate the off-state tribological behavior of journal bearings using MR fluids where the magnetic field is absent. There is not much research reported on the off state journal bearing lubricated by MR fluids.

3. To investigate the dynamic properties, stiffness and damping coefficients of the MR and NMR fluid lubricated journal bearing:
   3.1. by developing an NMR/MR fluid journal bearing test ring and measuring the dynamic properties
   3.2. by developing an appropriate code in order to calculate the stiffness and damping coefficients
   3.3. to create in situ Nanomagnetorheological fluids and to test them in the constructed test ring regarding the damping and stiffness coefficients.
   3.4. to investigate the stability of the dynamic system

4. To simulate and to study a rotor dynamic system, that is supported by on state MR and NMR fluids as well as the of state relevant situations. The relevant frequency responses were predicted and compared.

5. To investigate how artificial texturing affects the performance of magnetorheological fluid journal bearing.

6. To examine analytically the microrheology of the magnetohreological fluid, examine how exactly the particles interact with the carrier fluid, the bushing and the journal as well as with the magnetic field.
Outline of the Ph.D. Dissertation

The remainder of this dissertation is organized as follows: The theoretical background for the simulations is explained in detail in Chapter 2. More specifically, the rheology of the MR and NMR fluids was simulated using the Navier-Stokes equations which are described in Chapter 2.1.1. The Bingham model, discussed in 2.1.2, has been used for the approximation of the MR and NMR fluids apparent viscosity. Since the problem at hand has an electromagnetic component, the Gauss Law and the Maxwell's equations are described in Chapter 2.1.3. The boundary conditions of the various physical aspects of the MR and NMR fluid journal bearings are described in Chapter 2.2. More specifically, the boundary conditions for the fluid domain are described in Chapter 2.2.1, while the magnetic field boundary conditions are presented in Chapter 2.2.2. The static performance parameters are defined in Chapter 2.3. The stiffness and damping coefficients which describe the dynamic performance of the journal bearing are described in Chapter 2.4. Finally the equations that describe the dynamic behavior of the shaft are given in Chapter 2.5.

The static performance of the magnetorheological fluid journal bearing is described in Chapter 3, along with a study of the effect of the coils proportions on the distribution of the magnetic field inside the journal bearing. The capacity of the MR fluids to compensate for the wear present due to the contact of the journal and the bearing during start-up and shut-down is discussed in Chapter 4. In Chapter 5 the dynamic coefficients of the MR and NanoMR fluids are presented. The rotordynamic behavior of a shaft using conventional, MR and NMR fluids is discussed in Chapter 6. The effect of the artificial texturing on the bushing of a journal bearing using MR fluids is discussed in Chapter 7. The microrheology of the MR fluids is discussed in Chapter 8 along with the effect of temperature in the performance of the NMR/MR fluid journal bearing. An experimental investigation of a NMR fluid journal bearing is presented in Chapter 9. Finally, the most significant conclusions are summarized in Chapter 10.
Nomenclature

\( C \).radial clearance \( C = R_b - R_j \)
\( e \)eccentricity (m)
\( f_b \)friction coefficient on the bearing surface
\( f_j \)friction coefficient on the journal surface
\( f^* \)normalized friction coefficient \( f^* = f(R/C) \)
\( \dot{g} \)acceleration of gravity vector
\( h \)film thickness (m)
\( H \)magnetic field intensity (A/m)
\( \dot{B} \)magnetic field density
\( I \)unit tensor
\( I \)current (A)
\( \dot{j} \)current density (A/m²)
\( L \)bearing length (m)
\( N \)rotational speed of journal (rps)
\( p \)pressure (Pa)
\( p^* \)dimensionless pressure
\( p_{max} \)maximum pressure on the journal surface (Pa).
\( R_j, R_b \)journal and bearing radii (m)
\( S \)Sommerfeld number
\( S = \mu \cdot \omega \cdot R_j \cdot L(R_j/C)^2/(\pi \cdot W) \)
\( T_0 \)dimensionless yield stress
\( \tau_0 = \tau_0 \cdot C/(\mu_0 \cdot \omega \cdot R_j) \)
\( \ddot{u}, \ddot{v}, \ddot{w} \)fluid velocity vectors
\( W \)external force (N)
\( w^* \)dimensionless load carrying capacity
\( w^* = W \cdot C^2/(\mu \cdot \omega \cdot R_j^3 \cdot L) \)
\( U \)Journal velocity, parallel to the film
\( K_{ij} \)Stiffness properties \( i,j=x \) or \( y \)
\( K^* \)Dimensionless Stiffness
\( K^* = \frac{c \cdot K}{W} \)

\( C_{ij} \)Damping properties \( i,j=x \) or \( y \)
\( C^* \)Dimensionless Damping
\( C^* = \frac{C \cdot \omega \cdot e}{W} \)

Subscripts
\( 0 \)indicates the position of maximum film thickness at \( \theta = 0 \).
\( 1 \)indicates the position of minimum film thickness at \( \theta = \pi \).
\( b \)indicates the bearing.
\( j \)indicates the journal.

Greek symbols
\( \varepsilon \)relative eccentricity \( \varepsilon = e/C \)
\( \rho \)lubricant density (kg/m³)
\( \mu \)lubricant viscosity (Pa s)
\( \mu_a \)apparent viscosity (Pa s)
\( \mu_p \)plastic viscosity (Pa s)
\( \mu_0, \mu_f \)Newtonian component of apparent viscosity for Bingham model
\( \tau \)stress tensor
\( \tau_0 \)yield stress (Pa)

Patras 2015
2 Governing equations, assumptions and boundary conditions

The complex nature of the problem, i.e. the design of a magnetorheological fluid journal bearing, requires a series of multiphysics simulations. In order to calculate the flow of the lubricant the Navier-Stokes equations were solved, using the Bingham viscosity model in order to approximate the non Newtonian character of the MR fluid’s viscosity. The Bingham model parameters are determined based on a magnetic field simulation.

2.1 Navier Stokes equations

Concerning the lubricant flow, the Navier Stokes equations are used in the form of mass conservation:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{v}) = 0$$

(1)

and the momentum conservation equations:

$$\frac{\partial}{\partial t} \left( \rho \vec{v} \right) + \nabla \rho \vec{v} = -\nabla p + \nabla \cdot (\vec{f}) + \rho \vec{g} + \vec{F}$$

(2)

where $\rho \vec{g}$ and $\vec{F}$ are the gravitational and external body forces respectively.

The stress tensor $\vec{\tau}$ is given by:

$$\vec{\tau} = \mu \left[ \nabla \vec{v} + \left( \nabla \vec{v} \right)^{\top} - \frac{2}{3} \nabla \cdot \vec{v} \vec{I} \right]$$

(3)

The term $\frac{2}{3} \nabla \cdot \vec{v} \vec{I}$ is the effect of volume dilation.

2.2 The Bingham Viscosity Model

The viscosity of MR fluids can be approximated with the Bingham law for yield stress:

$$\tau = \tau_0(H) + \mu \dot{\gamma}$$

(4)

where $\tau$ is the shear stress of the material, $\tau_0$ the critical shear stress or yield stress and $\dot{\gamma}$ the shear rate. The relationship of the critical shear stress $\tau_0$ with the magnetic
field intensity $H$ can be estimated by experimental data. For certain MR fluids this relationship is available through manufacturer’s literature [75].

It is possible to obtain an equivalent or apparent viscosity:

$$\mu_a = \mu_f + \tau_y(H) \left| \frac{\partial \dot{u}}{\partial \gamma} \right|$$

where $\mu_a$ is the apparent viscosity of the material and $\mu_f$ is the Newtonian viscosity of the material when the shear stress overcomes the yield stress, in which case the material is flowing.

For the purposes of CFD simulation, the Bingham model is defined by two zones of viscosity. The first region is the plastic viscosity region where the material exhibits the Bingham solid behavior. In this region the viscosity takes a high value. This is the plastic viscosity or $\mu_p$. When the shear stress overcomes the yield threshold, the behavior of the Bingham material is described with the viscosity of flow or $\mu_f$. Thus the apparent viscosity of the Bingham model is mathematically defined as:

$$\mu = \begin{cases} 
\mu_f + \frac{\tau_p(H)}{\dot{\gamma}} \frac{\tau_p(H)}{\mu - \mu_f}, & \mu_f, \dot{\gamma} \leq \frac{\tau_p(H)}{\mu - \mu_f} \\
\mu_f, & \mu_f, \dot{\gamma} > \frac{\tau_p(H)}{\mu - \mu_f}
\end{cases}$$

Fig. 2. The bi-zone Bingham viscosity model
Typically the plastic viscosity is defined with a value of \( \mu_p = 100 \mu_f \) in order to replicate properly the Bingham behavior. The yield stress of the MR fluid is considered constant throughout the fluid volume for the purposes of the simulation. This assumption implies a homogenous magnetic field.

### 2.3 The Gauss Law and Maxwell's equation

The fundamental expression for the magnetic field is given by the Gauss law:

\[ \nabla \cdot \mathbf{B} = 0 \quad (7) \]

The differential form for Ampere's law including Maxwell’s correction is:

\[ \nabla \times \mathbf{H} = \mu_0 \mathbf{J} + \mu_0 \varepsilon_0 \frac{\partial \mathbf{E}}{\partial t} \quad (8) \]

where \( \mu_0 \) is the permeability of free space, \( \varepsilon_0 \) the free space permittivity, \( \mathbf{B} \) and \( \mathbf{H} \) becomes:

\[ \mathbf{B} = \mu_r \mu_0 \mathbf{H} \quad (9) \]

where \( \mu_r \) is the relative magnetic permeability of the material in which the magnetic density vector \( \mathbf{B} \) is calculated.

Equations (1) to (3) were solved using computational fluid dynamics (CFD). Also the equations (7) to (9) were solved with the finite element method.

Consequentially, concerning the magneto rheological fluid bearing, the pressure field was predicted using the set of equations (1) to (3), (4) to (6), and (7) to (9), combined the Bingham model with the magnetic field intensity.

### 2.4 Boundary conditions

#### 2.4.1 Fluid domain

The CFD model employed in this work is isothermal and the flow is considered to be under steady state. The boundary conditions involve the bushing, the journal and the lateral surfaces of the bearing, as shown in Fig. 3. The bearing is defined as a stationary wall with the velocity having zero velocity vectors in all directions. The journal is considered as a moving wall, in which there is only the tangential
component of the rotational velocity. Negative pressures are set to zero in order to account for cavitation. The pressure at the sides of the bearing is set equal to zero, functioning as a free flow boundary.

The model of the current work has been validated towards the work of Gertzos et. al [3]. In Fig. 4 we compare the results of the current work concerning relative eccentricity with those obtained in reference [3] for a bearing with non-newtonian lubricant in a range of Sommerfeld number values. The results concern a bearing with \(L/D=0.5\) with a fluid that exhibits dimensionless yield stress \(T_0=0.8\). The agreement of the acquired results with reference [3] is satisfactory.
2.4.2 Magnetic Field

In the magnetic part of simulation the main load is the current source density. Infinite space as a boundary condition lies on the outer limit of the simulated area and has zero magnetic potential. In other words we neglect any influence of the magnetic field outside the simulated space. Load capacity depends on the magnetic field distribution inside the bearing. One can observe that although the field in our case is not entirely homogenous, there is an adequate distribution within all the volume of lubricant inside the bearing. The relative boundary conditions for the magnetic simulation problem are shown in Fig. 5.

![Fig. 5. Boundary conditions of the magnetostatic simulation.](image)

The coils are with a certain load of current density $\vec{J}$. The most convenient way to express this load without taking into consideration the number of cables per unit area is the use of current density. The current density $\vec{J}$ is the ratio of total number of cables $n$ times the current intensity $\vec{I}$ of these cables divided by the cross section of the coil. A coil with higher number of cables has lower amperage for a certain value of current density. Current intensity has units of A/m$^2$.

The mean value of the magnetic field is the input used for the calculation of the apparent viscosity of the nano/magnetorheological fluid journal bearing. The calculation of the mean value of the magnetic field intensity $H$ is imperative since the field is not always homogenous. Thus it is reasonable to take into consideration the higher values of the magnetic field to a certain degree, developed locally within the volume of the lubricant’s film.
2.5 Microrheology simulation

In this simulation of the magnetorheological fluids, the rheological behavior is calculated by an iterative procedure.

The physical properties of the particles and the lubricant including viscosity and density along are introduced. The particles size \( (d=10^{-6}\text{m}) \) and the control volume dimensions are defined. The magnetic forces applied on the particles by an external homogenous magnetic field are calculated. The fluid forces applied on the particles by the lubricant are calculated in the next step through a CFD simulation. The kinematic state of the particles is computed then taking as a given the initial position and velocity of the lubricants, their size and mass and the forces applied on them. The final result occurs after a number of iterations during which the apparent viscosity is calculated. When the apparent viscosity value presents no significant change between two consecutive iterations, the calculation is considered complete and the simulation ends.
Fig. 7. The control volume simulated for the estimation of the rheological characteristics of the MR fluid.

As shown in Fig. 7, the size of the control volume was elected to be 180μm in length, 100μm in height and 100μm in width. In this scale the curvature of a radial journal bearing can be omitted. The boundary conditions used for the solution are also presented in the same figure. A homogenous magnetic field of 0.6 T is applied on the lubricant volume. The side of the bushing is considered a steady wall with no fluid velocity. The side of the bushing is considered a moving wall with the linear velocity of the journal. The rest of the areas of the control volume allow for the free flow of the lubricant. The particles are considered moving objects to which the hydrodynamic and magnetic forces are applied. Results have been calculated for two cases of rotational velocity of the journal at 2000 and 1500 rpm. The shaft has a radius \( R \), \( j=25\text{mm} \) and the slenderness ratio of the bearing is \( L/D=0.5 \).

2.5.1 Calculation of magnetic forces

The Maxwell magnetic forces are calculated as follows:

\[
\{F_{mx}\} = \frac{1}{\mu_0} \int \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \begin{bmatrix} n_1 \\ n_2 \end{bmatrix} ds
\]

(10)

where \( F_{mx} \) are the Maxwell forces, \( T_{11} = B_i^2 - \frac{1}{2} |B|^2 \), \( T_{12} = B_i B_j \), \( T_{21} = B_j B_i \), \( T_{22} = B_j^2 - \frac{1}{2} |B|^2 \) and \( n_i, n_j \) the unit surface normal in the global Cartesian coordinate system.

The kinematic of particles
The lubricant viscosity is considered a function of temperature. In this simulation physical properties of SAE30 are used. The problem is solved as a quasi static temperature problem. Viscosity values are considered for several temperature values in order to obtain solutions.

The Verlet integration method is used in order to calculate the movement of the particles inside the lubricant volume. The time step is defined as 1ns. The position of the particle is calculated as follows:

\[
\{x(t+1)\} = \{x(t)\} + \{v(t)\} \Delta t + \frac{1}{2} \{a(t)\} \Delta t^2
\]  

(11)

The particle velocity is computed as follows:

\[
\{v(t+1)\} = \{v(t)\} + \{a(t)\} \Delta t
\]

(12)

### 2.6 Performance parameters

The basic parameters that define the overall performance of the journal bearing are its load capacity and the generated friction. The load \( W \) that the journal bearing is able to support is calculated by integrating pressure \( p \) and the shear stress \( \tau \) along the circumference of the bearing. In a equilibrium position

\[
F_{pi} = \iint p_j dA_j = -W, \quad F_{pi} = 0
\]

(13)

where \( p_j \) is the pressure field in the circumference of the journal.

The friction force can be found as follows:

\[
F_{fi} = \iint \tau_i dA_i
\]

(14)

where \( F_{fi} \) is the friction force and \( i = j, b \) for journal or bearing respectively. The

The friction coefficient \( f_i \) is an important part of the performance characteristics of the journal bearing and is defined as follows:

\[
f_i = \frac{F_{fi}}{W}
\]

(15)
In order to describe the performance of the bearing in as general form as possible the dimensionless parameters of Sommerfeld number $S$, the normalized friction coefficient $f_i(R/C)$ and the dimensionless pressure $p^*$ have been used, having been defined as follows:

$$S = \mu \cdot \omega \cdot R \cdot L(R/C)^2 / (\pi \cdot W)$$

(16)

$$f_i(R/C) = F_{F,i} \cdot R \cdot W / C$$

(17)

$$p^* = (p - p_u) \cdot C^2 / (\mu \cdot \omega \cdot R_j^2)$$

(18)

Finally in order to establish that the flow is in laminar regime, the Reynolds number must be included in the calculations.

$$Re = \rho \cdot \omega \cdot R_j \cdot C / \mu$$

(19)

### 2.7 Numerical calculation of Dynamic Coefficients

The journal bearing system with its characteristics approximating its dynamic behavior is shown in Fig. 8.

![Diagram of the journal bearing](image)

Fig. 8. The stiffness and damping coefficients of the journal bearing.

The dynamic behavior of the journal bearing can be simplified with a substitute system of springs and dampers. We have the main and the coupled terms of stiffness $K_{ij} = \frac{\partial F_{ij}}{\partial i}$ and damping $C_{ij} = \frac{\partial F_{ij}}{\partial u_j}$ where $i,j=x$ or $y$. 

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The size of disturbance is a major issue when the Navier-Stokes equations are used. The logical choice would be to use a very small disturbance in order to achieve calculation of the dynamic coefficients with as minimum variation of journal translational speed as possible, thus calculating the derivative in the linear region. The use of such a disturbance yields very high values of the damping coefficients for all relative axes. This contradicts with the experimental results and the theoretically calculated values of the damping coefficient with the use of Reynolds equations. Thus there is a factor inside the formulation of the Navier Stokes equation that makes the difference between the two methods. The difference which occurs between the two methods has to be traced in the equations themselves. The continuity equation is expressed as:

\[ \nabla \vec{V} = 0 \]  

(20)

and the momentum equation as

\[ \frac{\partial \vec{V}}{\partial t} + \vec{V} \nabla \vec{V} = -\frac{1}{\rho} \nabla P + \frac{\mu}{\rho} \nabla^2 \vec{V} \]  

(21)

where \( \vec{V} \) is the vector of fluid velocity, \( P \) is fluid pressure, \( \mu \) is the fluid viscosity and \( \rho \) is the fluid density. Reynolds equation takes the form

\[ \frac{\partial}{\partial \theta} \left( h \frac{\partial P}{\partial \theta} \right) + \left( \frac{R}{L} \right) \frac{\partial}{\partial z} \left( h^3 \frac{\partial P}{\partial z} \right) = 6 \mu R \left[ \frac{h}{L} \frac{\partial}{\partial t} \right] + \frac{\partial h}{\partial \theta} + \frac{\partial h}{\partial t} \]  

(22)

where \( \theta \) is the circumferential coordinate of the bearing, \( h \) is the lubricant thickness, \( \omega \) is the rotational velocity, \( L \) is the bearing length and \( R \) is the journal radius.

It is evident that Reynolds equation does not include velocity. Furthermore, velocity in the case of momentum equation is related to its gradient. This means that when the disturbance is applied on the shaft, the velocity terms are changing non linearly as follows:

Additionally in the case of Reynolds equation, \( \frac{\partial P}{\partial \vec{V}} = 0 \). This is not the case for the Navier Stokes equations.
Thus the two methods could have a more similar behavior in the range of higher slenderness ratio values. Momentum equation for y axis can be expressed as:

\[
\rho \left( \frac{\partial v}{\partial t} + u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z} \right) = -\frac{\partial P}{\partial y} + \mu \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right) + \rho \cdot g \quad (23)
\]

where \( \rho \) is the lubricant density, \( u, v, w \) are the three velocity components for x, y and z axes respectively. In order to calculate the damping coefficients we apply a small disturbance on the shaft motion. Let us assume that an impact on the shaft accelerates it instantaneously on the y direction.

Since the velocity field is uniformly disturbed throughout the x and z axes expression (20) without loss of generality can be written as:

\[
\rho \left( \frac{\partial (v + v_o)}{\partial t} + (v + v_o) \frac{\partial (v + v_o)}{\partial y} \right) = -\frac{\partial P}{\partial y} + \mu \left( \frac{\partial^2 (v + v_o)}{\partial y^2} \right) + \rho \cdot g \quad (24)
\]

If we assume a minimal acceleration for the disturbance occurrence, steady state conditions could be assumed and the expression becomes:

\[
\rho \left( v + v_o \right) \frac{\partial (v + v_o)}{\partial y} = -\frac{\partial P}{\partial y} + \mu \left( \frac{\partial^2 (v + v_o)}{\partial y^2} \right) + \rho \cdot g \quad (25)
\]

Dividing by the projected area of the bearing A yields:

\[
\rho \left( v + v_o \right) \frac{A \cdot \partial (v + v_o)}{\partial y} = -\frac{A \cdot \partial P}{\partial y} + A \mu \left( \frac{\partial^2 (v + v_o)}{\partial y^2} \right) + \rho \cdot g \cdot A \quad (26)
\]

Multiplying with the finite time duration yields:

\[
\rho \left( v + v_o \right) \frac{A \cdot \partial (v + v_o) \cdot \partial t}{\partial y} = -\frac{A \cdot \partial P \cdot \partial t}{\partial y} + A \mu \left( \frac{A \cdot \partial^2 (v + v_o) \cdot \partial t}{\partial y^2} \right) + A \cdot \rho \cdot g \cdot \partial t \quad (27)
\]

By substituting \( v + v_o = \frac{\partial y}{\partial t} \) the relationship between pressure disturbance and pressure becomes:

\[
\rho \left( A \cdot \partial (v + v_o) \right) = -\frac{A \cdot \partial P}{v} + \mu \left( \frac{A \cdot \partial^2 (v + v_o) \cdot \partial t}{\partial y^2} \right) + A \cdot \rho \cdot g \cdot \partial t \quad (28)
\]
In other words the damping coefficient is equal to:

\[
\frac{A \cdot \partial P}{\partial v} = -\nu \rho \left( A \cdot \partial \left( v + v_o \right) \right) + \mu v \left( A \cdot \partial^2 \left( v + v_o \right) \cdot \partial t \right) + A \cdot \rho \cdot g \cdot v \cdot \partial t
\]  

(29)

It is evident that the apparent damping coefficient is a function of velocity and shaft's linear acceleration. The purpose of the analysis in the y direction is not to exclude what happens in the x direction but rather to establish that, without loss of generality (i.e. in both axes of movement), the velocity disturbance has a widely different effect on Navier Stokes equations than on Reynolds. Reynolds equation does not introduce velocity and the pressure gradient does not change over z direction. In other words if the movement of the shaft causes a disturbance that is not dismissible in comparison to the journal velocity, Navier Stokes equations treat this disturbance differently than the Reynolds equation does.

The disturbance used in order to calculate the damping coefficient should approximate the velocity and acceleration of the journal while on its stable orbit. The problem, finally, is to determine the proper disturbance which would permit the correct calculation of the stiffness and damping coefficients. In this work the disturbance of the shaft position and velocity was selected by comparison with the work of Glienicke et al. [14]. More specifically the proper disturbances were tested towards the stiffness and damping coefficients considering a bearing with \( L/D = 0.5 \).

In Fig. 9, the damping coefficients obtained are compared with those of reference [14] for \( L/D = 0.5 \).
The disturbances that have been used do not vary linearly and they are not the same for every term and direction. In Fig. 10 the disturbances are presented for a range of Sommerfeld number values.

The higher disturbance values are used for the coupled $C_{yx}$ term, while the coupled term $C_{xy}$ and the main term $C_{xx}$ are calculated with the lower values of disturbance. The disturbance is additionally a function of load. For the disturbance regarding the coupled $C_{yx}$ term the maximum value is required at $S=0.4$. The same trend appears for the disturbance used in the calculation of $C_{xx}$ term, with the maximum value required in the region of $S=0.4$ as well. For the $C_{xy}$ and $C_{xx}$ terms the trend is opposite with a minimum value of disturbance required at $S=0.26$. 

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2.8 Rotordynamics

The equations of motion for the journal bearing and the shaft can be formulated in a matrix form as follows:

$$[M][\ddot{X}]+[C][\dot{X}]+[K][X] = \{F\}$$  \hspace{1cm} (30)

where $[M]$, $[C]$ and $[K]$ are the total mass, damping and stiffness matrices including both the bearing and the rotor dynamic properties, $\{X\}$ is the displacement-rotation matrix and $\{F\}$ the forces-moments vector. The latter includes gravity. There is no imbalance involved in the calculations. The bearing forces are incorporated into equation (27) through the damping and stiffness matrices. The aforementioned matrices are assembled using the 6x6 matrices of all elements of the rotating shaft.

The elements used in this simulation are three node beam elements, as seen in Fig. 11 and are based on the Timoshenko beam theory. A fourth node is available for cross section orientation purposes only. For each rotor element the mass matrix is the sum of the translational and rotational mass matrices:

$$[M] = [M_t] + [M_r]$$  \hspace{1cm} (31)

where $[M_t]$ is the translational and $[M_r]$ the rotational mass matrix respectively.

$$[M_t] = \int_0^\rho [\Psi]^T [\Psi] dx$$  \hspace{1cm} (32)

where $[\Psi] = \begin{bmatrix} \frac{1}{2} (-s + s^2) & (s + s^2) & (1 - s^2) \end{bmatrix}$ is the translational shape function matrix and $\rho$ the density of the journal's material, $s = \frac{2x}{l_e} - 1$, $x$ is the longitudinal coordinate in the nodal coordinate system, $l_e$ is the element length.
where $\{\Phi\}$ is the rotational shape function matrix and $[\omega]$ the rotational matrix of the angular velocity of the shaft. For the case of the element used the shape functions matrix is formulated as follows:

$$\{\Phi\} = \begin{bmatrix} \frac{1}{2} (-s + s^2) & (s + s^2) & (1 - s^2) \end{bmatrix}$$

(34)

Due to the oscillation of the rotating shaft, Coriolis forces are bound to appear. The Coriolis forces are calculated as an external force acting on the element according to the following equation:

$$\{F_c\} = [G][\dot{X}]$$

(35)

where $\{F_c\}$ is the Coriolis forces vector, $[G]$ is the global Coriolis matrix and $[\dot{X}]$ is the nodal velocity vector. The global Coriolis matrix is calculated as follows:

$$[G] = \sum_i [G_i]$$

(36)

where $[G_i]$ is the element Coriolis matrix. The element Coriolis matrix is calculated as follows:

$$[G_i] = ([N_i] - [N_i]^T),$$

(37)

where $\{N_i\} = \int_0^L [\Phi]^T [\Phi] d\kappa$ and $J_p$ the polar moment of inertia of the cross section of the shaft. As shown in [30] the stiffness matrix due to bending for the Timoshenko element is:

$$[K_i] = EI\beta[x]^T [z_1 \dot{I} x]^T + EI\beta[x]^T [z_3 \ddot{I} x]^T + EI\beta[x]^T [w_4 \dot{I} x]^T + EI\beta[x]^T [w_4 \ddot{I} x]^T$$

(38)

where $E$ is the modulus of elasticity, $I$ is the second moment of area, $k$ is the shear coefficient, $G$ is the shear modulus and $A$ is the cross section area. The matrices are described in detail in the work of Gounaris and Papadopoulos [30]. Furthermore, if the rotor is supported by journal bearings, the stiffness and damping matrices have to be calculated and the global stiffness and damping matrices must be included.
\[ [K_r]_{\text{total}} = [K_r] + [K_h] \]  

(39)

Where \([K_h]\) is the bearing stiffness matrix at a node \(n\) and

\[ [C_r]_{\text{total}} = [C_{r_i}] + [C_r] \]  

(40)

where \([C_r]\) is the bearing damping matrix at node \(n\).

In this thesis, the rotordynamic behavior of a shaft with the configuration shown in Fig. 12 is presented. The simulated system consists of a rotor 432 mm long, with a diameter of 9.8 mm, supported on the one side by a ball bearing and on the other by a journal bearing with \(R_j=24.281\) mm and \(L=39.7\) mm. The bushing has a radius \(R_b=24.348\) mm. The shaft is supported on one roller bearing at the tip with the smallest diameter and on one journal bearing on the tip with the widest diameter as shown in Fig. 12.

Fig. 12 The rotor configuration.
2.9 Stability

The system of the equations (30) is purely nonlinear due to nonlinearly dependence of the $F_x$ and $F_y$ components of the displacements components $x, y$ and velocity components $\dot{x}, \dot{y}$. However, for small journal motions around the bearing equilibrium point, the $F_x$ and $F_y$ components can be expressed in the linear form of equation (30).

The linear system of second order differential equations for small displacements $x, y$ is obtained:

$$
\begin{bmatrix}
M & 0 \\
0 & M
\end{bmatrix}
\begin{bmatrix}
\ddot{x} \\
\ddot{y}
\end{bmatrix}
+ 
\begin{bmatrix}
c_{xx} & c_{xy} \\
c_{yx} & c_{yy}
\end{bmatrix}
\begin{bmatrix}
\dot{x} \\
\dot{y}
\end{bmatrix}
+ 
\begin{bmatrix}
k_{xx} & k_{xy} \\
k_{yx} & k_{yy}
\end{bmatrix}
\begin{bmatrix}
x \\
y
\end{bmatrix}
= 
\begin{bmatrix}
0 \\
0
\end{bmatrix}
$$

(41)

Let the non dimensional displacements be $\bar{X} = \frac{x}{c}$ and $\bar{Y} = \frac{y}{c}$, the dimensionless mass $\bar{M} = \frac{cM\Omega^2}{W}$ and the stiffness $K_{ij}$ and damping $C_{ij}$ properties in its non dimensional form, where $\Omega$ is the angular rotor speed and $C$ is the radial clearance. The set of equations (41) can be rewritten as:

$$
\begin{bmatrix}
\bar{M} & 0 \\
0 & \bar{M}
\end{bmatrix}
\begin{bmatrix}
\ddot{\bar{X}} \\
\ddot{\bar{Y}}
\end{bmatrix}
+ 
\begin{bmatrix}
c_{xx} & c_{xy} \\
c_{yx} & c_{yy}
\end{bmatrix}
\begin{bmatrix}
\dot{\bar{X}} \\
\dot{\bar{Y}}
\end{bmatrix}
+ 
\begin{bmatrix}
k_{xx} & k_{xy} \\
k_{yx} & k_{yy}
\end{bmatrix}
\begin{bmatrix}
\bar{X} \\
\bar{Y}
\end{bmatrix}
= 
\begin{bmatrix}
0 \\
0
\end{bmatrix}
$$

(42)

The system defined by the equations (33) is stable if the real parts of all the eigenvalues are negative. At the threshold of the stability, one pair of eigenvalues becomes purely imaginary and thus has a zero real part. Let put $\omega_r$ the vibration frequency and $\Omega_r$ the rotor angular speed at the threshold of the stability. The solution of the (42) is:

$$
\bar{X} = A \exp(i\omega_r \tau) \\
\bar{Y} = B \exp(i\omega_r \tau)
$$

(43)

Where $\bar{\omega}_r = \frac{\omega_r}{\Omega_r}$ and $\tau = \omega \tau$ the non-dimensional time.

Substituting the equation (43) in to system (42) it is obtained the following system of equations:
For a nontrivial solution of the above system, the determinant of the system matrix must be equal to zero. Using the fact that both real and imaginary parts of this determinant have to be equal to zero, it is obtained:

\[
\begin{align*}
\bar{M}_c \ddot{\omega}_{eq}^2 &= K_{eq} \frac{K_{xy} C_{yy} + K_{yx} C_{yy} - K_{xy} C_{yx} - K_{yx} C_{yx}}{C_{xx} C_{yy}} \\
\ddot{\omega}_{eq} &= \frac{(K_{eq} - K_{xx})(K_{eq} - K_{yy}) - K_{yx} K_{yy}}{C_{xx} C_{yy} - C_{yx} C_{yy}}
\end{align*}
\] (45)

It is evident that the exclusion of the coupled terms of the dynamic coefficients would overestimate the stability threshold of the system. Thus the computation of these terms is important for the correct estimation of the dynamic performance of the journal bearing.
3 Static Performance and MR/NMR fluid Bearing Design

The first step in the design process of a journal bearing is the evaluation of its static performance characteristics, which include the load capacity, the minimum film thickness, the lubricant side leakage and of course the friction coefficient. For the purposes of the simulation a rigid shaft is considered. The lubricant flow is simulated under isothermal conditions. In Fig. 13 the schematic of the MR fluid journal bearing offers an overview of its main geometrical features and its operating characteristics.

![Fig. 13. The MR fluid journal bearing schematic](image)

The main characteristics of the MR fluid journal bearing are the bearing radius $R_b$, the radial clearance $C$ and the length of the bearing $L$. A dimensionless expression of the proportions of the bearing may be obtained using the length to diameter ratio $L/D$. The shaft’s equilibrium position is expressed through the eccentricity $e$ and the attitude angle $\varphi$. The shaft rotates at constant rotational velocity $\omega$. The coil turns number and coil cable diameter affect significantly the magnetic field intensity $H$ and of course are taken into account in order to obtain the final geometry of the MR journal bearing system. The rest of the geometric parameters are extracted in relation to the basic design variables. For comparison a set of exclusive magnetic simulations was executed in which the number of coil turns changes, altering the coil’s outer diameter and all other related dimensions. In Table 1 the geometrical features of the bearing configuration used in the simulations that follow are presented.
Table 1. The main characteristics of the journal bearing in the following simulations

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bearing Radius</td>
<td>$R_b = 25\text{mm}$</td>
</tr>
<tr>
<td>Radial Clearance</td>
<td>$C = 235\mu \text{m}$</td>
</tr>
<tr>
<td>Journal Radius</td>
<td>$R_j = 24.765\text{mm}$</td>
</tr>
<tr>
<td>Coil Turns</td>
<td>4000</td>
</tr>
<tr>
<td>Coil Diameter</td>
<td>0.5mm</td>
</tr>
</tbody>
</table>

The LORD MRF-132DG magnetorheological fluid is used in order to obtain results. The Newtonian viscosity of this fluid at 40°C is $0.092 \pm 0.015 \text{ Pa s}$, with density $2930 \text{ kg/m}^3$ and the solids content is 80.98% by weight, and operating temperature from 40°C to 130°C.

Fig. 14. Resulting magnetic field flux lines.

In Fig. 14, we can see the resulting magnetic field intensity in the volume of the bearing and in Fig. 15 the resulting field within the magnetorheological fluid volume, respectively.
The density of the magnetic lines, within the magnetorheological fluid area, is constant. This indicates that the magnetic field in this area is homogeneous. The assumption that the viscosity is constant within the volume of the magnetorheological fluid is validated by these results.

### 3.1 Static Performance of a MR fluid journal bearing

![Graph showing eccentricity ε versus Sommerfeld number for various L/D ratios](image)

**Fig. 16.** Eccentricity $\varepsilon$ versus Sommerfeld number for $I = 0$ A and $L/D = 1/4$, $L/D = 1/2$, $L/D = 1$, $L/D = 2$. 
Fig. 17. Eccentricity $\varepsilon$ versus Sommerfeld number for $I = 14$ A and $L/D = 1/4, L/D = 1/2, L/D = 1, L/D = 2$.

Fig. 18. Eccentricity $\varepsilon$ versus Sommerfeld number for $I = 28$ A and $L/D = 1/4, L/D = 1/2, L/D = 1, L/D = 2$.

Fig. 16 to Fig. 18 depict the relationship between the eccentricity ratio and the Sommerfeld number for a given current intensity and several $L/D$ ratios. The results show the effect of the magnetic field in the load-carrying capability of the bearing. As an example and looking through Fig. 16 to Fig. 18, for an $L/D=1$ and $S = 0.28$ the value of eccentricity for $I = 0$ A is $\varepsilon = 0.4$, for $I = 14$ A is $\varepsilon = 0.38$ and for $I = 28$ A is $\varepsilon = 0.35$. So the observed decrement in eccentricity is 5% for $L/D=1$, $S = 0.28$ and going from $I = 0$ A to $I = 14$ A. From $I = 0$ A to 28 A the respective decrement is 12.5%. Also the respective decrement in eccentricity between $I = 14$ A and 28 A is 7.9%.
3.2 Friction coefficient of a MR fluid journal bearing

Friction coefficient in the journal is significantly increased by the presence of the magnetic field as shown in Fig. 19 to Fig. 21. The highest friction coefficient increase calculated was in the case of $L/D = 1/4$ and for the maximum current intensity.

Fig. 19. Normalized journal friction coefficient ($f_j$) versus Sommerfeld number for $I = 0$ A and $L/D = 1/4$, $L/D = 1/2$, $L/D = 1$, $L/D = 2$.

Fig. 20. Normalized journal friction coefficient ($f_j$) versus Sommerfeld number for $I = 14$ A and $L/D = 1/4$, $L/D = 1/2$, $L/D = 1$, $L/D = 2$. 
Fig. 21. Normalized journal friction coefficient ($f_j$) versus Sommerfeld number for $I = 28$ A and $L/D = 1/4$, $L/D = 1/2$, $L/D = 1$, $L/D = 2$.

$I = 28$ A. The effect is adversely weaker in higher length to diameter ratio and especially in the case of $L/D = 2$ in which case the journal friction coefficient is increased by approximately 4% for the current intensity increased from 0 to 28 A. Due to standard coil geometry considered, the effect of the magnetic field is lessened. Bearing geometry must be linked with coil geometry if we wish to have a uniform performance of the magnetorheological fluid bearing on different length to diameter ratios. The increase of current intensity is affecting dramatically the journal friction coefficient. Thus for a length to diameter ratio of $L/D = 1/4$ the increase of friction coefficient is 31% in the case of current intensity rise from 0 to 14 A and 34% in the case of 14–28 A for a Sommerfeld number of $S = 0.28$. In the later case the friction coefficient appears to be increasing, slightly more than in the first case. This trend is constant in all the different length to diameter ratios and it is related to the non-linear behavior of the lubricant in use.

The bearing friction coefficient is strongly linked with the current intensity, and consequently, the magnitude of the magnetic field. This is shown in Fig. 22 to Fig. 24. The highest bearing friction coefficient is found at $L/D = 1/4$, for the maximum current intensity of $I = 28$ A. The overall increase of bearing friction coefficient in comparison with the case of 0 A current intensity is 88%. The effect of the magnetic field is minimal in the case of $L/D = 2$, where the increase is found to be only 15%. As mentioned earlier the number of coil turns and the diameter of coil’s wire is considered constant in all the aforementioned cases.
The increase of bearing friction coefficient when the current intensity increases from 0 A to 14 A is 28% in the case of \( L/D = 1/4 \) and 0.9% in the case of \( L/D = 2 \). The increase of bearing friction coefficient when the current intensity increases from 14 A to 28 A is 28% in the case of \( L/D = 1/4 \) and 0.9% in the case of \( L/D = 2 \).
Fig. 24. Normalized bearing friction coefficient ($f_b$) versus Sommerfeld number for $I = 28$ A and $L/D = 1/4$, $L/D = 1/2$, $L/D = 1$, $L/D = 2$.

### 3.3 Lubricant flow rate

The lubricant flow rate at maximum lubricant thickness versus the Sommerfeld number is depicted in Fig. 25 to Fig. 27. It is widely affected by changes in the magnetic field within the volume of the bearing and the consequent changes in lubricant viscosity. For the maximum current intensity increment of 0–28 A the lubricant flow rate at the maximum thickness is 8% in the case of $L/D = 1/4$ and a Sommerfeld number of 0.28.

Fig. 25. Dimensionless flow rate at maximum lubricant thickness versus Sommerfeld number for $I = 0$ A and $L/D = 1/4$, $L/D = 1/2$, $L/D = 1$, $L/D = 2$. 
Change in lubricant flow rate at maximum lubricant thickness is decreasing with the increase of the length to diameter ratio. In the case of $L/D = 2$ lubricant flow rate at maximum lubricant thickness is decreased by approximately 1% when the current intensity is increased from 0 to 28 A.

![Graph showing the relationship between the side leakage over maximum flow rate ratio and Sommerfeld number. The increase of current intensity generally decreases side leakage. In the case of a length to diameter ratio $L/D = 1/4$ the decrease is approximately 2% when the current intensity is increasing from 0 to 14 A.](image)
for a Sommerfeld number $S = 0.28$. The decrease side leakage over maximum flow rate ratio remains 2% when the current intensity increases from 14 to 28 A.

![Graph](image1.png)

**Fig. 28.** Ratio of side leakage flow over flow rate $Q_s$ versus Sommerfeld number for $I = 0$ A and $L/D = 1/4$, $L/D = 1/2$, $L/D = 1$, $L/D = 2$.

![Graph](image2.png)

**Fig. 29.** Ratio of side leakage flow over flow rate $Q_s$ versus Sommerfeld number for $I = 14$ A and $L/D = 1/4$, $L/D = 1/2$, $L/D = 1$, $L/D = 2$.

The overall decrement of side leakage over maximum flow rate ratio when the current intensity increases from 0 to 28 A is decreasing by 4.3%. The maximum side leakage over maximum flow rate ratio decrement when $S = 0.28$, is observed in the case of $L/D = 1/2$ with a decrement of 6%. This indicates that in this length to diameter ratio the side leakage is strongly affected by the magnetorheological phenomenon.
Fig. 30 Ratio of side leakage flow over flow rate $Q_0$ versus Sommerfeld number for $I = 28$ A and $L/D = 1/4$, $L/D = 1/2$, $L/D = 1$, $L/D = 2$.

3.4 Attitude Angle

Fig. 31 to Fig. 33 shows the effect of increased load-carrying capability to the attitude angle. In lower Sommerfeld number values the attitude angle obtains lower values. This change in the attitude angle depicts a tendency of the shaft to take an offset position in higher load conditions on the one hand and on the other the resulting increment of the attitude angle with the increase of current intensity is in agreement with the general notion that a stronger magnetic field results in greater load capacity of the bearing.

Fig. 31. Attitude angle versus Sommerfeld number for $I = 0$ A and $L/D = 1/4$, $L/D = 1/2$, $L/D = 1$, $L/D = 2$. 
3.5 Design Considerations of a MR fluid journal bearing

In order to facilitate the design of this type of bearing and quantify the connection of the magnetic field with the performance of the bearing, the relationship between the current intensity, the proportions of the coils and the journal bearing itself, it is imperative to compare different cases seen in Fig. 34 to Fig. 37. The effect of the magnetic field is becoming more apparent. For the different length to diameter ratios used in the simulation, the number of coil turns remains constant. This means that for a greater length to diameter ratio we obtain a weaker magnetic field and thus the effect on the magnetorheological fluid is significantly lessened. On the other hand a
small length to diameter ratio is providing less volume for the magnetorheological fluid and although the magnetic field is stronger, the effect of the field to the bearing’s performance is limited by the lack of sufficient quantity of magnetorheological fluid.

![Fig. 34. Variation of relative eccentricity versus current intensity rise for $S = 0.28$ and $L/D = 1/4$, $L/D = 1/2$, $L/D = 1$, $L/D = 2$.](image)

Fig. 34 shows the effect of the current intensity rise on the relative eccentricity ($e$) for a given Sommerfeld number in all length to diameter ratios examined. In this case the smaller $L/D$ ratio is providing the least load capacity rise. Fig. 35 and Fig. 36 show the friction coefficient increment with the application of a higher current intensity. The effect of the smaller volume in the case of a $L/D = 1/4$ is obvious. The friction coefficient is rising faster when the volume of the lubricant is minimum.

![Fig. 35. Variation of bearing friction coefficient ($f_b$) versus current intensity rise for $S = 0.28$ and $L/D = 1/4$, $L/D = 1/2$, $L/D = 1$, $L/D = 2$.](image)
Fig. 36. Variation of journal friction coefficient ($f_j$) versus current intensity rise for $S = 0.28$ and $L/D = 1/4$, $L/D = 1/2$, $L/D = 1$, $L/D = 2$.

Fig. 37. Variation of flow rate $Q_0$ versus current intensity rise for $S = 0.28$ and $L/D = 1/4$, $L/D = 1/2$, $L/D = 1$, $L/D = 2$. 
Fig. 38. Variation of ratio $Q/Q_0$ versus current intensity rise for $S = 0.28$ and $L/D = 1/4, L/D = 1/2, L/D = 1, L/D = 2$.

Fig. 37 and Fig. 38 depict the drop of lubricant flow in the position of maximum lubricant thickness and the side leakage. The effect of the magnetic field to the ability of the lubricant to flow is evident. In the case of the shortest bearing, the drop of the flow is considerably higher than in higher $L/D$ ratios. This result is reasonable since the effect of the magnetic field is focused in a smaller fluid volume.

Fig. 39. Variation of attitude angle versus current intensity rise for $S = 0.28$ and $L/D = 1/4, L/D = 1/2, L/D = 1, L/D = 2$.

The attitude angle is rising when the current intensity is rising as shown in Fig. 39. This simply put means that the load capacity is most significantly improved in lower $L/D$ ratios. On the other hand the attitude angle of the journal is least affected by the
rise of current intensity when $L/D = 2$. The magnetic simulation gives an insight on the design requirements of the magnetorheological journal bearing. In Fig. 40 and Fig. 41, the line of 2000 A/m in the case of $I = 14$ A and the line of 4000 A/m in the case of $I = 28$ A show the increasing need for coil turns in order to achieve a relatively low value magnetic field intensity in longer bearings. The main priority is to keep the magnetic field as homogeneous as possible. A homogenous magnetic field in the lubricated area would result in homogenous physical properties and mechanical behavior throughout the length of the bearing. In case of non-uniform distribution of the magnetic field, the particle chains would be distributed in a non-uniform manner across the length of the bearing. This could cause uneven load distribution. Moreover, in case of wear creation due to the existence of magnetized particles could lead to excessive wear in portion of the bearing surface. Keeping this in mind, a homogenous magnetic field within the lubricated area is a desirable effect from a design and performance aspect.

![Graph showing correlation between coil's number of turns and magnetic field intensity for varying L/D ratios](image)

Fig. 40. Correlation between coil's number of turns and magnetic field intensity for $L/D = 1/4$, $L/D = 1/2$, $L/D = 1$, $L/D = 2$ and current intensity 14 A.

This is successfully achieved by increasing the height of the coil core and the number of the coil turns for higher length to diameter ratios. This technique is obviously cost intensive and the resulting benefits of the magnetorheological fluid bearing may in this manner be undermined. There is certainly a perspective for a minimum cost coil.
and core design which will produce as much a homogeneous magnetic field as possible even in long bearings.

Fig. 41. Correlation between coil’s number of turns and magnetic field intensity for $L/D = 1/4, L/D = 1/2, L/D = 1, L/D = 2$ and current intensity $28$ A.

3.6 Conclusions on the static performance of the MR fluid journal bearing

For a selected number of bearing states, several $L/D$ ratios, magnetic field variations, solutions were obtained in terms Sommerfeld number variation. The present results demonstrate that, in comparison to a normal bearing (lubrication without magnetic field), the presence of magnetic field can be beneficial for the bearing characteristics such as increased load-carrying capacity in terms of the magnetic field increment, whereas the results regarding the friction coefficient lead to a less beneficial function under the influence of the magnetic field. Thus, designing journal bearings under the excitement of the magnetic field should be realised taking provisions for the energy cost that occurs with the higher friction coefficient and the energy required to sustain the desired magnetic field. We can conclude that for a rise of current intensity from 0 to 28 A:

- The Sommerfeld ($S$) of the journal bearing decreases. The maximum decrement calculated was 36.39\% for $L/D = 1/4$ and eccentricity $\varepsilon = 0.8$. The minimum decrement is 1.64\% for $L/D = 2$ and the same value of $\varepsilon$. 

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The attitude angle ($\phi$) increases by 37.4% for a $L/D = 1/4$ $\varepsilon = 0.8$ and by 0.7% for a $L/D = 2$, $\varepsilon = 0.8$, respectively.

The friction coefficient ($f_b$) on the bearing surface increases. The maximum increment 22.2% is for $L/D = 1/4$ and eccentricity $\varepsilon = 0.8$ and the minimum increment is 0.6% for $L/D = 2$ and the same eccentricity.

The friction coefficient on the journal surface ($f_j$) increases, with the maximum increment (28%) to be for $L/D = 1/4$ and eccentricity $\varepsilon = 0.8$ and the minimum increment to be 0.2% for $L/D = 2$ and $\varepsilon = 0.8$.

The fluid flow is decreasing by 5.46% for $L/D = 1/4$ when $\varepsilon = 0.8$. Flow decrement drops to 0.16% in the case of $L/D = 2$ and the same $\varepsilon$.

3.7 Contribution

The general design implications of the coil’s geometry and the number of turns inside the coil on the performance of the magnetorheological fluid journal bearing has never before been established. Moreover, the static characteristics for this wide range of bearing proportions as well as the relationship between the amplitude of current with the friction coefficient and the relative eccentricity is also presented for the first time in this thesis.
4 Wear compensation

As seen previously, magnetorheological fluids have the ability to increase the load capacity of a journal bearing. It is reasonable to assume that this same ability can be used in order to compensate for the wear present in most cases on the bushing due to immediate contact between the shaft and the bearing during start up and shut down phases. The general form of this type of wear is depicted in Fig. 42 along with the relationship of the lubricant film thickness in the worn region of the bushing.

For the purposes of this work, a bearing of radius $R_b = 35$ mm was used with slenderness ratio $L/D = 0.5$ and a radial clearance $C = 285\mu m$. The ratio of wear depth to the radial clearance $\delta$ has been chosen to be $\delta = 0.5$. The viscosity of the newtonian lubricant is $\mu = 0.014$ Pa s and the density is $\rho = 890$ kg/m$^3$ as specified in [22]. The MRF viscosity in the non active state is $\mu = 0.112$ Pa s whereas the yield stress is $\tau_0 = 25000$ Pa and its density is $\rho = 2950$ kg/m$^3$. The properties of the MRF are those of LORD MRF-132DG. The properties refer to the state of the fluid under a magnetic field of intensity $H = 86$ kA/m.

![Fig. 42. Worn bearing schematic](image)

The developed model was qualified versus the results obtained by Hashimoto et al [22]. In Fig. 43 the attitude angle of the journal bearing is plotted versus the relative eccentricity for both the present work and reference [22]. The results show good agreement. In Fig. 44 relative eccentricity is plotted versus Sommerfeld number, for both the case of Newtonian and the case of MRF. The use of MRF seems to significantly increase the lubricant film thickness and consequently the load capacity.
of the journal bearing. While the load capacity seems to be widely improved with the use of MRF, the effect of the latter to the friction coefficient is adversely negative.

![Graph showing the friction coefficient comparison between Newtonian and MRF](image1)

**Fig. 43** Comparative results between present work and reference in [22] a bearing with $L/D=1$ and $Re=84$

In Fig. 45 the normalized friction coefficient of the bearing is presented for both the case of Newtonian and MRF used. The higher friction of the MRF is to be expected but somewhat limits the margins of using the specific material in its active state for long periods of time. The higher viscosity in the case of MRF is located in the area of lower shear rate values which is the worn area, as it is shown in Fig. 46. In other words the worn area fills with high viscosity lubricant, compensating in that manner the occurring wear.

![Graph showing eccentricity comparison](image2)

**Fig. 44.** Relative eccentricity $\varepsilon$ versus Sommerfeld number for both the case of a Newtonian lubricant and a MRF in a bearing with $L/D=1$ and $R_b=35mm$ and $\delta=0.5$
4.1 Conclusions on Wear Compensation

The magnetorheological fluid has the potential to increase the minimum film thickness to a degree that completely compensates possible wear. The intensity of the magnetic field should be optimized in terms of friction coefficient in order to avoid overcompensation and increase friction overwhelmingly in comparison to the plain bearing.

4.2 Contribution

The ability of magnetorheological fluids to compensate for the wear present on the bushing is a new concept never before being presented elsewhere.
5 Dynamic Coefficients of NMR, MR and Newtonian Fluid journal bearing using High Current

The dynamic behavior of a journal bearing is an integral part of its design. It is even more so in the case of the magnetorheological and nanomagnetorheological fluid journal bearing, given its capability of active control. For the calculation of the dynamic coefficients, the small disturbances method has been employed. The lubricant flow has been calculated using the Navier-Stokes equations, as mentioned earlier.

Since the disturbances being used to calculate the values of stiffness and damping coefficient are small, the accuracy of prediction of the hydrodynamic and friction forces is important. The error of the pressure CFD solution is 1E-6, which is adequate in terms of accuracy for the load prediction. This residual error is acceptably low in comparison to the relative literature [4]. Additionally the high number of elements used satisfies the demand for results irrelevant of the mesh used in the calculations. In the present paper there are 450 elements in the circumferential direction, 10 elements in the radial direction and 15 elements in the axial direction. The aforementioned configuration also corresponds with the standards set by literature for the minimum mesh requirements [3, 76].

The dynamic characteristics of the journal bearing are a significant aspect of the overall performance of the journal bearing. The high viscosity exhibited by the NMR fluid yields a significant damping capability along with increased stiffness of the journal bearing. In this section the dimensional values of the stiffness and damping properties are presented in detail, in order to have an absolute comparison between the Newtonian, MR and NMR fluids.

5.1 Stiffness Coefficients Validation

The following simulations consider a bearing with \( R_s = 49.999 \) mm, radial clearance \( C = 61 \) \( \mu \)m with journal rotating at 1000 rpm's. Three types of lubricant are used in this Chapter:

a. A typical newtonian lubricant with dynamic viscosity of 1.110E-02 Pa s at 40\(^\circ\)C and density 890kg/m\(^3\).
b. A typical MR fluid with viscosity is $0.112 \pm 0.02 \text{Pa s}$ at $40 \degree C$, with density $2.95-3.15 \text{g/cm}^3$ and the solids content is 80.98 % by weight, and the operating temperature varies from -40 $\degree C$ to 130 $\degree C$. With the magnetic field calculated in this paper the yield stress of the MRF is 25 kPa.

c. A NMRF which consists of $Fe$ nano particles with maximum yield stress of 20 Pa at 343 kA/m, with viscosity of 9.9Pa s at 25$\degree$C. The characteristics of the NMRF have been retrieved by the results presented in reference [66]. While there is significant difference between the suggested operating temperature of the fluids in comparison, the very high damping offered by the NMRF will not be substantially altered by a temperature increase of 20$\degree$C.

The capability of the developed code to accurately predict the dynamic characteristics of the journal bearing has been validated towards the work of Glienicke et al. [14] . The stiffness coefficients, in the case of $L/D=0.5$ are depicted in Fig. 47.

![Fig. 47. Stiffness coefficients for $L/D=0.5$ using a newtonian lubricant.](image)

The bearing stiffness coefficients are identical with those presented in reference [14].

### 5.2 Stiffness Coefficients Comparison

Stiffness coefficients are a significant measure of the dynamic behavior of the journal bearing system. The resulting stiffness of the MR fluid for a range of loads is
depicted in Fig. 13. Concerning the main terms of stiffness, the MR fluid yields higher values of the $K_{yy}$ term comparison to the newtonian lubricant (see Figs. Fig. 47 and Fig. 48). Most pronounced is the increase of stiffness at higher Sommerfeld number values. For $S=0.792$ the $K_{yy}$ term increases almost six times. On the contrary $K_{xx}$ term is adversely affected by MR fluid use. For instance the $K_{xx}$ is 41.8% lower for $S=0.792$ when MR fluid is used. $K_{yy}$ term is positively affected by the MR fluid with most significant benefits in the region of lower loads. For $S=0.796$, the use of Newtonian lubricant results in a 4.6 times increase of the $K_{yy}$ term. $K_{yx}$ term decreases in the case of MR fluid use. For example $K_{yx}$ decreases 80 times for a Sommerfeld number value of $S=0.198$.

Using the high viscosity NMR fluid yields even higher stiffness in comparison to the MR fluid, as shown in Figs. Fig. 48 and Fig. 49, where stiffness of the NMR fluid is presented for the same range of Sommerfeld number values and the same overall geometry of the bearing. The main terms of stiffness are higher in the case of NMR fluid for all Sommerfeld number values considered. The same applies for the $K_{yx}$ term. $K_{yx}$ term as an absolute value is higher in the case of NMR fluid use.
Comparing Figs. Fig. 47 to Fig. 49, the higher stiffness of the NMR fluid in the given range of Sommerfeld number values is evident. This trend is especially intense in the case of higher loads with the $K_{yy}$ term being two orders of magnitude higher than that of the MR fluid. This higher stiffness though is related with the much higher eccentricity of the journal in the case of NMR fluid and with the fact that Sommerfeld number is a function of apparent viscosity.

### 5.3 Damping Coefficients Validation

The damping coefficients for a bearing using newtonian lubricant with those presented in reference [14] are compared in Fig. 50 for $L/D=0.5$. The calculated coefficients show very good agreement with those presented in the aforementioned reference.
5.4 Damping Coefficients Comparison

MR fluid alters significantly the available damping of the journal bearing. Fig. 51 depicts the damping coefficients for a bearing with \( L/D = 0.5 \) using MR fluid. The higher values of damping are to be expected since in its active state, MR fluid has increased viscosity. With the only exception of \( C_{xy} \) for \( S = 0.158 \) which is slightly decreased by a 10.8% in the case of the MR fluid (see Figs. Fig. 50 and Fig. 51), the overall trend is a significant increase of the damping coefficients, especially on higher Sommerfeld number values. For instance \( C_{yy} \) increases by a 1.72E3% thanks to the high yield stress of the MR fluid. The same general tendency applies for all other terms.

![Fig. 51. Damping coefficients for a bearing with \( L/D = 0.5 \) using MR fluid.](image)

On the other hand the viscosity of the NMR fluid is also high in comparison to the newtonian lubricant. This results in high values of damping coefficients when this type of fluid is used. Fig. 52 shows damping coefficients for the bearing using NMR fluid. Similarly with the results obtained for the MR fluid, benefits to the damping coefficients in comparison to the case of newtonian fluid are more pronounced in the region of lower Sommerfeld number values. As an example of the magnitude of change, \( C_{xx} \) term increases by a 1.038E5% for \( S = 0.264 \) (see Figs. Fig. 50 and Fig. 52). Comparing Figs. Fig. 51 and Fig. 52 also yields some interesting results.
While $C_{yx}$ term is lower in the case of the NMR fluid, the other terms increase. For example the $C_{yy}$ term increases by 1.85E3% for $S=0.792$ in the case of NMR fluid while $C_{yx}$ drops by a 9.18E2% for $S=0.396$.

MR and NMR fluids can alter their properties under the influence of a magnetic excitation. The properties of these fluids end up to significantly alter the dynamic behavior of the journal bearing. These changes in their rheological behavior increase stiffness and damping, making them suitable candidates for bearing technological applications where sudden fluctuations of load can be significant and adaptiveness of the bearing is a prerequisite.

### 5.5 Conclusions on dynamic coefficients

The use of MR fluid with its high yield strength produced a very high rise of the stiffness coefficients, (e.g. for $S=0.792$ the $K_{yy}$ term was increased almost six times). The damping of the bearing was also higher when the MR fluid was used in its active state. In the case of NMR fluid the stiffness coefficient also rises as a result of its superior apparent viscosity. The most pronounced advantage of the NMR though is its high damping properties, which were the highest of the three lubricants used in this study. For instance, the $C_{yy}$ term increases by 1.85E3% for $S=0.792$ in the case of NMR, in comparison to the MR fluid. The effect of the NMR fluid on stability is also significant. The threshold of stability is widely enhanced.
Despite the beneficial effects of the MR and NMR fluids, on the dynamic behavior and stability of a journal bearing system, phenomena such as wear on moving parts, agglomeration of particles and high friction losses may be the focus of future research. Additionally these effects are dependent on the specific application.

The MR fluids can increase the stiffness and damping of the journal bearing. In their off state their rheological behavior approaches this of a newtonian fluid making it a suitable lubricant when both low friction and increased stability are required. On the other hand NMR fluids can benefit the journal bearing with even higher damping and stiffness in comparison to MR and newtonian fluids. Their only weakness is their controllability since in their off state they do not change their properties significantly.

5.6 Contribution

The study of the damping coefficients using the small perturbations method in combination with Navier Stokes and validated with another work is a considerable novelty. The dynamic characteristics of journal bearings using magnetorheological and nanomagnetorheological fluids is performed here for the first time.
6 Rotordynamics of MR, NMR and Newtonian Fluids using Low Current

The static characteristics of the MR fluid journal bearings have been presented in Chapter 3. It is necessary to compare the static performance of the MR and the NMR fluids in the lubrication of journal bearings. Moreover, the ability of control of the dynamic characteristics of a nano/magnetorheological fluid journal bearing forms the interaction of the bearing with the rotor. Nevertheless, how exactly the rotordynamic behavior of the system is affected, remains to be seen. The simulation of a rotor supported by a journal bearing using either a NMR, MR or Newtonian fluid is presented in this chapter making full use of the analytical tools developed earlier in this Thesis. Moreover, the current intensity values, the rotor configuration and the bearing geometry approximate those available during the experimental procedure described in Chapter 9.

6.1 System Description

As stated previously in the Introduction of this Thesis, the simulated system consists of a rotor 432 mm long, with a diameter of 9.8 mm, supported on the one side by a ball bearing and on the other by a journal bearing with $R_j=24.281$ mm and $L=39.7$ mm. The bushing has a radius $R_b=24.348$ mm. The bearing clearance is $C=67 \, \mu m$.

![Fig. 53 The rotor configuration.](image)

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The shaft is supported on one roller bearing at the tip with the smallest diameter and on one journal bearing on the tip with the widest diameter as shown in Fig. 53. For comparison one additional case has been simulated with a disk of 80mm diameter, 8 mm thick at the middle of the shaft. The lubricants used in the simulation were selected to be an MR and an NMR fluid. The MR fluid has a density of 2950 kg/m$^3$ and newtonian viscosity of 0.110 Pa s. The NMR fluid has a density of 1124 kg/m$^3$ and newtonian viscosity of 0.7 Pa s.

### 6.2 Magnetic field calculation

The magnetic field which activates the formation of the particles chains is calculated using an axisymmetric model of the journal bearing. The coils are fed with a maximum 2.39A of DC current at 17.5 Volt. The magnetic field is generated by two coils of 170 turns each. The cables are considered to be made of copper.

![Fig. 54. The distribution of the magnetic field intensity at the length of the journal bearing.](image)

The magnetic field intensity varies throughout the length of the shaft as shown in Fig. 54. The higher values of the magnetic field intensity are concentrated on the side of the bearing on which the rotor length is extended. Based on the physical properties of two typical MR and NMR fluids, the MR fluid has a yield stress of 15.27 Pa while the NMR fluid exhibits a yield stress of 0.6 Pa.
Fig. 55. The magnetic field density (T) inside the volume of the magnetorheological fluid journal bearing.

The magnetic field intensity depicted in Fig. 55 shows the influence of the shaft design on the form of the magnetic field. In effect the design of the shaft plays an important role in the formation of the magnetic field and should be taken into account in the design of the bearing.

6.3 Static Performance using MR and NMR fluids

The static performance of the journal bearing is considerably affected by the different fluids used for its lubrication. The Newtonian and MR fluids exhibit similar performance with small gains in the case of the MR fluid. This means that the magnetic field induced by the specific coils in the specific geometrical configuration is not capable of producing significant change in the static characteristics of the journal bearing. On the other hand the NMR fluid is not limited by the magnetic field intensity.
In Fig. 56 the relative eccentricity of the Newtonian, MR and NMR fluids is presented for a range of Sommerfeld number values 0.1 and 1. The use of MR fluid yields a 4.93% decrease of the relative eccentricity at $S=0.795$. There is no significant difference in dimensionless terms between the performance achieved by the Newtonian and NMR fluids. The high viscosity of the NMR fluid is a significant difference that is not directly apparent in this case. In other words, the high viscosity of the NMR differentiates widely the static performance for a given load at a given journal rotational velocity. In order to obtain a significant gain in terms of relative eccentricity, a high voltage feeding of the coils is required. With a DC supply of 4800 A, the MR fluid may obtain higher yield stress capacity and achieve significant benefits in terms of relative eccentricity. The same reasoning reappears on the normalized friction coefficient.
In Fig. 57 the normalized friction coefficient is depicted for the Newtonian, MR and NMR fluids. Produced friction is higher in the case of the MR fluid by 7.54% for $S=0.795$. The slight rise of the friction coefficient is consistent with the decrease of the relative eccentricity in the same Sommerfeld number value and is related with the higher shear stress developed in the case of the MR fluid.
6.4 Stiffness Coefficients

The stiffness coefficients generally are related with the static performance of the journal bearing. Higher stiffness values signify lower relative eccentricity and high load capacity.

Fig. 58. Stiffness coefficients over relative eccentricity for Newtonian fluid at 1000 rpm.

In Fig. 58 the stiffness coefficient in the case of Newtonian fluid is presented for a range of relative eccentricity's values. The stiffness coefficient results are consistent with those presented in [14]. The MR fluid's static performance shows small change in comparison to the use of Newtonian fluid. This trend applies in the case of stiffness coefficients as well.

Fig. 59. Stiffness coefficients over relative eccentricity for the MR fluid journal bearing at 1000 rpm.
In Fig. 59 the stiffness coefficients of the journal bearing using MR fluid are presented for a range of relative eccentricity values at 1000 rpm's. The comparison between Fig. 58 and Fig. 59 shows little difference, as expected since the static performance between the two cases has also small effect.

![Stiffness coefficients graph](image)

**Fig. 60.** The stiffness coefficients for the journal bearing using NMR fluid at 1000 rpm over a range of relative eccentricity values.

The NMR fluid with its high viscosity inflicts significant change in the stiffness coefficients of the journal bearing. In Fig. 60 the stiffness coefficients for the journal bearing using NMR fluid are depicted. The difference of the acquired values of the stiffness coefficients in comparison to those of the MR and Newtonian fluids is significant (see Fig. 61). The increase of the stiffness coefficient $K_{yy}$ reaches the 634% for relative eccentricity of 0.223.
Fig. 61 The percentage increase of the stiffness coefficients in the case of NMR fluid. Newtonian fluid performance is the basis of comparison.

6.5 Damping Coefficients

The damping capacity of the MR Fluid is expected to be higher than that of the Newtonian fluid, while the NMR fluid with its very high viscosity is expected to provide the highest damping capacity of all three candidate fluids.

Fig. 62 The damping coefficients for the journal bearing using Newtonian fluid at 1000 rpm.

In Fig. 62 the damping coefficients of the bearing using the Newtonian fluid are shown. The results concerning the Newtonian fluid are in accordance with the results presented in [14].
The MR fluid has indeed increased damping capacity in comparison to the Newtonian fluid (see Figs. Fig. 62 and Fig. 63). More specifically the MR fluid presents an 18.8% increase in the $C_{yy}$ term in comparison to the Newtonian fluid. A similar increase exists on the other terms as well.

The high viscosity of the NMR fluid is evidently increasing the damping coefficients of the journal bearing as shown in Fig. 64. For example for a relative eccentricity of 0.693 the main term $C_{yy}$ presents a six fold increase in comparison to the Newtonian fluid. The same applies for the $C_{xx}$ coefficient, in which case the MR and the newtonian fluids have a coefficient 79.3% lower than that of the NMR fluid. The
available damping capacity of the NMR is bound to produce a smoother function for the journal bearing. This trend is more apparent in Fig. 65 where the huge increase of damping capacity of the NMR fluid for $\varepsilon = 0.693$ is depicted.

![Figure 65: Increase of damping coefficients in the case of NMR fluid.](image)

Fig. 65 The increase of the damping coefficients in the case of NMR fluid. Newtonian fluid performance is the basis of comparison.

### 6.6 Damping capacity and Current Intensity

A significant difference of journal bearings using smart fluids is their capacity of vibrations control. This kind of control can be quantified when both the stiffness and damping coefficients of the journal bearing are calculated for every possible configuration of the journal bearing. In the case of the magnetorheological fluid journal bearing, for a suitable control law to be formed, the stiffness and damping coefficients must be expressed in relation with the two independent variables of the bearing: the journal's position and the current intensity on the coils of the bearing. Concerning the latter the damping coefficients are not linearly varying since the semi-solid core of the lubricant is not study and the geometry of the flow inside the lubricant volume is highly complex.
In Fig. 66 the relationship between the current intensity and the damping coefficients is established. The non linear nature of the viscosity of the MR fluid is the dominant factor that forms this relationship. In other words, the MR fluid's core alters the damping behavior of the fluid in a non-linear manner.

Fig. 67. The viscosity distribution on the bearing surface for $I=1.78\text{A}$ and $S=0.113$ a. in static conditions b. under disturbance using MR fluid.
In Fig. 67 the distribution of the MR fluid viscosity is shown within $\pi > \theta > 0$. In the middle a formation of high viscosity core alters the flow. In Fig. 67a the viscosity distribution is taken without any journal translational movement, while in Fig. 67b the viscosity distribution is shown while the shaft is accelerated vertically. Despite a small change of the viscosity when the shaft accelerates, the core remains present, altering the flow field inside the lubricant.

6.7 Response spectrum

The major advantage of the journal bearings using smart fluids is their ability to adjust their performance in different conditions of operation. In regions of rotational velocity of the shaft that are vibrations-intensive, high damping capabilities are required while in the low friction is the main priority.

![Response spectrum](image)

Fig. 68. The response spectrum of the shaft for all three fluids with and without disk placed on the shaft.

In Fig. 68 the response spectrum of the rotor bearing system is presented for newtonian, MR and NMR fluids used in the bearing, with and without disk on the shaft. The response is normalized over radial clearance of the journal bearing. As per API Standard 612 the initial and final speeds of the shaft can be located at 0.707 of peak amplitude for each critical speed. For instance for the critical speed at 2700 rpm an initial speed 2400 rpm's can be found for the cases Newtonian and MR fluids with disk. For the same cases the final speed is located at about 3100 rpm. The most
notable effect of the NMR fluids is the wide difference of maximum amplitude of the occurring vibrations. Regardless of the disk existence the NMR fluid journal bearing offers very low amplitude of oscillations of the shaft in the start-up region (low rpm). This means high tolerance towards wear due to contact of the shaft and the bushing. With the disk present on the shaft the critical speeds are subject to change. Based on the previous calculations it is evident that the vibrations on the shaft are at least lower in amplitude thanks to the beneficial influence of the NMR fluid. More specifically, in this particular case a 82.90% drop of oscillations amplitude is observed at 300 rpm. Additionally in the case of the NMR fluid, there is a slight transpose of the critical speed at 2800 rpm to 3000 rpm.

![Graph showing the rotor's orbit at 9000 rpm for Newtonian, MR and NMR fluids.](image)

**Fig. 69. The rotor's orbit at 9000 rpm for Newtonian, MR and NMR fluids.**

In Fig. 69 the rotor response at 9000 rpm for all three fluids examined in this work is depicted with the disk attached on the shaft. On the location of the journal bearing, the amplitude of the shaft's orbit is decreased by 79% in the case of the NMR fluid at 9000 rpm.

### 6.8 Conclusions on the rotordynamic performance of nano/magnetorheological fluid journal bearings

The use of NMR fluid on the journal bearing benefits significantly both the static and the dynamic performance of the journal bearing without significant power consumption on the coils. The six fold increase of the $C_{yy}$ and the $K_{yy}$ coefficient of...
the journal bearing using NMR fluid indicate the potential benefits of the NMR fluid use. This potential is evident in the amplitude of oscillations of the journal. For instance the amplitude of the orbit in the case of the NMR fluid at 1500 rpm is 42% lower than that of the newtonian and the MR fluid. The MR fluid on the other hand does not produce significantly beneficial results with the given magnetic field strength. The activation of the MR fluid requires high current density which is impractical to maintain and energy consuming. The practical advantage that the NMR fluids have to offer in terms of rotordynamics applications is the wider operating speed regions that may be used for safe operation of the supported rotating machinery.

6.9 Contribution

The low amplitude effect on the dynamic characteristics of the magnetorheological and nanomagnetorheological fluid journal bearings has never before been established. Moreover the rotordynamic behavior of a shaft supported by a journal bearing using these fluids is also considered for the first time here.

7 Artificial texturing in magnetorheological fluid journal bearings

MRF journal bearings present extremely improved performance in respect to the classical oil lubricated journal bearings in expense of much higher friction. Taking into consideration this limitation, the possibility to reduce the friction forces using artificial texturing was examined.
The basic geometry of the bearing considered in this work is depicted in Fig. 70. The bearing considered in the simulations has a radius \( R_b = 49.999 \) mm, and total length \( L = 49.999 \) mm. The radial clearance is \( C = 85.5 \) μm. The configuration of the texturing pattern is described by the number of dimples in circumferential and axial directions (N1 and N2 respectively). The bearing surface is only partially textured with the textured area being defined by the inlet and outlet angles \( \psi_1 \) and \( \psi_2 \) respectively.

Artificial texturing is a technique which employs very small dimples that can be created in the surface of the bearing. The principle of operation of this method, depicted in Fig. 71, is based in the fact that the flow entrance and exit of the dimples can create a small recirculating flow, acting as a microbearing underneath of the main flow, reducing locally the shear rate -consequently the shear stress- and the generated friction. In this work two shapes of dimples were examined: rectangular and egg-shaped dimples, as shown in Fig. 72.
Fig. 71. The flow inside the egg-shaped shaped dimple is reversed externally, acting as a microbearing.

Rectangular dimples have a length of 1.3 mm and a width of 1.5 mm. Egg-shaped dimples where defined by the following equation:

\[
 r(\psi, z) = \begin{cases} 
 R_b + d_{\text{dim}} \cdot e^{(\psi-a)/ \sqrt{2\sigma_1^2}} \cdot e^{(z-b)/ \sqrt{2\sigma_2^2}}, & \psi < a \\
 R_b + d_{\text{dim}} \cdot e^{(\psi-a)/ \sqrt{2\sigma_2^2}} \cdot e^{(z-b)/ \sqrt{2\sigma_1^2}}, & \psi \geq a
 \end{cases}
\] (4)
where $d_{\text{dim}}$ is the dimple depth, $a$ is the circumferential position of the dimple's center, $b$ is the axial coordinate of the dimple's center. The $\sigma_1$ and $\sigma_2$ parameters control the dimple overall length while $\sigma_3$ controls the dimple's width. In this work, $\sigma_1 = 1.2E-3$, $\sigma_2 = 6E-4$ and $\sigma_3 = 8E-4$.

In all cases examined in this work, the overall arc in which texturing is applied extends to 30 degrees and the dimples are uniformly distributed within the given length of the bearing. A single configuration of $N_1=6$ and $N_2=6$ was examined for both shapes. The rotational velocity of the journal is set to 1000 rpm. The lubricant used has a density of 2950 kg/m$^3$, yield stress of 25000 Pa and fluid (Newtonian) viscosity of 0.112 Pa s. The journal bearing with egg shaped texturing was modeled with 77220 hexahedral elements (89700 nodes). The journal bearing rectangular shape texturing was modeled with 260400 hexahedral elements (293088 nodes).

### 7.1 Validation

For the purposes of model validation, the results of the simulation of a plain bearing were compared with the results presented by Brito et al [8].

![Comparison of experimental with simulation results](image.png)

The bearing for which validation was performed has the same bearing radius and radial clearance as the ones used for the purposes of the simulations. The length is
$L=80\text{mm}$ and it was lubricated with ISO-VG32 lubricant. The viscosity of the lubricant was $0.0293 \text{ Pa s}$ at $40^\circ \text{C}$.

### 7.2 Influence of Angular Position of the Textured Area

The angular position of the textured area, as shown in Fig. 74, has been investigated in order to establish the relationship between $\psi_1$, load capacity and friction.

![Graph](image1)

**Fig. 74.** The relative eccentricity for four different $\psi_1$ values with rectangular dimples for load capacity of 6000N, 8000N and 10000N. The smooth bearing performance is present for comparison.

The results show minor improvement in terms of relative eccentricity with maximum improvement shown for $\psi_1 = 70^\circ$ where the relative eccentricity drop reaches 1.32% for the 6000N load.

![Graph](image2)

**Fig. 75.** The relative eccentricity for four different $\psi_1$ values with egg-shaped dimples for load capacity of 6000N, 8000N and 10000N. The smooth bearing performance is present for comparison.
The egg shaped texturing shows some significant improvement as depicted in Fig. 75. Although there is minor influence on the relative eccentricity of the journal bearing when the angular position of the textured area changes, there is a 3.08% decrease in the cases between 10 and 50 degrees for a 6000N load.

The friction coefficient is minimally affected by the angular position of the artificial texturing inside the bearing. In Fig. 76 the normalized friction coefficient is presented for a series of texturing circumferential position values. The results include the performance of the smooth bearing for comparison.

![Rectangular shape, \(T_0=25\text{kPa}\)](image)

**Fig. 76.** The normalized friction coefficient for smooth and textured bearing with rectangular dimples for various texturing circumferential locations under 6000N, 8000N and 10000N load

The friction coefficient exhibits a maximum increase of 0.14% in the case of 6000N load. In the case of 8000N there is a 0.05% decrease but these changes are negligible. The same trend appears in the case of egg-shaped artificial texturing, described in Fig. 75.
Fig. 77. The normalized friction coefficient of smooth and artificially textured journal bearing with egg-shaped dimples on four angular locations in the circumference of the bearing for loads of 6000N, 8000N and 10000N.

The friction coefficient does not change significantly with the change of the dimple geometry. There is a maximum increase of 0.95% of the friction coefficient for a load of 10000N. Overall there are negligible deviations from the values obtained for the smooth bearing. Another parameter of the overall geometry of the artificial texturing is the dimple depth. A comparison of the effect of the dimple depth on the relative eccentricity of the artificially textured journal bearing is presented in Fig. 78, for a load of 6000N.

Fig. 78. The effect of dimple depth on the relative eccentricity for a load of 6000N for both rectangular and egg-shaped dimples.

The difference of geometry between the two configurations induces different results on the performance of the artificially textured journal bearing. While the rectangular dimples depth increase has a negative impact on the relative eccentricity, the increase of the egg-shaped dimples depth improves the static performance of the bearing.
7.3 Conclusions on the artificial texturing being used in magnetorheological fluid journal bearings

The effect of artificial texturing on the performance of the journal bearing is positive although the extent of this effect seems to be minor in absolute values. The egg-shaped texturing is although promising as the increase of depth resulted in relative eccentricity improvement of 4.8% with a dimples density that is rather low, whereas the rectangular shaped texturing performance deteriorates. In other words there seem to be margins for further improvement of the performance benefits that the specific artificial texturing geometry has to offer.

7.4 Contribution

The effect of artificial texturing in journal bearings using magnetorheological lubricant is a new concept never before being examined.

8 Temperature influence

A constant homogenous magnetic field is applied on the lubricant volume. The magnetic field density reaches the 0.6 Tesla. There are eight particles inside the control volume, corresponding to a magnetorheological fluid with a 16.32% per weight content of iron particles. A first conclusion on the validity of non-linear viscosity models, concerning the approximation of the rheological behavior of a magnetorheological lubricant, can be drawn from the pressure distribution of the lubricant flow when the particles are present.

Fig. 79. Pressure distribution in the bearing surface for a journal velocity of \( v = 5.23 \) m/s at 49.36 °C
Since the journal and the bearing surfaces are parallel in this model, the pressure distribution is the result of the particles effect on the lubricant flow. A significant amount of pressure is then developed in the journal and bearing surfaces, as seen in Fig. 79.

![Shear stress distribution on the bearing surface at 49°C](image)

**Fig. 80. Shear stress distribution on the bearing surface at 49°C**

Since a certain amount of pressure is applied on the journal and the bearing surfaces, it is evident that shear stress will also be applied. In Fig. 80, the shear stress developed on the bearing surface is shown. The areas with the highest values of shear stress are located on the sides of the area where a particle is located. The exact same pattern appears on the velocity distribution of the lubricant, on the bearing surface, as shown in Fig. 81. In contrast with the smooth transition of the shear stress values when a non-linear viscosity model is used, this figure shows that the distribution of both pressure and shear stress in a magnetorheological fluid exhibits variations in the scale of the present simulation.
The maximum pressure on the bearing surface is related to temperature as shown in Fig. 82. As expected, the maximum pressure drops with the rise of temperature, since the carrier fluid’s viscosity is lower at higher temperatures. Additionally for higher journal speeds, the maximum pressure on the bearing surface increases. With a drop of journal velocity of 25%, the maximum pressure decreases by 26.89% for a temperature of 49°C. In higher temperature the drop is more significant reaching the 35.85% at 81°C.

The mean shear stress generated by the lubricant flow in the direction of motion is presented in Fig. 83. The drop of the carrier fluid’s viscosity causes the decreased shear stress developed on the bearing surface. The mean value of shear stress in the
direction of motion is directly connected with the amount of friction on the bearing surface. The drop of viscosity due to temperature causes a drop of the shear stress and thus friction. With lubricant temperature at 49°C the drop of shear stress reaches the 25.14%. For higher temperatures the drop of shear stress is increased. The drop of shear stress reaches the 27.81% for a lubricant temperature of 81°C.

Fig. 83. The increase of stiffness and damping coefficients for the case of 1500 rpm’s with \( L/D = 0.5 \). All coefficients are presented as percentage of the values in 81°C.

The stiffness and damping coefficients of a journal bearing rely significantly on the viscosity of the lubricant. Our code regarding the bearing dynamic properties have been validated with the values given by Glienecke et. al. in [14]. The stiffness and damping coefficients increase steadily with increasing temperature and a steady load for all cases concerned. The small perturbation techniques have been used in order to estimate the dynamic properties, as a function of the predicted apparent viscosity. Here, when we calculate the shear stresses through the aforementioned couple (hydrodynamic plus magnetic and particle dynamics) field problems, the apparent viscosity is accurately predicted. This means that the predictions of the rheological properties of the magnetorheological fluids from the micro to macroscale can be predicted more accurate.
In Fig. 84, the stiffness and damping coefficients are presented as ratio of their values in the case of temperature of 81°C, in order to describe the change of the dynamic coefficients over temperature. This comparison shows that the bearing stiffness and damping for the same load increases with temperature at the cost of decreased minimum film thickness. Most pronounced is the increase of the $K_{yy}$ term, which reaches the 88% between 49°C and 81°C. The reference values of the stiffness and damping properties at 81°C are for 1500 rpm's $K_{xx}=4.42E+05$ N/m, $K_{yy}=1.14E+06$ N/m, $C_{xx}=6.82E+07$ N s/m, $C_{yy}=3.04E+08$ N s/m and for 2000 rpm's, $K_{xx}=5.84E+05$ N/m, $K_{yy}=1.75E+06$ N/m, $C_{xx}=8.31E+07$ N s/m, $C_{yy}=4.14E+08$ N s/m. The same trend applies in the case of 2000 rpm's as well. The effect of temperature on the stiffness and damping coefficients for a rotational velocity of 2000 rpm's is shown in Fig. 85. As seen in the previous case of 1500 rpm's this increase of the stiffness and damping coefficients comes with the price of decrease of the minimum film thickness.
Fig. 85. The increase of stiffness and damping coefficients for the case of 2000 rpm’s with $L/D=0.5$. All coefficients are presented as percentage of the values in 81°C.

Again the stiffness coefficient $K_{yy}$ is the one term that presents the highest increase with the temperature. All the above results for the dynamic properties have been derived for an external load of 34.7N.

### 8.1 Conclusions on temperature effect

The existence of particles inside the lubricant volume causes a rise in pressure which is significant and could affect significantly the cavitation which appears in the lubricant. Additionally the non-linear viscosity models remain suitable for the description of the frictional forces acting in bearings lubricated with magnetorheological fluids but they tend to ignore the pressure rise developed in the lubricant and thus the overall performance of such fluids. Temperature is a factor which plays a significant role on the physical properties of magnetorheological fluids. The drop in the predicted pressure on the bearing between 49 and 81 ºC reaches the 70.58%, while the decrease of friction reaches the 78% for a journal velocity of 5.24m/s. The stiffness and damping coefficients are also widely affected by the temperature changes. The nominal temperature of the lubricant at the operating conditions must be taken into account in order to properly design and use a magnetorheological fluid journal bearing. This means that temperature may alter significantly the performance of bearings lubricated with magnetorheological lubricants.
8.2 Contribution
The microrheology of the MR fluids is a subject of ongoing research. However the results available in the literature concern either much lower scale of flow or they are concerned with the investigation of the yield stress of the MR fluids and not with the possibility of pressure induced by the existence of particles.

9 Experimental Investigation
The main concept in the design of the MR fluid bearing is the creation of a radial homogenous magnetic field which aligns the iron based particles inside the fluid into chains and changes its apparent viscosity.

The eccentricity ultimately becomes a function of the magnetic field. Of course the dynamic properties of the bearing change as well.

The equations of motion for the journal can be written as:

\[ F_x = m\ddot{x} + c_{sx}\dot{x} + c_{sy}\dot{y} + k_{sx}x + k_{sy}y \]  
\[ F_y = m\ddot{y} + c_{sy}\dot{y} + c_{sx}\dot{x} + k_{sx}x + k_{sy}y \]

In Fig. 86 the axes of motion along with the stiffness and damping forces are shown. If we treat the damping and stiffness coefficients as unknown factors, the system can be solved if four instances of journal's motion are considered. More specifically, for the x axis we consider:

\[ F_{x1} = m\ddot{x}_1 + c_{sx}\dot{x}_1 + c_{sy}\dot{y}_1 + k_{sx}x_1 + k_{sy}y_1 \]  
\[ F_{x2} = m\ddot{x}_2 + c_{sx}\dot{x}_2 + c_{sy}\dot{y}_2 + k_{sx}x_2 + k_{sy}y_2 \]
If we ignore the contribution of the coupled terms in the dynamic behavior of the bearing, the system changes into:

\[ F_{x1} - m\ddot{x}_1 = c_{xx}\dot{x}_1 + k_{xx}x_1 \]  \hspace{1cm} (5)

\[ F_{x2} - m\ddot{x}_2 = c_{xx}\dot{x}_2 + k_{xx}x_2 \]  \hspace{1cm} (6)

The solution of the above equations is found to be:

\[
c_{xx} = \begin{bmatrix}
F_{x1} - m\ddot{x}_1 & x_1 \\
F_{x2} - m\ddot{x}_2 & x_2 \\
\dot{x}_1 & x_1 \\
\dot{x}_2 & x_2 \\
\end{bmatrix}
\]  \hspace{1cm} (7)

\[
k_{xx} = \begin{bmatrix}
\dot{x}_1 & F_{x1} - m\ddot{x}_1 \\
\dot{x}_2 & F_{x2} - m\ddot{x}_2 \\
\dot{x}_1 & x_1 \\
\dot{x}_2 & x_2 \\
\end{bmatrix}
\]  \hspace{1cm} (8)

which after the calculation of the determinants can be written as:

\[
c_{xx} = \frac{(F_{x1} - m\ddot{x}_1)\cdot x_2 - (F_{x2} - m\ddot{x}_2)\cdot x_1}{(\dot{x}_1 \cdot x_2) - (\dot{x}_2 \cdot x_1)} \]  \hspace{1cm} (9)

\[
k_{xx} = \frac{(F_{x2} - m\ddot{x}_2)\cdot \dot{x}_1 - (F_{x1} - m\ddot{x}_1)\cdot \dot{x}_2}{(\dot{x}_1 \cdot x_2) - (\dot{x}_2 \cdot x_1)} \]  \hspace{1cm} (10)

Equations 9 and 10 can also be used for the calculation of \( c_{yy} \) and \( k_{yy} \) respectively. As mentioned in Eq. 9, the damping coefficients can be calculated through the measurement of force, velocity and displacement. With two sets of force measurements for each damping coefficient case, all the necessary data is available for calculation.

The experiments were conducted in a Bently Nevada Rk-4 rotorkit which is equipped with journal position and velocity sensors. The experimental set up is shown in Fig. 87.
The magnetorheological fluid bearing, shown in Fig. 88, constructed for the purposes of the experiments is equipped with two lateral coils. The bearing radius is 24.348E-3 m with radial clearance 67.5 μm and its slenderness ratio is $L/D=0.815$.

The coils of the magnetorheological fluid film bearing were supplied with a DC current of 2.4 Amperes, as seen in Fig. 89.
The magnetic field was used to create the proper excitation for the formation of chains of magnetic particles such as those depicted in Fig. 90.

Fig. 90.Chains of particles forming in the sides of the bearing with the magnetic field of the bearing activated. The lubricant flows freely in the sides.

In order to evaluate the journal bearing dynamic characteristics, a B&K hammer of Type 8202 with a force transducer sensor in its tip was used. Orbit characteristics along with force data were recorded in a UT-4000 digital oscilloscope. The main fluid used for the experiments was a SAE-10W oil based lubricant. In order to produce a suitable magnetorheological fluid for the experiments, the authors used iron particles powder, available for medical purposes. The mixture produced in vivo was given a 20% weight/weight content in iron particles. The overall lubricant volume was 700ml. No additives were used in the production of the lubricant. The diameter of the particles was confirmed to be, through measurements, up until 1 microns, with most of them being as large as 8 microns in diameter as shown in Fig. 91. The measurements were conducted in ICE-HT institute, of Patras Greece, using a Mastersizer S long bed of Malvern Instruments. The mixture produced in vivo was given a 20% weight/weight content in iron particles.

Fig. 91.Measurement of particle size distribution
9.1 Experimental Results

It was expected that the high yield stress of a magnetorheological fluid would enhance the dynamic characteristics of the journal bearing. This is evident in Fig. 92, where the orbit of the journal is presented for a rotational velocity of 250 rpm, with the magnetorheological fluid initially inactive and finally activated.

![Fig. 92. The journal orbit for 250 rpm's using magnetorheological fluid in its off and on state.](image)

With the activation of the coils, the orbit of the shaft is reduced by 75% in both axes of motion. In other words the available stability is significantly higher. The shaft operates without external loading (i.e. under its own weight).

The response of the magnetorheological fluid to the activation of the magnetic field is quick. In Fig. 93 the response time of the magnetorheological fluid can be evaluated through the change of the y axis journal motion before and after the activation of the magnetic field. As shown in the lower left corner of Fig. 93, the sampling period is 200ms. This means that within an overall 1.2 s the amplitude of the journal motion in the y axis at 250 rpm's is decreased by 45%.

![Fig. 93. The response of the journal motion to the activation of the magnetic field with the magnetorheological lubricant.](image)
The higher apparent viscosity of the MR fluid in its active state leads to significant benefits on the static performance of the journal bearing. In Fig. 94 journal orbits are experimentally depicted in a) the case of conventional function of the journal bearing (e.g. the coils are inactive) and b) the case of active magnetic field excitation. The lower orbits in both cases are orbits obtained with no load on the shaft other than its own weight. The higher orbits are obtained by the application of an upward load of 6kp (58.86N).

Relative eccentricity is apparently decreasing with the activation of the coils. More specifically there is a 5% decrease of relative eccentricity.

The dynamic properties of the journal bearing were calculated using a hammer with a force transducer. The input from the hammer was simultaneously compared with the effect on the journal position either on x or y axis. For simplicity reasons the procedure was limited to the main terms of damping and stiffness since the number of equations can be reduced if the coupled terms are neglected.

The stiffness of the journal bearing is widely enhanced by the use of the magnetorheological fluid. In Fig. 95, the $K_{xx}$ stiffness coefficient is presented for Newtonian and MR fluids. The MR fluid stiffness is calculated both in its active and its inert state. The results concern a steady load of 460 gr.
The results show the high stiffness achieved by the magnetorheological fluid in its active state.

The same behavior is shown in Fig. 96, where the $K_{yy}$ term is presented for the same range of journal rotational velocity. The active MR fluid offers higher stiffness in the y axis in comparison both to its inactive state and to the Newtonian fluid.

The active NMR fluid offers higher damping in the x axis, as shown in Fig. 97, where the $C_{xx}$ coefficient variation is compared for the Newtonian, the inactive and the active NMR fluids.
The benefits of the active magnetorheological fluid apply to the $C_{xy}$ damping coefficient as well, as shown in Fig. 98, where the $C_{xy}$ damping coefficient for a range of journal rotational velocity is depicted, using Newtonian active MR and inactive MR fluids.

Simulations performed for a nanomagnetorheological (NMR) fluid show that NMR fluids can also be applied as smart lubricants with promising results.

In the dimensional damping coefficients can be seen for a bearing with $R_b=49.999$ mm, radial clearance $C=61$ μm with journal rotating at 1000 rpm. The slenderness ratio is $L/D=0.8$. Using a NMR with properties as described in [66], the results for damping depicted in Fig. 100, show the potential of the NMR to absorb vibrations.
Fig. 99. Dimensional damping coefficient for a bearing with $L/D=0.8$ over a range of Sommerfeld number values.

Fig. 100. Dimensional damping coefficient for a bearing using NMR fluid with $L/D=0.8$ over a range of Sommerfeld number values.

It is observed that there is a reduction of the damping coefficients up to $S=0.2$ while a sharp gradient of one order of magnitude is obtained at $S=0.25$. The explanation for this phenomenon may be attributed to the specific conditions of this intermediate journal position. Ultimately there seems to be an intermediate position of the journal in medium values of relative eccentricity where damping is decreased. The non-linearities of the specific problem, especially due to the variable viscosity of the specific kind of fluids significantly affect the shape of the developed hydrodynamic pressure on the journal. In reference [3] the shape of the solid core of a non-newtonian fluid forming under various yield stress values is depicted. This shape changes significantly even for rather small changes in load/speed. Thus even the geometry of the lubricated volume is subject to changes dynamically when the MR fluid bearing operates.
9.2 Conclusions drawn from the Experimental Investigation

The use of smart fluids can enhance the availability of control over the dynamic properties of a journal bearing. The MR fluid produced for this specific set of experiments has shown that the effect of the magnetic field is strong and reliable. Simulations show that particles of diameter in the scale of nanometers could benefit the dynamic behavior of bearings lubricated with NMR fluids.

9.3 Contribution

While there are some efforts concerning the experimental investigation of magnetorheological fluid journal bearings [43] the main issue of the dynamic characteristics has never before been examined.

10 General Conclusions

Magnetorheological fluids can add the capacity of control when used for the lubrication of a journal bearing. When activated, magnetorheological fluids produce high friction. On the other hand, the ability of the magnetorheological fluids to change instantly the damping capacity of the journal bearing is substantial. Extreme vibrations in certain cases of shafts supported by journal bearings may be better handled by this type of a bearing without the permanent disadvantage of high friction (such as in the case of floating ring journal bearings).

Moreover the load capacity of the magnetorheological fluid journal bearing can be selectively increased. Thus this type of bearing may compensate the wear of the bushing due to the contact with the shaft. This capacity could be very useful in maritime applications where the maintenance procedures are hindered by difficulties in the flow of spare parts. A ship that can maintain its engine functionality during mid-sea operations or a drilling oil platform that needs a bearing replacement are two examples where the magnetorheological fluid journal bearings ability to compensate wear with increased load capacity and allow the operation of the rotating machinery for a certain period of time without harming any other part of these mechanisms, their supporting structures and any apparels being connected with the supported shaft.

The dynamic response of the rotor bearing system may also be affected and in certain cases the magnetorheological fluid journal bearings may permit the operation of the shaft with rotational velocity close to a critical speed value without the negative effect
of extreme and possibly hazardous vibrations. During the experimental investigation with the nanomagnetorheological fluid which was prepared in situ the activation of the fluid was almost instantaneous. The deactivation however would take a certain amount of time especially after a sequence of activation and deactivation cycles. Thus a certain semi-permanent magnetization effect was observed after a while. This may of course be related with the chemical composition of the paramagnetic particles.

Applying artificial texturing on a journal bearing may alleviate part of the friction, especially under starvation conditions, such as those prevalent during the start up and shut down phases of journal bearings operation. The use of artificial texturing is found to be beneficial and may improve both the friction coefficient and the load capacity. The said improvement though is somewhat limited. Additionally the dimples in a bushing with artificial texturing may function as particle reservoirs, with a suitably formed magnetic field.

The comparison of magnetorheological and nanomagnetorheological fluids shows that the latter may offer higher damping capacity but at the same time they lack the ability of control that the magnetorheological fluids may offer. This is linked with the porosity of the smaller paramagnetic particles of the nanomagnetorheological fluids, which leads to lower yield stress values. On the other hand, the lower sedimentation of the nanomagnetorheological fluids is a significant advantage, that ensures the reliability of the journal bearing in the long term.

The study of the microrheology of the magnetorheological fluid journal bearing yields two important conclusions. The first is that increase in temperature may indeed reduce minimum film thickness but the at the same time the damping capacity of the bearing is increased. The second is the value of the non linear viscosity models (such as Bingham) in the representation of the rheology of the MR and NMR fluids. While shear stress is present in the microflow simulations, some amount of pressure exists as well. This pressure induced by the existence of the particles in the flow of the lubricant is not predicted by the nonlinear viscosity models currently available. However it is a matter of further investigation how much this amount of pressure is significant for the application of the journal bearing.

The most important conclusions drawn from the experiments included in this thesis is the ability of the nanomagnetorheological fluid journal bearing to reduce the extent of
orbit, increase its damping capacity and activate rapidly. One major drawback that is matter of future research is the ability of the magnetorheological and nanomagnetorheological fluids to be used with some sort of filtering devices in the lubricant circuit. This would enhance the quality of the base fluid and would allow longer periods between maintenance inspections. The problem is that it would be difficult for the particles to pass the filter. One way around this problem would be a certain by pass that would permit the particles that come out of the journal bearing to return back. Their magnetic properties would be useful in order to discriminate them and make such a bypass possible. All in all, despite some practical aspects that remain to be solved, the ability of control of the load carrying capacity, damping and friction coefficient is significant and certainly worth of further development.

The future of the nano/magnetorheological fluids used in journal bearings is promising. The compensation of wear and the control of the rotordynamic behavior of the supported rotating machinery are two very strong incentives towards the continuation of the research of this type of bearings. In maritime applications for instance, where the repair of the journal bearing may mean the engine must be shut down and the ship may be left adrift, the specific type of bearing may very well compensate the wear of the bushing with the push of a button, letting the ship sail without imminent need of mid-sea repairs, which are most of the times cost intensive.

The main issues that remain to be resolved are concerned with the practical application of the MR and NMR fluids in real life journal bearings, where auxiliaries, such as filtering systems, existing lubricant circuits and limited available spaces for coils and the core of the MR fluid journal bearings may hinder their integration in current machine designs. While it is true that using this type of a bearing dictates a top to bottom redesign of the relevant machinery, the expected benefits overcome the problems that may arise in the application of this new type of bearing design.
References


Tribological Design of Nano/Magnetorheological Fluid Journal Bearings


Publications

Work on nano/magnetorheological fluid journal bearings

Journal papers


Conference papers


Charpter In Book


Work on journal bearings

Journal papers


Conference papers


Other research interests

Journal papers


**Conference papers**


ANNEX A

The mass inertia matrices for the Timoshenko beam elements are formulated for the two bending planes $xy$ and $xz$ as follows:

$$
[M^e_T] = [M^e_T]_{xy} + [M^e_T]_{xz}
$$

where

$$
[M^e_T]_{xy} = pA\int_0^l V(x)^T V(x)\,dx = pA[x]_{xy}^T \int_0^l f(x)^T f(x)\,dx[x]_{xy}^{-1}
$$

and finally:

$$
[M^e_T]_{xy} = pA[x]_{xy}^T [z]_{xy}[x]_{xy}^{-1}
$$

where:

$$
[z]_1 = \int_0^l f(x)^T f(x)\,dx = \begin{bmatrix}
\ell & 0 & 0 & \ell^2/2 & \ell^3/3 & 0 & 0 & \ell^4/4 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
\ell^2/2 & 0 & 0 & \ell^3/3 & \ell^4/4 & 0 & 0 & \ell^5/5 \\
\ell^3/3 & 0 & 0 & \ell^4/4 & \ell^5/5 & 0 & 0 & \ell^6/6 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
\ell^4/4 & 0 & 0 & \ell^5/5 & \ell^6/6 & 0 & 0 & \ell^7/7 \\
\end{bmatrix}
$$

and:

$$
[x]_{xy}^{-1} = \frac{1}{A} \begin{bmatrix}
\ell^4 + 12\beta\ell^2 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
-12\beta\ell & 0 & 0 & \ell^4 + 6\beta\ell^2 & 12\beta\ell & 0 & 0 & -6\beta^2 \\
3\ell^2 & 0 & 0 & -2\ell^3 - 6\beta\ell & 3\ell^2 & 0 & 0 & -\ell^3 + 6\beta\ell \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & \ell^2 & 0 & 0 & 0 & 0 \\
2\ell & 0 & 0 & -2\ell & 0 & 0 & \ell^2 & 0 \\
\end{bmatrix}
$$

Moreover, $A$ is the area of the cross section, $A = l^4 + 12\beta\ell^2$, $l$ is the element length and $\beta = EI/(kGA)$, $k$ is the shear coefficient, $I$ is the inertial moment, $G$ is the
shearing modulus. The inertia matrix of each element which is attributed to rotation is formulated as follows:

\[ [M^e_R] = [M^e_{R,xy}] + [M^e_{R,zz}] \]

The \([M^e_{R,xy}]\) is given by:

\[ [M^e_{R,xy}] = pI \int_0^l \theta(x) \theta(x) dx \]

and finally:

\[ [M^e_{R,xy}] = pI [z] [z]^{-1} \]

where:

\[ [z] = \int_0^l g(x) g(x) dx = \begin{bmatrix}
0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & \ell & \ell^2 & 0 & 0 \\
0 & 0 & \ell^2 & 4\ell^3/3 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 3\ell^3/3 + 6\ell\ell & 6\ell^4/4 + 6\ell\ell^2 & 9\ell^5/5 + 36\ell^3/3 + 12\ell\ell^3
\end{bmatrix} \]

The differential equation of the Timoshenko beam element can be expressed as:

\[
([M^e_t] + [M^e_R]) \cdot \{\ddot{q}^e_t\} - \Omega [G^e] \{\dot{q}^e_t\} + ([K^e_s] - [K^e_0]) \cdot \{q^e_t\} = \{\Omega^e\}
\]

Since \(\{q^e_t\}\) is the displacement of each degree of freedom the two matrices can be combined into one unified matrix of inertia.

\[ [M] = ([M^e_t] + [M^e_R]) \]

Due to the oscillation of the rotating shaft, Coriolis forces are bound to appear. The Coriolis forces are calculated as an external force acting on the element according to the following equation:

\[
[F_t] - [G] \dot{\vec{X}} = \{\Omega^e\} \quad \text{[18]}
\]

where \([F_t]\) is the Coriolis forces vector, \([G]\) is the global Coriolis matrix and \(\dot{\vec{X}}\) is the nodal velocity vector. The global Coriolis matrix is calculated as follows:
\[
[G] = \sum_{i}^{n} [G_i] \tag{19}
\]

where \([G_i]\) is the element Coriolis matrix. The element Coriolis matrix is calculated as follows:

\[
[G_i] = ([N_e] - [N_e]^T), \tag{20}
\]

where \([N_e] = \int J_p [\Phi]^T [\Phi] dx\) and \(J_p\) the polar moment of inertia of the cross section of the shaft. As shown in (25) the stiffness matrix due to bending for the Timoshenko element is:

\[
[K_e] = EI\beta[x]^T \begin{bmatrix} z_0 & 0 & 0 \\ 0 & z_0 & 0 \\ 0 & 0 & z_0 \end{bmatrix} + EI[x]^T \begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix} + EI\beta[x]^T \begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix} + EI[x]^T \begin{bmatrix} w_0 & 0 & 0 \\ 0 & w_0 & 0 \\ 0 & 0 & w_0 \end{bmatrix} \tag{21}
\]

where \(E\) is the modulus of elasticity, \(I\) is the second moment of area, \(k\) is the shear coefficient, \(G\) is the shear modulus and \(A\) is the cross section area. The matrices are described in detail in the work of Gounaris and Papadopoulos [30]. Furthermore, if the rotor is supported by journal bearings, the stiffness and damping matrices have to be calculated and the global stiffness and damping matrices must be included.

\[
[K_e]_{\text{total}} = [K_e] + [K_b]^T \tag{22}
\]

Where \([K_b]\) is the bearing stiffness matrix at a node \(n\) and

\[
[C_e]_{\text{total}} = [C_e] + [C_b]^T \tag{23}
\]

where \([C_b]\) is the bearing damping matrix at node \(n\).
JOURNAL PAPERS
Magnetorheological fluid journal bearing can be controlled by a steady magnetic field doing that very effective for attenuating and controlling the performance of the rotor bearing systems.

An integrated simulation study, of a magnetorheological (MRF) fluid journal bearing, via computational fluid dynamics (CFD) and finite element method (FEM) is presented in this paper. The journal bearing characteristics such as, eccentricity, attitude angle, oil flow and friction coefficients are calculated and presented as functions of the magnetic field, and L/D bearing ratios.

A specific procedure in order to simulate an MRF bearing operated in high eccentricity ratios is also presented and the meshing requirements are discussed.

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1. Introduction

1.1. Magnetorheological fluids in general

Magnetorheological fluid (MRF) is a manageable fluid that exhibits drastic changes in rheological properties adjustable and interchangeable to the applied magnetic field strength. The fluid is potentially advantageous to be employed in many applications. MRF is a kind of controllable or smart fluids whose rheological properties can be dramatically and reversibly varied by the application of an external magnetic field in a very short period of time. The MRF has the property of a normal viscosity in the absence of an external magnetic field, but in the presence of a strong magnetic field immediately solidifies to a grease state.

1.2. Application of magnetorheological fluids

The MRF’s are one of the most active “smart materials” of the current range. Most researches in the application of the MRF’s are focused on structural vibration control and flow power system. Stanway et al. [1] and Wang and Meng [2] made a survey study in the state of the MRF’s and the application of the MRF’s in several mechanical engineering systems. There are many papers dealing with the application of the magnetorheological fluids for controllable dampers [3–7] for seismic response control of frame structures [8] and vibration control of large structures [9]. The rapid, reversible and dramatic changes in its rheological properties provide a possibility of control in flow power systems [10,11].
1.3. Bearings and magnetorheological fluids

Congenital smart fluids to magnetorheological are the electrorheological fluids (ERF), which also exhibit drastic changes in rheological properties and interchangeable depending on the applied electric field strength. There are also, dozens of papers published which deal with the application of ER fluid in bearings for rotational machinery.

Vance and Ying [12] developed and demonstrated the dynamic behavior of the rotor systems supported on the multi-disk ER fluid damper. Dimarogonas and Kollias [13,14] studied stability of a rotor system supported by journal bearings with ER fluid theoretically and compared the capability of three kinds of ER fluid damper for controlling the rotor vibration.

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A_m$</td>
<td>magnetic vector potential (T)</td>
</tr>
<tr>
<td>$C,c$</td>
<td>clearance $C = R_b - R_j$ (m)</td>
</tr>
<tr>
<td>$D$</td>
<td>journal diameter</td>
</tr>
<tr>
<td>$e$</td>
<td>eccentricity (m)</td>
</tr>
<tr>
<td>$f_b$</td>
<td>friction coefficient on the bearing surface</td>
</tr>
<tr>
<td>$f_j$</td>
<td>friction coefficient on the journal surface</td>
</tr>
<tr>
<td>$F_{fr}$</td>
<td>frictional force (N)</td>
</tr>
<tr>
<td>$g$</td>
<td>gravity acceleration (m/s$^2$)</td>
</tr>
<tr>
<td>$h_1$</td>
<td>minimum film thickness (m)</td>
</tr>
<tr>
<td>$h_{b0}$</td>
<td>maximum film thickness (m)</td>
</tr>
<tr>
<td>$B$</td>
<td>magnetic field density (T)</td>
</tr>
<tr>
<td>$H$</td>
<td>magnetic field intensity (A/m)</td>
</tr>
<tr>
<td>$I$</td>
<td>unit tensor</td>
</tr>
<tr>
<td>$I$</td>
<td>electric current (A)</td>
</tr>
<tr>
<td>$j$</td>
<td>current density (A/m)</td>
</tr>
<tr>
<td>$k$</td>
<td>consistency factor (kg s$^{-2}$ m$^{-1}$)</td>
</tr>
<tr>
<td>$L$</td>
<td>bearing length (m)</td>
</tr>
<tr>
<td>$N$</td>
<td>rotational speed of journal (rps)</td>
</tr>
<tr>
<td>$p$</td>
<td>pressure</td>
</tr>
<tr>
<td>$p_a$</td>
<td>ambient pressure</td>
</tr>
<tr>
<td>$Q$</td>
<td>lubricant flow rate (m$^3$/s)</td>
</tr>
<tr>
<td>$Q_s$</td>
<td>lubricant side leakage flow rate (m$^3$/s)</td>
</tr>
<tr>
<td>$R_j,R_b$</td>
<td>journal and bearing radii</td>
</tr>
<tr>
<td>$S_0$</td>
<td>Sommerfeld number $S_0 = \mu_f \cdot \omega \cdot R_j \cdot L(R_j/C)^2/(\pi \cdot W)$</td>
</tr>
<tr>
<td>$\tau$</td>
<td>stress tensor</td>
</tr>
<tr>
<td>$\sigma_0$</td>
<td>yield stress (Pa)</td>
</tr>
<tr>
<td>$\vec{u}, \vec{v}, \vec{w}$</td>
<td>fluid velocity vectors</td>
</tr>
<tr>
<td>$W$</td>
<td>external force (N)</td>
</tr>
<tr>
<td>$U$</td>
<td>journal velocity, parallel to the film</td>
</tr>
</tbody>
</table>

Subscripts

0 indicates the position of maximum film thickness at $\theta = 0$
1 indicates the position of minimum film thickness at $\theta = \pi$
$b$ indicates the bearing
$j$ indicates the journal

Greek symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\gamma$</td>
<td>shear rate</td>
</tr>
<tr>
<td>$\varepsilon$</td>
<td>relative eccentricity $\varepsilon = e/C$</td>
</tr>
<tr>
<td>$\theta$</td>
<td>bearing angle ($^\circ$)</td>
</tr>
<tr>
<td>$\mu$</td>
<td>dynamic viscosity (Pa s)</td>
</tr>
<tr>
<td>$\mu_a$</td>
<td>apparent viscosity (Pa s)</td>
</tr>
<tr>
<td>$\mu_f$</td>
<td>fluid state viscosity (Pa s)</td>
</tr>
<tr>
<td>$\mu_p$</td>
<td>solid state viscosity</td>
</tr>
<tr>
<td>$\mu_o$</td>
<td>permeability of free space (m kg/(A s)$^2$)</td>
</tr>
<tr>
<td>$\mu_r$</td>
<td>relative permeability</td>
</tr>
<tr>
<td>$\rho$</td>
<td>lubricant density (kg/m$^3$)</td>
</tr>
<tr>
<td>$\tau_0$</td>
<td>yield stress (Pa)</td>
</tr>
<tr>
<td>$\phi$</td>
<td>attitude angle ($^\circ$)</td>
</tr>
<tr>
<td>$\omega$</td>
<td>rotational speed of journal (rad/s)</td>
</tr>
</tbody>
</table>

1.3. Bearings and magnetorheological fluids

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Nikolajsen and Hoque [15] presented a multi-disk ER fluid damper operating in shear flow mode and studied the effectiveness of the multi-disk ER fluid damper in controlling the vibration of rotor systems when passing through the critical speeds.

Nikolakopoulos and Papadopoulos [16] studied the dynamic characteristics of a controllable journal bearing lubricated with the ER fluid.

Based on the Bingham fluid theory, Tichy [17], Docier and Tichy [18] analyzed the dynamic characteristics of fluid film force and the existing conditions and manner of core in the ER fluid squeeze-film damper and journal bearings. Recently, Gertzos et al. [19] presented a CFD analysis in which for Bingham lubricated journal-bearing performance characteristics, such as relative eccentricity, attitude angle, pressure distribution, friction coefficient, lubricant flow rate, and the angle of maximum pressure, are derived and presented for several length over diameter (L/D) bearing ratios and dimensionless shear numbers of the Bingham fluid. The above diagrams presented in the form of Raimondi and Boyd charts, and can easily be used in the design and analysis of journal bearings lubricated with Bingham fluids.

From the above literature it is obvious that the ER fluids give the possibility of controllable journal bearings. However, in comparison with the properties of the ER fluid, an MRF inherently has higher yield strength; therefore it is capable of generating a greater fluid force. Furthermore, the MRF is activated by the application of an external magnetic field, which is easily produced by a simple, low-voltage electromagnetic coil and avoids probably arcing problems.

1.4. Simulation of the magnetorheological fluid journal bearing

Furthermore in the field of magnetorheological fluid lubricated journal bearings the published works are limited.

Zhu [20] presented an MRF squeeze-film damper operating in the squeeze film mode and showed that the MRF squeeze-film damper can effectively control the vibration of a rotor system, but an unbalanced magnetic pull force existing in the journal due to the eccentricity of the journal with respect to the damper housing may pull the journal to the damper housing and lock up the damper like a rigid support when the applied current in the coil is over a certain value.

Wang and Gordaninejad [4] combine a fluid mechanics-based approach and the Herschel–Bulkley constitutive equation to develop a theoretical model for predicting the behavior of field-controllable, magnetorheological, and electrorheological (ER) fluid dampers.

Hesselbach and Abel-Keilhack [21] investigated the influence of the magnetic field on the bearing gap of hydrostatic bearings with MRF’s. They found that, in a closed loop control, a nearly infinite stiffness, only limited by the resolution of the measuring system, can be achieved. The results showed that the concept of a hydrostatic bearing with MRF’s can overcome the drawbacks (stiffness and response time) of conventional hydrostatic bearing.

Kim et al. [22], presented a controllable semi-active smart fluid damper (SFD) using magnetorheological fluids, focusing on its design and modeling. It offers a comprehensive design method and an innovative experimental identification and modeling technique for MR-SFDs. They constructed a prototype MR-SFD and investigated how some of the critical design parameters affect the performance of the MR-SFD. Additionally they characterized the damper’s dynamic behavior experimentally using a novel excitation method that adopts active magnetic bearing (AMB) units. In modeling the dynamic behavior of the MR-SFD, they employed the describing function method. The describing function analysis effectively captured the non-linear dynamic behavior of the MR-SFD. Carmignani et al. [23] presented an analytical, numerical and experimental study off a magnetorheological squeeze-film damper. Numerical simulations were carried out in order to evaluate the dynamic behaviour of the damped rotor as a function of the current supplied to the adjustable device. A linear model that depicts the main characteristics of the system has been developed as a useful tool in damper and control design. They tested different fluids, and an optimal fluid has been singled out. The tests conducted on the selected fluid shown that it is possible to have the optimum conditions for each steady rotational speed.

Urreta et al. [24] summarizes the work carried out in the development of hydrodynamic lubricated journal bearings with magnetic fluids. Two different fluids have been analyzed, one ferrofluid from FERROTEC APG s10n and one magnetorheological fluid from LORD Corp., MRF122-2ED. Theoretical analysis has been carried out with numerical solutions of Reynolds equation, based on apparent viscosity modulation for ferrofluid and Bingham model for magnetorheological fluid. The authors in order to validate their model, designed, manufactured and set up a test bench, where their preliminary results shown that magnetic fluids can be used to develop active journal bearings.

The design of a magnetorheological squeeze-film damper is presented and discussed by Forte et al. [25]. A numerical simulation has been carried out in order to evaluate the dynamic behavior of the damped rotor as a function of the magnetic field strength. The authors made a test rig of a slender shaft supported by two oilite bearings and an unbalanced disk. The damper was interfaced with the shaft through a rolling bearing and the electric coils generate the magnetic field whose field lines cross the magnetorheological film.

1.5. Contribution of this paper

In this paper a simulation study via computational fluid dynamics is presented for a magnetorheological fluid lubricated journal bearing. Both, magnetic and rheological fields are solved. The aim of this paper is: (i) the presentation of a coupled model which solves the magnetic and rheological field giving a tool for the design of the MRF lubricated bearings, (ii) the
accurated calculation of the bearing attitude angle, eccentricity, friction coefficients and oil flow even for high eccentricities versus the magnetic field and (iii) the numerical techniques to get accurate results in high eccentricities.

The paper is organized as follows: The problem definition and computation approach are first presented early in Section 2, followed by an analytical validation of the model, in Section 3. Simulation results are then presented and discussed in Section 6, and the main findings are finally summarized and discussed in Section 7.

2. Problem formulation and validation

2.1. Geometrical model

In the present work, the bearing is considered to be rigid, and the flow steady and isothermal. The geometry of the bearing follows the model that is shown in Fig. 1; here, $O_b$ is the bearing centre, $O_j$ the journal centre, $R_b$ the bearing radius, $R_j$ the journal radius, $e$ the bearing eccentricity, $\phi$ the attitude angle, and $L$ the bearing length. The external load $W$ is assumed vertical (i.e. along the $y$-axis) and constant and a magnetic field $H$ is applied between rotor and the bearing.

2.2. Governing equations and assumptions

The following assumptions for the bearing model are used in this work:

A rigid aligned bearing with the geometry of Fig. 1 is considered; a steady-state operation is assumed; the flow is laminar and isothermal; a constant external vertical load $W$ is applied to the journal; the minimum pressure value is assumed to be above the vapor pressure; thus cavitation is not accounted for.

The viscosity of magnetorheological fluids can be approximated with the Bingham law (Fig. 2a) for yield stress:

$$\tau = \tau_0(H) + \mu \dot{\gamma}$$

where $\tau$ is the shear stress of the material, $\tau_0$ the critical shear stress or yield stress and $\dot{\gamma}$ the shear rate. The relation of the critical shear stress $\tau_0$ with the magnetic field intensity $H$ can be estimated by experimental data. For certain magnetorheological fluids this relation is available through manufacturer’s literature. It is possible to obtain an equivalent or apparent viscosity:

$$\mu_a = \mu_f + \tau_0(H) \frac{\partial \bar{u}}{\partial y}$$

where $\mu_a$ is the apparent viscosity of the material and $\mu_f$ is the Newtonian viscosity of the material when the shear stress overcomes the yield stress, in which case the material is flowing. During the magnetostatic simulation in ANSYS the Bingham model is defined somewhat differently by a bi zone viscosity model (Fig. 2b). So for the purposes of the simulation the apparent viscosity is a function of two separate viscosity regions. The first region is the plastic viscosity region where the material

![Fig. 1. General geometry and characteristics.](image-url)
exhibits the Bingham solid behavior. In this region the viscosity takes a high value. This is the plastic viscosity or $\mu_p$. When
the shear stress overcomes the yield threshold, the behavior of the Bingham material is described with the viscosity of flow or $\mu_f$. Thus the apparent viscosity of the Bingham model in the ANSYS simulation environment is mathematically defined as:

$$
\mu = \begin{cases} 
\mu_f + \frac{\tau_0}{\gamma}, & \dot{\gamma} > \frac{\tau_0}{\mu_f - \mu_p} \\
\mu_p, & \dot{\gamma} \leq \frac{\tau_0}{\mu_f - \mu_p}
\end{cases}
$$

Typically the plastic viscosity is defined with a value of $\mu_p = 100\mu_f$ in order to replicate properly the Bingham behavior. The yield stress of the magnetorheological fluid is considered constant throughout the fluid volume for the purposes of the simulation. This assumption implies a homogenous magnetic field.

The fundamental expression for the magnetic field is given by the Gauss law:

$$\nabla \cdot \mathbf{B} = 0$$

The differential form for Amperes law including Maxwell's correction is:

$$\nabla \times \mathbf{H} = \mu_0 \mathbf{J} + \mu_0 \varepsilon_0 \frac{\partial \mathbf{E}}{\partial t}$$

where $\mu_0$ is the permeability of free space, $\varepsilon_0$ the permittivity of free space, $\mathbf{B}$ the magnetic field density, $\mathbf{H}$ the magnetic field intensity, $\mathbf{E}$ the electric field and $\mathbf{J}$ the current density. In the case of magnetostatic analysis the relation between $\mathbf{B}$ and $\mathbf{H}$ becomes:
\[ \vec{B} = \mu_0 \mu_r \vec{H} \]  
(6)

where \( \mu_r \) is the relative magnetic permeability of the material in which the magnetic density vector \( \vec{B} \) is calculated.

The conservation equations for unsteady incompressible and isothermal flows are expressed by the mass conservation equation:

\[ \frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{u}) = 0 \]  
(7)

and the momentum equations,

\[ \frac{\partial (\rho \vec{u})}{\partial t} + \nabla \cdot (\rho \vec{u} \vec{u}) = -\nabla p + \nabla \cdot (\tau) + \rho \vec{g} + \vec{F} \]  
(8)

Fig. 3. Flowchart of simulation procedure.
where \( \rho \ddot{g} \) and \( \ddot{F} \) are the gravitational and external body forces, respectively. The stress tensor \( \ddot{\tau} \) is given by,

\[
\ddot{\tau} = \mu_s \left[ (\nabla \ddot{u} + (\nabla \ddot{u})^T) - \frac{2}{3} \nabla \cdot \ddot{u} \right]
\]  

(9)

The term \( \frac{2}{3} \nabla \cdot \ddot{u} \) is the effect of volume dilation.

Eqs. (7)–(9) were solved with the computational fluid dynamics code (CFD) Flotran of the ANSYS Multiphysics suite. Also, Eqs. (4)–(6) were solved with the finite element method using ANSYS Multiphysics for the magnetostatic case. Consequently, concerning the magnetorheological fluid bearing, the pressure field was predicted using the set of Eqs. (1)–(9), combined the Bingham model with the magnetic field intensity. It is true that the use of Navier–Stokes equations, instead of the generalized Reynolds equation, signifies greater computational costs. However, the work at hand is only the first step in obtaining solution to different problems in the field of magnetorheological fluid lubricated journal bearings, where the Reynolds equation has certain limitations. One of the main assumptions included in Reynolds equation is that the influence of the inertial force is omitted. Thus the use of Reynolds equation is limited when the clearance to diameter ratio becomes larger than a specific threshold. Moreover Navier–Stokes should be more suitable for high density materials, such as the MRF. For example, the density of a typical SAE-30 oil is approximately 890 kg/m\(^3\). The MRF-132DG which was used has a density of 2980 kg/m\(^3\). Given the magnitude of density for the specific application, Navier–Stokes equations have been chosen, giving a safe and generic tool for the simulation of a magnetorheological bearing.

The solution steps are illustrated in the flowchart of Fig. 3. The simulation process begins with the input of the main characteristics of the bearing and the coil, that consist the magnetorheological bearing. The magnetic problem is then solved. Given the magnetic field intensity in the area of MRF, the next step is the calculation of the occurring yield stress of the magnetorheological lubricant and its effective viscosity. Consequently the geometry of the lubricant volume for the given eccentricity is created. The mesh of the fluid volume is defined in sequence, with the proper density of nodes and elements. The space of minimum lubricant thickness is of specific interest at this point and the mesh thickness is specifically adjusted there. The problem of fluid flow is then solved. The resulting pressure is integrated around the journal and the bearing surfaces and the required calculations concerning the physical and other quantities of the flow, friction and load capacity are conducted here. Since the calculation of many different eccentricities ratios is required, the solution procedure is repeated for different eccentricity ratios, length to diameter ratios, and current intensity values.

### 2.3. Boundary conditions of the magnetostatic model

As shown in Fig. 3, the first step of these coupled simulations is the magnetostatic analysis. The boundary conditions of the magnetostatic analysis are shown in Fig. 4. The magnetic potential of the outer boundaries of the bearing assembly are set to be zero \( (A_z = 0) \). On the other hand the load of the coil is the current density. The current density of the coil’s cross section is

\[
J = \frac{N_{\text{turns}}I}{A_{\text{coil}}}
\]  

(10)

The properties of the components materials are considered as linear. The number of turns of the coil for all coupled simulations has a constant standard value of 4000 turns. The cable diameter is also constant for all coupled simulations with a value of \( d_{\text{cable}} = 0.5 \) mm.

![Fig. 4. Boundary conditions of magnetostatic model.](image)
2.4. Boundary conditions of the CFD model

We consider the bearing wall as stationary and the journal as rotating wall. The sides of the lubricant volume have been assigned with a zero pressure condition, meaning that the lubricant is free to flow there. The cavitation within the lubricant was modeled using the half Sommerfeld boundary condition.

\[ p - p_a \geq 0, \quad \pi \leq \theta \leq 2\pi, \quad z = 0, \quad z = L \]

The half Sommerfeld condition, utilized in the present work, neglects all negative pressures in the diverging part of the fluid film, which are physically unrealistic. The boundary conditions of the CFD problem are shown in Fig. 5. The half Sommerfeld condition, which is depicted as a darker area in Fig. 5, offers sufficient accuracy, fast convergence, and is selected in this work to accelerate the solution of the CFD problem. The use of zero pressure boundary condition at the sides of the bearing implies

Fig. 5. Boundary conditions of CFD model.

Fig. 6. The yield stress versus magnetic field strength \([26]\) (Lord Inc. MRF-132DG).
the leakage of the lubricant at the sides. The boundary condition for entry of the lubricant would simply be unnecessary because of the half Sommerfeld boundary condition. In other words, since the lubricant enters the bearing space at atmospheric pressure, a separate boundary condition for the lubricant inlet would be overlaid to half Sommerfeld and thus would be redundant. The Reynolds boundary condition, not utilized here, assumes that the positive pressure curve terminates with a zero gradient in the divergent part of the film; it gives in some cases more accurate results than the half Sommerfeld boundary condition. Nonetheless, it is still an approximation to the transition from single-phase flow to multi-phase flow, and is computationally more demanding.

2.5. Magnetorheological materials properties

The LORD MRF-132DG magnetorheological fluid is used in order to obtain results. The Newtonian viscosity of this fluid at 40 °C is 0.092 ± 0.015 Pa s, with density 2.93–3.18 g/cm³ and the solids content is 80.98% by weight, and operating temperature from −40 °C to 130 °C [26]. In Fig. 6, the yield stress of the magnetorheological lubricant is depicted in relation with the magnetic field intensity.

![Mesh of magnetostatic model and element PLAIN 53 geometry features.](image1)

**Fig. 7.** Mesh of magnetostatic model and element PLAIN 53 geometry features.

![Volume of lubricant divided into sectors.](image2)

**Fig. 8.** Volume of lubricant divided into sectors.
3. Simulation model

3.1. Magnetic field simulation meshing requirements

We use a standard element size of 1 mm in order to perform the basic meshing procedure. The need for high accuracy in our results within the magnetorheological fluid domain drives us to perform a secondary meshing in that particular area with a smaller element size.

The geometry is considered axisymmetric. In Fig. 7, the element’s colors depict the regions of different materials used for the analysis. The high mesh density in the region of the MRF is clearly depicted. We use the PLANE 53 element. The geometrical definition of PLANE 53 element and its features are shown in Fig. 7. A grid of a minimum 16,300 elements and 33,000 nodes was used for the solution, of the magnetic field and, a grid of 250,700 elements and 288,600 nodes was also used for the solution, of the rheological field.

3.2. Meshing and eccentricity requirements in CFD analysis

It is evident that the small gap formed by the bearing and the shaft when the eccentricity of the shaft is significant, must be well described, with a relatively dense mesh. This requirement poses the need for an adaptive meshing strategy that will balance our need for the least computational cost possible without any compromise on the accuracy of the CFD analysis. The meshing procedure is adaptive in relation to eccentricity and has been divided in two parts. The first part, in the region of maximum lubricant thickness, is meshed with a standard number of one circumferential divisions per degree. In the region of minimum lubricant thickness the circumferential divisions exceed the five divisions per degree. This number of divisions is subject to change in accordance with the respective eccentricity ratio. In order to achieve the adaptive and regionally refined meshing, the volume of the lubricant was divided into several volumes. Fig. 8 shows the geometrical model and the circular sectors that assemble the lubricant volume. Both the divisions parallel to the axis of the bearing and the divisions of the lubricant thickness are set to standard values of 16 and 12, respectively.

Fig. 9 shows the mesh formed for an eccentricity ($e$) of 80%. The element used in the CFD simulation is the FLUID 142, the geometry and the features of which are shown in Fig. 8.

3.3. Solution of the equilibrium

The calculation of forces acting on the journal is performed with the integration of pressure in the area of the journal. The result is the total force acting on the journal. The model’s kinematic behavior is totally static. This means that journal’s position is defined a priori and that the force calculated is equal in size and in opposite direction of the load of the bearing. Each CFD simulation was executed for specific eccentricities. The same technique was used for both the Newtonian and the Bingham fluid. The attitude locus can be obtained by coupling the results of Sommerfeld number, eccentricity ($e$) and attitude angle ($\phi$).
4. Numerical model validation

To validate the present computational approach, we test our tool for the case of: (i) a journal bearing operating with Bingham fluids and (ii) we validated our results regarding the magnetic field simulation with a squeeze-film damper lubricated with magnetorheological fluids. We compare our results in case (i) to those reported in [19] for a similar problem setup, see Figs. 10a–11b, and for case (ii) to those reported in [25], see Figs. 12a and 12b.

In case (i) of Bingham validation with Ref. [19] the following data used: $\mu = 0.2 \text{ Pa s}$, $R_b = 25 \times 10^{-3} \text{ m}$, $c = 235 \times 10^{-6} \text{ m}$, $N = 120 \text{ rps}$.

The Sommerfeld number is presented versus the relative eccentricity ($\epsilon$) in Figs. 10a and 10b for $T_0 = 0$ and $T_0 = 0.4$, respectively. The journal friction coefficient ($f_j$) is presented versus the Sommerfeld number in Figs. 11a and 11b for $T_0 = 0$ and $T_0 = 0.4$, respectively. The bearing slenderness ratio is $L/D = 1$. As it is observed, very good agreement is obtained.

In case of validation with Ref. [25], we plot in Fig. 12a the variation of the magnetic field density inside the volume of the magnetorheological fluid damper, while in Fig. 12b the same results from the simulation of present paper. The data given in [25] are: $L = 2 \times 10^{-2} \text{ m}$, $c = 2.5 \times 10^{-3} \text{ m}$, $R_b = 3.2 \times 10^{-2} \text{ m}$.

The analysis of the magnetic behavior of the bearing was carried out by using a finite element code ANSYS relating the value of the electric current running in the coils and the value of the magnetic field in the fluid. All material magnetic properties were assumed linear, i.e. constant relative magnetic permeability, $\mu_r = 2000$ for steel and $\mu_r = 5$ for the MRF.

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![Fig. 10a. Validation diagram with Ref. [19], eccentricity ($\epsilon$) versus Sommerfeld number for $T_0 = 0$ and $L/D = 1$.](image1.png)

![Fig. 10b. Validation diagram with Ref. [19], eccentricity ($\epsilon$) versus Sommerfeld number for $T_0 = 0.4$ and $L/D = 1$.](image2.png)
The bearing magnetic permeability was neglected because of the discontinuous structure (i.e., balls and plastic cage). It is also observed a very well agreement. The maximum magnetic flow intensity is 1.978 T and 1.952 T in the present work. The values are almost identical.

5. Performance characteristics

The relations below were used in order to obtain solution. The friction force is calculated in the equilibrium position by integrating the shear stress \( \tau \) over the bearing (or journal) area:

\[
F_{f_i} = \int \int_{A_i} \tau_i dA_i, \quad i = j \text{ or } b
\]  

where \( i = j \) refers to the journal and \( i = b \) refers to the bearing and \( A_i \) is the total area of journal or bearing.

The friction coefficient is calculated by the relation:

\[
f_i = \frac{F_{f_i}}{W}
\]
The load-carrying capacity that the bearing will support is found by integrating the pressure around the journal. The boundary condition defined by Eq. (11) is included in the integration to prevent sub-atmospheric pressure. At equilibrium position, the load capacity of the journal equals the external vertical load.

$$F_{pl} = \int \int_{A_j} p dA_j = -W, \cdots F_{pl} = 0$$  \hspace{1cm} (14)

The lubricant flow rates, $Q_0$ at the maximum position and $Q_1$ at the minimum position of the film thickness, are calculated by integrating the tangential velocity of the lubricant:

![Fig. 12a. Resulting magnetic field contour from paper [25].](image1)

![Fig. 12b. Resulting magnetic field contour from present paper (validation diagram).](image2)
\[ Q_0 = \int \int_{A_{h0}} u_{h0} \, dA_{h0}, \ldots Q_1 = \int \int_{A_{h1}} u_{i} \, dA_{i}, \]  
\hfill (15)

The rate at which the lubricant is lost due to side leakage is:
\[ Q_s = Q_0 - Q_1 \]  
\hfill (16)

According to the coordinate system of Fig. 1, the attitude angle is computed by
\[ \varphi = 90 - \tan^{-1}\left(\frac{e_x}{-e_y}\right) \]  
\hfill (17)

6. Results

The magnetic analysis provides the magnitude of the magnetic field within the volume of the bearing. The main characteristics of the magnetorheological journal bearing are the bearing radius \( R_b \), the radial clearance \( C \) and the length to diameter ratio and thus the length of the bearing \( L \) itself. The coil turns number and coil cable diameter have constant values throughout the coupled magnetic-CFD simulation. The rest of the geometric parameters are extracted in relation to the basic design variables. For comparison a set of exclusive magnetic simulations was executed in which the number of coil turns changes, altering the coil’s outer diameter and all other related dimensions.

In Fig. 13a, we can see the resulting magnetic field intensity in the volume of the bearing and in Fig. 13b the resulting field within the magnetorheological fluid volume, respectively. The density of the magnetic lines, within the magnetorheological fluid area, is constant. This indicates that the magnetic field in this area is homogeneous. The assumption that the viscosity is constant within the volume of the magnetorheological fluid is validated by these results.

Figs. 14a–14c depicts the relationship between the eccentricity ratio and the Sommerfeld number for a given current intensity and several \( L/D \) ratios. The results show the effect of the magnetic field in the load-carrying capability of the bearing. As an example and looking through Figs. 14a–14c, for an \( L/D = 1 \) and \( S_0 = 0.28 \) the value of eccentricity for \( I = 0 \) A is \( \varepsilon = 0.4 \), for \( I = 14 \) A is \( \varepsilon = 0.38 \) and for \( I = 28 \) A is \( \varepsilon = 0.35 \). So the observed decrement in eccentricity is 5% for \( L/D = 1, S_0 = 0.28 \) and going from \( I = 0 \) A to \( I = 14 \) A. From \( I = 0 \) A to 28 A the respective decrement is 12.5%. Also the respective decrement in eccentricity between \( I = 14 \) A and 28 A is 7.9%.

Friction coefficient in the journal is significantly increased by the presence of the magnetic field as shown in Figs. 15a–15c. The highest friction coefficient increase calculated was in the case of \( L/D = 1/4 \) and for the maximum current intensity

![Magnetic Field Flux Lines](image-url)
\( I = 28 \) A. The effect is adversely weaker in higher length to diameter ratio and especially in the case of \( L/D = 2 \) in which case the journal friction coefficient is increased by approximately 4\% for the current intensity increased from 0 to 28 A. Due to standard coil geometry considered, the effect of the magnetic field is lessened. Bearing geometry must be linked with coil geometry if we wish to have a uniform performance of the magnetorheological fluid bearing on different length to diameter ratios.

The increase of current intensity is affecting dramatically the journal friction coefficient. Thus for a length to diameter ratio of \( L/D = 1/4 \) the increase of friction coefficient is 31\% in the case of current intensity rise from 0 to 14 A and 34\% in the case of 14–28 A for a Sommerfeld number of \( S_o = 0.28 \). In the later case the friction coefficient appears to be increasing, slightly more than in the first case. This trend is constant in all the different length to diameter ratios and it is related to the non-linear behavior of the lubricant in use.
Fig. 14b. Eccentricity ($\varepsilon$) versus Sommerfeld number for $I = 14 \, \text{A}$ and $L/D = 1/4$, $L/D = 1/2$, $L/D = 1$, $L/D = 2$.

Fig. 14c. Eccentricity ($\varepsilon$) versus Sommerfeld number for $I = 28 \, \text{A}$ and $L/D = 1/4$, $L/D = 1/2$, $L/D = 1$, $L/D = 2$.

Fig. 15a. Normalized journal friction coefficient ($f_J$) versus Sommerfeld number for $I = 0 \, \text{A}$ and $L/D = 1/4$, $L/D = 1/2$, $L/D = 1$, $L/D = 2$. 
Fig. 15b. Normalized journal friction coefficient ($f_j$) versus Sommerfeld number for $I = 14\ A$ and $L/D = 1/4, L/D = 1/2, L/D = 1, L/D = 2$.

Fig. 15c. Normalized journal friction coefficient ($f_j$) versus Sommerfeld number for $I = 28\ A$ and $L/D = 1/4, L/D = 1/2, L/D = 1, L/D = 2$.

Fig. 16a. Normalized bearing friction coefficient ($f_b$) versus Sommerfeld number for $I = 0\ A$ and $L/D = 1/4, L/D = 1/2, L/D = 1, L/D = 2$. 

Fig. 16b. Normalized bearing friction coefficient ($f_b$) versus Sommerfeld number for $I = 14\,\text{A}$ and $L/D = 1/4, 1/2, 1, 2$.

Fig. 16c. Normalized bearing friction coefficient ($f_b$) versus Sommerfeld number for $I = 28\,\text{A}$ and $L/D = 1/4, 1/2, 1, 2$.

Fig. 17a. Dimensionless flow rate at maximum lubricant thickness versus Sommerfeld number for $I = 0\,\text{A}$ and $L/D = 1/4, 1/2, 1, 2$.
**Fig. 17b.** Dimensionless flow rate at maximum lubricant thickness versus Sommerfeld number for $I = 14$ A and $L/D = 1/4, 1/2, 1, 2$.

**Fig. 17c.** Dimensionless flow rate at maximum lubricant thickness versus Sommerfeld number for $I = 28$ A and $L/D = 1/4, 1/2, 1, 2$.

**Fig. 18a.** Ratio of side leakage flow over flow rate $Q_0$ versus Sommerfeld number for $I = 0$ A and $L/D = 1/4, 1/2, 1, 2$. 
Fig. 18b. Ratio of side leakage flow over flow rate $Q_0$ versus Sommerfeld number for $I = 14$ A and $L/D = 1/4, L/D = 1/2, L/D = 1, L/D = 2$.

Fig. 18c. Ratio of side leakage flow over flow rate $Q_0$ versus Sommerfeld number for $I = 28$ A and $L/D = 1/4, L/D = 1/2, L/D = 1, L/D = 2$.

Fig. 19a. Attitude angle versus Sommerfeld number for $I = 0$ A and $L/D = 1/4, L/D = 1/2, L/D = 1, L/D = 2$. 

Fig. 19b. Attitude angle versus Sommerfeld number for $I = 14$ A and $L/D = 1/4$, $L/D = 1/2$, $L/D = 1$, $L/D = 2$.

Fig. 19c. Attitude angle versus Sommerfeld number for $I = 28$ A and $L/D = 1/4$, $L/D = 1/2$, $L/D = 1$, $L/D = 2$.

Fig. 20. Variation of relative eccentricity versus current intensity rise for $S_o = 0.28$ and $L/D = 1/4$, $L/D = 1/2$, $L/D = 1$, $L/D = 2$. 
Fig. 21. Variation of bearing friction coefficient ($f_b$) versus current intensity rise for $S_o = 0.28$ and $L/D = 1/4$, $L/D = 1/2$, $L/D = 1$, $L/D = 2$.

Fig. 22. Variation of journal friction coefficient ($f_j$) versus current intensity rise for $S_o = 0.28$ and $L/D = 1/4$, $L/D = 1/2$, $L/D = 1$, $L/D = 2$.

Fig. 23. Variation of flow rate $Q_0$ versus current intensity rise for $S_o = 0.28$ and $L/D = 1/4$, $L/D = 1/2$, $L/D = 1$, $L/D = 2$. 
The bearing friction coefficient is strongly linked with the current intensity, and consequently, the magnitude of the magnetic field. This is shown in Figs. 16a–16c. The highest bearing friction coefficient is found at $L/D = 1/4$, for the maximum current intensity of $I = 28$ A. The overall increase of bearing friction coefficient in comparison with the case of $0$ A current intensity is $88\%$. The effect of the magnetic field is minimal in the case of $L/D = 2$, where the increase is found to be only $15\%$. As mentioned earlier the number of coil turns and the diameter of coil’s wire is considered constant in all the aforementioned cases. The increase of bearing friction coefficient when the current intensity increases from $0$ A to $14$ A is $28\%$ in the case of $L/D = 1/4$ and $0.9\%$ in the case of $L/D = 2$. The increase of bearing friction coefficient when the current intensity increases from $14$ A to $28$ A is $28\%$ in the case of $L/D = 1/4$ and $0.9\%$ in the case of $L/D = 2$.

The lubricant flow rate at maximum lubricant thickness versus the Sommerfeld number is depicted in Figs. 17a–17c. It is widely affected by changes in the magnetic field within the volume of the bearing and the consequent changes in lubricant viscosity. For the maximum current intensity increment of $0$–$28$ A the lubricant flow rate at the maximum thickness is $8\%$ in the case of $L/D = 1/4$ and a Sommerfeld number of $0.28$. Change in lubricant flow rate at maximum lubricant thickness is decreasing with the increase of the length to diameter ratio. In the case of $L/D = 2$ lubricant flow rate at maximum lubricant thickness is decreased by approximately $1\%$ when the current intensity is increased from $0$ to $28$ A.

Figs. 18a–18c depicts the relationship between the side leakage over maximum flow rate ratio and Sommerfeld number. The increase of current intensity generally decreases side leakage. In the case of a length to diameter ratio $L/D = 1/4$ the decrease is approximately $2\%$ when the current intensity is increasing from $0$ to $14$ A for a Sommerfeld number $S_o = 0.28$. The decrease side leakage over maximum flow rate ratio remains $2\%$ when the current intensity increases from $14$ to $28$ A. The overall decrement of side leakage over maximum flow rate ratio when the current intensity increases from $0$ to $28$ A is.

![Fig. 24. Variation of ratio $Q_s/Q_0$ versus current intensity rise for $S_o = 0.28$ and $L/D = 1/4, L/D = 1/2, L/D = 1, L/D = 2$.](image)

![Fig. 25. Variation of attitude angle versus current intensity rise for $S_o = 0.28$ and $L/D = 1/4, L/D = 1/2, L/D = 1, L/D = 2$.](image)
decreasing by 4.3%. The maximum side leakage over maximum flow rate ratio decrement when \( S_o = 0.28 \) is observed in the case of \( L/D = 1/2 \) with a decrement of 6%. This indicates that in this length to diameter ratio the side leakage is strongly affected by the magnetorheological phenomenon.

Figs. 19a–19c shows the effect of increased load-carrying capability to the attitude angle. In lower Sommerfeld number values the attitude angle obtains lower values. This change in the attitude angle depicts a tendency of the shaft to take an offset position in higher load conditions on the one hand and on the other the resulting increment of the attitude angle with the increase of current intensity is in agreement with the general notion that a stronger magnetic field results in greater load capacity of the bearing.

In Figs. 20–24, the effect of the magnetic field is becoming more apparent. For the different length to diameter ratios used in the simulation, the number of coil turns remains constant. This means that for a greater length to diameter ratio we obtain a weaker magnetic field and thus the effect on the magnetorheological fluid is significantly lessened. On the other hand a small length to diameter ratio is providing less volume for the magnetorheological fluid and although the magnetic field is stronger, the effect of the field to the bearing’s performance is limited by the lack of sufficient quantity of magnetorheological fluid.

Fig. 20 shows the effect of the current intensity rise on the relative eccentricity (\( e \)) for a given Sommerfeld number in all length to diameter ratios examined. In this case the smaller \( L/D \) ratio is providing the least load capacity rise. Figs. 21 and 22 show the friction coefficient increment with the application of a higher current intensity. The effect of the smaller volume in the case of a \( L/D = 1/4 \) is obvious. The friction coefficient is rising faster when the volume of the lubricant is minimum. Figs. 23 and 24 depict the drop of lubricant flow in the position of maximum lubricant thickness and the side leakage. In Figs. 23 and 24, the effect of the magnetic field to the ability of the lubricant to flow is depicted. In the case of the shortest bearing, the drop of the flow is considerably higher than in higher \( L/D \) ratios. This result is reasonable since the effect of the magnetic field is focused in a smaller fluid volume. The attitude angle is rising when the current intensity is rising as shown in Fig. 25. This simply put means that the load capacity is most significantly improved in lower \( L/D \) ratios. On the other hand the attitude angle of the journal is least affected by the rise of current intensity when \( L/D = 2 \).

The magnetic simulation gives an insight on the design requirements of the magnetorheological journal bearing. In Figs. 26 and 27, the line of 2000 A/m in the case of \( I = 14 \) A and the line of 4000 A/m in the case of \( I = 28 \) A show the increasing need for coil turns in order to achieve a relatively low value magnetic field intensity in longer bearings. The main priority is to keep the magnetic field as homogeneous as possible. A homogenous magnetic field in the lubricated area would result in homogenous physical properties and mechanical behavior throughout the length of the bearing. In case of non-uniform distribution of the magnetic field, the particle chains would be distributed in a non-uniform manner across the length of the bearing. This could cause uneven load distribution. Moreover, in case of wear creation due to the existence of magnetized particles could lead to excessive wear in portion of the bearing surface. Keeping this in mind, a homogenous magnetic field within the lubricated area is a desirable effect from a design and performance aspect.

This is successfully achieved by increasing the height of the coil core and the number of the coil turns for higher length to diameter ratios. This technique is obviously cost intensive and the resulting benefits of the magnetorheological fluid bearing may in this manner be undermined. There is certainly a perspective for a minimum cost coil and core design which will produce as much a homogeneous magnetic field as possible even in long bearings.

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**Fig. 26.** Correlation between coil’s number of turns and magnetic field intensity for \( L/D = 1/4 \), \( L/D = 1/2 \), \( L/D = 1 \), \( L/D = 2 \) and current intensity 14 A.
7. Conclusions

This paper has addressed the problem of magnetorheological fluid lubricated journal bearings operating. To this end, we developed a tool for solving the coupled magnetic-rheological flow problem.

For a selected number of bearing states, several \(L/D\) ratios, magnetic field variations, solutions were obtained in terms Sommerfeld number variation. The present results demonstrate that, in comparison to a normal bearing (lubrication without magnetic field), the presence of magnetic filed can be beneficial for the bearing characteristics such as increased load-carrying capacity in terms of the magnetic field increment, whereas the results regarding the friction coefficient lead to a less beneficial function under the influence of the magnetic field.

Thus, designing journal bearings under the excitement of the magnetic field should be realised taking provisions for the energy cost that occurs with the higher friction coefficient and the energy required to sustain the desired magnetic field. A mechanism or a method that could lessen the friction coefficient of the magnetorheological fluid, leaving the rest of its beneficial effects intact, could be a perspective for future research.

Thus, designing journal bearings under the application of the magnetic field we can conclude that for a rise of current intensity from 0 to 28 A:

- The Sommerfeld \((S_o)\) of the journal bearing decreases. The maximum decrement 36.39% is for \(L/D = 1/4\) and eccentricity \(\varepsilon = 0.8\) and the minimum decrement is 1.64% for \(L/D = 2\) and the same value of \(\varepsilon\).
- The attitude angle \((\varphi)\) increases by 37.4% for a \(L/D = 1/4\) \(\varepsilon = 0.8\) and by 0.7% for a \(L/D = 2\), \(\varepsilon = 0.8\), respectively.
- The friction coefficient \((f_b)\) on the bearing surface increases. The maximum increment 22.2% is for \(L/D = 1/4\) and eccentricity \(\varepsilon = 0.8\) and the minimum increment is 0.6% for \(L/D = 2\) and the same eccentricity.
- The friction coefficient on the journal surface \((f_j)\) increases, with the maximum increment (28%) to be for \(L/D = 1/4\) and eccentricity \(\varepsilon = 0.8\) and the minimum increment to be 0.2% for \(L/D = 2\) and \(\varepsilon = 0.8\).
- The fluid flow is decreasing by 5.46% for \(L/D = 1/4\) when \(\varepsilon = 0.8\). Flow decrement drops to 0.16% in the case of \(L/D = 2\) and the same \(\varepsilon\).

The appropriate meshing techniques described in Section 3.2, could be used for the simulation of the magnetorheological fluid bearing in high eccentricities.

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References

Journal Bearing Stiffness and Damping Coefficients Using Nanomagnetorheological Fluids and Stability Analysis

The integrity and reliability of a rotor depend significantly on the dynamic characteristics of its bearings. Bearing design has evolved in many ways in order to achieve higher damping and stiffness. A promising field in terms of vibrations control and overall performance improvement for the journal bearings is the use of smart lubricants. Smart lubricants are fluids with controllable properties. A suitable excitation, such as an electric or a magnetic field, is applied to the lubricant volume and changes its properties. Magnetorheological (MR) fluids consist one category of lubricants with controllable properties. Magnetic particles inside the MR fluid volume are coerced by a magnetic field. These particles form chains which hinder the flow of the base fluid and alter its apparent viscosity. According to the magnetic particle size, there are two subcategories of magnetorheological fluids: the regular MR fluids with particles sizing some tens of micrometers and the nanomagnetorheological (NMR) fluids with a particle size of a few nanometers. The change of magnetorheological fluid’s viscosity is an efficient way of control of the dynamic characteristics of the journal bearing system. In this work, the magnetic field intensity inside the volume of lubricant is calculated through finite element analysis. The calculated value of the magnetic field intensity is used to define the apparent viscosity of both the MR and the NMR fluids. Using computational fluid dynamics (CFD) method, the pressure developed inside the journal bearing is found. Through this simulation with the use of a suitable algorithm, the stiffness and damping coefficients are calculated and stability charts of Newtonian, MR, and NMR fluid are presented and discussed. [DOI: 10.1115/1.4027748]

Keywords: journal bearings, magnetorheological, nanomagnetorheological fluids, stiffness, damping properties, stability

1 Introduction

MR fluids are a category of fluids, the viscosity of which can be controlled by a suitable magnetic field. The physical mechanism through which properties control becomes possible is provided by a dispersion of paramagnetic particles. Under the influence of an externally applied magnetic field, the aforementioned particles form chains which hinder the flow of the carrier fluid, changing in this manner its apparent viscosity. This allows the use of these fluids in various applications, such as brakes and dampers [1,2], in which the control of their rheological behavior can be exploited. Wiltsie et al. [3] present a robot which uses a MR fluid as an adhesive matter which permits the robot to climb. Hesselbach and Abel-Keilhack [4] presented an application of the MR fluids in hydrostatic bearing, exploiting the beneficial effects of these controlled liquids on the bearing behavior.

Moving a step forward, NMR fluids have also been proposed for certain applications. Safarik et al. [5] studied the applicability of magnetic nanoparticles in biomedical applications, while Pijal-kowski [6] introduces a novel internal combustion engine, which uses a NMR mechatronic commutator as a replacement of the crankshaft and the connecting rod mechanisms. The particle size of the paramagnetic particles, inside the MR fluid volume, plays a significant role in the rheological behavior of the MR fluids and their physical properties. It is important to suggest how this parameter can contribute to the performance of this kind of fluids.

Zhang et al. [7] study the role of sedimentation after a long time of lack of motion on the effectiveness of a MR fluid damper for automotive applications. Chaudhuri et al. [8] presented the effects of substitution of micrometer-sized powder by nanometer-sized powder in MR fluids. They found that by mixing a reasonable percent of nano iron powder in the NMR fluid, a substantial change in the rheological characteristics is obtained. Vekas [9] examines the similarities and main differences between the MR and NMR fluids through various examples. Kim et al. [10] have conducted a comparative study on the effect of particle size to the properties of a MR fluid. More specifically, they compared the performance of a MR and a NMR fluid on apparent viscosity, yield stress, and other parameters. López-López et al. [11] investigated the interaction between micrometer-sized magnetizable particles dispersed in a ferrofluid upon application of a magnetic field. NMR fluids contain small particles \((d < \approx 30 \text{ nm})\) made of \(\text{Fe}_2\text{O}_3\). This results in a weaker MR effect than the one present in conventional MR fluids. On the other hand, the sedimentation stability of these magnetic fluids is higher [12].

The simulation of rotating machinery is always an active item for research. Li et al. [13] applied computational fluid dynamics coupled with rotordynamics simulation of a rotor. Chouksey et al. [14] performed modal analysis taking into account the fluid film forces and the rotor material damping. In Ref. [15], San Andres discusses the role of inertia in the calculation of rotodynamic force coefficients and their effect on the stability of a rotor–bearing system. Kirk et al. [16] evaluated the stability and transient response of a high-speed automotive turbocharger. They used various computer models with different bearing designs and properties to obtain the linear stability threshold speeds and also the

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nonlinear transient response, in the high speed automotive turbochargers.

Many researchers have also done work on the active stability control of rotor–bearing systems using smart lubricants. The damping characteristics of MR fluids have motivated research toward rotor dynamics. Forte et al. [17] studied the performance improvements induced by the use of a MR fluid damper in the case of rotating machinery. The effect of use of MR fluids in journal bearings is discussed in Ref. [18].

Using Navier–Stokes equations in the calculation of fluid film pressure on a journal bearing is a demanding computational task. On the other hand, Reynolds equation, the most widespread method of calculating the lubricant pressure, ignores the inertial phenomena. The inclusion of the inertial phenomena in the Navier–Stokes equations provides an accurate method of calculation of the journal bearing’s dynamic characteristics and their stability [19]. The rheological behavior of MR and NMR fluids can be simulated using the Bingham plastic model. Tichy [20] proposed a differential method with an iterative algorithm to study Bingham plastic flow in a journal bearing.

In this work, the effect of use of MR and NMR fluids in journal bearings is compared, using a combined finite element and computational fluid dynamics approach. The properties of an adequately strong magnetic field are calculated and in turn the static and dynamic characteristics of the journal bearing system are obtained. For the calculation of the dynamic properties of the bearing, the method of small perturbations has been modified in order to be applied correctly with the use of Navier–Stokes equations, especially in the case of the damping coefficients calculation. The different static and dynamic performance characteristics of MR, NMR, and Newtonian lubricant are compared for two length over diameter ratios and a range of loads. The relevant stability charts are also obtained.

2 Problem Formulation and Validation

2.1 Geometrical Model. In the present work, the bearing is considered to be rigid, and the flow steady and isothermal. The geometry of the bearing follows the model that is shown in Fig. 1; here, \( O_b \) is the bearing center, \( O_j \) is the journal center, \( R_b \) is the bearing radius, \( R_j \) is the journal radius, \( e \) is the bearing eccentricity, \( \phi \) is the attitude angle, and \( L \) is the bearing length. The external load \( W \) is assumed vertical (i.e., along the \( y \)-axis) and constant and a magnetic field \( H \) is applied between rotor and the bearing.

2.2 Governing Equations and Assumptions. The following assumptions for the bearing model are used in this work:

A rigid aligned bushing with the geometry of Fig. 1 is considered; a steady-state operation is assumed; the flow is laminar and isothermal; a constant external vertical load \( W \) is applied to the journal; the minimum pressure value is assumed to be above the vapor pressure; thus cavitation is not accounted for.

The viscosity of MR fluids can be approximated with the Bingham law for yield stress

\[
\tau = \tau_0 + \mu_f \dot{\gamma}
\]

(1)

where \( \tau \) is the shear stress of the material, \( \tau_0 \) is the critical shear stress or yield stress, and \( \dot{\gamma} \) is the shear rate. The relationship of the critical shear stress \( \tau_0 \) with the magnetic field intensity \( H \) can be estimated by experimental data. For certain MR fluids, this relationship is available through manufacturer’s literature [21]. It is possible to obtain an equivalent or apparent viscosity

\[
\mu_a = \mu_l + \frac{\tau_0(H)}{\dot{\gamma}} \\
\mu_f
\]

(2)

where \( \mu_a \) is the apparent viscosity of the material and \( \mu_f \) is the Newtonian viscosity of the material when the shear stress overcomes the yield stress, in which case the material is flowing. Typically, the plastic viscosity is defined with a value of \( \mu_f = 100 \mu_l \) in order to replicate properly the Bingham behavior. The yield stress of the MR fluid is considered constant throughout the fluid volume for the purposes of the simulation. This assumption implies a homogenous magnetic field.

The fundamental expression for the magnetic field is given by the Gauss law

\[
\nabla \cdot \mathbf{H} = 0
\]

(4)

The differential form for Amperes law including Maxwell’s correction is

\[
\nabla \times \mathbf{H} = \mu_0 \mathbf{J} + \mu_0 \varepsilon_0 \frac{\partial \mathbf{E}}{\partial t}
\]

(5)

where \( \mu_0 \) is the permeability of free space, \( \varepsilon_0 \) the free space permittivity, \( \mathbf{B} \) and \( \mathbf{H} \) becomes

\[
\mathbf{B} = \mu_0 \mathbf{H}
\]

(6)

where \( \mu_r \) is the relative magnetic permeability of the material in which the magnetic density vector \( \mathbf{B} \) is calculated.

Concerning the lubricant flow, the Navier–Stokes equations are used in the form of mass conservation
and the momentum conservation equations

\[ \frac{\partial \rho}{\partial t} \mathbf{v} + \nabla (\rho \mathbf{v}) = -\nabla p + \nabla \cdot (\bar{\tau}) + \rho \mathbf{g} + \mathbf{F} \]  

where \( \rho \mathbf{g} \) and \( \mathbf{F} \) are the gravitational and external body forces, respectively.

The stress tensor \( \bar{\tau} \) is given by

\[ \bar{\tau} = \mu \left[ \left( \nabla \mathbf{v} + (\nabla \mathbf{v})^T \right) - \frac{2}{3} \nabla \cdot \mathbf{v} \right] \]  

In this work, we solve Eqs. (7)–(9) with the code CFD FLORTRAN of the ANSYS MULTIPHYSICS suite. Also the Eqs. (4)–(6) were solved with the finite element method using ANSYS MULTIPHYSICS for the magnetostatic case.

Consequentially, concerning the MR and NMR fluid bearings, the boundary conditions was predicted using the set of Eqs. (1)–(9), linking the magnetic field intensity magnetic field intensity with the Bingham viscosity model.

2.3 Performance and Operational Characteristics Calculation. We calculate the sum of forces applied on the journal by integrating the pressure on the journal surface.

\[ F = \int_{A_j} p \cdot dA_j \]  

Frictional forces are calculated by integration of the shear stresses on the journal and bearing surfaces

\[ F_{fr} = \int_{A_j} \tau \cdot dA_j, \quad \text{where } i = j \text{ or } b \]  

We calculate the normalized friction coefficient as follows:

\[ f_i(R_j/C) = \frac{F_{fr}}{W}, \quad \text{where } i = j \text{ or } b \]  

where \( W \) is the load which is applied on the bearing.

We relate most of the performance parameters as well as the stiffness and damping coefficients with the Sommerfeld number

\[ S = \mu \cdot \omega \cdot R_j \cdot L\left( R_j/C \right)^2 / (\pi \cdot W) \]  

where \( \mu \) is the fluid viscosity, \( \omega \) is the angular velocity of the journal, \( R_j \) is the journal radius, \( L \) is the bearing length, and \( C \) is the radial clearance. Sommerfeld number as a dimensionless parameter serves as a criterion of comparison for many different configurations of journal bearings. Low values of Sommerfeld number would point to a bearing with high load capacity.

A rigid rotor is considered and supported by two identical journal bearings. The system is assumed to be completely symmetrical and has no imbalance. In hydrodynamic regime, an oil film is created between the journal and the bearing. Hydrodynamic film reactions are thus applied on the journal.

We calculate the critical rotational velocity as follows [22]:

\[ \omega_c^2 = \frac{K_{eq}}{C_{eq}} = \frac{K_{xx}C_{yy} + K_{yy}C_{xx} - K_{xy}C_{yx}}{C_{xx} + C_{yy}} \]  

\[ \frac{M_c}{C_{eq}} = \frac{(K_{eq} - K_{xx})(K_{eq} - K_{yy}) - K_{xy}K_{yx}}{C_{xx}C_{yy} - C_{xy}C_{yx}} \]  

With the bearing dynamic properties calculated, as a function of Sommerfeld number and the applied external magnetic field, for the cases of Newtonian, MR, and NMR fluids lubrication, the equivalent stiffness \( K_{eq} \) can be computed, and the \( \omega_c \) can be also computed using the Eq. (14). Further, the critical mass \( M_c \) can be calculated as \( M_c = K_{eq}/\omega_c^2 \). The stability charts can be easily presented and comparisons between Newtonian, MR, and NMR lubrication can be achieved.

2.4 Boundary Conditions and Validation of the CFD Model. The CFD model employed in this work is isothermal and the flow is considered to be under steady state. The boundary conditions involve the bushing, the journal, and the lateral surfaces of the bearing, as shown in Fig. 2. The bearing is defined as a stationary wall with the velocity having zero velocity vectors in all directions. The journal is considered as a moving wall, in which there is only the tangential component of the rotational velocity. Negative pressures are set to zero in order to account for cavitation. The pressure at the sides of the bearing is set equal to zero, functioning as a free flow boundary.

The model of the current work has been validated toward the work of Gertzos et al. [23]. In Fig. 3, we compare the results of the current work concerning relative eccentricity with those obtained in Ref. [23] for a bearing with non-Newtonian lubricant in a range of Sommerfeld number values. The results concern a bearing with \( R_b = 49.999 \text{ mm} \) and radial clearance \( C = 61 \text{ \mu m} \) with journal rotating at 1000 rpm.

2.5 Magnetostatic Simulation Boundary Conditions. In the magnetic part of simulation, the main load is the current source density. Infinite space as a boundary condition lies on the outer limit of the simulated area and has zero magnetic potential. In other words, we neglect any influence of the magnetic field outside the simulated space. As discussed in Ref. [24], load capacity depends on the magnetic field distribution inside the bearing. The relative boundary conditions for the magnetic simulation problem are shown in Fig. 4.

2.6 Fluids Properties. We use three types of lubricant in the present work:

(a) A typical Newtonian lubricant with dynamic viscosity of \( 1.110 \times 10^{-2} \text{ Pa·s} \) at \( 40^\circ\text{C} \) and density of \( 890 \text{ kg/m}^3 \).

(b) A typical MR fluid with viscosity is \( 0.112 \pm 0.02 \text{ Pa·s} \) at \( 40^\circ\text{C} \), with density of \( 2.95–3.15 \text{ g/cm}^3 \) and the solids content is 80.98% by weight, and the operating temperature varies from \(-40^\circ\text{C}\) to \(130^\circ\text{C}\). With the magnetic field calculated in this paper, the yield stress of the MR Fluid is \( 25 \text{ kPa} \).
2.7 Damping and Stiffness Calculation. The stiffness and damping properties of a journal bearing can be treated in a manner analogous to a spring and viscous damper from simple vibration theory. The journal bearing system with its characteristics approximating its dynamic behavior is shown in Fig. 5.

2.7.1 Disturbance and Influence on Navier–Stokes Equations. The damping coefficients are calculated with the disturbance of the journal in the x and y directions separately. Through the disturbance of the journal, the dynamic coefficients can be obtained by the following equations:

\[ C_{ij} = \frac{\partial F_{ij}}{\partial h_{ij}} \]  

where \( i, j = x \) or \( y \).

The size of disturbance is a major issue when the Navier–Stokes equations are used. The logical choice would be to use a very small disturbance in order to achieve calculation of the dynamic coefficients with as minimum variation of journal translational speed as possible, thus, calculating the derivative in the linear region. The use of such a disturbance yields very high values of the damping coefficients for all relative axes. This contradicts with the experimental results and the theoretically calculated values of the damping coefficient with the use of Reynolds equations. Thus, there is a factor inside the formulation of the Navier–Stokes equation that makes the difference between the two methods. The difference which occurs between the two methods has to be traced in the equations themselves. The continuity equation is expressed as

\[ \nabla \mathbf{V} = 0 \]  

and the momentum equation as

\[ \frac{\partial \mathbf{V}}{\partial t} + \mathbf{V} \nabla \mathbf{V} = -\frac{1}{\rho} \nabla P + \mu \nabla^2 \mathbf{V} \]  

where \( \mathbf{V} \) is the vector of fluid velocity, \( P \) is fluid pressure, \( \mu \) is the fluid viscosity, and \( \rho \) is the fluid density. Reynolds equation takes the form

\[ \frac{\partial}{\partial h} \left( \frac{h}{L} \frac{\partial P}{\partial h} \right) + L \left( \frac{h}{L} \frac{\partial P}{\partial h} \right)^2 \frac{\partial}{\partial z} \left( \frac{h}{L} \frac{\partial P}{\partial h} \right) = 6\mu L \left[ \frac{\omega}{\rho} \frac{\partial h}{\partial h} \right] + \frac{\partial h}{\partial h} \]  

where \( \theta \) is the circumferential coordinate of the bearing, \( h \) is the lubricant thickness, \( \omega \) is the rotational velocity, \( L \) is the bearing length, and \( R \) is the journal radius.

It is evident that Reynolds equation does not include velocity. Furthermore, velocity in the case of momentum equation is related to its gradient. This means that when the disturbance is applied on the shaft, the velocity terms are changing nonlinearly as follows:

Additionally, in the case of Reynolds equation, \( \partial P/\partial h = 0 \).

This is not the case for the Navier–Stokes equations. Thus, the two methods could have a more similar behavior in the range of higher slenderness ratio values.

Momentum equation for y-axis can be expressed as

\[ \rho \left( \frac{\partial v}{\partial h} + u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z} \right) = -\frac{\partial P}{\partial y} + \rho \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right) + \rho \cdot g \]  

where \( \rho \) is the lubricant density and \( u, v, \) and \( w \) are the three velocity components for \( x, y, \) and \( z \)-axes, respectively. In order to calculate the damping coefficients, we apply a small disturbance on the shaft motion. Let us assume that an impact on the shaft accelerates it instantaneously on the y direction.

Since the velocity field is uniformly disturbed throughout the \( x \)- and \( z \)-axes expression (19) without loss of generality can be written as
Multiplying with the finite time duration yields

\[ \rho \left( \frac{(v + v_0)}{\partial t} + (v + v_0) \frac{\partial(v + v_0)}{\partial y} \right) = -\frac{\partial P}{\partial y} + \mu \left( \frac{\partial^2(v + v_0)}{\partial y^2} \right) + \rho \cdot g \tag{20} \]

If we assume a minimal acceleration for the disturbance occurrence, steady state conditions could be assumed and the expression becomes

\[ \rho \left( v + v_0 \right) \frac{\partial(v + v_0)}{\partial y} = -\frac{\partial P}{\partial y} + \mu \left( \frac{\partial^2(v + v_0)}{\partial y^2} \right) + \rho \cdot g \tag{21} \]

Dividing by the projected area of the bearing \( A \) yields

\[ \rho \left( v + v_0 \right) \frac{A \partial(v + v_0)}{\partial y} = -A \frac{\partial P}{\partial y} + A \mu \left( \frac{\partial^2(v + v_0)}{\partial y^2} \right) + \rho \cdot g \cdot A \tag{22} \]

Multiplying with the finite time duration yields

\[ \rho \left( v + v_0 \right) \frac{A \cdot \partial(v + v_0)}{\partial y} = -A \frac{\partial P}{\partial y} + \left( \frac{A \cdot \partial^2(v + v_0)}{\partial y^2} \right) + A \cdot \rho \cdot g \cdot \partial t \tag{23} \]

By substituting \( v + v_0 = \frac{\partial y}{\partial t} \), the relationship between pressure disturbance and pressure becomes

\[ \rho \left( A \cdot \partial(v + v_0) \right) = -\frac{A \cdot \partial P}{\partial y} + \left( \frac{A \cdot \partial^2(v + v_0)}{\partial y^2} \right) + A \cdot \rho \cdot g \cdot \partial t \tag{24} \]

In other words, the damping coefficient is equal to

\[ \frac{A \cdot \partial P}{\partial v} = -v_p \left( \frac{A \cdot \partial(v + v_0)}{\partial y} \right) + \mu v \left( \frac{A \cdot \partial^2(v + v_0)}{\partial y \cdot \partial y^2} \right) + A \cdot \rho \cdot g \cdot v \cdot \frac{\partial t}{\partial y} \tag{25} \]

It is evident that the apparent damping coefficient is a function of velocity and shaft’s linear acceleration. The purpose of the analysis in the \( y \) direction is not to exclude what happens in the \( x \) direction but rather to establish that, without loss of generality (i.e., in both axes of movement), the velocity disturbance has a widely different effect on Navier–Stokes equations than on Reynolds. Reynolds equation does not introduce velocity and the pressure gradient does not change over \( z \) direction. In other words, if the movement of the shaft causes a disturbance that is not dismissible in comparison to the journal velocity, Navier–Stokes equations treat this disturbance differently than the Reynolds equation does. The disturbance used in order to calculate the damping coefficient should approximate the velocity and acceleration of the journal while on its stable orbit. The problem, finally, is to determine the proper disturbance which would permit the correct calculation of the stiffness and damping coefficients. In this work, the disturbance of the shaft position and velocity was selected by comparison with the work of Glienicke et al. [25]. More specifically, the proper disturbances were tested toward the stiffness and damping coefficients considering a bearing with \( L/D = 0.5 \). In Fig. 6, the damping coefficients obtained are compared with those of Ref. [25] for \( L/D = 0.5 \).

The disturbances that have been used do not vary linearly and they are not the same for every term and direction. In Fig. 7, the disturbances are presented for a range of Sommerfeld number values.

The higher disturbance values are used for the coupled \( C_{xy} \), while the coupled term \( C_{yx} \) and the main term \( C_{xx} \) are calculated with the lower values of disturbance. The disturbance is additionally a function of load. For the disturbance regarding the coupled \( C_{xy} \) term, the maximum value is required at \( S = 0.4 \). The same trend appears for the disturbance used in the calculation of \( C_{xx} \) term, with the maximum value required in the region of \( S = 0.4 \) as well. For the \( C_{xx} \) and \( C_{xy} \) terms, the trend is opposite with a minimum value of disturbance required at \( S = 0.26 \).

### 3 Results

#### 3.1 Grid Sensitivity and Convergence Criteria

Since the disturbances we use to calculate the values of stiffness and damping coefficient are small, the accuracy of prediction of the hydrodynamic and friction forces is important. The error of the pressure CFD solution is \( 1 \times 10^{-6} \), which is adequate in terms of accuracy for the load prediction. This residual error is acceptably low in comparison to the relative literature [26]. Additionally, the high number of elements used satisfies the demand for results irrelevant of the mesh used in the calculations. In the present paper, there are 450 elements in the circumferential direction, 10 elements in the radial direction, and 15 elements in the longitudinal direction. The aforementioned configuration also corresponds...
with the standards set by literature for the minimum mesh requirements [23,27].

3.2 Magnetic Field Simulation. An adequate magnetic field is a prerequisite for the proper employment of MR and NMR fluids. The main goal is the achievement of a high yield stress. This goal has to be compromised along with the available coil volume, the available coil cable overall length, and finally the cost of the overall design. Without having in mind any cost reservations, a suitable coil that would induce magnetic field intensity high enough to produce yield stress of 25 kPa in the MR fluid is shown in Fig. 8 for $L/D = 0.5$. One can observe that although the field in our case is not entirely homogenous, there is an adequate distribution within all the volume of lubricant inside the bearing.

3.3 Static Performance. The high yield stress values of the MR fluid result in higher load capacity of the journal bearing. In Fig. 9, relative eccentricity is depicted for a range of Sommerfeld number values, with a slenderness ratio of 0.5. The MR and NMR fluids are excited with 300 A.

In Fig. 9, we observe wide enhancement in terms of load capacity. The bearing using MR fluid exhibits the lower relative eccentricity values, while in the case of Newtonian and NMR fluids the relative eccentricity has no significant difference. The benefits of the MR fluid use are higher for $S = 0.7928$ with a 88.31% lower relative eccentricity in comparison to the cases of Newtonian and NMR fluids.

3.4 Dynamic Characteristics. The dynamic characteristics of the journal bearing are a significant aspect of the overall performance of the journal bearing. The high viscosity exhibited by the NMR fluid yields a significant damping capability along with increased stiffness of the journal bearing. In this section, the dimensional values of the stiffness and damping properties are presented in detail, in order to have an absolute comparison between the Newtonian, MR, and NMR fluids.

3.4.1 Stiffness Coefficients Validation. The capability of the developed code to accurately predict the dynamic characteristics of the journal bearing has been validated toward the work of Glienicke et al. [25]. The stiffness coefficients, in the case of $L/D = 0.5$, are depicted in Fig. 10.

The bearing stiffness coefficients are identical with those presented in Ref. [25].

3.4.2 Stiffness Coefficients Comparison. Stiffness coefficients are a significant measure of the dynamic behavior of the journal bearing system. The resulting stiffness of the MR fluid for a range of loads is depicted in Fig. 13. Concerning the main terms of stiffness, the MR fluid yields higher values of the $K_{yy}$ term comparison to the Newtonian lubricant (see Figs. 10 and 11). Most pronounced is the increase of stiffness at higher Sommerfeld number values. For $S = 0.792$, the $K_{yy}$ term increases almost six times. On the contrary, $K_{xx}$ term is adversely affected by MR fluid use. For instance, the $K_{xx}$ is 41.8% lower for $S = 0.792$ when MR fluid is used. $K_{xy}$ term is positively affected by the MR fluid with most significant benefits in the region of lower loads. For $S = 0.796$, the use of Newtonian lubricant results in a 4.6 times increase of the $K_{xy}$ term. $K_{xx}$ term decreases 80 times for a Sommerfeld number value of $S = 0.198$.

Using the high viscosity, NMR fluid yields even higher stiffness in comparison to the MR fluid, as shown in Figs. 11 and 12, where stiffness of the NMR fluid is presented for the same range of Sommerfeld number values and the same overall geometry of the bearing. The main terms of stiffness are higher in the case of NMR fluid.
fluid for all Sommerfeld number values considered. The same applies for the $K_{yx}$ term. $K_{xy}$ term as an absolute value is higher in the case of NMR fluid use.

Comparing Figs. 10–12, the higher stiffness of the NMR fluid in the given range of Sommerfeld number values is evident. This trend is especially intense in the case of higher loads with the $K_{yy}$ term being two orders of magnitude higher than that of the MR fluid. This higher stiffness though is related with the much higher eccentricity of the journal in the case of NMR fluid and with the fact that Sommerfeld number is a function of apparent viscosity.

3.4.3 Damping Coefficients Validation. We compare the damping coefficients for a bearing using Newtonian lubricant with those presented in Ref. [25], with which they show very good agreement. In Fig. 13, the damping coefficients are presented.

3.4.4 Damping Coefficients Comparison. MR fluid alters significantly the available damping of the journal bearing. Figure 14 depicts the damping coefficients for a bearing with $L/D = 0.5$ using MR fluid. The higher values of damping are to be expected since in its active state, MR fluid has increased viscosity. With the only exception of $C_{xy}$ for $S = 0.158$ which is slightly decreased by a 10.8% in the case of the MR fluid (see Figs. 13 and 14), the overall trend is a significant increase of the damping coefficients, especially on higher Sommerfeld number values. For instance, $C_{yy}$ increases by a $1.72 \times 10^4\%$, thanks to the high yield stress of the MR fluid. The same general tendency applies for all other terms.

On the other hand, the viscosity of the NMR fluid is also high in comparison to the Newtonian lubricant. This results in high values of damping coefficients when this type of fluid is used. Figure 15 shows damping coefficients for the bearing using NMR fluid. Similarly with the results obtained for the MR fluid, benefits to the damping coefficients in comparison to the case of Newtonian fluid are more pronounced in the region of lower Sommerfeld
number values. As an example of the magnitude of change, $C_{xy}$ term increases by $1.038 \times 10^3\%$ for $S = 0.264$ (see Figs. 13 and 15). Comparing Figs. 14 and 15 also yields some interesting results.

While $C_{xy}$ term is lower in the case of the NMR fluid, the other terms increase. For example, the $C_{yx}$ term increases by $1.85 \times 10^3\%$ for $S = 0.792$ in the case of NMR fluid while $C_{xy}$ drops by a $9.18 \times 10^2\%$ for $S = 0.396$.

3.5 Stability. The results obtained show that the NMR fluids can benefit the stability offered by a journal bearing. This can be more intuitively depicted in Fig. 16 where the product of dimensionless stability of the system for all the range of Sommerfeld number values concerned. The marked region in Fig. 16 offers additional stable operating conditions in comparison to the Newtonian lubricant and it is the main benefit from the use of NMR fluid in the lubrication of the journal bearing.

4 Conclusions

MR and NMR fluids can alter their properties under the influence of a magnetic excitation. The properties of these fluids end up to significantly alter the dynamic behavior of the journal bearing. This changes in their rheological behavior increase stiffness and damping, making them suitable candidates for bearing technological applications where sudden fluctuations of load can be significant and adaptiveness of the bearing is a prerequisite.

The use of MR fluid with its high yield strength produced a very high rise of the stiffness coefficients (e.g., for $S = 0.792$ the $K_{yy}$ term was increased almost six times). The damping of the bearing was also higher when the MR fluid was used in its active state. In the case of NMR fluid, the stiffness coefficient also rises as a result of its superior apparent viscosity. The most pronounced advantage of the NMR though is its high damping properties, which were the highest of the three lubricants used in this study. For instance, the $C_{yy}$ term increases by $1.85E3\%$ for $S = 0.792$ in the case of NMR, in comparison to the MR fluid. The effect of the NMR fluid on stability is also significant. The threshold of stability is widely enhanced.

Despite the beneficial effects of the MR and NMR fluids, on the dynamic behavior and stability of a journal bearing system, phenomena such as wear on moving parts, agglomeration of particles, and high friction losses may be the focus of future research. Additionally, these effects are dependent on the specific application.

The MR fluids can increase the stiffness and damping of the journal bearing. In their off state their rheological behavior approaches this of a Newtonian fluid making it a suitable lubricant when both low friction and increased stability are required. On the other hand, NMR fluids can benefit the journal bearing with even higher damping and stiffness in comparison to MR and Newtonian fluids. Their only weakness is their controllability since in their off state they do not change their properties significantly. The control of the concentration of the particles inside the lubricant could be an interesting field for future research. The chemical composition and the molecular geometry of the particles of these fluids may yield new fluids with better characteristics in the future. The experimental validation of the simulation results will qualify the method followed in this paper. The magnetic simulation can also be improved through the use of a three-dimensional model.

Nomenclature

- $B =$ magnetic field density
- $C =$ radial clearance $C = R_b - R_j$
- $C_s =$ $C \cdot \frac{c}{W}$ = dimensionless damping
- $K_{xx}, K_{xy}, K_{yx}, K_{yy} =$ stiffness properties
- $L =$ bearing length (m)
- $N =$ rotational speed of journal (rpm)
- $p =$ pressure (Pa)
- $p_{\text{max}} =$ maximum pressure on the journal surface (Pa)
- $R_j, R_b =$ journal and bearing radii (m)
- $S =$ Sommerfeld number
- $S = \frac{\mu_t \cdot \omega \cdot R_j \cdot L \cdot (R_j/C)^2}{(\pi \cdot W)}$
- $T_0 =$ dimensionless yield stress
- $T_0 = \frac{\tau_0 \cdot C}{(\mu_t \cdot \omega \cdot L)}$
- $U =$ journal velocity, parallel to the film
- $u, v, w =$ fluid velocity vectors
- $W =$ external force (N)
- $W^* =$ dimensionless load carrying capacity $W^* = W / (\mu_t \cdot c^2 / (\mu_t \cdot \omega^2 \cdot L))$
- $\mu_t =$ apparent viscosity $\mu_t = \mu_t + \tau_0 (H) / |\partial u / \partial y |$
- $\mu_t =$ Newtonian viscosity
- $\omega =$ rotational velocity (rad/s)
- $\tau_0 =$ yield stress (Pa)
- $\tau =$ stress tensor

Subscripts

- $0 =$ indicates the position of maximum film thickness at $\theta = 0$
- $1 =$ indicates the position of minimum film thickness at $\theta = \pi$
- $b =$ indicates the bearing
- $j =$ indicates the journal

References


Temperature Influence on the Behavior of a Magnetorheological Fluid Journal Bearing

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Abstract  Active control of vibrations is an important capability of bearings using magnetorheological fluids as lubricants. Magnetorheological fluids are a suspension of micron sized iron particles in a carrier fluid, usually a mineral oil. These particles, polarized under the influence of a magnetic field, form chains inside the lubricant volume, they hinder the flow of the fluid, change its apparent viscosity and offer active control on the available damping of the bearing. Magnetorheological fluid’s apparent viscosity relies heavily on the viscosity of the base fluid. The base fluid’s viscosity depends on temperature. On the other hand the existence of particles is a factor that may influence the lubricant’s temperature. In this paper the dynamic characteristics of a journal bearing lubricated with magnetorheological fluids are investigated for a range of temperature and load conditions.

Keywords  Magnetorheological fluid · Journal bearings · Simulation

1 Introduction

Rotating machinery support is a crucial issue for a number of applications. The use of hydrodynamic journal bearings offers the advantages of high damping and low friction at high rotating velocities of the rotor. The calculation of the dynamic characteristics of journal bearings using Reynolds equations is presented in [1]. Gertzos et al. [2] make use of Navier Stokes equations in the calculation of the static performance of journal bearings. The static performance characteristics of bearings using magnetorheological fluids have been studied by several researchers [3–5]. Since temperature may affect significantly the viscosity of the carrier lubricant in magnetorheological fluids, it is also evident that the performance of bearings using
these fluids will also be affected. Ghaednia et al. [6] study the performance of magnetorheological fluid journal bearings under the influence of temperature. In most cases of simulation, the behavior of the magnetorheological fluid is approximated with a non-linear viscosity model. In this work the aforementioned technique is compared with results obtained when the dispersed particles and the lubricant of the magnetorheological fluid are simulated separately and the macroscopic properties of the fluid are not approximated with a particular viscosity model but rather with the simulation of the flow in microscopic scale. The calculation of the macroscopic properties based on these simulation results and the estimation of performance characteristics, such as pressure distribution, shear stresses, and as a result the apparent viscosity which affects the dynamic properties of a journal bearing, are strongly influenced by this information.

2 Theory

In this simulation of the magnetorheological fluids, the rheological behavior is calculated by an iterative procedure (Fig. 1):

The physical properties of the particles and the lubricant including viscosity and density along are introduced. The particles size (d = 10e-6 m) and the control volume dimensions are defined. The magnetic forces applied on the particles by an external homogenous magnetic field are calculated. The fluid forces applied on the particles by the lubricant are calculated in the next step through a CFD simulation. The kinematic state of the particles is computed then taking as a given the initial position and velocity of the lubricants, their size and mass and the forces applied on them. The final result occurs after a number of iterations during which the apparent viscosity is calculated. When the apparent viscosity value presents no significant change between two consecutive iterations, the calculation is considered complete and the simulation ends.

As shown in Fig. 2, the size of the control volume was elected to be 180 μm in length, 100 μm in height and 100 μm in width. In this scale the curvature of a radial journal bearing can be omitted. In Fig. 2, the boundary conditions used for the solution are also presented. A homogenous magnetic field of 0.6 T is applied on the lubricant volume. The side of the bushing is considered a steady wall with no fluid velocity. The side of the bushing is considered a moving wall with the linear velocity of the journal. The rest of the areas of the control volume allow for the free flow of the lubricant. The particles are considered moving objects to which the hydrodynamic and magnetic forces are applied. Results have been calculated for two cases of rotational velocity of the journal at 2000 and 1500 rpm’s. The shaft has a radius $R_j = 25$ mm and the slenderness ratio of the bearing is $L/D = 0.5$. 
During the magnetic simulation the magnetic field is calculated according to the Amperes law:

$$\nabla \times \vec{H} = \mu_0 \vec{J} + \mu_0 \varepsilon_0 \frac{\partial \vec{E}}{\partial t}$$

(1)

where $\mu_0$ is the permeability of free space, $\varepsilon_0$ the free space permittivity, $\vec{B}$ and $\vec{H}$ becomes:

$$\vec{B} = \mu_0 \mu_r \vec{H}$$

(2)

The Maxwell magnetic forces are calculated as follows:

$$\{F_{mx}\} = \frac{1}{\mu_0} \int s \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \begin{bmatrix} n_1 \\ n_2 \end{bmatrix} ds$$

(3)
where $F_{mx}$ are the Maxwell forces, $T_{11} = B_x^2 - \frac{1}{2}|B|^2$, $T_{12} = B_xB_y$, $T_{21} = B_xB_y$, $T_{22} = B_y^2 - \frac{1}{2}|B|^2$ and $n_1, n_2$ the unit surface normal in the global Cartesian coordinate system.

The Navier-Stokes equations are employed in order to calculate the interaction between the lubricant and the particles as well as to calculate the developing shear stress and the apparent viscosity of the MR fluid. The mass conservation and the momentum conservation equations are described as follows:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \bar{v}) = 0 \quad (4)$$

$$\frac{\partial}{\partial t} (\rho \bar{v}) + \nabla (\rho \bar{v} \bar{v}) = -\nabla p + \nabla \cdot (\bar{\tau}) + \rho \bar{g} + \bar{F} \quad (5)$$

The lubricant viscosity is considered a function of temperature. In this simulation physical properties of SAE30 are used. The problem is solved as a quasi static temperature problem. Viscosity values are considered for several temperature values in order to obtain solutions.

The Verlet integration method is used in order to calculate the movement of the particles inside the lubricant volume. The time step is defined as 1 ns. The position of the particle is calculated as follows:
\[ \{x(t + 1)\} = \{x(t)\} + \{v(t)\} \Delta t + \frac{1}{2} \{a(t)\} \Delta t^2 \] (6)

The particle velocity is computed as follows:

\[ \{v(t + 1)\} = \{v(t)\} + \{a(t)\} \Delta t \] (7)

The dynamic behavior of the journal bearing system can be evaluated if the bearing is approximated by a system of springs and dampers. The dynamic behavior of the journal bearing system can then be expressed as:

\[ [M] \cdot [\ddot{x}] + [C] \cdot [\dot{x}] + [K] \cdot [x] = [F] \] (8)

where \([M]\) is the matrix expression of journal’s inertia, \([C]\) is the damping matrix and \([K]\) is the stiffness matrix of the journal bearing. If we apply a small disturbance on journal position and a small force in each direction we can calculate the stiffness and damping coefficients:

\[ K_{ij} = \frac{F_{ij}}{x_{ij}} \] (9)

\[ C_{ij} = \frac{F_{ij}}{\dot{x}_{ij}} \] (10)

In order to attain solution regarding the flow problem the convergence criterion of pressure change between two consecutive iterations is satisfied when \(\Delta p < 5\%\), while for the magnetic field problem the material properties are considered linear. Finally the Verlet algorithm is applied and the velocities of the solid particles change according to the forces applied to each one of them. All the above described algorithms are correlated using the flow chart presented in Fig. 1. Overall the model employs 526,885 elements.

3 Results

A constant homogenous magnetic field is applied on the lubricant volume. The magnetic field density reaches the 0.6 T. There are eight particles inside the control volume, corresponding to a magnetorheological fluid with a 16.32 % per weight content of iron particles. A first conclusion on the validity of non-linear viscosity models, concerning the approximation of the rheological behavior of a magnetorheological lubricant, can be drawn from the pressure distribution of the lubricant flow when the particles are present.
Since the journal and the bearing surfaces are parallel in this model, the pressure distribution is the result of the particles effect on the lubricant flow. A significant amount of pressure is then developed in the journal and bearing surfaces, as seen in Fig. 3.

Since a certain amount of pressure is applied on the journal and the bearing surfaces, it is evident that shear stress will also be applied. In Fig. 4, the shear stress developed on the bearing surface is shown. The areas with the highest values of

**Fig. 3** Pressure distribution on the bearing surface for a journal velocity of $v = 5.23$ m/s at a temperature of $49.36^\circ$C

**Fig. 4** Shear stress distribution on the bearing surface at a temperature of $49.36^\circ$C
shear stress are located on the sides of the area where a particle is located. The exact same pattern appears on the velocity distribution of the lubricant, on the bearing surface, as shown in Fig. 5. In contrast with the smooth transition of the shear stress values when a non-linear viscosity model is used, this figure shows that the distribution of both pressure and shear stress in a magnetorheological fluid exhibits variations in the scale of the present simulation.

The maximum pressure on the bearing surface is related to temperature as shown in Fig. 6. As expected, the maximum pressure drops with the rise of temperature, since the carrier fluid’s viscosity is lower at higher temperatures. Additionally for

![Image](image_url)

**Fig. 5** The fluid velocity distribution on the bearing surface

![Image](image_url)

**Fig. 6** The maximum pressure on the bearing surface for two cases of journal velocity with a magnetic field intensity of 0.6 T
higher journal speeds, the maximum pressure on the bearing surface increases. With a drop of journal velocity of 25\%, the maximum pressure decreases by 26.89\% for a temperature of 49 °C. In higher temperature the drop is more significant reaching the 35.85\% at 81 °C.

The mean shear stress generated by the lubricant flow in the direction of motion is presented in Fig. 7. The drop of the carrier fluid’s viscosity causes the decreased shear stress developed on the bearing surface. The mean value of shear stress in the direction of motion is directly connected with the amount of friction on the bearing surface. The drop of viscosity due to temperature causes a drop of the shear stress and thus friction. With lubricant temperature at 49 °C the drop of shear stress reaches the 25.14\%. For higher temperatures the drop of shear stress is increased. The drop of shear stress reaches the 27.81\% for a lubricant temperature of 81 °C.

The stiffness and damping coefficients of a journal bearing rely significantly on the viscosity of the lubricant. Our code regarding the bearing dynamic properties have been validated with the values given by Glienicke et al. in [1]. The stiffness and damping coefficients increase steadily with increasing temperature and a steady load for all cases concerned. The small perturbation techniques have been used in order to estimate the dynamic properties, as a function of the predicted apparent viscosity. Here, when we calculate the shear stresses through the aforementioned couple (hydrodynamic plus magnetic and particle dynamics) field problems, the apparent viscosity is accurately predicted. This means that the predictions of the rheological properties of the magnetorheological fluids from the micro to macro scale can be predicted more accurate.

In Fig. 8 the stiffness and damping coefficients are presented as ratio of their values in the case of temperature of 81 °C, in order to describe the change of the dynamic coefficients over temperature. This comparison shows that the bearing stiffness and damping for the same load increases with temperature at the cost of decreased minimum film thickness. Most pronounced is the increase of the $K_{yy}$
term, which reaches the 88 % between 49 and 81 °C. The reference values of the stiffness and damping properties at 81 °C are for 1500 rpm’s $K_{xx} = 4.42\times10^5$ N/m, $K_{yy} = 1.14\times10^6$ N/m, $C_{xx} = 6.82\times10^7$ N ∙ s/m, $C_{yy} = 3.04\times10^8$ N ∙ s/m and for 2000 rpm’s, $K_{xx} = 5.84\times10^5$ N/m, $K_{yy} = 1.75\times10^6$ N/m, $C_{xx} = 8.31\times10^7$ N ∙ s/m, $C_{yy} = 4.14\times10^8$ N ∙ s/m. The same trend applies in the case of 2000 rpm’s as well. The effect of temperature on the stiffness and damping coefficients for a rotational velocity of 2000 rpm’s is shown in Fig. 9. As seen in the previous case of 1500 rpm’s this increase of the stiffness and damping coefficients comes with the price of decrease of the minimum film thickness.

Again the stiffness coefficient $k_{yy}$ is the one term that presents the highest increase with the temperature. All the above results for the dynamic properties have been derived for an external load of 34.7 N.

**Fig. 8** The increase of stiffness and damping coefficients for the case of 1500 rpm’s with L/D = 0.5. All coefficients are presented as percentage of the values in 81 °C

**Fig. 9** The increase of stiffness and damping coefficients for the case of 2000 rpm’s with L/D = 0.5. All coefficients are presented as percentage of the values in 81 °C
4 Conclusions

The existence of particles inside the lubricant volume causes a rise in pressure which is significant and could affect significantly the cavitation which appears in the lubricant. Additionally the non-linear viscosity models remain suitable for the description of the frictional forces acting in bearings lubricated with magnetorheological fluids but they tend to ignore the pressure rise developed in the lubricant and thus the overall performance of such fluids. Temperature is a factor which plays a significant role on the physical properties of magnetorheological fluids. The drop in the predicted pressure on the bearing between 49 and 81 °C reaches the 70.58 %, while the decrease of friction reaches the 78 % for a journal velocity of 5.24 m/s. The stiffness and damping coefficients are also widely affected by the temperature changes. The nominal temperature of the lubricant at the operating conditions must be taken into account in order to properly design and use a magnetorheological fluid journal bearing. This means that temperature may alter significantly the performance of bearings lubricated with magnetorheological lubricants.

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Static Performance of Surface Textured Magnetorheological Fluid Journal Bearings

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\textbf{A B S T R A C T} Previous studies of journal bearings with artificial texturing on the bearing surface show potential benefits in certain cases. These benefits are usually focused on a specific operating area of the bearing, whereas under certain operating conditions the performance of the bearing is deteriorating due to the surface texturing. Gaining control over the viscosity of the lubricant may become a useful tool in order to take advantage of the surface texturing in a wider range of loads and journal velocities. One way to achieve this control is the use of magnetorheological fluid journal bearings. Magnetorheological fluids are solutions of iron based paramagnetic particles in conventional lubricant. Under the influence of an external magnetic field, these particles form chains, they hinder the flow of the lubricant and they ultimately alter its apparent viscosity. In this work the two techniques are combined in order to optimize the behaviour of the journal bearing in as much a variety of operating conditions as possible. Different shapes applied on the surface texturing will be examined.

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1. \textbf{INTRODUCTION}

The use of surface modification in order to achieve performance improvement in journal bearings has been the subject of numerous works \cite{1,2}. The main issue with the application of surface texturing techniques in journal bearings is the fact that possible benefits are limited in a specific set of operating conditions. Lu and Khonsari \cite{3} have shown that dimples may affect positively the journal bearing performance especially under boundary lubrication regime. On the other hand, the application of artificial surface texturing in thrust bearings and parallel sliders has shown higher potential benefits in comparison to journal bearings as shown in \cite{4,5}.

Since the benefits from artificial texturing application occur under specific operating conditions \cite{6}, making the most out of this technique requires active control over the operating conditions themselves.

Magnetorheological fluid journal bearings offer the opportunity of control over the viscosity of the lubricant. Magnetorheological fluids consist of paramagnetic iron-based particles dispersed in
conventional lubricant. A magnetic field polarizes the particles, forces them into forming chains and changes the apparent lubricant viscosity. Magnetorheological fluid journal bearings have been the object of recent research [7,8] but the potential of using the magnetorheological fluids in conjunction with artificial texturing has not been investigated previously.

The simulation of the complex flow induced in a journal bearing with artificial texturing is a task of which Navier Stokes are better suited than Reynolds since inertia effects in this case are important [9,10]. This effect is even higher in the case of magnetorheological fluids due to their high density.

In this work the capacity of the magnetorheological fluids to take advantage of the artificial texturing in journal bearing is examined. The static performance of various configurations of artificial texturing is considered.

2. THEORY

The magnetorheological fluids exhibit a rheological behaviour which can be described by the Bingham rheological model.

\[
\mu_a = \begin{cases} 
\mu_f + \frac{\tau_0(H)}{\dot{\gamma}}, & \dot{\gamma} > \frac{\tau_0(H)}{\mu - \mu_f} \\
\mu_p, & \dot{\gamma} \leq \frac{\tau_0(H)}{\mu - \mu_f}
\end{cases}
\]  

where: \(\mu_f\) is the Newtonian viscosity of the magnetorheological fluid, \(\mu_p\) is the plastic viscosity, \(\tau_0(H)\) is the yield stress under the influence of a field with intensity \(H\) and \(\dot{\gamma}\) is the shear rate.

The calculation of the journal bearing performance is been accomplished using the continuity

\[
\frac{\partial \rho}{\partial t} + \nabla (\rho \dot{v}) = 0
\]  

and momentum equations:

\[
\frac{\partial}{\partial t} (\rho \dot{v}) + \nabla (\rho \dot{v} \dot{v}) = -\nabla p + \nabla (\vec{\tau}) + \rho \ddot{g} + \vec{F}
\]

where: \(\rho\) is the fluid density, \(\dot{v}\) is the velocity vector, \(p\) is the pressure, \(\vec{\tau}\) is the stress tensor and \(\vec{F}\) is the external forces vector. The basic geometry of the bearing considered in this work is depicted in Fig. 2.

Fig. 1. The Bingham rheological model.

The Bingham viscosity model, as shown in Figure 1, describes a semi-solid material which flows under a certain value of shear stress. This behaviour describes the rheology of the magnetorheological fluid under the excitation of an external magnetic field. The apparent viscosity of the magnetorheological fluid according to Bingham fluid is described as follows:

Fig. 2. Schematic of the journal bearing’s geometry.
The bearing considered in the simulations has a radius \( R_b = 49.999 \) mm, and total length \( L = 49.999 \) mm. The radial clearance is \( C = 85.5 \) \( \mu \)m. The configuration of the texturing pattern is described by the number of dimples in circumferential and longitudinal directions (N1 and N2, respectively). The bearing surface is only partially textured with the textured area being defined by the inlet and outlet angles \( \psi_1 \) and \( \psi_2 \), respectively. In this work two shapes of dimples were examined: rectangular and egg-shaped dimples, as shown in Fig. 3.

Rectangular dimples have a length of 1.3 mm and a width of 1.5 mm. Egg-shaped dimples where defined by the following equation:

\[
\begin{align*}
    r(\psi, z) &= \begin{cases} 
    R_b + d_{\text{dim}} \cdot e^{(2\sigma_1 z^3)} \cdot e^{(2\sigma_3 z^3)}, & \psi < a \\
    R_b + d_{\text{dim}} \cdot e^{(2\sigma_2 z^3)} \cdot e^{(2\sigma_3 z^3)}, & \psi \geq a
    \end{cases}
\end{align*}
\]

where: \( d_{\text{dim}} \) is the dimple depth, \( a \) is the circumferential position of the dimple’s center, \( b \) is the longitudinal coordinate of the dimple’s center. The \( \sigma_1 \) and \( \sigma_2 \) parameters control the dimple overall length while \( \sigma_3 \) controls the dimple’s width. In this work, \( \sigma_1 = 1.2E-3, \sigma_2 = 6E-4 \) and \( \sigma_3 = 8E-4 \).

3. RESULTS

In all cases examined in this work, the overall arc in which texturing is applied extends to 30 degrees and the dimples are uniformly distributed within the given length of the bearing. A single configuration of N1 = 6 and N2 = 6 was examined for both shapes. The rotational velocity of the journal is set to 1000 rpm. The lubricant used has a density of 2950 kg/m\(^3\), yield stress of 25000 Pa and fluid (Newtonian) viscosity of 0.112 Pas. The journal bearing with egg-shaped texturing was modelled with 77220 hexahedral elements (89700 nodes). The journal bearing rectangular shape texturing was modelled with 260400 hexahedral elements (293088 nodes).

2.1 Validation

For the purposes of model validation, the results of the simulation of a plain bearing were compared with the results presented by Brito et al [11].

![Fig. 4. Comparison of experimental with simulation results.](image)

The bearing for which validation was performed has the same bearing radius and radial clearance as the ones used for the purposes of the simulations. The length is \( L = 80 \) mm and it was lubricated with ISO VG 32 lubricant. The viscosity of the lubricant was 0.0293 Pas at 40 °C.

2.2 Angular position of texturing

The angular position of the textured area, as shown in Fig. 5, has been investigated in order to establish the relationship between \( \psi_1 \), load capacity and friction.
The results show minor improvement in terms of relative eccentricity with maximum improvement shown for all textured cases where the relative eccentricity drop reaches 1.32 % for the 6000 N load.

The egg-shaped texturing shows some significant improvement as depicted in Fig. 6.

Although there is minor influence on the relative eccentricity of the journal bearing when the angular position of the textured area changes, there is a 3.08 % decrease in the cases between 10 and 50 degrees for a 6000 N load.

The friction coefficient is minimally affected by the angular position of the artificial texturing inside the bearing. In Fig. 7 the normalized friction coefficient is presented for a series of texturing circumferential position values. The results include the performance of the smooth bearing for comparison.

The friction coefficient exhibits a maximum increase of 0.14 % in the case of 6000 N load. In the case of 8000 N there is a 0.05 % decrease but these changes are negligible. The same trend appears in the case of egg-shaped artificial texturing, described in Fig. 8.
The friction coefficient does not change significantly with the change of the dimple geometry. There is a maximum increase of 0.95% of the friction coefficient for a load of 10,000 N. Overall there are negligible deviations from the values obtained for the smooth bearing.

2.3 Dimple depth

Another parameter of the overall geometry of the artificial texturing is the dimple depth. A comparison of the effect of the dimple depth on the relative eccentricity of the artificially textured journal bearing is presented in Fig. 9, for a load of 6000 N.

![Graph showing the effect of dimple depth on the relative eccentricity for a load of 6000 N for both rectangular and egg-shaped dimples.](image)

**Fig. 9.** The effect of dimple depth on the relative eccentricity for a load of 6000 N for both rectangular and egg-shaped dimples.

The difference of geometry between the two configurations induces different results on the performance of the artificially textured journal bearing. While the rectangular dimples depth increase has a negative impact on the relative eccentricity, the increase of the egg-shaped dimples depth improves the static performance of the bearing.

4. CONCLUSIONS

The effect of artificial texturing on the performance of the journal bearing is positive although the extent of this effect seems to be minor in absolute values. The egg-shaped texturing is although promising as the increase of depth resulted in relative eccentricity improvement of 4.8% with a dimples density that is rather low, whereas the rectangular shaped texturing performance deteriorates. In other words there seem to be margins for further improvement of the performance benefits that the specific artificial texturing geometry has to offer.

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