A SELF-CIRCULATING POROUS BEARING WITH A WRAPPED-AROUND RESERVOIR

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A SELF-CIRCULATING POROUS BEARING WITH A WRAPPED-AROUND RESERVOIR

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ABSTRACT

All tribological systems consisting of moving/rotating parts need continuous lubrication, temperature control, reduced friction as means to ensure a smooth operation regime. Common for these systems is an external pressure source that carries lubricant to the system and withdraws the used and heated lubricant away from the system. This circuit usually consists of motor actuated pumps, pipes and tubes. The research herein offers a feasibility study for a new, revolutionary type of bearing, which excludes such an external circuit; instead, it is being characterized by a self-contained lubricating mechanism.

This bearing has a stationary porous bushing whose inner diameter faces the bearing clearance, while the outer one faces a wrap-around reservoir. The eccentric shaft generates pressure difference ensuring the fluid to circulate naturally between these two regions. This circulating mechanism is described and numerically simulated using the full 3D Navier-Stokes Equations for the fluid flow in the bearing clearance and the adjacent reservoir. The flow inside the porous matrix is modeled using the Darcy law, the added Forchheimer term to account for the inertial effects when the speed increases, and the Brinkman term to account for the hydrodynamic boundary condition effects that appear at the fluid/porous media interface and for the added shear effects inside the porous media. The numerical simulations concluded that the load capacity decreases and
the attitude angle increases as the permeability increases. In addition, the increase of the reservoir depth results in a reduction of load capacity. The thermal effects have been studied, and it was concluded that the bearing parameters have the same influencing effects on the load capability as for the isothermal case; however, due to the decreased dynamic viscosity of the lubricant film the generated pressures and loads are significantly smaller. The static and dynamic characteristics of the bearing have been mapped and discussed as a function of the bearing geometric features and included the thermal effects. The results of these studies proved the feasibility of the proposed system.

In the second part of the dissertation, the design and testing of the bearing prototype was described. Different reservoir depth configurations have been considered, various angular speeds, and various static loading conditions. Pressure and temperature measurements have been recorded circumferentially at the symmetry plan of the bearing and the shaft orbits have been monitored. The results of these experiments concluded that proposed the new bearing design was feasible and functioning according to the theoretical predicted model.

The numerical results and the experimentally obtained data have been in good agreement with each other.

*The application with the presented characteristics has been filed for a patent:*

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CHAPTER I

INTRODUCTION

Most bearing applications, high-speed and high temperature bearings in particular, require the use of temperature controlled systems to provide (in a closed circulating loop) the necessary amount of lubricant both for load carrying capability and thermal management necessary for safe and continuous operation.

The progressively high temperature lubrication environments for today’s demanding applications require both novel types of lubricants and innovative ways for their dispensation, circulation and thermal conditioning. Many alternatives have been proposed in response to these requirements, most prominent amongst them being the oil-less (air based), powder, and liquid metal lubrication systems.

The research herein intends to prove the feasibility of a new type of bearing that does not need an external pressure source to ensure its operation. This self-circulating, self-lubricating, zero-leakage, tribological system were the fluid circulation is a result of continuity, momentum, natural and forced convection laws and in the absence of an external pumping mechanism or external pressure source is proven numerically and
tested experimentally. A porous medium shaped as a bushing is used as a divider between a high pressure zone, where the pressure is generated by the eccentric shaft and a lower pressure zone situated on the other side of the porous wall in the form of a wrapped around reservoir. The lubricant is impregnated in the porous material and is present in the bearing clearance as well as inside the reservoir. Due to the pressure gradient created, the lubricating fluid circulates freely in and out of the reservoir guarantying a closed loop, self-contained and self-lubricated circuit.

Present work uses oil as working fluid and represents a feasibility study for an application adapted to high temperature, high speed bearings using low temperature melting earth metals (gallium, indium and tin) for lubrication. In the recent past, military nuclear reactor applications have used sodium and potassium as lubricating fluids.

This innovative self-circulating, self-lubricating system has significant potential to improve the life expectancy of bearings functioning in complex installations where they are difficult to replace and monitor. Indeed, if such a device would be adopted by industry: jet engine manufacturers, steam and power gas turbine manufacturers, compressor and pump manufacturers, even automobile manufacturers, economies of scale could be realized through the elimination of the forced pumping system. Overall engines would be reduced in size and the maintenance procedures would be radically changed compared to the conventional lubricating systems.
1.1 Porous media modeling considerations

Given the topic covered, the focus of the literature review is on the theory of porous mediums and the technology of existing porous bearings. The flow through porous media represents a complex physical phenomenon whose pertinent fundamental physics are reviewed below.

1.1.1 On porosity, permeability and their relationship

While porosity is an intrinsic property associated with the physical properties of the porous medium, representing the fraction of the void volume to the total volume of the porous medium, permeability is an extrinsic property that characterizes the ease with which a fluid passes through that medium due to an imposed pressure gradient. Thus, permeability, a macroscopic property defined by the Darcy’s law, is also dependent on the geometry of the porous structure (porosity $\varepsilon$, and tortuosity $\tau$). Much work has been performed in order to determine the nature of permeability and how the porous structure affects it: (Kozeny [1], Brooks and Purcell [2], Bundine et al. [3], Bear [4]). Fundamentally, it has been concluded that permeability is proportional with the mean square of the pore size, porosity; tortuosity and inverse proportional with the specific surface (see Bear [4], Chap 5).

In normally occurring mediums, porosity is of a spatially random distributed nature and is treated with statistical methods (Chalkley et al. [5]). There is no one general relationship between the effective porosity and permeability, even though the porosity is clearly a dependent variable in the permeability calculation. For instance, the
Kozeny-Carman equation states that for tightly packed spheres the permeability is
\[ k = \frac{\delta}{180 (1-\varepsilon)^2} \varepsilon^3 \] where \( \delta \) is a mean sphere diameter (Carman [6]). Of course, for a well-defined medium where such \( \delta \) can be easily calculated (e.g. a matrix of spheres) \( k \) and \( \varepsilon \) are intrinsically related and their unique relationship can be defined. To avoid uncertainty due to the empiricism of the formulas, the numerical simulations presented herein treat porosity and permeability as two independent properties that can be varied separately. Still, the ranges for these values are chosen realistically with respect to each other: porosities in the range of 0.1 to 0.5 and permeabilities of 1E-10m\(^2\) to 1E-13m\(^2\).

1.1.2 The Darcy, Beavers-Joseph, Brinkman, and Forchheimer approaches

The Darcy pressure drop law: \( u = -\frac{k \partial P}{\mu \partial x} \) is the most fundamental building block for flow in a porous medium modeling and it was obtained empirically in the nineteenth century (Nield and Bejan [7]). \( \partial P/\partial x \) is the pressure gradient in the direction of the flow, \( \mu \) is the dynamic viscosity of the fluid, and \( k \) is the specific permeability. The Darcy formulation has been validated by the results of many experiments and, in addition, has been obtained theoretically with the use of deterministic and statistical methods (Whitaker [8], Ene and Polisevki [9]). If one draws a parallel with the Fourier Law one can state that the ratio \( \frac{k}{\mu} \) known as ‘hydraulic conductivity’ (Bear [4]), plays the same role as the thermal conductivity in the Fourier’s law.

Nonetheless, the Darcy model alone cannot predict either the effects of viscous shear that appear at the interface with a solid boundary (especially when high surface
roughness is present), nor the flow development and pressure drop in a high velocity regime.

In 1967 experimental work reported by Beavers and Joseph [10] for a two-dimensional Poiseuille flow described an increase in the mass flow over a permeable block when compared to an impermeable block. They found that when a viscous fluid passes a porous solid, tangential stress entrain the fluid below the interface with a velocity that is slightly greater than that of a fluid in the bulk of a porous bearing, concluding the presence of a boundary layer in the permeable block. They suggested replacing the effect of this boundary layer with a slip velocity proportional to the tangential stress. Hence, the velocity gradient has the form: \( \frac{du}{dy} \bigg|_{y=0} = -\frac{1}{\sqrt{k}} (u_S - u_D) \), where \( u_S \) represents the slip velocity, \( u_D \) is the Darcy velocity inside the porous, and \( \alpha \) is a dimensionless constant depending on the geometry of the interstices, the flow direction at the interface, and the permeability of the medium. Since the determination of \( \alpha \) and \( u_S \) is difficult, the implementation of Beavers and Joseph formula is rather complex.

In 1947 Brinkman [11] pointed out that using Darcy inside the porous media (a first order differential equation) and Navier-Stokes Equations (NSE) in the fluid region (a second order differential equation) makes impossible to formulate ‘rational’ and continuous boundary conditions. He suggested an extension to Darcy’s law to describe the flow through a dense bed of particles stating the equilibrium between the pressure gradient, the divergence of the viscous stress tensor, and the damping force caused by the porous mass. His objective was to extend the Stokes drag force on a sphere (a sphere placed in an infinite plain domain) to include the effect of the neighboring spheres. He
proposed the following form: \( \nabla P = -\frac{\mu^*}{k} \vec{u} + \mu^* \nabla^2 \vec{u} \), superimposing the viscous penetration flow (Stokes flow) and Darcy flow, where \( \mu^* \) is the apparent dynamic viscosity in the porous medium representing the contribution of the solid particles through shear to the transport of the momentum in fluid. He suggested that the value of \( \mu^* \) may be different than that of \( \mu \) and initially used the Einstein formula for the effective viscosity of suspension: \( \mu^* = \mu \left( 1 + 2.5(1 - \varepsilon) \right) \). Experimental results obtained by Carman and Brinkman’s formulation are in good agreement for values of \( \varepsilon > 0.4 \) and using \( \mu^* = \mu \). This extension of Darcy’s formula is used by some authors to describe the flow in the boundary layer in the porous medium, rather than using the Beaver-Joseph [12] slip velocity boundary. Neale and Nader [13] arrived to the same result both by using Brinkman equation and by using the Beaver-Joseph condition for the problem of a flow in a channel bounded by a thick porous wall. They also recommended that the ratio \( \frac{\mu^*}{\mu} \) to be taken as unity.

A graphical representation on the modeling of the porous media/fluid interface using the presented models is shown in Figure 1.1. The differences between models are clearly stated. There is a discontinuity in the velocity field when using Darcy equation; the Beavers-Joseph slip boundary condition reconciles this discontinuity and finally the Brinkman model ensures a smooth transition between the fluid and the porous region with the presence of a thin boundary layer located underneath the surface, inside the porous block.
Figure 1.1 Different approaches for modeling the fluid/porous media interface using Darcy, Beaver-Joseph and Brinkman formulations

Another important and widely used extension to the Darcy’s law is the Forchheimer term (Forchheimer [14]) which accounts for the inertial effects at higher velocities through an increase in drag. Experimental observations do indicate that the pressure drop in the porous medium at higher flow velocities is proportional to a linear (Darcy’s contribution) and a square combination of the flow velocity (inertial effects). The form proposed by Forchheimer is: \( \nabla P = -\frac{\mu}{k} \overline{u} - b\overline{u}^2 \), where the value of coefficient \( b \) is empirically determined and is a function of the structure of the porous media and Reynolds number, \( Re_k \). The determination of the Forchheimer coefficient can be either empirical, or analytical when based on the characterization of the porous skeletal material. There are many different expressions proposed for this \( b \) coefficient due to the fact that different flow conditions and assumptions have led to different expressions, Sobieski [15]. In 1982 Joseph et al. [16] proposed the generally accepted form: \( \nabla P = -\frac{\mu}{k} \overline{u} - b\overline{u}^2 \).
where \( c_f \) is a dimensionless form-drag constant. The transition from the linear Darcy regime to the Forcheimer regime occurs for Reynolds number, \( 1<Re_k = \frac{\rho u k^{3/2}}{\mu} <10 \), (Ward [17]), where the characteristic length \( k^{1/2} \) is based on the permeability \( k \), and has the units of \([m^2]\).

In 1990 Vafai and Kim [18] presented an exact solution for the interface region between a porous medium and a fluid by solving the Navier-Stokes Equations (NSE) inside the fluid region and the Darcy-Brinkman-Forchheimer \( \nabla P = -\frac{\mu}{k} \vec{u} + \mu \nabla^2 \vec{u} - \frac{c_f \rho r}{\sqrt{k}} |\vec{u}|\vec{u} \) formulation inside the porous media and assuming the continuity of velocity and shear stress at the porous media/fluid film interface. Note that the viscosity used in both microscopic and macroscopic viscous terms, is the fluid viscosity. This formulation is widely accepted today because it clearly relates the flows inside and outside the porous medium.

Nield [19] noticed that even though the shear stress is continuous over the pore hollow section of the interface, that is not the case over the solid portion of the porous medium, and the averaged shear stress does not necessarily match at the interface. He concluded that the use of shear stress continuity at the interface might result in over-determining the physical problem.
1.2 Porous bearing modeling (Literature review)

Morgan and Cameron [20] were first to propose a modified Reynolds equation that accounts for the entry of lubricant from the porous medium into the gap between bearing and journal. They solved for the pressure distribution in a short porous bearing and obtained the load capacity. Their calculations satisfied the Reynolds equation in the film and the Laplace equation in the porous region, while respecting continuity of pressure and shear stress. The proposed modified Reynolds equation was:

$$\frac{\partial}{\partial z} \left( \frac{h^3}{12} \frac{\partial p}{\partial z} \right) = \frac{\partial}{\partial x} \left( \frac{uh}{2} \right) + (V_h - V_0)$$

where $V_0$ was the velocity into the pores, $V_0 = -\left( \frac{\partial p}{\partial y} \right) \frac{k}{\eta}$. The final form of the Reynolds equation obtained was:

$$\frac{\partial}{\partial z} \left( \frac{h^3}{12} \frac{\partial p}{\partial z} \right) = 6U_h \frac{\partial h}{\partial x} \left( \frac{uh}{2} \right) + 12k \left( \frac{\partial p}{\partial y} \right)_h$$

where $\left( \frac{\partial p}{\partial y} \right)_h$ was the pressure gradient at the boundary of the porous material and was given by the solution of Laplace $\nabla^2 p = 0$ throughout the porous matrix.

Cameron et al. [21] showed that the porous bearings can reach eccentricity ratios of 1.0 for a finite load. They reported high coefficients of friction even though the lubrication continued to be fully hydrodynamic and determined the critical conditions below which the lubrication stops being fully hydrodynamic. Rhodes and Rouleau [22] presented the results for a short porous bearing (the short bearing assumption: $\frac{\partial p}{\partial z} = 0$ : the gradient of the axial pressure is zero) with end seals. They observed the decrease in the load carrying capability factor that took into account the effect of the short bearing assumption compared to the finite dimensions bearing. Cusano and Phelan [23] presented experimental results that confirmed the functionality of porous bearings at
higher eccentricity ratios compared to solid bearings, findings that correlated with the permeability effects on the load carrying capability. These contributions are the first and most important to the development and numerical modeling of the porous bearings. The numerical results were based on the Darcy model for a full hydrodynamic lubrication, assuming a linear pressure drop in the porous media and no slip boundary conditions at the porous/fluid film interface.

Goldstein and Braun [24] were first to apply the slip boundary conditions formulated by Beavers and Joseph to analysis of porous bearings. Murti [25] derived a modified Reynolds equation applicable to finite porous bearings that also included the slip velocity effect. He concluded that this slip effect was predominant for low values of permeability. Other contributions were made by Puri and Patel [26] and Rao [27]. Their approach used a Beavers and Joseph-type slip boundary condition that considers the existence of a thin boundary layer beneath the porous/fluid interface, layer where the viscous shear is the dominant mechanism of slip flow; this approach allowed a smooth transition from the velocity of the fluid film to the Darcy velocity inside the porous medium. Both papers ([26] and [27]) showed a decrease in the load capacity as a result of using the slip velocity condition. Kumar ([28], [29]) analyzed the hydrodynamic lubrication using the Reynolds modified equation for porous journal bearings with slip velocity in the turbulent regime. He obtained the performance characteristics in an exact closed form solution.

Using the stochastic theory for a rough porous bearing developed by Christensen [30], Prakash and Tiwari [31] concluded that the surface roughness has a strong effect on the bearing performance. This effect can either increase, or decrease the pressure build-
up depending on the bearing type, nominal geometry, roughness type and the working conditions of the contact. Gururajan and Prakash [32] also reported a strong dependence between the roughness of the porous material and the slip effects and hence the performance of the bearing.

Lin and Hwang ([33], [34]) derived a modified Reynolds equation using the Brinkman-extended Darcy formulation with velocity continuity and stress jump at the interface. Following numerical simulations, they reported an increase in the load capacity and a reduction in the friction coefficient. The jump stress approach replaces the slip velocity condition at the fluid/porous interface and provides closure for the system of equations. The consideration of the viscous shear effects will cause a significant reduction in the eccentricity ratio and better performance characteristics for the system: higher threshold speed for stability (a more stable bearing) and increased static attitude angle.

Another important contribution to the porous bearing modeling was the consideration of the non-uniformity that led to heterogeneous permeability of the porous media. Cusano [23] presented analytical solutions for a porous bearing with variable permeability in the axial and radial direction. The equations used were approximate representations of the permeability variation curves given by Morgan [35].

Murti ([36], [37]), Jenicek, and Schenk [38] described methods to determine the permeability of porous metal bearings. According to their observations, the liquid permeability is dependent with time and pressure and the gas permeability is not dependent.
Cusano ([39], [23]) noted that adopting techniques of local reduction of permeability, the coefficient of friction was reduced and the load carrying capacity increased. Quan and Wang [40] concentrated their research in the direction of improving the load capacity and the dynamic performance of the porous bearing. Their approach considered using a porous media with a permeability that varies circumferentially. Their results showed that having lower permeability in the high pressure region of the bearing reduced the migration of the fluid inside the porous and hence increased the total load. In the low pressure region an porous matrix with a higher permeability was used. The bigger the permeability ratio between the two regions, the higher the load capacity and lower friction was obtained. It was also mentioned that practical limitations are imposed on this ratio, since a higher ratio will make the bearing susceptible to rupture.

Chen, et al. [41] concluded that an increase in stress jump parameter provides a decrease in load capacity, an increase in the attitude angle, and an increase in the coefficient of friction. The cavitated region was solved using the Elrod algorithm.

Contributions to the study of the turbulent flow inside porous media have been done by: Kumar ([42],[43]) who used a linearization technique and Burton’s law of wall, and derived lubrication equations including slip velocity and end leakage for self-acting hydrodynamic porous bearings operating in turbulent regimes. In addition, Pedras and Lemos [44] studied the turbulent regime in a porous medium using Reynolds-averaged equations. More contributions have been added by Orselli and Lemos [45], who studied the influence of a porous insert in a incompressible turbulent flow in a pipe that suffered a sudden contraction.
The static and dynamic properties of porous bearings were obtained by Lin and Hwang J [46] and the effect of viscous shear stresses was proved theoretically to increase the load capacity and static attitude angle and decrease the coefficient of friction.

Elsharkawy and Nassar [47] presented closed form analytical solutions for three types of porous bearings: parallel surface bearing of infinite width, journal bearings and parallel circular plates. Results showed the decrease in load carrying capacity with the increase of permeability up to a point when the effect of porous can be neglected. For dimensionless permeability parameter \(\psi = k_z h_p / h_0^3\) of less then 0.001, the effect of the porous layer on the hydrodynamic lubrication of squeeze-film porous bearings can be neglected. \(h_p\) was the thickness of the porous layer, \(k_z\) was the permeability of the porous media in the z direction and \(h_0\) was the fluid film thickness.

1.3 Scope of work

The scope of the present work is to demonstrate that a loaded shaft, in an eccentric position, is able to pump fluid from the bearing clearance, through the porous medium into the external reservoir and create the condition in the passive space for the fluid to be pumped back in the clearance space. This closed self circulating, self lubricating process is made possible due the positive pressure created by the hydrodynamic effects in the convergent section and is subsequently governed by the pressure differential between the active and passive spaces of the bearing. The present work uses oil as the working fluid and represents a feasibility study for a high
temperature application that will use low temperature melting earth metals such as gallium and lithium for lubrication.

The elements of the bearing are detailed in Figure 1.2. The working fluid fills the bearing clearance (1), the porous matrix (2) and the wrapped-around reservoir (3). The shaft (4) fulfills the role of the circulating pump, generating pressure in the fluid film due to its eccentricity. The porous medium (2) is homogeneous, having the same flow resistance in all directions and separates the active space (1) from the passive space (3), allowing the fluid to circulate between the two spaces. Spiral groove seals (5) have been

![Solid model of the porous self-circulating bearing](image)

**Figure 1.2** Solid model of the porous self-circulating bearing (1-bearing clearance; 2-porous medium; 3-reservoir; 4-shaft; 5- spiral groove seal; 6-outer casing with heat exchanging fins, 7-lip seals)
machined on the surface of the shaft providing a back-pumping effect towards the center of the bearing. To ensure that even after passing through the spiral groove seal the fluid will not leak sideways, high speed lip seal (7) were considered at the ends of the bearing. These lip seals will also ensure that there is no fluid leakage when the bearing is not in function.

Since the removal of the pump driven circulating system eliminates any chance of active cooling, the thermal management strategy proposed is based on an enhanced natural convection with the addition of heat exchanging fins located on the exterior surface of the casing (6).

Figure 1.3 presents all the bearing components in a disassembled view to complete the information in Figure 1.2, which presented the assembly of the proposed

![Disassembled solid model of the porous self-circulating bearing](image)

Figure 1.3 Disassembled solid model of the porous self-circulating bearing (1- bearing clearance; 2-porous medium; 3-reservoir; 4-shaft; 5- spiral groove seal; 6-outer casing with heat exchanging fins, 7-lip seals)
configuration. The designed prototype did incorporate all the elements presented. The numerical modeling did not take into account the effects of the seals contributing to the pressure increase.

To explain the relationship between the pressures generated as the main mechanism responsible for the fluid self-circulation, Figure 1.4 presents, in a circumferential cross section, the schematic of the system concept for the solid model concept presented in Figure 1.2. Due to the rotation of the eccentric shaft (4), the hydrodynamic pressure rises in the convergent region ((1)- $p_{1c}$) of the bearing, and while

Figure 1.4 Concept schematic for the self-circulating bearing: (1) clearance ($p_{1c}$—pressure in the clearance in the convergent zone, $p_{1d}$—pressure in the clearance in the divergent zone); (2) porous (3) reservoir ($p_{2c}$—pressure in the reservoir in the convergent zone, $p_{2d}$—pressure in the reservoir in the divergent zone), (4) shaft
some of the fluid travels in the circumferential and axial directions, some of it exits radially through the homogeneous porous medium (2) and enters into the stationary reservoir (3). The prerequisite for such flow to occur is that the hydrodynamic pressure generated in the active space to be higher than the pressure in the passive space, hence providing the pressure differential required for the fluid to traverse the porous medium. The pressure in the reservoir, $p_{2c}$, is thus related to, and dependent of the levels of pressure in the clearance $p_{1c}$.

In the passive space there is a circumferential and axial pressure variation between the sections lying directly under the convergent and respectively divergent regions of the active space, where the necessary functional relationships require that $p_{1c} > p_{2c}$ in the convergent, and $p_{2d} > p_{1d}$ in the divergent sections. These differences need to be maintained for the circulation between the active space and the reservoir to be continuous.

Figure 1.5 presents a schematic of typical pressure distribution between the pressures in the active space and the passive space of the hydrodynamic porous bearing. Note that the cavitation effects are also accounted for in the divergent zone. The distribution of the pressure gradients in combination with the necessity of the entire system mass conservation will force the fluid to circulate in and out of the active and passive bearing spaces. From the study of Figure 1.5 one should note that the pressure in the reservoir follows the pressure patterns in the bearing active space but at different magnitudes, the reservoir serving as a connected vessel. The pressure in the reservoir is fundamentally controlled by the pumping action of the eccentric shaft in combination with the resistance of the porous medium. The active-passive space pressure differential
Figure 1.5 Concept schematic of the pressure differential mechanism. $p_{1c}$, $p_{2c}$ – pressure distribution in the active(1) and passive (2) spaces in the convergent regions; $p_{1d}$, $p_{2d}$ – pressure distribution in the active(1) and passive (2) spaces in the divergent region

relationship shows that the higher pressures in the active space ($p_{1c} > p_{2c}$) help move the fluid through the porous medium to the reservoir, while in the divergent region, where cavitation is a factor, the higher pressure in the reservoir ($p_{2d} > p_{1d}$), helps the fluid from the reservoir back.
1.4 Previous work

The construction of a mathematical model for a self-circulating porous bearing with a contiguous reservoir was first attempted by Johnston et al. [48]. For simplicity, the 2-D geometry of the actual journal bearing (axial and circumferential directions) was unwrapped, and periodic boundary conditions were considered in the circumferential direction. The approach used Reynolds equation for the bearing’s clearance zone, the Darcy’s law to account for the pressure drop across the porous region and Navier-Stokes Equations to model the fluid in the reservoir. The model did not use slip boundary conditions at the fluid/porous material interface and assumed the continuity of velocity at the interfaces. The authors have presented parametric solutions for pressure distributions in the film and reservoir regions, including the cavitation effects. The concept of the lubricant being re-circulated in and out of the external reservoir and back into the active clearance region was numerically proven for the first time.

1.5 Objectives

The present work continues work done in Johnston et al. [48] by using a three-dimensional cylindrical geometry with fully termed 3-D continuity and Navier-Stokes Equations (NSE) applied both in the active space of the bearing (the clearance) and the passive space (the reservoir), see Figure 2.6. The flow in the porous medium is modeled using the Darcy-Brinkman-Forcheimer formulation. This formulation integrates seamlessly the bearing active space, the porous medium and the recirculation reservoir.
The performance optimization presents sophisticated problems that couple fluid dynamics, porous media modeling and heat transfer modeling to allow for the desired flow recirculation and thermal management.

To address these problems, this research combines theory, modeling, design, building of prototypes, and experimental testing with the following tasks:

1. Develop a three dimensional finite geometry model to use for numerical calculations. Use the complete termed system of equations:
   - The full Navier Stokes Equations for the fluid areas (incorporating thus the inertial effects)
   - The full Darcy, Forchheimer, and Brinkman pressure drop formulation for the flow inside the porous media (this will ensure the incorporation of viscous shear effects and the high speed effects inside the porous).

2. Develop a thermo-hydrodynamic model to include the heat generation due to the motion of the shaft through shear at solid/fluid boundary, the effects of lowering the viscous shear forces inside the fluid due to temperature rise, and the heat loss due to the effects of an array of heat dissipating fins. Test multiple configurations of heat dissipating fins.

3. Develop a method to estimate the static and dynamic coefficients of the film of lubricant using the computational fluid dynamics software CFD-ACE+ and
using the fully termed equations mentioned in objective 1. Perform the analysis for both isothermal and thermo-hydrodynamic conditions, varying the permeability and the reservoir depth. Investigate the effects of the bearing features on the bearing stability.

4. Build experimental bearing prototype to test the concept, monitor the built-up pressures and temperatures for varying loads, varying running speeds, and different reservoir depths. Monitor the orbits of the shaft.

5. Investigate the correspondence between of the numerical and the experimental results.

6. In conclusion, review the achievements of this work in view of the programmatic laid out above.
CHAPTER II

NUMERICAL MODELING USING CFD-ACE+

The present research is presenting the numerical simulation on a 3-D model, both isothermal and adiabatic numerical analysis using the multi-physics computational commercial software package CFD-ACE+ [49]. The software includes preprocessing tools for geometry and grid generation (CFD-GEOM), solver set-up interface to input the model boundary and volume conditions and an advanced solver (CFD-ACE-GUI), and a post-processor to compile and visualize the results (CFD-VIEW).

The simulated system includes a stationary porous bushing whose inner diameter faces the bearing clearance while the outer diameter faces the wrapped-around reservoir. The study uses the complete 3D Navier-Stokes Equations (NSE) for the fluid motion in the bearing clearance and the adjacent reservoir. The modules used for computation were the flow module, the heat transfer module, and the cavitation module. Explanations for each module and the associated equations are presented, together with the boundary conditions and the way they relate to the computation. The flow inside the porous matrix is modeled using the Brinkman formulation with the added pressure ‘penalties’ in the form of the Darcy and Forchheimer terms. The bearing operates in the fully hydrodynamic lubricating regime and surface roughness effects are not considered.
The cavitation model proposed by Singhal [50] is utilized in the divergent region for the isothermal model, and Gumbel cavitation model is utilized for the thermo-hydrodynamic model.

2.1 The flow module for fluid regions.

The flow module provides the solution of velocity and pressure fields inside the domain by solving the conservation of momentum equations coupled with the continuity equation.

The form of the Navier-Stokes equations considered is:

\[
\frac{\partial (\rho U)}{\partial t} + \nabla \cdot (\rho U U) = - \frac{\partial p}{\partial x_i} + \nabla \cdot (\mu \nabla U) + S_{Mx_i}
\]  

where \((x_i = x, y, z)\), and \(S_{Mx_i}\) represents the source term. The associated continuity equation is:

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

CFD-ACE solver applies the NSE in a cartesian body fitted system of coordinates. The equations are discretized using the finite volume method whereby the NSE and continuity are integrated across a computational domain divided into a number of control volumes (cells). A typical representation of such a cell is shown in Figure 2.1a), while a multitude of adjacent cells is shown in Figure 2.1(b). The cell center is denoted by \(P\), while the adjacent cells centers are marked \(W, E, S\) and \(N\). The algorithm uses a collocated cell-centered arrangement, whereby all dependent variables and material properties are stored at the cell center \(P\), as the average values of all the
properties contained inside the respective cell. Then the momentum governing equations can be expressed in the form of a generalized arbitrary variable $\Phi$ (for $u, v, w$) as:

$$\frac{\partial \rho \Phi}{\partial t} + \nabla \cdot (\rho \vec{U} \Phi) = \nabla \cdot (\Gamma \nabla \Phi) + S_\Phi$$

where $\Gamma$ represents the transport properties ($\mu, k$), while $\Phi$ will take the values of 1 (for continuity) and $\vec{U}$ for momentum. Integrating this equation over a control-volume cell $d\mathcal{V}$, one can write:

$$\int_{\mathcal{V}} \frac{\partial \rho \Phi}{\partial t} d\mathcal{V} + \int_{\mathcal{V}} \nabla \cdot (\rho \vec{U} \Phi) d\mathcal{V} = \int_{\mathcal{V}} \nabla \cdot (\Gamma \nabla \Phi) d\mathcal{V} + \int_{\mathcal{V}} S_\Phi d\mathcal{V}$$

Because the arbitrary dependent variable is calculated at the cell-centers, its values at the finite volume faces (where conservation rules are applied) have to be interpolated. Different interpolation schemes are imbedded with the code and they can be chosen as a function of the level of numerical accuracy and stability desired: first order upwind, central difference, or second order upwind. For the calculations presented herein a first order upwind was used. Figure 2.1 (b) notations help understand the construction of the upwind concept for this problem, as they are presented in the equations below.

In this scheme, $\Phi_e$ is evaluated at the cell face, $e$, (Figure 2.1 (b)) but depending on the flow direction, $\Phi_e$ will take the values at grid locations P or E ($\Phi_P$ or $\Phi_E$) depending whether $U^n_e$ is positive or negative. Mathematically then, $\Phi$ can be expressed as:

$$\Phi_e = \Phi_P \text{ if } U^n_e > 0$$

$$\Phi_e = \Phi_E \text{ if } U^n_e < 0$$
Figure 2.1 a) Typical single finite volume computational cell; b) assembly of a multitude of single finite volume cells. The arrow indicates the diffusion or convection of an arbitrary property across a face separating two contiguous cells.

This scheme has first-order accuracy. The numerically integrated momentum equations can then be expressed as a system of algebraic linear equations of the form:

\[(a_p - S_p)\Phi_P = \sum_{nb} a_{nb}\Phi_{nb} + S_U \quad 2.6\]

where the subscript \((nb)\) denotes values at neighboring cells, \(a_{nb}\) being the neighbor coefficients to \(a_P\) (the coefficient for the variable currently being calculated). This system of finite-volume difference equations, each representing the conservation of the property \(\Phi\) (velocity \(\vec{U}\) in this case) for an individual cell, is non-linear because of the structure of the \(a_{nb}\) coefficients.

An iterative method has to be employed for the implementation of the numerical solution. Fluid density and velocities are calculated only at cell centers, but for conservation purposes they need to be known also on the cell faces. Therefore a linear interpolation is necessary, the latter decoupling the velocity and pressure fields, thus causing potentially instability problems. In CFD-ACE+, the interpolation problem is
treated by the method proposed by Rhie and Chow [51] and by Peric [52]. According to these authors, the cell-face mass flux is evaluated by averaging the momentum equation at the cell faces and relating the velocities at the cell faces to the local pressure. The CFD-ACE solver uses the Semi-Implicit Method for Pressure-Linked Equations Consistent (SIMPLEC) computational scheme which is an enhancement of Patankar and Spalding [53] SIMPLE algorithm. SIMPLEC was proposed by Van Doormal and Raithby [54], and introduces a pressure-correction equation by way of the continuity equation, while the momentum equations solve for \((u, v, w)\).

\[ a_p u_p = \left( \sum_{n_b} a_{nb} u_{nb} + S_U \right)_p - \left( \sum_e p_e A_e n_{xe} \right)_p \]  \hspace{1cm} (2.7)

In this equation, the velocity \(u\) is evaluated at point \(P\) while a pressure \(P\) has to be introduced at face \(e\). The pressure differential between the faces of the control volume causes the flow and therefore the velocity at the arbitrary point \(P\). However, this pressure is unknown and a ‘guess’ value is at the root of the iterative nature of the solution. If one designates the guessed pressure as \(P^*\), one can rewrite the above equation for a resulting value of the iterated \(u^*\) as:

\[ a_p u_p = \left( \sum_{n_b} a_{nb} u_{nb}^* + S_U \right)_p - \left( \sum_e p_e^* A_e n_{xe} \right)_p \]  \hspace{1cm} (2.8)

A resulting \(u^*\), generated by 2.8, will not be a true solution as long as the residual generated is not below a low acceptable value. Thus, iteratively corrected values have to be calculated for \(u, v, w\) and \(P\). If \(u'\) and \(p'\) are the respective corrections, then the newly iterated values become:
$$u = u^* + u'; \quad v = v^* + v'; \quad w = w^* + w'$$

$$\bar{P} = P^* + p'$$

The expression for $u'$ can be obtained by subtracting equation 2.8 from 2.7. The scheme of solution is iterative in nature and continues until the residual values of the correcting quantities $u', v', w', p'$ are below a low acceptable limit. The reader is referred to the CFD-ACE+ user manual [49] for further details.

The residuals of continuity and momentum equations are required to be below $10^{-4}$ in order to achieve acceptable convergence. To improve convergence the under-relaxation factors are adjusted individually for each one of the primitive variables ($u, v, w, p$). For both velocity and pressure, the typical under relaxation factors vary between 0.1 and 0.7.

2.2 The flow module for porous media

The equations and the associated boundary conditions are set up in such a way as to allow the treatment of the active domain (clearance), the porous medium and the passive space (reservoir) as a continuum; that is, all three domains use the same equations but with different values for porosity and permeability parameters. When $\varepsilon = \frac{\alpha_{void}}{\alpha_{total}} = 1$ ($k \to \infty$), calculations are performed in the reservoir and the active space, and equations 2.1 and 2.2, the standard NSE, are the governing equations. For values of $\varepsilon \in (0,1)$ a certain level of porosity is present and calculations are then performed for the porous region. When $\varepsilon = 0$, that represents a solid wall and continuity equation is identical 0. Thus for $\varepsilon \in (0,1)$ one can re-write the continuity equation, 2.2 as
\[
\frac{\partial \varepsilon \rho}{\partial t} + \nabla \cdot (\varepsilon \rho \vec{U}) = 0
\]

and the momentum equation 2.1 as

\[
\frac{\partial \varepsilon \rho \vec{U}}{\partial t} + \nabla \cdot (\varepsilon \rho \vec{U} \vec{U}) = -\varepsilon \nabla P + \nabla \cdot (\varepsilon \vec{\tau}) - \frac{\varepsilon^2 \mu}{k} \vec{U} - \frac{\varepsilon^3 \mu}{\sqrt{k}} C \rho |\vec{U}| \vec{U}
\]

Equation 2.12 is generally known as the Brinkman-Forchheimer extended Darcy equation, and together with equation 2.11 make it clear that \(\varepsilon\), which represents the porosity of the stationary bushing allows the application of the same equations for all three spaces. The differentiation between the porous and clear spaces is being made only through the value adopted by \(\varepsilon\). In equation 2.12, the last two terms, represent the Darcy and Forchheimer ‘penalty pressure drops’ which are active only in the porous medium. These terms represent added resistances to the flow in the porous medium thus adding to the original \(\nabla P\) that characterizes pressure drops in the non-porous medium. When \(\varepsilon = 1\), \((k \to \infty)\), the Darcy \((\frac{\varepsilon^2 \mu}{k} \vec{U})\) and Forchheimer \((\frac{\varepsilon^3 \mu}{\sqrt{k}} C \rho |\vec{U}| \vec{U})\) terms tend to zero allowing equation to regain the form of equation 2.1. In the Forchheimer term the constant \(C\) is a dimensionless drag universal constant, with an empirical value of 0.55 (Ward [17] and Joseph et al. [16]).

2.3 The cavitation module

Implementing a cavitation model as it applies to the physics of the bearing has been, and continues to be a worthy endeavor. Starting with Gumbel [55] and Swift-Stieber [56] in the early decades of the 1900s who offered non-conserving mass models and ending with Elrod [57], Vijaygharavan and Keith [58], and Kumar and Booker [59]
who proposed mass conserving models, many researchers implemented cavitation models for application with the Reynolds equation. Three-dimensional simulations have been made possible by both the computer revolution and the introduction of commercial algorithms like the one used in the present paper. These codes need properly tailored, bearing physically compatible modules to appropriately model cavitation. Singhal et al. [50] noted that such early models lacked numerical robustness and some of the physics, since they did not distinguish between vaporous and gaseous cavitation. The causes and modeling of these two types of cavitation are vastly different. Braun and Hannon [60] have recently offered a comprehensive review of the state of cavitation simulation and discussed these two types of cavitation. The multi-physics CFD-ACE+ offers a Full Cavitation Model (Singhal et al. [50]) that addresses both the vaporous and the gaseous cavitation. The latter is referred to, as non-condensable gases (NCG) type cavitation. The NCG cavitation is most probable to occur when pressures fall below the partial pressures of the NCG dissolved in the working fluid, and the short time constant characterizing short transient dynamics behavior is not applicable. In this context, a conservation advection-diffusion equation for the gas mass fraction \( f_{gas} \) is added to the NSE and continuity equations.

\[
\frac{\partial (\rho_{gas} f_{gas})}{\partial t} + \nabla \cdot (\rho_{gas} \bar{u} f_{gas}) = \nabla \cdot (\Gamma f_{gas})
\]

where \( \Gamma \) is the effective transport coefficient, and \( f_{gas} \) is the actual property transported.

The density of the air mass fraction is computed using the perfect gas law:

\[
\rho_{gas} = \frac{W_{gas} p}{RT}
\]

where \( W_{gas} \) is the molecular weight, \( p \) is the gas partial pressure (not vaporization pressure), \( R \) the universal gas constant, and \( T \) is the gas temperature. The model
implemented in this paper does not allow for additional diffusion of gas from the oil, beyond an initially specified fraction, $f_{gas}$. Even though the value of $f_{gas}$ is kept constant throughout the computations, its distribution in the bearing active space depends on the local cells pressure and temperature. The advective left hand side of the Eq. 2.13 plays the dominant role in distributing the gas mass fraction around the circumference. Thus, more of the gas fraction will be concentrated in the low-pressure region with very little gas in the high-pressure region. It is in this way that the cavitation region is naturally defined.

The density of the mixture can be calculated using:

$$\frac{1}{\rho} = \frac{f_{gas}}{\rho_{gas}} + \frac{1 - f_{gas}}{\rho_{liquid}} \quad 2.15$$

while gas fraction volume is expressed according to:

$$\alpha_{gas} = f_{gas} \frac{\rho}{\rho_{gas}} \quad 2.16$$

For the computations the fraction of liquid by volume ($\alpha_{liquid}$), the following formula is employed:

$$\alpha_{liquid} + \alpha_{gas} = 1 \quad 2.17$$

For the numerical implementation, the threshold for gaseous cavitation is considered to lie at 70kPa and the gas initial and fixed mass fraction is set to 6.95 E-5. The distribution of mass fraction around the bearing circumference will vary, based on the local pressure. Figure 2.2 illustrates the coupling between the cavitation module and the flow module.
2.4 The heat transfer module

In CFD-ACE, the heat transfer process in the purely fluid region is computed using the equation for the conservation of energy in fully conservative form. The model can be used to produce the temperature field and energy transfer characteristics of the model. The solution adopted has the form of an enthalpy equation [49]:

Figure 2.2 Flowchart for flow and cavitation modules coupled in CFD-ACE+
\[
\frac{\partial (\rho h_0)}{\partial t} + \nabla \cdot (\rho \vec{V} h_0) = \nabla \cdot (k_{\text{eff}} \nabla T) + \frac{\partial p}{\partial t} + \\
\left[ \frac{\partial (u \tau_{xx})}{\partial x} + \frac{\partial (u \tau_{yx})}{\partial y} + \frac{\partial (u \tau_{zx})}{\partial z} \right] + \\
\left[ \frac{\partial (v \tau_{xy})}{\partial x} + \frac{\partial (v \tau_{yy})}{\partial y} + \frac{\partial (v \tau_{zy})}{\partial z} \right] + \\
\left[ \frac{\partial (w \tau_{xz})}{\partial x} + \frac{\partial (w \tau_{yz})}{\partial y} + \frac{\partial (w \tau_{zz})}{\partial z} \right] + S_k
\]

2.18

The first term is the time transient component, second is the enthalpy flux. On the right hand side, we recognize the conduction term, followed by the compressibility term, heat generation term, and finally the source term.

\[ h_0 \text{ is the total enthalpy defined as } h_0 = i + \frac{p}{\rho} + \frac{1}{2} (u^2 + v^2 + w^2) \]

\[ i \text{ is the internal energy and is function of the state variables } \rho \text{ and } T \]

\[ k_{\text{eff}} \text{ is the effective thermal conductivity of the material.} \]

\[ p \text{ is the static pressure} \]

\[ \tau \text{ is the viscous stress tensor} \]

\[ S_k \text{ is the source term which can contain additional components such as: reactions, radiation, spray, body forces.} \]

Equation 2.20 is discretized using the same procedure as the continuity and the momentum equations; procedure was detailed in section 2.1.
For the porous region, the equation used to model heat transfer, heat generation, and transport is:

\[
\frac{\partial (\varepsilon \rho h_0)}{\partial t} + \nabla \cdot (\varepsilon \rho \vec{V} h_0) = \lambda \cdot q + \varepsilon \varepsilon \cdot \nabla \vec{V} + \varepsilon \frac{\partial p}{\partial t}
\]

2.19

For the porous region we recognize the same terms as for the purely fluid region with the difference that \( \lambda \) is the thermal conductivity of the porous medium which represents and effective thermal conductivity for the pores and solid regions in combination:

\[
\lambda = -2\lambda_S + \frac{1}{\frac{\varepsilon}{2\lambda_S + \lambda_F} + \frac{1-\varepsilon}{3\lambda_S}}
\]

2.20

\( \lambda_S \) is the thermal conductivity of the fluid

\( \lambda_F \) is the thermal conductivity of the solid.

The heat transfer module cannot work in conjunction with the cavitation model presented in the previous section. As a result of this, for the thermal simulations presented in this research a classical Gumbel model for cavitation was used. Figure 2.3 illustrates the coupling between the thermal module and the flow module.
Figure 2.3 Flowchart for flow and thermo-hydrodynamic modules coupled in CFD-ACE+

1. Initialize pressure field $p$
2. Solve momentum equation for $u$
3. Solve momentum equation for $v$
4. Solve momentum equation for $w$
5. Check mass continuity
6. Repeat $N_{ITER}$ times until convergence
7. Solve $p'$
8. Update $u'$, $v'$, $w'$
9. Solve the energy equation for enthalpy $h$
10. Solve the state equations of the fluid
11. Apply Gumbel formulation for cavitation: $p_{min} = p_{cav}$
2.5 The grid deformation module

In CFD-ACE, the grid deformation module is used for moving/deforming boundaries and grid, consequently. The module was used in the chapter regarding the numerical computations of the dynamic coefficients of the bearing.

CFD-ACE+ provides two methods for automatic re-meshing when the boundaries are moving using the grid deformation module: transfinite interpolation (TFI) and solid-body elasticity analogy (SEA). The TFI scheme was chosen because of time economy and good applicability to structured grids. The second method is recommended for more complicated, unstructured grids.

The linear theory for dynamic coefficients assumes small displacement and velocity perturbation in the vicinity of an equilibrium position. For these types of simulations, the grid deformation module was employed to control the deformation of the fluid region due to an small velocity perturbation applied time dependently to a steady state equilibrium solution computed priory.

The module uses a standard transfinite interpolation scheme to re-mesh the domain after the moving of its boundaries. This feature was used for both thermal and isothermal simulations, and does not need other type of boundary conditions. Figure 2.4 illustrates the coupling between the grid deformation module with the flow and thermal module.
Figure 2.4  Flowchart for flow, thermo-hydrodynamic and grid deformation modules coupled in CFD-ACE+
2.6 The geometric model and discretization

The three zones (active, porous, and passive) are discretized using an orthogonal mesh, Figure 2.5. The grid point distribution, produced with the preprocessor CFD-GEOM is non-uniform, with higher density in the clearance region and at the porous/fluid interface, on both the passive and active sides. The number of grid points in the radial direction used for the clearance is 11, and in the porous media 51. In the circumferential direction there are 361 nodes resulting in 1 element for each angular degree. In the axial direction, 51 nodes are considered. The deep reservoir is discretized using 26 nodes in the radial direction for a total number of 1,620,000 cells for this bearing configuration. The shallow reservoir bearing configuration uses 11 nodes in the radial direction resulting in 1,260,000 cells for this bearing entire discretization.

| Radial direction (bearing clearance) | 11 |
| Radial direction (porous region)    | 51 |
| Radial direction (reservoir)        | Deep 26 |
|                                    | Shallow 11 |
| Axial direction                     | 51 |
| Circumferential direction           | 361 |

The surface of the rotor is considered a rigid wall, with an imposed angular velocity and no slip boundary conditions at its surface. The porous media is considered as a rigid volume with isotropic properties. At the interfaces between the clearance and
Figure 2.5 Model geometry for the numerical simulation with orthogonal mesh. Details (b) and (c) show the orthogonal mesh axially and circumferentially respectively, detail (d) is illustrating the clearance region.
the porous medium, and the porous medium and the reservoir, due to the formulation of Equations 2.11 and 2.12, there are no ‘no-slip condition’, but rather continuity of velocities and shear stresses with conservation of mass. The eccentricity of the bearing is a geometric input characteristic.

2.7 The boundary and volume conditions

The boundary conditions used to set up the model are presented in Figure 2.6. The fluid/porous media interface is computationally contiguous with mass, momentum, and energy being conserved across. No special boundary conditions are considered for these interfaces. The boundary type conditions used are: (a) wall, (b) outlet, and (c) symmetry.

(a) A wall boundary type condition is a bounding surface through which there is not flow passing. The shaft is considered as an rotating wall boundary condition having a linear velocity. The angular velocity, expressed in rpm (rotations per minute), is transformed into a tangential velocity. The walls of the enclosure are stationary/ no velocity.

- For isothermal computations, the temperature of the rotating shaft (rotating wall BCs) was set a priory and did not vary during the computations.

- For the thermal simulations, the rotating shaft was considered to be an adiabatic surface. This condition means that there is no heat flux through this boundary and its temperature was allowed to float and be computed by the solver.
For the isothermal simulations, the temperature of the exterior wall of the bearing was set priory and did not vary during the computations.

For the thermal simulations, the exterior wall was considered to have convective heat transfer with the exterior. The heat transfer at the wall was calculated and implemented using the formula: 

\[ q_w = h_c A (T_a - T_w) \]

where \( h_c \) is the external heat transfer coefficient per unit surface and \( T_a \) is the temperature of the exterior ambient (the temperature from the area outside of the computational grid system). By balancing the external and the internal heat flux the temperature of the wall, \( T_w \) was obtained. The procedure for determining \( h_c \) is presented in section 3.3.

(b) The outlet boundary condition is a bounding surface from which the fluid is expected to flow outside of the domain. The outlet boundary condition was considered laterally in the area between the rotating shaft and the porous wall (also known as the bearing clearance) and it was set as a pressure type condition. Its pressure and temperature values corresponded to the atmospheric conditions. The other properties are allowed to adjust themselves according to the solution of the momentum and continuity equations. There are no fundamental differences between the settings for isothermal and thermal model simulations.

(c) The symmetry boundary condition is employed as a mean to shorten the computation domain and hence the computation time. Pressure and velocity fields are give a condition of the form \( \frac{\partial (\rho u)}{\partial x_i} \bigg|_{x=0} = 0 \). For heat transfer purposes, there is no heat allowed to cross the symmetry boundary condition so it effectively behaves as an adiabatic wall.
2.8 Grid and numerical schemes convergence tests

Grid convergence tests were performed to decide the optimum grid density/CPU time usage for both isothermal and thermal models.

The grid convergence test for the isothermal model was performed by increasing the number of grid cells by a factor of 1.5 in each direction (resulting 2.1 mil cells), respectively decreasing the number of grid cells by a factor of 1.5 in each direction for the shallow reservoir configuration (resulting 400k cells). The value of the total load acting on the rotor was chosen as the global variable to judge whether grid convergence was achieved. The different values obtained for the 3 meshes configurations varied by 1.84 to -3.15 % compared to the original discretization (see Table 2.2). The results show
good convergence around the 1.2 mil grid density, leading to choosing this grid density for all computations.

The added advantage of using the reference grid density, rather than the coarser grid, resided with obtaining a better flow resolution and more detailed flow patterns without prohibitive CPU time usage.

Table 2.2 Proof of convergence for forces for isothermal case.

<table>
<thead>
<tr>
<th>Number of grid cells</th>
<th>Total load on the shaft</th>
<th>Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>400k</td>
<td>2.41E+02</td>
<td>-3.15%</td>
</tr>
<tr>
<td>1.2mil</td>
<td>2.49E+02</td>
<td>-</td>
</tr>
<tr>
<td>2.1mil</td>
<td>2.54E+02</td>
<td>1.84%</td>
</tr>
</tbody>
</table>

The grid convergence for the thermal model was performed on various grid configurations and due to the increased complexity of the problem resulted by adding the energy equation, the upwind scheme, the central scheme and second order upwind scheme have been tested.

The values obtained for the tested grid configurations are presented in Table 2.3. Again, the results show a good convergence for the 1.2 mil grid density, and no significant difference between the different order schemes. This result is supported by the discussion on the local Peclet number.

Peclet number (Pe) is relevant to the study of transport phenomena in fluid flows by indicating the complexity of solution models that can be adopted. It represents the ratio of advection to diffusion. In applications where Peclet number is larger than 2
Table 2.3 Proof of convergence for forces and heat fluxes.

<table>
<thead>
<tr>
<th></th>
<th>Load upwind scheme</th>
<th>Load central scheme</th>
<th>Load second order scheme</th>
<th>Heat flux upwind scheme</th>
<th>Heat flux central scheme</th>
<th>Heat flux second order scheme</th>
</tr>
</thead>
<tbody>
<tr>
<td>55k</td>
<td>50.79%</td>
<td>50.94%</td>
<td>50.82%</td>
<td>-25.44%</td>
<td>-25.45%</td>
<td>-25.49%</td>
</tr>
<tr>
<td>400k</td>
<td>22.62%</td>
<td>22.63%</td>
<td>22.61%</td>
<td>-1.19%</td>
<td>-1.19%</td>
<td>-1.22%</td>
</tr>
<tr>
<td>620k</td>
<td>22.62%</td>
<td>22.63%</td>
<td>22.61%</td>
<td>-1.19%</td>
<td>-1.19%</td>
<td>-1.19%</td>
</tr>
<tr>
<td>800k</td>
<td>11.50%</td>
<td>11.51%</td>
<td>11.50%</td>
<td>0.27%</td>
<td>0.24%</td>
<td>0.27%</td>
</tr>
<tr>
<td>1.2mil</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>2.1mil</td>
<td>-1.84%</td>
<td>-1.84%</td>
<td>-1.84%</td>
<td>-1.68%</td>
<td>-1.68%</td>
<td>-1.68%</td>
</tr>
</tbody>
</table>

Simple upwind schemes are pertinent. \( \text{Pe} = \frac{LU}{\alpha} \) where \( L \) is the characteristic length, \( U \) the velocity, \( \alpha \) is the thermal diffusivity \( \alpha = \frac{k}{\rho C_p} \), \( k \) is the thermal conductivity, \( \rho \) is the density of the fluid, and \( C_p \) is the heat capacity. The local Peclet number for the current application was estimated in the smallest grid cell, located in the clearance region. In the rest of the computational domain, the local Peclet number is expected to be of higher value. For \( L = 5.08 \times 10^{-7} \text{m} \), \( U = 15000 \ \text{rpm= 39.9 m/s} \), \( k = 0.15 \ \text{W/m K} \), \( C_p = 1800 \ \text{J/kg K} \), \( \rho = 863 \ \text{kg/m}^3 \) the local Peclet number obtained is \( \text{Pe}=209 \).

In this situation, the dependency of the flow upon downstream locations is diminished, and the variables in the flow tend to become 'one-way' properties. The use of an upwind scheme is justified.
CHAPTER III

NUMERICAL RESULTS

The parameters varied during the simulations are: angular velocity, permeability, porosity, reservoir depth, porous bushing width, bearing concentric clearance, and shaft eccentricity. The results include the flow patterns, pressure maps, velocity maps, temperature maps and attitude angles, and are presented on a parametric basis. Because an optimal configuration would require large fluid circulation through the porous bed (for cooling purposes) combined with a large load carrying capability, an interactive parametric analysis is essential in order to optimize the load carrying capacity versus geometric and operational parameters. The matrices for the numerical experiments performed are illustrated in Table 3.1 and Table 3.2. The properties of the oil used as lubricant have been determined experimentally. The complete procedure of obtaining these properties as a function of the temperature increase is detailed in section 4.3. The geometric and operational inputs, as presented in Table 3.3 and Table 3.4, have an effect on the two most important output quantities that have an inverse relationship: fluid circulation between the active and passive regions and the load carrying capability. For these parametric input values the optimized output variables are the absolute pressure, attitude angle, and fluid circulation between the active (clearance) and passive (reservoir)
spaces. The aspect ratio (L/D) is 1.5. These geometric parameters correspond to a real model prototype tested in our facility.

Table 3.1 Matrix of numerical experiments (angular velocity and permeability are varied)

<table>
<thead>
<tr>
<th>Deep</th>
<th></th>
<th>Shallow</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>k=1E-10m²</td>
<td>2krpm</td>
<td>k=1E-10m²</td>
<td>2krpm</td>
</tr>
<tr>
<td></td>
<td>5krpm</td>
<td></td>
<td>5krpm</td>
</tr>
<tr>
<td></td>
<td>10krpm</td>
<td></td>
<td>10krpm</td>
</tr>
<tr>
<td></td>
<td>15krpm</td>
<td></td>
<td>15krpm</td>
</tr>
<tr>
<td>k=1E-11m²</td>
<td>2krpm</td>
<td>k=1E-11m²</td>
<td>2krpm</td>
</tr>
<tr>
<td></td>
<td>5krpm</td>
<td></td>
<td>5krpm</td>
</tr>
<tr>
<td></td>
<td>10krpm</td>
<td></td>
<td>10krpm</td>
</tr>
<tr>
<td></td>
<td>15krpm</td>
<td></td>
<td>15krpm</td>
</tr>
<tr>
<td>k=1E-12m²</td>
<td>2krpm</td>
<td>k=1E-12m²</td>
<td>2krpm</td>
</tr>
<tr>
<td></td>
<td>5krpm</td>
<td></td>
<td>5krpm</td>
</tr>
<tr>
<td></td>
<td>10krpm</td>
<td></td>
<td>10krpm</td>
</tr>
<tr>
<td></td>
<td>15krpm</td>
<td></td>
<td>15krpm</td>
</tr>
<tr>
<td>k=1E-13m²</td>
<td>2krpm</td>
<td>k=1E-13m²</td>
<td>2krpm</td>
</tr>
<tr>
<td></td>
<td>5krpm</td>
<td></td>
<td>5krpm</td>
</tr>
<tr>
<td></td>
<td>10krpm</td>
<td></td>
<td>10krpm</td>
</tr>
<tr>
<td></td>
<td>15krpm</td>
<td></td>
<td>15krpm</td>
</tr>
</tbody>
</table>
Table 3.2 Matrix of numerical experiments (eccentricity and permeability are varied)

<table>
<thead>
<tr>
<th>Deep</th>
<th>( k=1\times10^{-10}m^2 )</th>
<th>( k=1\times10^{-11}m^2 )</th>
<th>( k=1\times10^{-12}m^2 )</th>
<th>( k=1\times10^{-13}m^2 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>e/c=0.1</td>
<td>e/c=0.1</td>
<td>e/c=0.1</td>
<td>e/c=0.1</td>
<td>e/c=0.1</td>
</tr>
<tr>
<td>e/c=0.2</td>
<td>e/c=0.2</td>
<td>e/c=0.2</td>
<td>e/c=0.2</td>
<td>e/c=0.2</td>
</tr>
<tr>
<td>e/c=0.3</td>
<td>e/c=0.3</td>
<td>e/c=0.3</td>
<td>e/c=0.3</td>
<td>e/c=0.3</td>
</tr>
<tr>
<td>e/c=0.4</td>
<td>e/c=0.4</td>
<td>e/c=0.4</td>
<td>e/c=0.4</td>
<td>e/c=0.4</td>
</tr>
<tr>
<td>e/c=0.5</td>
<td>e/c=0.5</td>
<td>e/c=0.5</td>
<td>e/c=0.5</td>
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</tr>
<tr>
<td>e/c=0.7</td>
<td>e/c=0.7</td>
<td>e/c=0.7</td>
<td>e/c=0.7</td>
<td>e/c=0.7</td>
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<tr>
<td>e/c=0.8</td>
<td>e/c=0.8</td>
<td>e/c=0.8</td>
<td>e/c=0.8</td>
<td>e/c=0.8</td>
</tr>
<tr>
<td>e/c=0.9</td>
<td>e/c=0.9</td>
<td>e/c=0.9</td>
<td>e/c=0.9</td>
<td>e/c=0.9</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Shallow</th>
<th>( k=1\times10^{-10}m^2 )</th>
<th>( k=1\times10^{-11}m^2 )</th>
<th>( k=1\times10^{-12}m^2 )</th>
<th>( k=1\times10^{-13}m^2 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>e/c=0.1</td>
<td>e/c=0.1</td>
<td>e/c=0.1</td>
<td>e/c=0.1</td>
<td>e/c=0.1</td>
</tr>
<tr>
<td>e/c=0.2</td>
<td>e/c=0.2</td>
<td>e/c=0.2</td>
<td>e/c=0.2</td>
<td>e/c=0.2</td>
</tr>
<tr>
<td>e/c=0.3</td>
<td>e/c=0.3</td>
<td>e/c=0.3</td>
<td>e/c=0.3</td>
<td>e/c=0.3</td>
</tr>
<tr>
<td>e/c=0.4</td>
<td>e/c=0.4</td>
<td>e/c=0.4</td>
<td>e/c=0.4</td>
<td>e/c=0.4</td>
</tr>
<tr>
<td>e/c=0.5</td>
<td>e/c=0.5</td>
<td>e/c=0.5</td>
<td>e/c=0.5</td>
<td>e/c=0.5</td>
</tr>
<tr>
<td>e/c=0.7</td>
<td>e/c=0.7</td>
<td>e/c=0.7</td>
<td>e/c=0.7</td>
<td>e/c=0.7</td>
</tr>
<tr>
<td>e/c=0.8</td>
<td>e/c=0.8</td>
<td>e/c=0.8</td>
<td>e/c=0.8</td>
<td>e/c=0.8</td>
</tr>
<tr>
<td>e/c=0.9</td>
<td>e/c=0.9</td>
<td>e/c=0.9</td>
<td>e/c=0.9</td>
<td>e/c=0.9</td>
</tr>
</tbody>
</table>
### Table 3.3 Operating parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dynamic Viscosity of the lubricating oil, $\mu$</td>
<td>0.01097 kg/m·s (10 cP, light oil)</td>
</tr>
<tr>
<td>Density of the lubricating oil, $\rho$</td>
<td>818.718 kg/m³, (51.11 lb/ft³)</td>
</tr>
<tr>
<td>Rotational speed</td>
<td>1, 2, 10, 15 (krpm)</td>
</tr>
<tr>
<td>The gaseous cavitation pressure threshold</td>
<td>70 kPa (9.8 Psi)</td>
</tr>
<tr>
<td>The outlet pressure (atmospheric conditions)</td>
<td>100 kPa (14.7 Psi)</td>
</tr>
</tbody>
</table>

### Table 3.4 Geometric parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shaft diameter, $D$</td>
<td>24.8412 mm (0.996 in)</td>
</tr>
<tr>
<td>Length of the bearing, $L$</td>
<td>38.1 mm (1.5 in)</td>
</tr>
<tr>
<td>Concentric clearance, $c$</td>
<td>0.0508 mm (0.002 in)</td>
</tr>
<tr>
<td>Depth of the re-circulating reservoir (Figure 1.4, (3))</td>
<td>Deep: 3.175mm (0.125 in)</td>
</tr>
<tr>
<td>Shaft eccentricities, ($e/c$)</td>
<td>0.1 to 0.9</td>
</tr>
<tr>
<td>Width of the porous media</td>
<td>3.175 mm (0.125 in)</td>
</tr>
<tr>
<td>Porosity of the porous insert, $\varepsilon$</td>
<td>0.1, 0.2, 0.3, 0.4, 0.5</td>
</tr>
<tr>
<td>Permeability of the porous insert, $k$</td>
<td>$10^{-10}$, $10^{-11}$, $10^{-12}$, $10^{-13}$ (m²)</td>
</tr>
</tbody>
</table>
3.1 Isothermal hydrodynamic model

The Reynolds number based on the bearing concentric clearance is \( Re_c = \frac{\rho (\omega R) c}{\mu} \), and is varied from 9.67 to 72.55 as the rotational velocity varies between 2 and 15 krpm. According to Szeri [61], a clearance based Reynolds number \( Re_c < 2000 \) indicates a laminar operational regime.

The velocity field presented in Figure 3.1 is obtained for a deep reservoir 3.175 mm (0.125 in) configuration with the shaft rotating counterclockwise at 15 krpm (\( Re_c = 72.55 \)). The permeability of the porous media is 1E-11 m² and the porosity is 0.5. The direction of the velocity vectors clearly supports the operational logic described in Figure 1.4. The fluid is leaving the high pressure zone of the clearance (Detail (B)), and after travelling through the reservoir reenters the low pressure zone (Detail (D)). Zone (A) and (C) are zones of mixed fluid motion, that is, these zones are marked at intersection of regions where flow changes directions of motion from (clearance ➔ porous and porous ➔ reservoir) to (reservoir ➔ porous and porous ➔ clearance). This mechanism also allows the fluid to exchange heat with the environment contiguous to the outer boundary of the reservoir and thus cool and reject the heat picked up while in the active space of the bearing. The exact same process can be concluded to be active for the case of the shallow reservoir configuration, see Figure 3.4.
Figure 3.1 Normalized vector field for a deep reservoir configuration; eccentricity \( e/c = 0.5 \); porosity \( \varepsilon = 0.5 \); permeability \( k = 1 \times 10^{-11} \text{m}^2 \); \( \text{Re}_c = 72.55 \text{ (15 krpm)} \) (see details Figure 3.2).
Figure 3.2 Details of flow in the regions A, B, C, D of Figure 3.1
Detail (A) -> Flow in the region where the divergent zone ends and the convergent zone begins; Detail (B) -> Flow from reservoir into the porous medium (midway in the convergent zone); Detail (C) -> Flow in the region where the convergent zone ends and the divergent zone begins; Detail (D) -> Flow from porous medium into the reservoir (midway in the convergent zone)
Figure 3.3 Additional details B’, B”, D’, D” of flow in the regions A, B, C, D of Figure 3.2
Figure 3.4 Normalized vector field for a shallow reservoir configuration; eccentricity \((e/c)=0.5\); porosity \((\varepsilon)=0.5\); permeability \((k)=1\times10^{-11}m^2\); \(Re_c=72.55\) (15 krpm); (see details Figure 3.5)
Figure 3.5 Details of flow in the regions A, B, C, D of Figure 3.4 with additional details B', B'', D', D''
Figure 3.6 Additional details B’, B’”, D’, D’” of flow in the regions A, B, C, D of Figure 3.5
3.1.1 The Influence of rotational speed

Figure 3.7 presents the pressure distribution for the deep reservoir case, for permeabilities of 1E-10 m$^2$ and 1E-11 m$^2$, and speeds varying from 2 to 15 kRPM. Porosity and eccentricity are kept constant at $\varepsilon = 0.5$ and $e/c=0.5$. Figure 3.8 presents the pressure profiles for the same deep reservoir case, for permeabilities of 1E-12 m$^2$ and 1E-13 m$^2$ and speeds varying from 2 to 15 kRPM.

In each of these figures for constant porosity, permeability, and eccentricity the pressure build-up increases with angular velocity. As permeability is decreased from 1E-10 m$^2$ in Figure 3.7 to 1E-13 m$^2$ in Figure 3.8, for the same velocity, the pressure in the convergent zone continue increasing and reaches a maximum pressure for the lowest permeability. The introduction of the Singhal [50] cavitation model limits the pressure drops in the divergent zone to the partial pressure of the gas released, Nitrogen at 70kPa. When the liquid pressure falls below the saturation pressure, the gases dissolved in the lubricant are being released. Each component has its own partial pressure, and the Nitrogen is the first one to be released.

It can be easily observed that the decrease in permeability is instrumental in increasing the absolute pressure in the active zone.
Figure 3.7 Pressure distribution on the shaft at the axial plane of symmetry for a deep reservoir configuration; eccentricity (e/c) = 0.5; porosity (ε) = 0.5; concentric clearance = 0.002 in; Re_c = 9.67 (2 krpm); 24.18 (5 krpm); 48.36 (10 krpm); 72.55 (15 krpm), permeability $k=1\times 10^{-10} \text{m}^2$, $1\times 10^{-11} \text{m}^2$-isothermal model
Figure 3.8 Pressure distribution on the shaft at the axial plane of symmetry for a deep reservoir configuration; eccentricity (e/c)= 0.5; porosity (ε)= 0.5; concentric clearance = 0.002 in; Reₚ=9.67 (2 krpm); 24.18 (5krpm); 48.36 (10 krpm); 72.55 (15 krpm), permeability $k=1E-12 m^2$, $1E-13 m^2$ - isothermal model
Figure 3.9 Pressure distribution on the shaft at the axial plane of symmetry for a shallow reservoir configuration; eccentricity (e/c)= 0.5; porosity (ε)= 0.5; concentric clearance = 0.002 in; $Re_c$=9.67 (2 krpm); 24.18 (5 krpm); 48.36 (10 krpm); 72.55(15 krpm), permeability $k=1E-10m^2$, $1E-11m^2$ - isothermal model
Figure 3.10 Pressure distribution on the shaft at the axial plane of symmetry for a shallow reservoir configuration; eccentricity \((e/c) = 0.5\); porosity \((\varepsilon) = 0.5\); concentric clearance = 0.002 in; \(Re_c\) = 9.67 (2 krpm); 24.18 (5 krpm); 48.36 (10 krpm); 72.55 (15 krpm), permeability \(k = 1 \times 10^{-12} m^2\), \(1 \times 10^{-13} m^2\) - isothermal model
The effect of the shallow reservoir (relative to the deep one) upon the pressure magnitudes is noteworthy and is presented on a comparative basis in Figure 3.9 and Figure 3.10. For the highest permeability studied (1E-10 m$^2$), Figure 3.7 (a) and Figure 3.9 (a) the effect of the reservoir depth is quite pronounced, as the maximum pressure increases from 107 kPa in the deep reservoir case to 126 kPa for the shallower reservoir. All other parameters being the same, the pressure magnitudes, at all velocities are higher in Figure 3.9 (a) compared to Figure 3.7 (a) due to the shallowness of the reservoir. Cavitation pressure levels (70kPa) are not reached for these two cases.

As permeability is decreased from 1E-10 m$^2$ to 1E-11 m$^2$ while keeping eccentricity and porosity constant, the same physical trends are noticed in the comparative study of Figure 3.7(b) and Figure 3.9(b). The difference resides in the maximum values that the pressures are reaching when velocities are increased. Thus, the maximum pressure for the shallow reservoir reaches 180 kPa as compared with the equivalent case of the deep reservoir (maximum pressure, 165 kPa). When these results are compared to those of the pair Figure 3.7(a) and Figure 3.9(a), it is obvious that the decrease in permeability and reservoir depth both have the effect of increasing the pressures in the active space. However, as the permeability is further decreased to 1E-12 m$^2$ and 1E-13 m$^2$ respectively, the effect of reservoir depth appears to become irrelevant, its role appearing to be preempted by the decrease of bushing’s permeability, Figure 3.8, Figure 3.10. For the case when the permeability is 1E-13 m$^2$, the porous media acts almost like a solid boundary.
Figure 3.11 Load carrying capability function of angular velocity for a deep reservoir configuration; eccentricity (e/c)=0.5; porosity (\( \varepsilon \))= 0.5, permeabilities (\( k \)) ranging from 1E-10 m\(^2\) to 1E-13 m\(^2\)-isothermal model;

Figure 3.12 Load carrying capability function of angular velocity for a shallow reservoir configuration; eccentricity (e/c)=0.5; porosity (\( \varepsilon \))= 0.5, permeabilities (\( k \)) ranging from 1E-10 m\(^2\) to 1E-13 m\(^2\)-isothermal model.
The load carrying capability is obtained by integrating the pressure distribution on the surface of the shaft. One has to keep in mind that the computations have been performed on a half domain, using the symmetry boundary conditions, therefore the result of the integration needs to be multiplied by a factor of two to give the total load of the actual bearing. The values for loads have been presented as a function of the running angular velocity in Figure 3.11 for the deeper reservoir configuration and Figure 3.12 for the shallower reservoir configuration. For these geometric configurations, the load capacity does vary for large permeabilities of $1 \times 10^{-10}$ m$^2$ and $1 \times 10^{-11}$ m$^2$. For the low permeabilities cases, of $1 \times 10^{-12}$ m$^2$ and $1 \times 10^{-13}$ m$^2$, the bearing is supporting the same amount of load regardless of its reservoir depth. It should be noted, that without means of increasing the load (such as controlled permeability of the bushing or the use of spiral groove seals), the load capacity of the porous bearing is generally “low”.

For higher permeability of $1 \times 10^{-10}$ m$^2$ and shallow reservoir configuration the maximum carrying load supported by the configuration is 36N (3.6 kg), while the maximum obtained for the small permeability of $1 \times 10^{-13}$ m$^2$ is 300N (30 kg).

Figure 3.13 and Figure 3.14 present for both deep and shallow reservoir configuration the magnitude (color scale) of the pressure distribution throughout the clearance/porous/reservoir assembly in the axial symmetry plane, in both the circumferential and radial directions. This information complements the 1-D pressure curves presented in Figure 3.7, Figure 3.8, Figure 3.9 and Figure 3.10 for the angular velocity of 15 krpm. Figure 3.13 (a) and Figure 3.14 (a) confirm that the higher permeability for the deep reservoir translates into a lower pressure in the active region compared to the shallow reservoir and significant difference in pressure distribution both
radially and circumferentially. Most interestingly, one can follow the development and penetration of the pressure field radially into the porous medium. The radial pressure gradient is indicated by the change in color scale and confirms the radial flow out of the active space and into the reservoir over most of the circumferential length of the convergent zone. Both these cases are free of cavitation, which appears to be caused by the higher permeability. Figure 3.13 (c), (d) and Figure 3.14 (c) and (d) present information for the lower permeabilities of 1E-12m² and 1E-13m² respectively. They reinforce the findings from Figure 3.8 and Figure 3.10 where it was shown that as the permeability decreases, the pressure increases. This is due to the fact that less fluid is being allowed to leaves the active zone. It is noteworthy that this increase in pressure is accompanied by a much shorter circumferential zone of high radial pressure gradient \( \frac{p_{1c}-p_{2c}}{\Delta r} > 0 \); this shorter zone is also responsible for the lesser amount of fluid that leaves the active zone.

At the low permeabilities, if one compares Figure 3.13 (c), (d) and Figure 3.14 (c), (d), there seems to be very little difference in the pressure patterns between the shallow and the deep reservoirs cases. Again, the explanation resides with the reduced amount of fluid allowed out of the active space and into the reservoir, fact that makes irrelevant the depth of the reservoir wrapped around the bearing.
Figure 3.13 Circumferential-radial pressure fields in the plane of symmetry for deep reservoir configuration with permeabilities ($k$) ranging from $1E-10$ m$^2$ to $1E-13$ m$^2$; eccentricity (e/c)= 0.5; c=0.002 in; porosity ($\varepsilon$)= 0.5; Re$_c$=72.55 (15 krpm) -isothermal model.
Figure 3.14 Circumferential-radial pressure fields in the plane of symmetry for shallow reservoir configuration with permeabilities \( k \) ranging from \( 1 \times 10^{-10} \text{ m}^2 \) to \( 1 \times 10^{-13} \text{ m}^2 \); eccentricity \( e/c = 0.5 \); \( c = 0.002 \text{ in} \); porosity \( \varepsilon \) = 0.5; \( \text{Re}_c = 72.55 \) (15 krpm) - isothermal model.
3.1.2 The Influence of eccentricity ratio

Figure 3.15 through Figure 3.18 present the effect of the bearing shaft eccentricity variations on the pressure distribution. It is noteworthy the fact that the high permeability case of 1E-10 m² is functioning cavitation free for all configurations. The reverse of this desirable phenomenon is the very little pressure built up, which translates into very little load carrying capability.

Fundamentally, when changing the eccentricity ratio, the velocity gradient in the film is changed. Since the shear between the layers of the lubricant film is the main mechanism for pressure generation, when decreasing this gap, the velocity gradient between the velocity of the shaft and the small velocity value at the entrance in the porous media is increased. The increase of shaft eccentricity is resulting consistently into an increase in pick pressures values for all reservoir configurations and porous bushing permeability configurations proposed.

It is documented in the literature that experimental runs on regular porous bearings suggest that they function at very high eccentricities. This is due to the fact that the porous media is allowing the fluid film to migrate instead of being “squeezed”. From the study of Figure 3.15 one can conclude that for low permeabilities of 1E-10m² the levels for cavitation to occur are not met for the deep reservoir case. For the shallow reservoir case one can notice a slightly small influence mostly for higher eccentricities. Decreasing the permeability to 1E-11m², Figure 3.16 and further, results in large areas of film rupture.
Figure 3.15 Pressure distribution on the shaft at the axial plane of symmetry when the shaft eccentricity is changed parametrically for a deep reservoir; porosity ($\varepsilon$) = 0.3; concentric clearance = 0.002 in; $Re_c := 72.55$ (15 krpm), permeability ($k$) = $1E-10 m^2$-isothermal model.

Figure 3.16 Pressure distribution on the shaft at the axial plane of symmetry when the shaft eccentricity is changed parametrically for a deep reservoir; porosity ($\varepsilon$) = 0.3; concentric clearance = 0.002 in; $Re_c := 72.55$ (15 krpm), permeability ($k$) = $1E-11 m^2$-isothermal model.
Figure 3.17 Pressure distribution on the shaft at the axial plane of symmetry when the shaft eccentricity is changed parametrically for a deep reservoir; porosity ($\varepsilon$) = 0.3; concentric clearance = 0.002 in; $Re_c = 72.55$ (15 krpm), permeability ($k$) = $1E-12m^2$-isothermal model.

Figure 3.18 Pressure distribution on the shaft at the axial plane of symmetry when the shaft eccentricity is changed parametrically for a deep reservoir; porosity ($\varepsilon$) = 0.3; concentric clearance = 0.002 in; $Re_c = 72.55$ (15 krpm), permeability ($k$) = $1E-13m^2$-isothermal model.
Figure 3.19 Pressure distribution on the shaft at the axial plane of symmetry when the shaft eccentricity is changed parametrically for a shallow reservoir; porosity ($\varepsilon$) = 0.3; concentric clearance = 0.002 in; $Re_c = 72.55$ (15 krpm), permeability ($k$) = $1E-10m^2$ - isothermal model

Figure 3.20 Pressure distribution on the shaft at the axial plane of symmetry when the shaft eccentricity is changed parametrically for a shallow reservoir; porosity ($\varepsilon$) = 0.3; concentric clearance = 0.002 in; $Re_c = 72.55$ (15 krpm), permeability ($k$) = $1E-11m^2$ - isothermal model
Figure 3.21 Pressure distribution on the shaft at the axial plane of symmetry when the shaft eccentricity is changed parametrically for a shallow reservoir; porosity ($\varepsilon$)= 0.3; concentric clearance = 0.002 in; $Re_c := 72.55$ (15krpm), permeability ($k$)=1E-12m$^2$-isothermal model

Figure 3.22 Pressure distribution on the shaft at the axial plane of symmetry when the shaft eccentricity is changed parametrically for a shallow reservoir; porosity ($\varepsilon$)= 0.3; concentric clearance = 0.002 in; $Re_c := 72.55$ (15krpm), permeability ($k$)=1E-13m$^2$-isothermal model
The pressure profiles are further integrated and the load carrying capability is obtained when the eccentricity ratio is varied parametrically (Figure 3.23 and Figure 3.24). The dependence is linear for low permeabilities of $1\times10^{-10} \text{ m}^2$ and $1\times10^{-11} \text{ m}^2$. Overall, the shallow reservoir case produced higher load capability when compared to the deep reservoir case. This result is closely related to the observation that the clearance-porous-reservoir system is behaving as a “virtual clearance”. In the shallow reservoir case, this clearance is smaller; therefore, the load carrying capability generated is higher.

For low permeabilities cases, $1\times10^{-12} \text{ m}^2$ and $1\times10^{-13} \text{ m}^2$, the bearing is supporting the same amount of load regardless of its reservoir depth, and the dependence is exponential. The maximum load that the geometry proposed can carry for eccentricity of 0.9 is 1600N (160kg) for a permeability of $1\times10^{-13} \text{ m}^2$ and 400N (40 kg) for permeability of $1\times10^{-12} \text{ m}^2$. 
Figure 3.23  Load carrying capability for a deep reservoir configuration when the
shaft eccentricity is changed parametrically; porosity(ε)= 0.3, permeabilities (k) ranging
from 1E-10 m² to 1E-13 m²; eccentricities (e/c) ranging from 0.1 to 0.9,
Rec=72.55(15krpm); -isothermal model

Figure 3.24  Load carrying capability for a shallow reservoir configuration when
the shaft eccentricity is changed parametrically; porosity(ε)= 0.3, permeabilities (k)
ranging from 1E-10 m² to 1E-13 m², eccentricities (e/c) ranging from 0.1 to 0.9,
Re_c=72.55(15krpm) -isothermal model
3.1.3 The influence of porous bushing porosity

Figure 3.25 presents a comparison between pressure profiles in deep (a), versus shallow (b) reservoir configurations as porosity is changed parametrically (typical values for actual materials vary between 0.1 and 0.5), while permeability (1E-11 m²), angular velocity (10 krpm) and eccentricity (0.9) are the kept constant. At constant permeability an increase in porosity has the same effect as an increase in permeability at constant porosity since it makes it easier for the fluid to ‘permeate’ in and out of the porous material. For purpose of generalization, these two properties are considered unrelated in the present study, so that their effects can be studied in isolation.

An increase in porosity resulted in a decrease in the resistance encountered by the fluid while passing throw the porous media and hence a decrease of the pressure build up, effect observed for both configurations. The increase in porosity makes the active space act as if the passive space is part of it, that is, there is a formation of an equivalent ‘virtual clearance’ that incorporates both spaces; this, in turn, contributes to a significant reduction in the pressure buildup. As porosity $\epsilon \to 1$ and permeability $k \to \infty$, certainly there is clarity that the active and passive spaces become undistinguishable from each other. In as far as the effect of the homogeneous cavitation model, it limits the pressure low side to the cavitation pressure (gas partial pressure), while the actual extent of the cavity zone is a function of the initial mass of gas, $f_g$, considered to be present in the oil at the beginning of the computation.
Figure 3.25 Pressure distribution on the shaft at the axial plane of symmetry when the porosity of the bushing is changed parametrically; concentric clearance = 0.002 in; $Re_c = 48.36$ (10 krpm), permeability $(k) = 1E-11 \text{m}^2$-isothermal model
3.1.4 The influence of porous bushing permeability

Figure 3.26 presents the effect of permeability change under constant porosity ($\varepsilon = 0.3$) and angular velocity (15 krpm). The effect of eccentricity is also factored in by changing it from 0.5 to 0.9. For the deep reservoir, Figure 3.26(a) and (b), it is seen that pressure increases as permeability decreases when all other parameters are kept constant. For the shallow reservoir, Figure 3.27 (a) and (b) the same trends are in effect. As it was concluded from Figure 3.8 and Figure 3.10, at very low permeability $1E-12 \text{ m}^2$ to $1E-13 \text{ m}^2$ the depth of the reservoir is not a factor in pressure variation any longer, since the porous bushing allows very limited flow circulation and is majorly responsible for the increase in pressure. This conclusion is supported by a visual inspection and comparison of the pair of Figure 3.26 and Figure 3.27 respectively, showing that at low permeability both the deep and shallow reservoirs exhibit the same pressure magnitudes.

As expected, for the same permeability, but with an increase in eccentricity, pressures also will increase. Thus, in Figure 3.26(a), for an eccentricity of 0.5, permeability $1E-13 \text{ m}^2$ and angular speed of 15 krpm the maximum pressure reaches values of 870 kPa, while in Figure 3.26(b), by comparison, when eccentricity has been increased to 0.9, while keeping all other parameters unchanged, a maximum of 4600 kPa was obtained. On further comparison with Figure 3.27 (a) and (b) it shows that for both reservoirs configuration, (shallow or deep) the eccentricity has the effect of raising the pressure.
Figure 3.26 Pressure distribution on the shaft at the axial plane of symmetry when the permeability of the bushing is changed parametrically for a deep reservoir; eccentricity (e/c) = 0.5 and 0.9; porosity (ε) = 0.5; Re_c = 72.55 (15 krpm) - isothermal model
Figure 3.27 Pressure distribution on the shaft at the axial plane of symmetry when the permeability of the bushing is changed parametrically for a shallow reservoir; eccentricity (e/c) = 0.5 and 0.9; porosity (ε) = 0.5; Rec = 72.55 (15 krpm) - isothermal model
3.1.5 The influence of reservoir depth

Figure 3.28 presents pressure profiles in the active space (clearance) and passive space (reservoir) superimposed, for the cases of the deep (Figure 3.28(a)) and shallow (Figure 3.28(b)) reservoirs respectively. In Figure 3.28 (a), the shaft is subject to an eccentricity of 0.5 and angular velocity of 10 krpm. The pressure curve reaches a maximum value of 165 kPa in the convergent zone, while the divergent zone is, for most part, cavitating. This profile is reminiscent of the Half- Sommerfeld curve, with the exception that a variable pressure is calculated in the divergent zone. The pressure profile in the deep reservoir is approximately constant at 108 kPa. The relationship between the two computed profiles verifies the originally advanced concept (Figure 1.5): fluid circulating out of the convergent zone into the reservoir \( \Delta p = p_{1c} - p_{2c} > 0 \), and returning into the active zone in the divergent zone: \( \Delta p = p_{2d} - p_{1d} > 0 \). The arrows on the figure indicate the flow direction. These pressure curves are consistent with and confirm the flow vector patterns presented in Figure 3.1 and Figure 3.2 which visualize the flow out of the active space and into the reservoir passive space and vice versa.
Figure 3.28 Pressure distribution on the shaft at the axial plane of symmetry when the depth of the reservoir is changed parametrically; eccentricity (e/c) = 0.5; porosity (ε) = 0.5; permeability (k) = 1E-11 m²; Re_c = 72.55(15krpm) - isothermal model
Figure 3.28 (b) represents the cases of the shallower reservoirs: the 0.01in reservoir case that was considered throw the entire study as shallow reservoir and an even smaller case of 0.005in as illustrated in the graphs legend. While the circulatory concept proven in Figure 3.28 (a) is reconfirmed here, there are both qualitative and quantitative differences due to the reservoir shallowness: (i) the pressure profile in the reservoir follows the “ups” and “downs” of those in the active space. In fact the shallower the reservoir, the more alike are the pressure in the active and passive spaces respectively; (ii) the magnitudes of the pressure are higher than those shown in Figure 3.28 (a). For a constant parameters, as the reservoir becomes shallower and shallower the difference Δp between the active and the passive space decreases. This is consistent with the discussion of Figure 1.5 where the notion of a ‘virtual clearance’ (clearance + depth of reservoir) was introduced. As the ‘virtual clearance’ becomes smaller, due to the decrease in the reservoir depth the pressures will go higher: the pressures both in the active space and passive spaces get enhanced. As the pressure in the reservoir gets larger, the resulting smaller Δp between the active and the passive space, for the same operating conditions (eccentricity, permeability, angular velocity, porosity) translate into lower flow circulation between the two regions.
3.1.6 The influence of the porous bushing thickness

Figure 3.29 and Figure 3.30 present the variation of pressure when the thickness of the porous bushing is varied. The increase in the thickness of the porous bushing translated into a higher resistance to flow for the fluid to overcome when travelling radially. The fluid will have to build higher pressure when the porous bushing is thicker for it to be able to have sufficiently high pressure gradient needed to pass thought the porous matrix barrier.

The increase in the porous bushing thickness has the same effect on the maximum pressures generated as the other parameters that influence the flow resistance thought the porous media: permeability and porosity. The results of these influences have been discussed at large in Sections 3.1.3 and 3.1.4. Is not by accident that all these three parameters are contained in the Darcy law, as they related the flow resistance with the pressure drop across a porous matrix.

The numerical simulations were computed for a shallow reservoir case for both low 1E-11 m$^2$ and 1E-13 m$^2$ high permeability, with the same findings: increase in the maximum pressure generated when the porous bushing thickness was increased.
Figure 3.29 Pressure distribution on the shaft at the axial plane of symmetry when the porous bushing thickness is changed parametrically for a shallow reservoir; eccentricity \((e/c)=0.9\); permeability\((k) = 1E-11 \, \text{m}^2\); porosity\((\varepsilon)=0.5\); concentric clearance\((c)=0.002\); \(\text{Re}_c=72.55\) (15krpm) - isothermal model

Figure 3.30 Pressure distribution on the shaft at the axial plane of symmetry when the porous bushing thickness is changed parametrically for a shallow reservoir; eccentricity \((e/c)=0.9\); permeability\((k)= 1E-13 \, \text{m}^2\); porosity\((\varepsilon)=0.5\); concentric clearance\((c)=0.002\); \(\text{Re}_c=72.55\) (15krpm) - isothermal model.
3.1.7 The influence of the concentric clearance of the bearing

In all previous calculations, the concentric clearance was set to a value of 5.08E-5m (0.002 inches). Therefore, the pressure magnitude was the interplay between the resistance of the fluid being pushed into the porous region and into the reservoir and the circumferential resistance of the flow. The latter is the main load carrying capability developer. Keeping the concentric clearance and eccentricity of the shaft at a constant value meant that the circumferential resistance was constant. When varying the concentric clearance and depending on the value of the porous bushing permeability, one can observe an interesting trend.

Figure 3.31 corresponds to a permeability of 1E-11 m², and in this case, for a gradual increase in the concentric clearance a gradual increase in pressure magnitudes was observed and not intuitively expected. Figure 3.32 corresponds to a permeability of 1E-12 m², and in this case, for a gradual increase in the concentric clearance, and up to a value of 0.003 inches a gradual increase in the pressure magnitude was observed. After passing this threshold, a further increase will result in lower pressure build-up. Figure 3.33 corresponds to a permeability of 1E-13 m², and in this case, for a gradual increase in the concentric clearance up to a value of 0.002 inches a gradual increase in pressure magnitudes was observed. After passing this threshold towards 0.001 and 0.0005, lower pressure build-up is noted. This characteristic behavior past the threshold is closer to that of a regular journal bearing.
Figure 3.31 Pressure distribution on the shaft at the axial plane of symmetry when the concentric clearance is changed parametrically for a shallow reservoir; eccentricity \( \frac{e}{c} = 0.9 \); permeability \( k = 1 \times 10^{-11} \text{ m}^2 \); porosity \( \varepsilon = 0.5 \); \( \text{Re}_c = 72.55 \) (15 krpm) - isothermal model.

Figure 3.32 Pressure distribution on the shaft at the axial plane of symmetry when the concentric clearance is changed parametrically for a shallow reservoir; eccentricity \( \frac{e}{c} = 0.9 \); permeability \( k = 1 \times 10^{-12} \text{ m}^2 \); porosity \( \varepsilon = 0.5 \); \( \text{Re}_c = 72.55 \) (15 krpm) - isothermal model.
Figure 3.33 Pressure distribution on the shaft at the axial plane of symmetry when the concentric clearance is changed parametrically for a shallow reservoir; eccentricity (e/c)=0.9; permeability(k)= 1E-13 m²; porosity (ε)=0.5; Reₖ:= 72.55 (15 krpm) - isothermal model

![Pressure Distribution Diagram]

Figure 3.34 Radial (Porous), circumferential and axial mass flow schematics for a control volume (Q₁-total mass debit entering the control volume, Qₑ-mass debit going circumferentially, Qₐ-mass debit going axially, Qₚ-mass debit going into the porous)
The explanation resides with the relationship between the hydraulic resistance to
the axial flow, the hydraulic resistance to the flow into the porous media and finally the
hydraulic resistance of the circumferential flow. The complex relation between these
three resistances is controlling the pressure generation mechanism. The flow resistance is
the factor that subsequently governs the mass flow of the lubricant in that direction. The
pressure loses and the fluid shear are the two factors contributing to the pressure
magnitude.

The mass flow in the axial direction can be approximated using the theory of flow
between parallel flat plates as \( Q_a = \frac{2h^3}{3\mu l} \left( -\frac{dp}{dx} \right) \). Therefore the axial hydraulic resistance
can be considered as \( R_a = \frac{3\mu l}{2h^3} \). When the concentric clearance (represented by the
parameter \( h \) in this formulas) is increased, the axial resistance is decreased translating
into higher pressure losses. In addition, the concentric clearance increase resulted into a
slightly smaller velocity gradient, hence smaller pressure generation. Once can conclude
that both parameters at play (axial resistance and velocity gradient) act in the direction of
decreasing the pressure with the increase of concentric clearance.

Furthermore, the mass flow in the radial direction can be approximated roughly
using the Darcy law \( Q_r = \frac{k}{\mu} \left( -\frac{dp}{dx} \right) \). The radial hydraulic resistance in this case was
estimated as \( R_p = \frac{\mu}{k} \). One can clearly see that the concentric clearance variation does not
directly contribute to the resistance of the flow in the radial direction.

An investigation into the mass flow percentages for axial, circumferential and
radial (into the porous) flow for a control volume located in the convergent part of the
bearing (Figure 3.34) is further conducted in order to explain the trends in Figure 3.31,
Figure 3.32 and Figure 3.33. Table 3.5 through Table 3.7 support the logic that an increase in the concentric clearance translated to more fluid being “leaked” axially. In addition, an important observation is that with the increase in the concentric clearance the fluid flow into the porous matrix is decreasing; therefore, more fluid remains in the clearance and contributes to the pressure generation. It is obvious that competing factors are at play in generating pressure, and the prevailing ones depend on the bearing parameters.

Table 3.5 Mass flow for a control volume for permeability 1E-11 m²

<table>
<thead>
<tr>
<th>cc</th>
<th>Qt</th>
<th>Qc</th>
<th>Qa</th>
<th>Qp</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.01</td>
<td>100%</td>
<td>18.55%</td>
<td>27.10%</td>
<td>54.34%</td>
</tr>
<tr>
<td>0.002</td>
<td>100%</td>
<td>12.26%</td>
<td>6.55%</td>
<td>81.20%</td>
</tr>
<tr>
<td>0.0005</td>
<td>100%</td>
<td>13.39%</td>
<td>0.42%</td>
<td>86.19%</td>
</tr>
</tbody>
</table>

Table 3.6 Mass flow for a control volume for permeability 1E-12 m²

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<th>Qc</th>
<th>Qa</th>
<th>Qp</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.01</td>
<td>100%</td>
<td>26.61%</td>
<td>50.25%</td>
<td>23.15%</td>
</tr>
<tr>
<td>0.003</td>
<td>100%</td>
<td>14.36%</td>
<td>23.03%</td>
<td>62.61%</td>
</tr>
<tr>
<td>0.002</td>
<td>100%</td>
<td>13.19%</td>
<td>14.16%</td>
<td>72.64%</td>
</tr>
</tbody>
</table>

Table 3.7 Mass flow for a control volume for permeability 1E-13 m²

<table>
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<th>Qc</th>
<th>Qa</th>
<th>Qp</th>
</tr>
</thead>
<tbody>
<tr>
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<td>100%</td>
<td>33.71%</td>
<td>62.55%</td>
<td>3.75%</td>
</tr>
<tr>
<td>0.003</td>
<td>100%</td>
<td>20.92%</td>
<td>46.52%</td>
<td>32.55%</td>
</tr>
<tr>
<td>0.002</td>
<td>100%</td>
<td>17.73%</td>
<td>33.38%</td>
<td>48.90%</td>
</tr>
</tbody>
</table>
3.1.8 The correlations between the velocity fields and pressure fields

Figure 3.35 presents the circumferential-radial flow in the active and passive spaces of a deep reservoir configuration. As the aim is to visualize the flow field, all velocity vectors are represented by equal size vectors. The superimposed pressure field allows correlation of the pressure gradients Δp described in Figure 3.28 (a) with the flow direction. Section (a-a) is a location in the convergent region where flow comes into the reservoir (similar to location of Detail (B) in Figure 3.1 and Figure 3.2). The location of the (a-a) line corresponds to the maximum in the pressure curve shown in the insert figure (see Figure 3.28 (a)). The location of lines (b’-b’) and (b’’-b’’) correspond to the regions where the pressure in the clearance and pressure in the reservoir are equal, hence the pressure difference necessary to make the fluid pass between the two regions is deficient. These are the so called ‘no-flow lines’ and they are visible in Details (A) and (C) of Figure 3.1 and Figure 3.2. In these regions, while there is fluid circulating in the reservoir, the fluid in the porous region is either radially stagnating or slowly changing direction of flow. Section (c-c) is located in the divergent region (see also Figure 3.1 and Figure 3.2, Detail (D)), where the pressure gradient is inverted making possible for the fluid to re-enter the bearing active space. The pressure color map indicates a maximum pressure differential (Figure 1.5) $p_{1c}-p_{2c}>0$ in the (a-a) section, and $p_{2d}-p_{1d}>0$ in section (c-c) making possible the flow circulation described above.
Figure 3.35 Normalized vector field superimposed on the pressure field for a deep reservoir configuration in the axial plane of symmetry; eccentricity (e/c) = 0.5; porosity (ε) = 0.5; permeability (k) = 1E-11 m²; Reₜ = 72.55(15krpm) - isothermal model
Figure 3.36 Velocity magnitude field for a deep reservoir configuration in the axial symmetry plane. (Velocity fields in the clearance were shown in Figure (8) Details (B') and (D')); eccentricity (e/c) = 0.5; porosity (ε) = 0.5; permeability (k) = 1E-11 m²; Re = 72.55(15krpm) - isothermal model
Figure 3.36 is associated with Figure 3.35 to represent the actual velocity magnitude fields that the vector maps did not indicate. These velocity fields in conjunction with the vector map of the previous figure characterize completely the flow in the porous medium and reservoir. The velocity magnitudes and the extent of their circulation pressure profile are associated with the “high” permeability (1E-11 m$^2$) of this experiment. Figure 3.37 presents similar information as Figure 3.36 but the permeability has been reduced to 1E-13 m$^2$ (the “low” permeability of this experiment). Comparing the two figures one can assess in a two dimensional view the effect that permeability has on the actual velocities in the porous medium and in the reservoir. The flow will reach velocities of 0.1 m/s when permeability is 1E-11 m$^2$, while in the case of permeability 1E-13 m$^2$ the velocity magnitude is one order of magnitude smaller (0.05 m/s). For the lower permeability case the maximum pressure generated is 905.6 kPa (see insert Figure 3.37) compared to 160kPa (see insert Figure 3.36). The comparison between these figures shows that the higher permeability engenders higher velocity flows in the reservoir, even though the pressure in Figure 3.36 (insert) is lower than in Figure 3.37 (insert). While the pressure difference is a clear indicator of the intensity of flow circulations between the active and passive region, if the resistance the fluid needs to overcome in the porous medium is too large the result is low communication between the two regions and a lower circulation in the reservoir and hence lower cooling capability.
Figure 3.37  Velocity magnitude field for a deep reservoir configuration in the axial symmetry plane. (Velocity fields in the clearance were shown in Figure 3.3 Details (B’) and (D’)); eccentricity (e/c) =0.5; porosity (ε)= 0.5; permeability (k) = 1E-13 m²; Reₜ = 72.55(15krpm) - isothermal model
3.1.9 The 3D effects on pressure fields

Figure 3.38 illustrates the tridimensional pressure plots on the bearing shaft surface for a high permeability case \((k) = 1E-11 \text{ m}^2\). This allows the reader to understand the overall development of the pressure in the bearing. The convergent high-pressure region as well as the cavitation are now visualized both in the axial and circumferential directions. While the circumferential profile has an expected behavior, the axial profile merits additional discussion. Even though the bearing has an L/D ratio of 1.5, this profile is rather flat, reminiscing of a pressure distribution usually associated with a long bearing. This flatness of the axial profile can be attributed to the fact that the high permeability causes a larger mass of fluid to leave the active space and circulate mostly through the porous medium and the reservoir both in the radial and circumferential directions. That leaves little fluid to leak axially. For the same reason (less fluid in the active space) the pressures are lower than what is seen in the case of the lower permeability presented in the next figure. Therefore, again, the mechanism of pressure build-up inside this porous bearing is interplay of three resistances (radial, axial and circumferential resistance).

Figure 3.39 presents the same type of information as Figure 3.38, but this time small permeability conditions are implemented \((k) = 1E-13 \text{ m}^2\). All other parameters, geometric and operational are kept the same. The decreased permeability causes much higher circumferential pressures compared to the previous case. The pressure differential across the porous medium is now much larger but it is accompanied by a higher resistance to flow. The axial pressure profile has regained its parabolic-like shape, as a
Figure 3.38 Tridimensional pressure carpet plot on the shaft: eccentricity (e/c)=0.5, porosity (ε)= 0.5; permeability (k) = 1E-11 m$^2$; $Re_c=72.55$ (15krpm) (Cavitation threshold at 70kPa) - isothermal model

result of which, there is increased axial flow through the clearance and diminished radial and axial flow in the porous medium. In this case, the lubricating oil is being leaked out axially through the outlets, mostly in the site of minimum clearance.

Figure 3.40 presents a 3D view of both flow patterns and pressures throughout the bearing. The color map is indicative of the pressure variation axially and circumferential-
Figure 3.39 Tridimensional pressure carpet plot on the shaft: eccentricity (e/c)=0.5, porosity (ε)= 0.5; permeability (k) = 1E-13 m$^2$; Re$_c$=72.55(15krpm) (Cavitation threshold at 70kPa) - isothermal model

...ly at the surface of the rotating shaft, while the superimposed vectors indicate the flow patterns in 3D at eighth selected locations. This approach allows concomitant description of the flow and pressures in a three dimensional manner. The pressure map details clearly the zones of high pressure in the convergent zone and cavitation in the divergent one. The axial cross-sections as well as the circumferential ones show the radial-axial flow direction both in the porous medium and the reservoir. The flow characteristics are...
eminently three-dimensional. Even though the porous medium has isotropic properties, the flow in the porous region is strongly radial proving furthermore that the mechanism driving the flow is the radial pressure difference between the active and the passive zone. The flow on the shaft surface and in the reservoir region is strongly circumferential.

Figure 3.40 Tridimensional vector field including the flow on the shaft emphasizing the exclusively circumferential flow of the fluid in the reservoir and exclusively radial flow in the porous region.
3.1.10 The effects of closed outlets on pressures profiles

All numerical simulations presented up to this section, had the outlets of the bearing open to atmospheric pressure. The actual design contains inward pumping spiral-groove seals, doubled by lip-seals located at the axial ends. This concept ensures that the same fluid is circulated endlessly with no axial leakage.

Concerns regarding the effect of axial flow are investigated considering the case of closed outlets. For the purpose of comparison, the open outlets were set to a pressure of 0 [Pa]. This change was needed to compare these results with the closed ends results, since the computer code used 0 [Pa] as a reference pressure in the absence of any reference to anchor (it is a closed system now, with no reference from exterior).

Comparisons are presented here for the cases when the outlets are open versus the case when both axial outlets are fully closed (Figure 3.41 and Figure 3.42). It was noted that the closed outlets cases correspond to higher-pressure build-up. The increase in pressure is due to more fluid being present in the clearance region and contributing to pressure build up. While the pressure curves magnitudes are somewhat different between the open and close ends cases, the trends and shapes remained intact; therefore the circulation process, which is the main scope of the present research remains unchanged. Since the actual design includes spiral groove seals and lip seals, it is recommended that future work to include these effects as well.
Figure 3.41 Pressure distribution on the shaft at the axial plane of symmetry when the outlet is closed compared to outlet opened; eccentricity (e/c)=0.5; permeability (k) = 1E-13 m²; porosity (ε)=0.5; Re_c:= 72.55 (15 krpm) - isothermal model.
Figure 3.42 Pressure distribution on the shaft at the axial plane of symmetry when the outlet is closed compared to outlet opened; eccentricity (e/c)=0.9; permeability (k)=1E-13 m$^2$; porosity (ε)=0.5; $Re_c=72.55$ (15 krpm) - isothermal model
3.2 Thermo-hydrodynamic model

While an isothermal flow represents a model for which the heat exchange with the surroundings and the heat generated are in perfect equilibrium such that the system temperature remains constant, in the thermodynamic case part of the heat generated by shear is being transmitted to the environment and part is contributing to the increase in the internal energy of the system.

Since the pressures generated by the hydrodynamic film are important to determination of all bearing major performance parameters, the thermal effects are an important aspect when studying hydrodynamic bearings.

The differences between the isothermal numerical results and thermo-hydrodynamic results are expected to be significant due to the oil dynamic viscosity drop with the increase of temperature. A lower dynamic viscosity translates into a lower resistance to the flow, which is the main mechanism for pressure generation.

In journal bearings, when considering the energy generation and transfer equations, the temperature of the bearing expected to increase above the temperature of the supply by 50 to 100 deg F. Therefore, adequate means of cooling for the lubricant are needed, since bearing surfaces are subjected to degradation at very high temperatures and the risk of the oil lubricant flashing due to over-heating is also high. For the proposed bearing configuration, the removal of the pump driven circulating system eliminates any chance of active cooling. The cooling strategy proposed herein is based on enhanced natural convection supported by the addition of circumferential pin fins located on the exterior surface of the doughnut reservoir, Figure 3.43.
3.2.1 The influence of heat transfer coefficient at the wall

The overall effect of the fins are estimated and the convective heat transfer ratio obtained incorporated into the numerical model. Heat is transferred from the lubricant film to the reservoir wall by convection, from the surface of the reservoir to the system of fins by conduction and from the fins to the surrounding environment by convection (see Figure 3.44).
To create a simplified equation for the heat transfer of a fin, the following assumptions have been made: constant heat transfer properties for materials, one-dimensional conduction, uniform cross-sectional area for fins and uniform convection across the surface area.

1) Liquid-solid wall convection: \( q_1 = h_l A_l (T_i - T_{w1}) \)

2) Wall conduction: \( q_2 = \frac{k_W A_l}{t} (T_{w1} - T_{w2}) \)

3) Solid wall-air convection: \( q_3 = h_a A_a \eta_{fin} (T_{w2} - T_a) \)

After simple algebraic manipulation: \( q_{\text{total}} = A_l U_l (T_i - T_a) \), where: \( U_l = \frac{1}{\frac{1}{h_l} + \frac{t}{k_W} + \frac{A_l}{A_a h_a \eta_{fin}}} \)

where \( h_l \) represents the free convective heat transfer coefficient between the lubricant and the metal casing with typical values between 20 and 100 \( \text{W/m}^2 \text{K} \) (a value of 50 \( \text{W/m}^2 \text{K} \) was considered for this coefficient).
$h_a$ represents the free convective heat transfer coefficient between the metal casing and the air with typical values between 5 and 25 $W/m^2K$ (a value of 12.5 $W/m^2K$ was considered for this coefficient). The solid wall material was considered to be made out of steel with a conducting heat transfer coefficient: $k_{steel} = 43 \ W/mK$, the thickness of the wall ($t = 3mm$), $A_l$ the lateral area of the bearing exterior wall ($A_l = 0.004408 m^2$), $A_a$ the total lateral area of the fins. Each fin has the geometric dimensions of 50x4x4 mm resulting in a total area open to air of $A_{fin} = 8E-4m^2$, the total area $A_a$ was hence a function of the number of fins considered. The efficiency for a straight rectangular fin is $\eta_{fin} = 0.9$.

Three configurations have been considered: no fins (baseline), 30 fins and 100 fins. The overall heat transfer coefficients obtained are presented in Table 3.8.

<table>
<thead>
<tr>
<th>Configuration</th>
<th>The overall heat transfer coefficient of the system</th>
</tr>
</thead>
<tbody>
<tr>
<td>No fins (baseline)</td>
<td>10 $W/m^2K$</td>
</tr>
<tr>
<td>30 fins</td>
<td>27.47 $W/m^2K$</td>
</tr>
<tr>
<td>100 fins</td>
<td>40.05 $W/m^2K$</td>
</tr>
</tbody>
</table>
Figure 3.45 through Figure 3.47 present temperature fields for the same geometric and functional parameters of the bearing, when parametrically changing the overall heat transfer coefficient. One can follow the development and penetration of the temperature field radially from the area where it is produced (at the fluid/porous media interface) and into the porous medium. The flow maintained its mainly radial flow direction inside the porous media and strongly circumferential flow direction in the reservoir region. The zones where the fluid leaves the clearance space, the zones where the fluid turns-around and the zones where the fluid reenters the bearing clearance have the exact same location for no fin configuration (Figure 3.45), 30 fins configuration (Figure 3.46), and 100 fins configuration (Figure 3.47). The difference between the three cases supports the fundament that the more heat is removed the lower the temperature. This difference comes from the fact that the heat generated by the shear forces is the same for all three configurations; however, the amount of heat removed from the system is different, translating to different internal energies stored and hence different temperatures. Figure 3.45 thought Figure 3.47 have different scales to capture the temperature gradient across the radial and circumferential direction; however the maximum temperature reached for each case is stated at the end of the upper scale.

From the study of Figure 3.45, one can observe for the baseline case (no fins) a maximum temperature reached at 467K. This represents an increase of 157K over the reference values of 300K. In Figure 3.46, while the heat transfer is enhanced, still an unsatisfactory maximum temperature is obtained (435K, an increase of 135 compared to the reference). Figure 3.46 offers a satisfactory temperature field with a maximum temperature reached of 357K (a 57K increase compared to the reference value).
The configuration with 100 pin fins (heat transfer coefficient of 40.05 \( \text{W/m}^2\text{K} \)) makes the passive thermal management proposed solution a realistic one since the working fluid is not reaching high temperatures. The obtained temperatures are farther away from flashing or boiling point levels (477 to 533 K depending on the oil type). In addition, and importantly, this bearing concept is also applicable to the use of liquid metal lubricants (gallium and indium). In these cases the cooling capabilities (convection + conduction) of the working fluid are greatly enhanced when compared to that of oil. Even more significant is the capability of such a bearing to function in extremely hot environments (up to 1500K). It is well known that using oil lubricated bearings (rolling element or hydrodynamic) in such environments has been a continuously thorny issue due to the possibility of oil flashing and that powder lubrication was one attempt at solving this problem.

To conclude, it is expected that a further increase in the number of fins would result in a lower temperature for the fluid lubricant. However, the cost of machining these fins compared to the benefits in lowering the temperature is not justified. For example a configuration using 150 fins provides an overall heat transfer coefficient of 42.85 \( \text{W/m}^2\text{K} \), and a further increase to 200 fins would result in a heat transfer coefficient of 44.85 \( \text{W/m}^2\text{K} \). Other type of improvements can be made by increasing the lateral area of the fins, and hence increasing the area for convective heat with the environment. These optimizations do not represent a main objective for the present research. For the purpose of this research, a 100 fins configuration providing an overall
Figure 3.45 Normalized vector field superimposed on the temperature field for a deep reservoir configuration in the axial plane of symmetry; eccentricity \( (e/c) = 0.5 \); porosity \( (\varepsilon) = 0.5 \); permeability \( (k) = 1E-11 \text{ m}^2 \); \( \text{Re}_c = 72.55(15\text{krpm}) \); the base line case-no fins \( U_1 = 10 \text{ W/m}^2\text{ K} \)
Figure 3.46 Normalized vector field superimposed on the temperature field for a deep reservoir configuration in the axial plane of symmetry; eccentricity (e/c) =0.5; porosity (ε)= 0.5; permeability (k) = 1E-11 m$^2$; Re$_c$=72.55(15krpm); 30 fins configuration

$U_2 = 27.47 \text{ W/m}^2 \text{ K}$
Figure 3.47 Normalized vector field superimposed on the temperature field for a deep reservoir configuration in the axial plane of symmetry; eccentricity (e/c) =0.5; porosity (ε)= 0.5; permeability (k) = 1E-11 m²; Re=72.55(15krpm); 100 fins configuration $U_3 = 40.05 \, \text{W/m}^2\,\text{K}$
heat transfer coefficient of $40.05 \, \text{W/m}^2\text{K}$ was satisfactory and was used for all numerical computations presented in this chapter.

Therefore, as a conclusion for this section, in the case of a porous bearing with a wrapped around reservoir, the mechanisms for energy generation and transfer are:

- The shear stress within the film produces viscous losses which generate heat;
- The heat convection occurring along the lubricant film due to mass transfer;
- The heat conduction occurring across and along the fluid film in the clearance;
- The heat conduction and convection occurring through the porous bushing;
- The heat conduction and convection occurring in the reservoir;
- The free convection from the bearing wall to the ambient;
- The increase in internal energy.

All these mechanism are contributing to the generation and transfer of heat, resulting in a system at equilibrium.

3.2.2 The influence of angular speed

Figure 3.48 through Figure 3.51 present the pressure dependence for the deep reservoir and for permeabilities ranging from $1\times10^{-10}$ m$^2$ to $1\times10^{-13}$ m$^2$ and speeds varying from 2 to 15 krpm. The concentric clearance ($c=0.002$ inches), porosity ($\varepsilon = 0.5$), porous media thickness ($d=0.1$ inches) and eccentricity ($e/c=0.5$) respectively, are kept constant. In each of these figures at constant porosity, permeability and eccentricity, the pressure
increases with angular velocity. Cavitation pressure levels for all cases are considered using a Gumbel algorithm ($p_{\text{min}} = p_{\text{cav}}$) to set the minimum pressure at 70kPa.

As permeability is decreased from 1E-10 m$^2$ in Figure 3.48 to 1E-13 m$^2$ in Figure 3.51, for the same range of velocities (2-15 krpm), the pressure in the convergent zone keeps increasing and reaches a maximum for the lowest permeability. As expected, the pressure increased with angular speed for constant permeability. These exact same findings were established using the isothermal model. The effect of the heat generation and dissipation is rather upon the pressure magnitude of these pressure curves then upon the pressure shapes. For the highest permeabilities studied (1E-10 m$^2$) and (1E-11 m$^2$) the cavitation levels for pressure, again, similarly to the isothermal case are not met.

When looking at the maximum pressure generated for the deep reservoir case, the permeability of 1E-10 m$^2$, and the rotational speed of 15krpm the pressure drop is from 107 kPa in Figure 3.7 (a) to 101.2 kPa of Figure 3.48 (a 6% reduction). For the permeability of 1E-11 m$^2$ the maximum pick pressure decrease is more significant: from 168 kPa (Figure 3.7 (b)) to 113kPa (Figure 3.49), therefore a 33% reduction. For permeability of E-12 m$^2$ the maximum pick pressure decrease is from 440 kPa (Figure 3.8 (a)) to 195kPa (Figure 3.50), a 55% reduction. For permeability of 1E-13 m$^2$ the maximum pick pressure decrease is from 900 kPa (Figure 3.8 (b)) to 425kPa (Figure 3.51), a 53% reduction. The findings of extremely high differences (55% and 53% reduction) for low permeability ranges of 1E-12 m$^2$ and 1E-13 m$^2$ are consistent with previous stating suggesting that for these conditions very little fluid travels through the porous media. Therefore, the fluid trapped in the clearance region was heated due to the viscous shear and resulted to the same rate of viscosity decrease and the same rate of
pressure build-up decrease. Comparing the shallow reservoir case results illustrated in Figure 3.52 through Figure 3.55, the same trends of pressure increase with the angular velocity increase and porous media permeability decrease are recovered. For low permeability of the porous media, the maximum pressures built for deep and shallow reservoirs are identical. This property has been conserved for both isothermal and thermo-hydrodynamic model.

When comparing the pressure drops between the isothermal and the thermal model, one can observe the following: for the permeability of 1E-10 m² the maximum pressure pick has decrease from 126 kPa (Figure 3.9 (a)) to 105 kPa (Figure 3.52), therefore a 17% reduction. For the permeability of 1E-11 m² the maximum pressure pick decreases from 180 kPa (Figure 3.9 (b)) to 117 kPa (Figure 3.53), a 35% reduction. For the permeability of 1E-12 m² the maximum pressure pick decreases from 440 kPa (Figure 3.10 (a)) to 195 kPa Figure 3.54, a 55% reduction. For the permeability of 1E-13 m² the maximum pressure pick decreases from 900 kPa (Figure 3.10 (b)) to 425 kPa (Figure3.55), an 53% reduction. The discussion for the deep reservoir case is valid for the shallow reservoir case: lower permeabilities translated to higher differences between the thermal and isothermal model. We can conclude that the change in temperature notably affected the pressure magnitude within the film. Further investigations into the effects of the temperature increase on the other characteristics of the bearing are justified.
Figure 3.48 Pressure distribution on the shaft surface at the axial plane of symmetry for a deep reservoir configuration when the angular velocity of the shaft is changed parametrically; eccentricity (e/c) = 0.5; permeability(k) = 1E-10 m$^2$; porosity(ε) = 0.5-thermal model

Figure 3.49 Pressure distribution on the shaft surface at the axial plane of symmetry for a deep reservoir configuration when the angular velocity of the shaft is changed parametrically; eccentricity (e/c) = 0.5; permeability(k) = 1E-11 m$^2$; porosity(ε) = 0.5-thermal model
Figure 3.50 Pressure distribution on the shaft surface at the axial plane of symmetry for a deep reservoir configuration when the angular velocity of the shaft is changed parametrically; eccentricity (e/c) = 0.5; permeability (k) = 1E-12 m²; porosity (ε) = 0.5-thermal model

Figure 3.51 Pressure distribution on the shaft surface at the axial plane of symmetry for a deep reservoir configuration when the angular velocity of the shaft is changed parametrically; eccentricity (e/c) = 0.5; permeability (k) = 1E-13 m²; porosity (ε) = 0.5-thermal model
Figure 3.52 Pressure distribution on the shaft surface at the axial plane of symmetry for a shallow reservoir configuration when the angular velocity of the shaft is changed parametrically; eccentricity (e/c)= 0.5; permeability(k)= 1E-10 m$^2$; porosity(ε)= 0.5-thermal model

Figure 3.53 Pressure distribution on the shaft surface at the axial plane of symmetry for a shallow reservoir configuration when the angular velocity of the shaft is changed parametrically; eccentricity (e/c)= 0.5; permeability(k)= 1E-11 m$^2$; porosity(ε)= 0.5-thermal model
Figure 3.54 Pressure distribution on the shaft surface at the axial plane of symmetry for a shallow reservoir configuration when the angular velocity of the shaft is changed parametrically; eccentricity (e/c)= 0.5; permeability($k$)= 1E-12 m$^2$; porosity($\varepsilon$)= 0.5-thermal model

Figure 3.55 Pressure distribution on the shaft surface at the axial plane of symmetry for a shallow reservoir configuration when the angular velocity of the shaft is changed parametrically; eccentricity (e/c)= 0.5; permeability($k$)= 1E-13 m$^2$; porosity($\varepsilon$)= 0.5-thermal model
Figure 3.56 and Figure 3.57 present the variation of load with the increase of angular velocity. For high permeabilities the load carrying capability is extremely small (values of less than 20 N corresponding to less than 2 kg load) and is not being improved with the increase of the speed. For smaller permeabilities of $1 \times 10^{-12}$ m$^2$ and $1 \times 10^{-13}$ m$^2$, the load carrying capability is 40N (4kg) for 2krpm to 180 N (18kg) corresponding to a speed of 15krpm.

Similar to the isothermal case, the shallow reservoir has produced higher load capability compared to the deep reservoir case for the high permeability cases. Again, for low permeabilities cases, $1 \times 10^{-12}$ m$^2$ and $1 \times 10^{-13}$ m$^2$, the bearing is supporting the same amount of load regardless of its reservoir depth. The maximum load that the geometry proposed can carry for eccentricity of 0.9 is 180N (18kg) for a permeability of $1 \times 10^{-13}$ m$^2$ and 80N (8 kg) for permeability of $1 \times 10^{-12}$ m$^2$.

While these values might seem low, there are expected for a porous bearing with small diameter (1 inch), 1.5 L/D ratio and no means of sealing at the axial ends considered for this model. The pressure values have dropped drastically due to the decrease in the kinematic viscosity. Porous bearings, in general, have the disadvantage of low carrying capability, and the addition of the wrapped around reservoir is acting in the direction of exacerbating this even more.
Figure 3.56 Load carrying capability for a deep reservoir configuration when the angular velocity of the shaft is varied parametrically; eccentricity (e/c)=0.5; porosity (\(\varepsilon\))=0.5, permeabilities (\(k\)) ranging from 1E-10 m\(^2\) to 1E-13 m\(^2\)-thermal model.

Figure 3.57 Load carrying capability for a shallow reservoir configuration when the angular velocity of the shaft is varied parametrically; eccentricity (e/c)=0.5; porosity (\(\varepsilon\))=0.5, permeabilities (\(k\)) ranging from 1E-10 m\(^2\) to 1E-13 m\(^2\)-thermal model.
Figure 3.58 and Figure 3.59 present the magnitude (color scale) of the pressure distribution throughout the clearance/porous/reservoir assembly in the axial symmetry plane for both deep and shallow reservoir configuration, in the circumferential radial directions. This information confirms that the higher permeability for the deep reservoir translates into a lower pressure in the active region and significant difference in pressure distribution both radially and circumferentially, findings confirmed for the isothermal model. The radial pressure gradient is indicated by the change in color scale and confirms the radial flow out of the active space and into the reservoir over most of the circumferential length of the convergent zone showing that as the permeability decreases, the pressure increases. For the low permeability cases, similar with the isothermal model, there seems to be very little difference in the pressure patterns between the shallow and the deep reservoirs cases. Again, the explanation resides with the reduced amount of fluid allowed out of the active space and into the reservoir, fact that makes the influence of the reservoir depth minimal.
Figure 3.58 Circumferential-radial pressure fields in the plane of symmetry for a deep reservoir configuration with permeabilities ranging from $1E-10$ m$^2$ to $1E-13$ m$^2$; eccentricity ($e/c$) = 0.5; $c$=0.002in; porosity ($\epsilon$)= 0.5; $Re_c$=72.55 (15 krpm) -thermal model
Figure 3.59 Circumferential-radial pressure fields in the plane of symmetry for a shallow reservoir configuration with permeabilities ranging from $1 \times 10^{-10}$ m$^2$ to $1 \times 10^{-13}$ m$^2$; eccentricity (e/c) = 0.5; c=0.002in; porosity ($\varepsilon$) = 0.5; Re$_c$=72.55(15krpm) - thermal model
3.2.3 The influence of eccentricity

Figure 3.60 through Figure 3.67 present the pressure distribution on the shaft for various eccentricities, permeabilities of the porous bushing and reservoir depths. The finding for the isothermal model remain valid: the increase in eccentricity was translated into an increase in pressure magnitude. It is noteworthy that permeability increase has a dual effect in the case of thermo-hydrodynamic simulations. Firstly, the decrease in permeability results into an increase in pressure magnitude though the increase in flow resistance, and secondly the increase of permeability translates to an increase of shear inside the porous region, mechanism responsible for heat generation and viscosity drop. These two effects are balancing each other, and the overall effect is the increasing pressure magnitude when the permeability decreases (result obtained in the isothermal case in section 3.1.4).

The comparison between the characteristics of a porous hydrodynamic bearing and the thermo-hydrodynamic bearing show that the influence of the temperature on pressure fields is in the direction of lowering the maximum pressure generated. Again, this is due to the fact that under higher temperatures, the dynamic viscosity of the lubricant decreases dramatically and with it the value of the shear between the layers of the fluid. This shear stress is the property “responsible” for generating pressure and subsequently, load carrying capability.

The increase of shaft eccentricity is resulting into an increase in pick pressures values for all configurations proposed, similar to the isothermal case. From the study of Figure 3.60, Figure 3.61, Figure 3.64 and Figure 3.65 one can conclude that for low permeabilities of 1E-10m² and 1E-11m² the levels for cavitation to occur are not met for
both the deep and shallow reservoir case, and for all eccentricities. Decreasing the permeability to 1E-12 m$^2$ (Figure 3.62 and Figure 3.66) results in large cavitating areas.

For the isothermal hydrodynamic model when the eccentricity was (e/c)=0.9, the rotational speed was 15 krpm, and the permeability of the porous bushing was 1E-13 m$^2$, the maximum generated load was 1600N (both deep and shallow configurations, see Figure 3.23 and Figure 3.24). In the thermo-hydrodynamic model, when using the same geometric and functional conditions the maximum load is 360N (36 kg), therefore a 77.5% reduction (see Figure 3.68). For a high permeability of 1E-10 m$^2$ and the same working conditions and a shallow reservoir depth, the reduction in load carrying capability due to the incorporation of thermal effects is 75% (from 16N to 4N, see Figure 3.23 and Figure 3.69).
Figure 3.60 Pressure distribution on the shaft at the axial plane of symmetry for a deep reservoir configuration when the shaft eccentricity is changed parametrically; permeability ($k$)= 1E-10 m$^2$; porosity ($\varepsilon$)= 0.3; concentric clearance ($c$)=0.002; $Re_c$=72.55 (15 krpm) -thermal model

Figure 3.61 Pressure distribution on the shaft at the axial plane of symmetry for a deep reservoir configuration when the shaft eccentricity is changed parametrically; permeability ($k$)= 1E-11 m$^2$; porosity ($\varepsilon$)= 0.3; concentric clearance ($c$)=0.002; $Re_c$=72.55 (15 krpm) -thermal model
Figure 3.62 Pressure distribution on the shaft at the axial plane of symmetry for a deep reservoir configuration when the shaft eccentricity is changed parametrically; permeability \( k = 1 \times 10^{-12} \text{ m}^2 \); porosity \( \varepsilon = 0.3 \); concentric clearance \( c = 0.002 \); \( \text{Re}_c = 72.55 \) (15 krpm) - thermal model.

Figure 3.63 Pressure distribution on the shaft at the axial plane of symmetry for a deep reservoir configuration when the shaft eccentricity is changed parametrically; permeability \( k = 1 \times 10^{-13} \text{ m}^2 \); porosity \( \varepsilon = 0.3 \); concentric clearance \( c = 0.002 \); \( \text{Re}_c = 72.55 \) (15 krpm) - thermal model.
Figure 3.64 Pressure distribution on the shaft at the axial plane of symmetry for a shallow reservoir configuration when the shaft eccentricity is changed parametrically; permeability \( k \)= 1E-10 m\(^2\); porosity \( \varepsilon \)= 0.3; concentric clearance \( c \)=0.002; \( \text{Re}_c:=72.55 \) (15 krpm) -thermal model

Figure 3.65 Pressure distribution on the shaft surface at the axial plane of symmetry for a shallow reservoir configuration when the shaft eccentricity is changed parametrically; permeability \( k \)= 1E-11 m\(^2\); porosity \( \varepsilon \)= 0.3; concentric clearance \( c \)=0.002; \( \text{Re}_c:=72.55 \) (15 krpm) -thermal model
Figure 3.66 Pressure distribution on the shaft at the axial plane of symmetry for a shallow reservoir configuration when the shaft eccentricity is changed parametrically; permeability \(k\) = 1E-12 m\(^2\); porosity \(\varepsilon\) = 0.3; concentric clearance \(c\) = 0.002; \(Re_c:=72.55\) (15 krpm) - thermal model

Figure 3.67 Pressure distribution on the shaft at the axial plane of symmetry for a shallow reservoir configuration when the shaft eccentricity is changed parametrically; permeability \(k\) = 1E-13 m\(^2\); porosity \(\varepsilon\) = 0.3; concentric clearance \(c\) = 0.002; \(Re_c:=72.55\) (15 krpm). - thermal model
Figure 3.68 Load carrying capability for a deep reservoir configuration; porosity ($\varepsilon$) = 0.3, eccentricities (e/c) ranging from 0.1 to 0.9; permeabilities ranging from $1E-10$ m$^2$ to $1E-13$ m$^2$, Rec=72.55 (15 krpm) -thermal model

Figure 3.69 Load carrying capability for a shallow reservoir configuration; porosity ($\varepsilon$) = 0.3, eccentricities (e/c) ranging from 0.1 to 0.9; permeabilities ranging from $1E-10$ m$^2$ to $1E-13$ m$^2$, Rec=72.55 (15 krpm) -thermal model
3.2.4 Temperature maps as a function of permeability

Figure 3.70 and Figure 3.71 present the temperature maps for both deep and shallow configurations for an angular velocity of the shaft of 15krpm and a heat convective coefficient at the wall of $40.05 \, \text{W/m}^2\text{K}$. It is observed that the heat generated by shear is transported with ease by the lubricant into the porous matrix. Then changing the permeability of the medium, the solid part of the porous matrix had the same conducting effects, however the convective effects are very different due to the ease with which the fluid is passing through this media being different. In the deep reservoir case Figure 3.70, after exiting the clearance region, where it is being produced, the heat is being transferred by both conduction and convection inside the porous matrix and from there in the reservoir where the convection effects are strong. From the study of Figure 3.70, one can establish that the heat transported by convection in the reservoir is directed back towards the porous wall facing the reservoir, which initially had a lower temperature. From here is transferred by conduction across the porous media, hence resulting in an extended area with high temperature. In the shallow reservoir Figure 3.71, the heat generated by the shear is existing the bearing clearance and going into reservoir, and because of its shallowness it is immediately transferred thought the walls with the convective heat flux properties imposed on the boundary. It appears that a higher reservoir will ensure a more even temperature distribution in the bearing, avoiding areas of extreme temperatures, but also a higher temperature reached overall. Common to all configurations is the location of the maximum temperature: located at the film/porous bushing interface since. In this area the shear effects are highest.
The temperature field distributions presented in Figure 3.70 and Figure 3.71 had different scales in order to better capture the temperature distribution inside the bearing. The minimum temperature for all cases was 300K representing the temperature of the outlet boundary condition and the reference temperature. The maximum of the scale is the maximum temperature reached for the specific simulations, and it varies significantly from a deep to a shallow configuration and for a high to a small permeability of the porous bushing. Therefore, one can conclude that the maximum temperature reached was a function of the bearing characteristics.

Counter intuitively, the highest temperatures (375K) are obtained for a deep reservoir configuration with a high permeability of 1E-10 m$^2$. This result is related with the discussion on section 3.1.9. where radial, axial and circumferential flow was discussed. For high permeability ranges, the fluid will most likely enter the porous region instead of travelling axially towards the outlets, and therefore removing less amounts of heat. The decrease of permeability is resulting in an increase in the viscous shear and therefore an increase in temperature. However, this heat is removed axially at higher rate that in the high permeability cases, resulting in lower temperatures of 340K for permeability of 1E-12 m$^2$ and 328 K for 1E-13 m$^2$.

In the shallower reservoir case, the overall temperatures are smaller than for the deep reservoir case. This is due to the fact that more heat was removed axially in the shallow reservoir case. The effect of the permeability decrease will translate into an increase to flow resistance in the radial direction, and a higher leakage axially. The same findings as in the deep reservoir case are reinforced. In section 3.2.5 the case when the outlets are closed will be investigated to eliminate the axial losses effects.
Another interesting result is that, similar to the isothermal case, where small permeability of $1E{-12}$ m$^2$ and $1E{-13}$ m$^2$ for the porous bushing translated into pressure distributions independent of the reservoir depth. In the thermo-hydrodynamic case the effect of the reservoir upon maximum temperature generated is also small: for a deep reservoir configuration and permeability of $1E{-12}$ m$^2$, the temperature reaches a value of 340K and for a shallow reservoir configuration and same permeability is reaches 338.7K. For a permeability of $1E{-13}$ m$^2$ the deep configuration reached 328.6K when the shallow reached 326.9K. Both cases illustrate a less than 2 degrees difference. This reinforces the known fact that there is a strong connection between the velocity, pressure, and temperature fields. Section 3.1.1 concluded that pressure fields in the clearance region are the same for high permeability ranges independent of the reservoir depth.

Since the heated fluid is exiting the clearance axially in the region of minimum clearance, in order to respect continuity low temperature fluid (300K-the reference temperature) is entering the bearing system through the outlets in the higher clearance region. Therefore, further investigations into the case of open outlet compared to the closed outlet axial boundary conditions are justified. The aim is to isolate this effect and to simulate a real technical application that is axially sealed.
Figure 3.70  Circumferential-radial temperature fields in the plane of symmetry for a deep reservoir configuration with permeabilities ranging from $1 \times 10^{-10} \text{m}^2$ to $1 \times 10^{-13} \text{m}^2$; eccentricity $(e/c)=0.5$; $c=0.002 \text{ in}$; porosity $(\varepsilon)=0.5$; $Re_c=72.55 (15 \text{krpm})$
Figure 3.71 Circumferential-radial temperature fields in the plane of symmetry for a shallow reservoir configuration with permeabilities ranging from $1E-10 \text{ m}^2$ to $1E-13 \text{ m}^2$; eccentricity $(e/c)=0.5$; $c=0.002 \text{ in}$; porosity $(\varepsilon)=0.5$; $Re_c=72.55$ (15 krpm).
3.2.5 Temperature maps for closed outlets case

The case when the axial outlets of the bearing are closed has been investigated. The heat generated by the shear forces in the clearance of the porous is therefore mostly transported radially through the porous and into the reservoir. The numerical simulations have been performed for a rotating speed of 15krpm, permeabilities of 1E-10m\(^2\) to 1E-12m\(^2\), eccentricity (e/c) 0.5, porosity 0.5, and concentric clearance 0.002 inches.

When decreasing the permeability the overall temperature in the bearing is increasing due to the added shear effect generated inside the porous. When the permeability is lower the viscous shear effects are more pronounced due to the added resistance to the flow. Contrary to the open outlets case, the highest temperature is reached for the lowest permeability (629K for a permeability of 1E-12m\(^2\) compared to 600K for a permeability of 1E-10m\(^2\)). When keeping the permeability constant, the amount of heat generated by shear is the same for both reservoir configurations, however the maximum temperature is different. From the inspection of Figure 3.72 one can conclude that for the case of a shallow reservoir the temperature in the bearing is higher and not as evenly distributed as in the deep reservoir configuration. In the deep reservoir configuration more heat is being transferred through the exterior wall due to the reservoir enlarged area. (this does not include the area of the fins which is considered the same in both bases). A convective heat transfer coefficient at the wall is 40.05 \(\frac{W}{m^2 K}\).

The cooling of this case is not efficient since temperatures of 600 to 643K are obtained. A strategy to improve the thermal management can include the use of different
Figure 3.72 Temperature distributions when the outlet is closed

(a) permeability 1E-10m$^2$
(b) permeability 1E-11m$^2$
(c) permeability 1E-12m$^2$
(d) permeability 1E-10m$^2$
(e) permeability 1E-11m$^2$
(f) permeability 1E-12m$^2$
fin configurations that can remove more heat or the increase the heat transfer coefficient with the air by forced convection.

3.2.6 Temperature maps for a low heat conducting porous bushing

Zirconia, is a ceramic type material that blocks heat effectively and has a low thermal conductivity coefficient (1/10 that of stainless steel). This material is considered for the porous bushing in order to estimate the influence of the conduction in solid part of porous on the overall temperature. The same heat transfer coefficient at the bearing wall is considered.

The radial temperature gradient is indicated by the change in the color scale and confirms a radial flow out of the active space and into the reservoir over most of the circumferential length of the convergent zone. The effect of increasing the temperature is not as dramatic one might expect to the presence of fluid lubricant inside the pores. This lubricant will still carry some of the heat by conduction and convection mechanisms; therefore the system will not become excessively heated as it would have in the case of a regular plain bearing. Nonetheless, the increase in temperature is significant (a 64K increase) when compared to the case when the porous bushing was made out of steel Figure 3.47.

This simulation goes further to demonstrate that the thermal model implemented is correct, and that the porous bearing provides better cooling capabilities compared to a regular journal bearing.
Figure 3.73 Normalized vector field superimposed on the temperature field for a deep reservoir configuration in the axial plane of symmetry; eccentricity (e/c) = 0.5; porosity (ε) = 0.5; permeability (k) = 1E-11 m²; (15krpm); U = 40.05 W/m²K; 
k_{zirconia} = 4.3 \frac{W}{mK}.
3.3 Static and dynamic characteristics

In real applications, bearings or any other mechanical devices do not operate at fixed positions. There are usually small variations around an equilibrium position. The characteristics of the bearing are static: eccentricity and phase angle and dynamic: stiffness and damping coefficients.

Analytical investigations on the performance of porous bearings under turbulent lubrication regimes have been done by Kumar [28] using the Reynolds equation. The dimensionless load capacity and attitude angles were determined theoretically. The theoretical results considered eccentricities of 0.4 and 0.6 and turbulent Reynolds numbers of 3000 and 5000, obtaining dimensionless load capacities from 0.5388 to 1.3043 and attitudes angles in the range 60.1908 up to 69.4818.

Lin and Hwang [46] have investigated the static and dynamic characteristics of long porous journal bearings and concluded that the Brinkman term added shear effect which increased the load capacity and the static attitude angle compared to the Beavers and Joseph slip velocity or the Darcy formulation. Using a modified Reynolds equation to account for the effect of the porous matrix and adding the viscous shear effects resulted in a significant reduction of the friction parameter.

D’Agustino et al. [62] obtained the stability threshold of the journal for non-steady simulations. They determined the stiffness and damping coefficients for the case when the porous bearing was lightly loaded using a Reynolds equations and Darcy formulation for the porous media. Also, a Reynolds based formulation using the Beaver-
Joseph criterion for the effect of slip flow of couple stress fluids was presented by Guha and Chattapadhyay [63].

It is important to determine the stiffness and the damping of the film in order to determine the dynamic motion, the synchronous response, and the linear stability analysis.

In Figure 3.74, the relationship between the system of coordinates attached to the line of centers and the system of coordinates attached to the line of load is presented. Commonly, in analytic computations, the load is an input and the eccentricity is obtained after solving the pressure equation as an output.

Figure 3.74 Relationship between the coordinate system attached to the line of centers (r,t) and the system attached to the line of load (x,y).
The rotational matrix from the coordinate system (r,t) to (x,y) as they are represented in Figure 3.74 is constructed as follows:

\[
\begin{bmatrix}
F_x \\
F_y
\end{bmatrix} = \begin{bmatrix}
\cos \varphi & -\sin \varphi \\
\sin \varphi & \cos \varphi
\end{bmatrix} \begin{bmatrix}
F_r \\
F_t
\end{bmatrix}
\]

3.3.1 The static characteristics.

When using CFD-ACE+, the eccentric position of the bearing system was established and the geometry discretized, making the eccentricity an input variable and the load carrying capability value and direction an output variable. Therefore the line of load was obtained from the radial and tangential components using the equation:

\[
\varphi = \arctan \frac{F_r}{F_t}
\]

Figure 3.75 presents the attitude angles obtained for a configuration with porosity 0.3, porous media thickness 0.1 inches, angular velocity of 15krpm, for both reservoirs depths deep and shallow.

From the study of Figure 3.75 one can conclude that while the permeability of the porous bushing makes significant difference in the values of the attitude angles, the effect of the reservoir appears to be minimal. This is consistent with the circumferential pressure profiles developments which have similar shapes (the tangential and radial components of the pressure have a similar proportionality factor). Higher permeabilities \((1E-10 \text{ m}^2 \text{ and } 1E-11\text{m}^2)\) ensure higher values for the attitude angles which translated into a configuration prone to being unstable. For the lower permeabilities \((1E-12 \text{ m}^2 \text{ and} \)
Figure 3.75 Attitude angle change for a shallow (i) and a deep (ii) reservoir as a function of permeability, porosity(\( \varepsilon \)) = 0.3, Re_0 = 72.55(15krpm)- isothermal model

1E-13 m^3, the attitude angles have values that tend asymptotically to the behavior of a solid bearing. Common for all permeabilities is the situation of eccentricities smaller the 0.5 as a highly unstable configuration, due to the high values for the attitude angle.

One can conclude that while the reservoir depth has minimal influence on the stability of the system, the permeability of the porous bushing plays an important role.

The results obtained are in agreement with the findings of Cameron [20] when comparing the effects on bearing attitude angle and implied stability of a solid bushing versus a porous one. For a solid bushing when eccentricity reaches 1, the force along the line of centers trends towards an infinite value (because of infinite pressure values) while the attitude angle moves towards 0 deg. The porous medium changes this situation by
Figure 3.76 Attitude angle change for a shallow (i) and a deep (ii) reservoir as a function of permeability, porosity(ε)= 0.3, Re_c = 72.55(15krpm)- thermal model allowing the fluid to ‘leak’ through it, thus preventing infinite pressures and loads along the line of centers, and causing higher attitude angles than the solid bushing. Consequently, one can conclude that the porous bushing bearing offers reduced stability compared to the solid bearing.

The attitude angles obtained using the thermal model are presented in Figure 3.76 and show very little differences between the shallow and the deep reservoir cases.

While comparing the thermal and the isothermal effects on the attitude angle, it can be concluded that the isothermal model predicted smaller attitude angle, and therefore a more stable configuration.
3.3.2 The dynamic characteristics.

The dynamic coefficients are obtained applying small amplitude perturbations to the equilibrium position (Figure 3.77). They are the gradients of the reaction force evaluated in the equilibrium position.

\[
\begin{align*}
K_{rr} &= -\frac{\partial F_r}{\partial r}, \\
K_{tt} &= -\frac{\partial F_t}{\partial t}, \\
K_{rt} &= -\frac{\partial F_r}{\partial t}, \\
K_{tr} &= -\frac{\partial F_t}{\partial r}, \\
C_{rr} &= -\frac{\partial F_r}{\partial r}, \\
C_{tt} &= -\frac{\partial F_t}{\partial t}, \\
C_{rt} &= -\frac{\partial F_r}{\partial t}, \\
C_{tr} &= -\frac{\partial F_t}{\partial r},
\end{align*}
\]

Figure 3.77 Small amplitude motions around the equilibrium position.

The computation procedure flowchart for computing the stiffness coefficients using CFD-ACE+ is presented in Figure 3.78 and the damping coefficients in Figure 3.79. The results obtained correspond to radial and tangential dynamic coefficient.

The choice of the magnitude of the perturbation in the calculations of the dynamic coefficients represents a delicate dilemma. Choy and Braun [64] advised choosing a value of the perturbation magnitude according to ones interest. If the accuracy near the
equilibrium position was of interest, then small values for perturbation are appropriate, if
the accuracy was needed further away from the equilibrium position, then a large value
for the magnitude of perturbation was advised. According to the authors their calculations
have been performed using a value of 0.2% of the bearing radial clearance. Lund and
Thomsen [65] suggest that in practice perturbation amplitudes as large as up to 50% of
the radial clearance renders valid results in most applications.

In order to determine the magnitude of the perturbation a convergence test was
performed on the damping coefficient using values of 0.001 m/s to 0.00001 m/s. The
results show that for 0.001 m/s to 0.0001 m/s there is very little variation in the values
obtained. However, for values of 0.00001 m/s ones notice a sudden jump. A similar result
was acknowledged by Choy and Braun [64] who stated that calculations with very small
perturbations endanger the accuracy at, and close to the equilibrium position.

Based on the results shown in Table 3.9, a value of 0.001 m/s was considered
suitable. The orientation of the perturbations applied with respect to the system of
coordinates was presented in Figure 3.77

<table>
<thead>
<tr>
<th>Velocity (m/s)</th>
<th>$C_{rr}$</th>
<th>$C_{rt}$</th>
<th>Error $C_{rr}$</th>
<th>Error $C_{rt}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.00001</td>
<td>75.45</td>
<td>9868.24</td>
<td>1485.08%</td>
<td>988.58%</td>
</tr>
<tr>
<td>0.0001</td>
<td>637</td>
<td>905.1</td>
<td>2.27%</td>
<td>0.16%</td>
</tr>
<tr>
<td>0.0003</td>
<td>626</td>
<td>903.63</td>
<td>0.54%</td>
<td>0.32%</td>
</tr>
<tr>
<td>0.0005</td>
<td>624</td>
<td>904.18</td>
<td>0.21%</td>
<td>0.26%</td>
</tr>
<tr>
<td>0.0008</td>
<td>623</td>
<td>905.53</td>
<td>0.04%</td>
<td>0.11%</td>
</tr>
<tr>
<td>0.001</td>
<td>623</td>
<td>906.52</td>
<td>--</td>
<td>--</td>
</tr>
</tbody>
</table>
Figure 3.78  Flow chart diagram for calculating of stiffness coefficients using CFD-ACE+
Figure 3.79 Flow chart diagram for calculating damping coefficients using CFD-ACE+

Set $\varepsilon$

Steady state computation to obtain the equilibrium position

Obtain: $F_{1r}, F_{1t}, \varphi = \tan\frac{F_{1r}}{F_{1t}} W_1$

Impose velocity $\dot{\varphi}$ for shaft

Obtain: $F_{4r}, F_{4t}$

Compute:

$C_{rr} = \frac{F_{4r} - F_{1r}}{\dot{\varphi}}$

$C_{tt} = \frac{F_{4t} - F_{1t}}{\dot{\varphi}}$

Impose velocity $\dot{t}$ for shaft

Obtain: $F_{5r}, F_{5t}$

Compute:

$C_{rt} = \frac{F_{5r} - F_{1r}}{\dot{t}}$

$C_{tt} = \frac{F_{5t} - F_{1t}}{\dot{t}}$

Transient computation using grid deformation module to obtain new equilibrium position
The computations for dynamic coefficients have been mapped as a function of eccentricity, for a rotational speed of 15kRPM, concentric clearance 0.002 inches. The parameters varied have been geometrical (the reservoir depth) and functional (the permeability of the porous bushing). The matrix of numerical experiments that have been performed is presented in Table 3.10. The numerical simulations have been performed for both isothermal and thermo-hydrodynamic models.

Table 3.10 Matrix of numerical experiments for stiffness and damping coefficients (isothermal and thermo-hydrodynamic simulations)

<table>
<thead>
<tr>
<th></th>
<th>k=1E-11m²</th>
<th>e/c=0.1</th>
<th>e/c=0.3</th>
<th>e/c=0.5</th>
<th>e/c=0.7</th>
<th>e/c=0.9</th>
</tr>
</thead>
<tbody>
<tr>
<td>deep</td>
<td>k=1E-13m²</td>
<td>e/c=0.1</td>
<td>e/c=0.3</td>
<td>e/c=0.5</td>
<td>e/c=0.7</td>
<td>e/c=0.9</td>
</tr>
<tr>
<td></td>
<td>k=1E-11m²</td>
<td>e/c=0.1</td>
<td>e/c=0.3</td>
<td>e/c=0.5</td>
<td>e/c=0.7</td>
<td>e/c=0.9</td>
</tr>
</tbody>
</table>


Due to the fact that the journal equilibrium position shifts with operating conditions, the system of coordinates attached to (r, t) coordinates changes orientation and is inconvenient in rotor dynamics calculations. The matrices of rotation employed to transform the obtained values from (r,t) system of coordinates to the fixed (x,y) system are:

\[
\begin{bmatrix}
K_{xx} & K_{xy} \\
K_{yx} & K_{yy}
\end{bmatrix} = \begin{bmatrix}
\cos \varphi & -\sin \varphi \\
\sin \varphi & \cos \varphi
\end{bmatrix} \begin{bmatrix}
K_{rr} & K_{rt} \\
K_{tr} & K_{tt}
\end{bmatrix} \begin{bmatrix}
\cos \varphi & \sin \varphi \\
-\sin \varphi & \cos \varphi
\end{bmatrix}
\]

\[
\begin{bmatrix}
C_{xx} & C_{xy} \\
C_{yx} & C_{yy}
\end{bmatrix} = \begin{bmatrix}
\cos \varphi & -\sin \varphi \\
\sin \varphi & \cos \varphi
\end{bmatrix} \begin{bmatrix}
C_{rr} & C_{rt} \\
C_{tr} & C_{tt}
\end{bmatrix} \begin{bmatrix}
\cos \varphi & \sin \varphi \\
-\sin \varphi & \cos \varphi
\end{bmatrix}
\]

Figure 3.80 The dynamic coefficients of the bearing ($K_{xx}$ $K_{yy}$ Principal direction stiffness coefficients, $K_{xy}$ $K_{yx}$ Cross coupled stiffness coefficients, $C_{xx}$ $C_{yy}$ Principal direction damping coefficients, $C_{xy}$ $C_{yx}$ Cross coupled damping coefficients).
The dynamic coefficients have been non-dimensionalized according to the equations:

\[ \bar{K} = \frac{Kc}{W} \quad 3.6 \]

\[ \bar{C} = \frac{Cc\omega}{W} \quad 3.7 \]

Where \( c \) is the concentric clearance, \( \omega \) is the rotational speed and \( W \) is the load. For simplicity the bar associated with the non-dimensional value is dropped.

Figure 3.81 indicates the stiffness and damping coefficient for a bearing with a deep reservoir and relatively high permeability (1E-11 m\(^2\)) for the porous bushing. It becomes apparent that the direct stiffness coefficients for this configuration \( K_{xx} \) and \( K_{yy} \) are almost zero. This characteristic is directly related to the porous bushing permeability, property that is allowing the fluid to penetrate inside the porous if variations on the clearance gap occur. When the shaft moves, the fluid film is leaked into the porous matrix instead of being squeezed as in the case of solid bearings. Therefore, this new type of bearing is not providing elastic support. The cross coupled stiffness coefficients are perfectly symmetrical; therefore the bearing has high stability. Both stiffness and damping coefficients have higher values at small eccentricity positions. This characteristic can be interpreted as follows: a perturbation at small eccentricity will create a larger response force from the lubricant film then a perturbation occurring at higher eccentricity. This response can potentially have a destabilizing effect on the bearing. In addition, low eccentricity ratios are associated with low loads on the bearing, and hence we recover another known fact for regular bearings that functioning under light loads usually creates unstable configurations. For eccentricities above 0.3 the dynamic coefficients are linear and have low values, therefore the bearing is stable.
While increasing the permeability of the bushing to 1E-13 m² and investigating the deep reservoir configuration, the elastic response of the bearing can be observed from the study of Figure 3.82. For this case the coefficients are non-zero, reinforcing the statement that permeability is the contributing factor for the elastic response of the bearing. The values of the direct stiffness coefficients are positive and slowly increasing with the increase of eccentricity. It is note-worthy, that the direct damping coefficients at eccentricity of 0.1 are negative; this translates into an unstable position. At eccentricities over 0.3 the direct damping coefficients recover to a positive value. For all eccentricities, the cross-coupled coefficients are perfectly symmetrical, which translates to a stable bearing configuration.

From the study of Figure 3.83 and Figure 3.84 one can conclude that the dynamic response of the shallow bearing is very similar to the one with deep reservoir. It appears that both the static and dynamic characteristic are not influenced by the depth of the recirculating reservoir.
Figure 3.81 Non-dimensional dynamic coefficients for deep reservoir configuration as a function of eccentricity ($k$): 1E-11 m$^2$ -isothermal model
Figure 3.82 Non-dimensional dynamic coefficients for deep reservoir configuration as a function of eccentricity ($k$): 1E-13 m$^2$-isothermal model
Figure 3.83 Non-dimensional dynamic coefficients for shallow reservoir configuration as a function of eccentricity \((k)\): 1E-11 m\(^2\)-isothermal model
Figure 3.84 Non-dimensional dynamic coefficients for shallow reservoir configuration as a function of eccentricity \((k)\): 1E-13 m\(^2\)-isothermal model
Figure 3.85 through Figure 3.88 present the stiffness and damping coefficients computed using the thermo-hydrodynamic model, and for the same geometric and functional parameters as Figure 3.81 through Figure 3.84.

The bearing continues to have an elastic response close to zero for high permeabilities cases, which is independent of the reservoir depth. When permeability is decreased one observe the same effects as in the isothermal case: the elastic response is non-zero and has small positive values. The cross-coupled coefficients are symmetrical, hence the bearing is stable.

The values for the non-dimensionalized stiffness and damping coefficients for thermal and isothermal models are very similar due to the fact that they are non-dimensionalized with respect to the generated load.
Figure 3.85 Non-dimensional dynamic coefficients for deep reservoir configuration as a function of eccentricity ($k$): 1E-11 m$^2$-thermal model
Figure 3.86 Non-dimensional dynamic coefficients for deep reservoir configuration as a function of eccentricity ($k$): 1E-13 m$^2$-thermal model
Figure 3.87 Non-dimensional dynamic coefficients for shallow reservoir configuration as a function of eccentricity ($k$): 1E-11 m$^2$-thermal model
Figure 3.88 Non-dimensional dynamic coefficients for shallow reservoir configuration as a function of eccentricity \( (k) \): 1E-13 m\(^2\)-thermal model
Previous work presented the theoretical feasibility of a new porous bearing concept. A test rig was designed and fabricated in order to investigate and validate the results predicted by the numerical experiments.

Due to the difficulty of monitoring the pressure distribution in the clearance of a rotating shaft, the first goal was to monitor the pressure and temperature distributions in the wrapped around reservoir. These pressures and temperatures are monitored with pressure tabs and thermocouples placed around the circumference in the symmetry plane. The second goal was to monitor the displacements of both ends of the shaft with the use of proximity sensors and obtain the orbits. Third goal was to monitor the torque produced by the bearing. Measurements have been recorded for different loads and running speeds.

Figure 4.1 Schematic of the test installation
Figure 4.1 illustrates the schematic of the overall test bearing installation. The test loop contains a high-speed rotor (spindle) connected to a gearbox multiplier, connected to an electric motor. The high-speed spindle was manufactured by Whitnon Manufacturing. The electric motor is a 3-HP continuous duty AC induction motor fabricated by Louis Allis Co. The test loop provided consistent variable speeds for up to 5000 rpm, which represents the highest speed limit for which the bearing was tested as part of this dissertation research.

The design of the components of the test head was made so that they can be easily assembled and disassembled. The CAD software SolidWorks has been used for design and execution drawings generation. The complete execution drawings for all the components are presented in APPENDIX O.

The design of the bearing starts with the dimensions of the porous bushing. These porous cylinders have been custom made by Mott Corporation following a sintering process of metal particles with different diameters ranging from 0.2 microns to 100 microns. The bushings have been molded in a cast with the following dimensions: ID-1 inch, OD-1.25 inches and length-1.5 (see Figure 4.4). Therefore, the L/D ratio of the bearings is 1.5.

A complimentary part of the experimental work was the determination the porous media characteristics for the samples provided by the Mott Corporation (see paragraph 4.2).
4.1 Design of the test head for the porous bearing insert

An ensemble view of the test head is illustrated in Figure 4.2. The way the various components interact with each other is illustrated in Figure 4.3.

Figure 4.2 Isometric view of the test section assembly (SolidWorks model)
Figure 4.3 Cross-sectional view of the test section assembly (SolidWorks model)

Figure 4.4 Porous bushings grade 100 (a) and grade 40 (d)
4.1.1 Design of the shaft

The bearing shaft (Figure 4.5) was machined out of 4340 PHT Steel to a diameter of 0.996 inches, hence allowing for a concentric clearance of 0.002 inches.

The shaft is the fixed part; the casing is the component that is “floating” to adjust the relative position between the shaft and the casing when load and angular speed are varied. Both ends of the shaft are resting on super precision SKF rolling bearings-MM204K (Figure 4.3) enclosed into two uprights (Figure 4.2) custom designed particularly for this experiment. At one end, the ball bearing is assembled with a preload class C of 200 N (according to the manufacturer specifications). For this purpose, 4 precision compression springs 9435ZK48 have been used. Each spring could hold a load of 51N at a compressed length of 12.7 mm.

Spiral groove type seals have been machined on the shaft surface and designed in such a way to have an inward pumping and sealing effect. The design of the spiral groove seals is presented in section 4.1.3. To ensure that even after passing the spiral groove the fluid will not leak sideways, high speed lip seals SKF-9706 CR 25X35X7CRW1 V were considered (Figure 4.3). These seals have the rotor diameter fitted on the shaft and will slightly deform with the shaft rotation. The manufacturer specs have been followed to accommodate the bore diameter and width in the casing where the seals are press fitted (see executions drawings in APPENDIX O). These seals will also ensure that there is no fluid leaking when the bearing is not functioning. The maximum speed allowed for the rotor surface when using this particular set of seals was 7000rpm. After a limited time of test runs, due to visible wear, the lip seals were being replaced with new ones to maintain zero leakage conditions throughout the entire matrix of experimental runs.
4.1.2 Design of the casing

Two casings machined out of aluminum with different reservoir depths (a deep configuration of 0.125 inches and a shallow configuration of 0.01 inches) were designed to enclose the porous bushing, see Figure 4.6. The dimensions for the reservoir depths correspond to the ones in the numerical model.

The porous bushing can be inserted and removed from one end. This end of the bearing is closed with a flange, which has machined in it the bore diameter of the static lip seal. The flange is connected to the bearing housing and secured with screws. These components do not move with respect to each other during the experimental runs. The relatively simple way in which they are relating to each other is clearly illustrated in Figure 4.3.
Figure 4.6 Bearing casing (SolidWorks model and physical model)
For instrumentation purposes, at one end of the casing and on the flange two \( \Phi \frac{3}{4} \) holes placed at 90 degrees with respect to each other have been machined so that they can accommodate the proximity sensors. Eight pressure ports have been drilled circumferentially at equal distance of each other, in the middle plane of the bearing (one every 45 degrees). The diameter of these holes was \( \Phi 0.0670 \) inches. In the axial direction four holes have been drilled with the purpose of measuring the performance of the spiral groove seal.

For temperature measurements, eight holes with \( \Phi 0.0938 \) were drilled in the middle plane location, circumferentially, also spaced at equal distance of 45 degrees between the centers of the holes.

The physical components of the solid work model presented in Figure 4.3 are illustrated in Figure 4.7. Figure 4.8 illustrates the overall installation as seen on the schematic of Figure 4.1. Figure 4.9 through Figure 4.11 illustrate the test head together with the instrumentation lines seen from lateral and top views.

The entire test rig is fixated to a rigid metal base plate to minimize vibration.
Figure 4.7 Test section components
Figure 4.8 Test section (general view)
Figure 4.9 Test section (lateral view right)
Figure 4.10 Test section (lateral view left)
Figure 4.11 Test section (top view)

Oil reservoir
4.1.3 Active sealing using spiral groves

In bearing technology, the use of seals is required in order to prevent the side leakage of the lubricant. To assess this problem two types of seals were considered.

The spiral groove seals are contactless seals that have an advantage of a long life and non-damaging effects on the shaft while having very low leakage rates. Their usage offers the following advantages: greater stability under light loading conditions, the self pressuring effect of the grooves tends to eliminate the cavitation effect, the self pumping effect on the lubricant towards the center of the bearing eliminates the need of a pressurized supply source and the need to end seals. They consist of a recurrent pattern of grooves situated on one of the bearing surfaces. In 1951 Whipple [66] solved the hydrodynamic theory of a thrust bearing consisting of a plane disc rotating above a stationary disc on which a set of grooves arranged in a herring bone pattern was machined. The lubricant was being pumped towards the center of the bearing, while the smooth disc was rotating providing thrust. Essential design parameters were provided for herring-bone and spiral grooved types of bearing: the minimum separation between the discs, the width of the herringbone pattern, the width, depth and pitch of the grooves in such that the thrust produced reaches a maximum. Whipple pointed out the theoretically close connection between spiral grooved thrust bearings and viscosseals.

An experimental investigation paper focused on gas lubricated hydrodynamic-type bearings published in 1957 by Ford et al. [67] contained one of the first published discussions on the subject of spiral grooves. Essential design parameters were determined such that the maximum pressure generated within the bearing were obtained both for herring-bone and for spiral grooved designs. More work in the theory of gas lubricated
hydrodynamic bearings was performed recognizing that grooves are improving the load-carrying capacity and the stability of these bearings, Elrod [68], Whipple [66] and Muijderman [69].

In 1965 Stair [70] presented a theoretical analysis on the visco-type shaft seal. The analysis showed sealing coefficient and dissipation function variation for an concentric laminar flow case as a function of seal various geometries.

More recent work under the guidance of Yoshimoto is focusing on the application of the spiral grooves to the water lubricated conical hydrostatic bearings. They too concluded based on experimental and analytical results the contribution of the spiral groove to the load carrying and to the stability of the rotating shaft, as well as the influence of the design parameters on its stability Yoshimoto et al. [71][72]. Yoshimoto et al. [73] also concluded that the bearings achieve larger load capacity with numerous spiral grooves rather than with the conventional four spirals design.

Jang and Chang [74] analyzed the herringbone grooved journal bearing of a spindle motor in a computer hard disk drive using Reynolds equation and incorporating an Elrod type cavitation algorithm. They concluded that the load capacity and bearing torque are increasing with the increase of eccentricity, length to diameter ratio and the decrease of the groove width ratio. They also noted a significant decrease of the cavitation region due to the inclusion of the herringbone grooves. The maximum load capacity was found to occur for a groove angle of 30 degrees.

In our present case, we considered machining the grooves on the shaft surface, hence the rotating surface. For the case of an incompressible lubricant the case of grooves
inscribed on the stationary member will translate in the same performance characteristics. The considered configuration was chosen for easier machining considerations.

The grooves orientation and the shaft rotation are coupled as such that the fluid is being pumped towards the center of the bearing. Only a certain orientation for the grooves direction with respect to the rotating shaft will ensure inwards pumping of the fluid and hence sealing capabilities. A shaft rotating in an opposite direction will have an outward pumping effect.

Design considerations where followed according to Whipple [66] , Muijderman [69] , Chow and Vohr [75].

The grooves located at the two axial ends are positioned symmetrically with respect of the center of the bearing. According to the mentioned authors, the performance of the helical-grooved bearing depends on the values of groove parameters such as: groove angle ($\beta_g$), the ratio of groove width to total width ($\alpha_g$), ratio of groove clearance to ridge clearance ($\Gamma_g$), and ration of length of grooving to total length of the bearing ($Y_g$).

![Spiral groove seal schematic](image)

Figure 4.12 Spiral groove seal schematic
Following the guidelines in Chow and Vohr [75], for a laminar regime (Re < 500) the optimum parameters are: \( \alpha_g = \frac{a_g}{a_g + a_r} = 0.5 \), \( \beta_g = 151.5 \) degrees, \( Y_g = \frac{L_1}{L} = 0.75 \) and \( \Gamma_g = \frac{h_g}{h_r} = 2.1 \).

The groove depth is the computed accordingly to be 0.0022 inches to correspond to a bearing concentric clearance of 0.002 inches. The length of the groove is determined to be 0.5625 inches. Considering a number of 20 grooves (one groove every 18 degrees) computing the arc length of 18 degrees for our bearing case and using the formula for \( \alpha_g \) ones obtains a groove width of 0.065, while the spacing between grooves is 0.065 inches according to the expression for \( \alpha_g \). The complete execution drawing of the spiral groove on the shaft is presented in APPENDIX O.

4.2 Experimental determination of permeability and porosity.

Figure 4.13 presents various porous material samples available for choosing the material for the porous bushing. These amorphous, sinterized, isotropic, easy to machine materials are manufactured by Mott Corporation in Farmington, CT.

An installation was designed with the goal of measuring the permeability for each of these samples. According to the Darcy law, permeability is a measure of the flow conductance of the matrix. To determine the permeability of the materials an installation presented in Figure 4.15 was constructed. The two things that needed to be treated with precaution where the bypass flows around the sample and maintaining a flow regime that is laminar.
Positive displacement Flow Meter (Figure 4.14)

An Omega FPD 2024 positive displacement flow meter (Figure 4.14), connected to a 12.4 V supply source was used to estimate the flow rate. The accuracy of the flow meter was ±0.5% of the displayed value. The output of the flow meter is a square wave of varying frequency (TTL) related to the angular speed of the internal gear, hence the name positive displacement flow meter. This frequency output is related to the flow rate measured.

Details on the pressure transducers have been given in section 4.4. The experimental procedure was the following: for each sample tested the mass flow was increased. This increase affected the pressures at the inlet and outlet (Figure 4.15) as the flow through the porous media was adjusting to the new flow debit. The pressure drop across the sample was calculated.

The permeability value was calculated from the Darcy equation for the pressure drop across a porous media: \( \Delta p = \frac{Q\mu L}{kA} \), where \( \Delta p \) and \( Q \) are recorded values, \( A = 0.000504683 \text{ m}^2 \) is the effective area (see Figure 4.16), \( L = 0.00199134 \text{ m} \) the thickness of the porous sample and \( \mu = 0.02 \text{ N s/m}^2 \) the dynamic viscosity of the fluid.

The values obtained are presented in Figure 4.17. For grain sizes of 5 micrometers the permeability obtained was 1.1E-12 m², for grain sizes of 10, 20 and 40, the permeability was 2.5E-12 m², 7E-12 m², and 1.4E-11 m². For samples with grain sizes 5, 0.5 and 0.2 presented in Figure 4.13, the pressure needed to push fluid through was not possible to reach using the pump and equipment in our lab facility.
Figure 4.13 Porous media grades from 0.2 to 100 (the numbers on the picture indicate the grain size in micrometers)

Figure 4.14 Omega positive displacement flow meter
Figure 4.15 Experimental apparatus for permeability testing

Figure 4.16 Experimental apparatus for permeability testing (inside view)
Based on these results a sample of grain size 40 was selected for the bearing prototype.

The overall uncertainty for the measurement was the combination of the errors of each component in the deterministic equation 

\[ e_k = \pm \sqrt{e_p^2 + e_Q^2 + e_{\mu}^2 + e_L^2 + e_A^2} = \pm \sqrt{\left(\frac{u_p}{p}\right)^2 + \left(\frac{u_Q}{Q}\right)^2 + \left(\frac{u_{\mu}}{\mu}\right)^2 + \left(\frac{u_L}{L}\right)^2 + \left(\frac{u_A}{A}\right)^2} \]

\[ u_p = 0.25\% \text{ of the full scale (0.25PSI), } u_p = \frac{1}{2} \text{ of the caliper resolution (0.000127m),} \]
\[ u_{\mu} = 1.5\% \text{ of the displayed value (0.0003 Ns/m}^2), \ u_Q = 0.5\% \text{ of the displayed value (0.0125 gal/min). All these contributions add to a maximum error estimate of } u_k=6.77\%. \]

The estimation of porosity for samples illustrated in Figure 4.4 was made by comparing the weighs of the samples to the weight of identical sized sample made of unsintered material. The results of these measurements are presented in Table 4.1

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Figure 4.17 Experimentally determined permeability
4.3 Experimental determination of viscosity and density for the lubricating oil.

Experimental measurements for viscosity and density as a function of temperature have been performed.

The working fluid was DOW Corning 200 (polydimethylsiloxane). According to the product information page this oil has an excellent viscosity-temperature characteristic, is thermally and chemically stable, has shear-breakdown resistance, and is compatible with rubber (information of high interest, since the bearing application uses viton lip seals). It has little change in its physical properties over a wide temperature span (a relatively flat viscosity-temperature slope and serviceability from 40°C to 200°C). Boiling point is located at 140°C and flash point is 326°C (Open Cup) and 100°C (Closed Cup).

Viscometer (Figure 4.18).

The viscometer is a Cambridge Applied Systems with an accuracy of ±1.5% of the displayed viscosity. Cambridge Applied Systems calibrated this viscometer for a specific viscosity range. The measurements have been performed while the temperature was increased using a special constant temperature baths. The second sets of measurements have been performed when the oil temperature was on the decreasing slope (Figure 4.20).

Hygrometer (Figure 4.19)

The density of the oil was calculated using hydrometers (tool made out of glass and metal used for measuring the "specific gravity" of liquids that is, determining whether a liquid is more or less dense than water). The oil was placed in a glass
Figure 4.18 Setup for establishing the oil viscosity variation with temperature.

Figure 4.19 Hygrometer
container and hydrometers were immersed. The reading on the scale of the floating hydrometer was recorded. As the temperature increased the density of the oil decreased and the hydrometer was slowly lowering, new readings being recorded. The error of the measurement is \( \frac{1}{2} \) of the graded interval (0.0005-the increments on the scale where 0.001). The oil density was obtained by multiplying the specific gravity reading on the hydrometer with the density of water (1000 kg/m\(^3\)).

![Graph showing viscosity and density as a function of temperature.](image)

Figure 4.20  Experimental measurement of oil viscosity and density as a function of temperature
4.4 Instrumentation and calibrations procedures

4.4.1 Pressure transducers

Two types of pressure transducers were used to measure the circumferential pressure (1 to 8 illustrated in the upper picture of Figure 4.21 and the pressure transducers used for the active sealing zone 9 to 12). These pressure transducers are being manufactured by Transducers direct, Cincinnati. The circumferential pressure transducers are: four TDH30CG001503D004 with range 0-15 psig, 1% accuracy, and four TDH30CG002503D004 with a range of 0-25 psig and 1% accuracy. A voltage of 5 V is
supplied. This type of pressure transducers operate by having resistance strain gages placed on the non wetted side of a diaphragm surface. The diaphragm is exposed to the fluid on wetted side and to a known reference pressure on the other (a vacuum or atmospheric pressure). The difference in pressure from the two sides creates a strain on the diaphragm which is measured as a voltage change which is then converted to a pressure reading through use of a calibration curve which can be found in APPENDIX A.

Figure 4.22 Circumferential location of the pressure ports in the symmetry plane

4.4.2 Thermocouples

Eight Omega type-K thermocouples (accuracy of ±2 °F) have been used in the experiment. They are distributed at even distance (Figure 4.23) along the entire circumference at the median plan of the bearing.
4.4.3 Proximity sensors

The 4 proximity sensors used (Figure 4.24) are Bentley Nevada 8 mm model, with a 5 m extension cable. Proximeters 1 and 2 are located on the motor side (proximeter 1 is horizontal, proximeter 2 is vertical), while proximeters 3 and 4 are located at the opposite side (proximeter 3 is horizontal, proximeter 4 is vertical). These proximity sensors are located at exactly 90 degrees with respect to each other. They required calibration since the voltage reading is sensitive to the material and specific shape of the target whose displacement is being measured. The supply voltage was -24V. The calibration set up is illustrated in Figure 4.25. The resolution of the measurement is 0.00001 inches, and the error is 0.000005 inches. The results of the calibration are included in APPENDIX B.
Figure 4.24 Proximity sensors signal and power units

Figure 4.25 Proximity sensor calibration
4.4.4 Strain gage

The force applied to the bearing, the torque is measured using a strain gage. The strain gages allow the measurements of the strain caused when an applied force induces stress in the object. When stretched, a wire will increase in length and thin in cross-section. This causes an increase in resistance of the wire constant material resistivity. A Wheatstone bridge was used to convert the resistance to voltage that was recorded. Wheatstone bridge circuits are based on the principle of comparing the voltage drop across parallel resistance “legs”. If all the resistances in the bridge are the same, then the voltage drop across them is equal and the bridge output voltage is zero. If a small imbalance exists between the resistances, then the bridge generates a proportional voltage output.

Before measurements are taken, the resistances in the Bridge circuit must be balanced. Pressing the CAL button while C is set to 10 would simulate a strain of \( \varepsilon_{cal} = 961.5 \times 10^{-6} \), compared to the voltage read out which is the gain \( V_{out} = 2.06 \) one obtains a calibration factor of \( K = 2142.48 \text{V} \) \( (V_{out} = K\varepsilon_{cal}) \). This calibration is performed without load on the bearing. When the system is loaded, the output voltage will be proportional to the applied strain and the factor of proportionality is \( K \) \( (V_{measured} = K\varepsilon_{axial}) \).

Using Hook’s Law \( \varepsilon_{axial} = \frac{\sigma_{axial}}{E_{steel}} \) one can obtain the axial stress \( \sigma_{axial} \) knowing \( E_{steel} = 193.053 \times 10^9 \text{N/m}^2 \). For the simple bending of a cantilever beam, the axial stress is related to the bending moment \( M \), the moment of inertial \( I = \frac{bh^3}{12} \) and the thickness of the beam \( h \) through the formula: \( \sigma_{axial} = \frac{Mh}{2I} \). Therefore the bending moment which is equal
with the torque in the system can be estimated. \( M = \frac{V_{measured}}{K} E \frac{bh^2}{6} \). The error is estimated \( e_M = \pm \sqrt{e_V^2 + e_b^2 + e_h^2} = \pm 0.00086 \text{Nm} \)

4.4.5 Speed sensors Monarch Instruments

The speed sensor is a Monarch Instruments LED Remote Optical Sensor and is detecting a reflective tape adhered to the coupling as seen in Figure 4.26.

Connected to a 5V supply, the speed sensor is emitting a pulse for every rotation of the shaft by detecting a reflecting target. The speed range of the sensor is 1-250,000 rpm. The output of the speed sensor is a TTL signal counted through the data acquisition system.

4.4.6 System Neff-470

System Neff-470 (Figure 4.27) is a data acquisition “front end” subsystem controlled externally by a host computer and internally by microprocessor-operated ROM software. A variety of System 470 function cards accommodate both analog and digital I/O signals. Continuous or single scan acquisition can be selected with sample rates of 1 or 10 kHz. Sixteen card slots are provided for the installation of I/O function cards. The resolution of each channel is 16 bits for an input voltage of \( \pm 5 \text{mV} \) to \( \pm 10 \text{mV} \). For noise removal, the analog input cards have built in filters.
Figure 4.26 Optical speed sensor
Figure 4.27 Data acquisitions and storage unit
CHAPTER V

RESULTS AND DISCUSSION

The experimental research is required in order to validate the numerical results obtained in previous chapters.

The test installation and the test head presented in Chapter IV have been experimentally investigated. The displacement of the casing with respect to the shaft, the torque, pressure, and temperature distributions in the wrapped around reservoir of the bearing when the angular speed and the static load have been parametrically changed have been recorded. The results have been analyzed and found to present essential similarities when compared to the results of the numerical model.

5.1 Experimental procedure

Before performing the experiments, the test head was disassembled, each component diligently cleaned and the test head reassembled. Throughout the entire matrix of experiments one porous bushing was used, therefore it was very important to clean the debris that might have been worn during the previous tests and might have been
resting in the porous material. This debris in the porous bushing was flushed out using break cleaner and then the bushing was dried using high-pressure air. With all components assembled, the lubricant was inserted into the system through the small reservoir seen in Figure 4.11. The air was carefully “bled” out from all the pressure lines which led to pressure transducers. The reference position of the proximity sensors was recorded and later used in the data analysis.

For safety precautions and to ensure that there is no touch down between the shaft and the bearing bushing, a small pressure of 2 PSI was applied to the bearing system through the oil reservoir. When interpreting the results this added pressure was subtracted, to show only the pressure built by the system. Throughout the matrix of experiments there was no axial leakage, and due to continuity the oil levels in the reservoir used to fill the system did not vary. The oil that was impregnated in the porous and filled the clearance and reservoir gaps have been continuously circulated during the experiments. The small reservoir containing oil was used only for precaution measures, and did not affect the system.

The room temperature was not constant throughout the experiments; differences of up to 10 F have been experienced.

The matrix of experimental testing (Table 5.1) was designed such that for each loading configuration considered, the speed was increased with 1krpm increments, and the test was ran for 1 hour at each speed. The variable inputs have been geometrical: two reservoir configurations: shallow and deep and operational: the angular velocity of the shaft and the applied load on the casing. The shaft had an angular velocity counterclockwise. The output variables have been: the circumferential temperatures and
Table 5.1 Matrix of experimental runs

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500rpm
pressures in the reservoir at the plane of bearing symmetry, the torque and the
displacement of the casing with respect to the shaft measured at both ends. The results for
deep and shallow reservoir configurations for 5lb loads are being further presented and
analyzed. Complete results for all loading configurations considered in the matrix of
experiments are presented in appendices: pressure distribution in APPENDIX C, D;
temperature distribution in APPENDIX E, F; orbits in APPENDIX G, H; torque in
APPENDIX K and L.

5.2 Pressure distribution in the reservoir

The circumferential variation of pressure in the bearing reservoir, at the plane of
symmetry, for various journal speeds with an applied static load of 5lb for a deep
reservoir configuration is shown in Figure 5.1. The pressure distribution for a shallow
reservoir using the same operational parameters is shown in Figure 5.2. The
circumferential direction is represented by the readings of the 8 equally spaced pressure
transducers.

As predicted from the numerical experiments, the pressure distribution in the
reservoir for deep case configuration is linear and constant throughout the circumferential
direction, while in the shallow case is a Sommerfeld type curve. The pressure magnitudes
are higher in the deep reservoir case compared to the shallow reservoir for all running
speeds. This result is consistent with the finding in section 3.1.5. When increasing the
speed, the magnitude of the pressure in the deep reservoir case appears to be decreasing
slightly, while for the shallow reservoir case, the pressure is slightly increasing. This
Figure 5.1 Circumferential pressure for a deep reservoir configuration; 5-lb load
Figure 5.2 Circumferential pressure for a shallow reservoir configuration; 5-lb load
Figure 5.3 Circumferential pressure for a shallow reservoir configuration; 3000 rpm
characteristic is recovered throughout the entire matrix of experiments (see APPENDIX C and APPENDIX D). The results for shallow reservoir case are consistent with the numerical findings presented in section 3.1.1. The numerical experiments did not take into consideration the effects of spiral groove seals or the effects of the lip seals, therefore an exact match between experimental and numerical model is not realistic.

This property of the pressure curves (constant pressure for the deep configuration and Sommerfeld type curve for shallow configuration) is consistent throughout the entire matrix of experiments (APPENDIX C and APPENDIX D).

Furthermore, the pressure distribution in the shallow bearing reservoir for different applied loads and constant rotational speed is presented in Figure 5.3. The increase in load translated into an increase in eccentricity and further into an increase in the pressure in the clearance of the bearing and consequently the pressure in the reservoir. These results are consistent with the numerical findings presented in section 3.1.2.

5.3 Temperature variation in the reservoir

Figure 5.4 and Figure 5.5 illustrate the temperature evolution with time when the bearings system was loaded with 5lb and the speed of the shaft was parametrically increased to 5000 rpm. The deep configurations reached a maximum value of 333K after 1 hour of running at 5000 rpm and the shallower case reached the value of 342K. These values are similar to the ones predicted numerically (see Figure 3.70 and Figure 3.71). The differences come from the fact that the numerical model had the axial outlets opened
Figure 5.4 Temperature increase when the velocity was changed parametrically for a deep reservoir; 5-lb load

Figure 5.5 Temperature increase when the velocity was changed parametrically for a shallow reservoir; 5-lb load
to the reference temperature, which was a source of heat loss, while the practical application was sealed with viton lip seals. In addition, the experimental model has a larger casing to incorporate the seals and does not incorporate a system of fin pins heat exchangers. All this factors contribute to the difference between the numerical model and the experimental results. However, after five hours of continuum testing the temperature was stabilizing for each increasing shaft speed, therefore the thermal management is successful and the proposed system is feasible.

Figure 5.6 and Figure 5.7 present the circumferential temperature variation in the reservoir at the symmetry plan of the bearing for deep and shallow configurations. Both configurations present a circumferential temperature variation below 0.4K. The strong convective effects inside the reservoir are responsible for this temperature homogenization. The maximum temperature reading is at temperature ports 2 and 3 (see Figure 4.23 for location of the temperature ports). This section represents the area of minimum clearance were the shear effects are most pronounced and there the fluid exists the clearance region (see Figure 3.2). As a reminder: the shaft is the fixed part of the bearing, and the housing is the floating component. When the housing is loaded the region of minimum clearance is located in the upper side and not the lower region as in the case of conventional, fixed housing-loaded shaft configurations. The temperature variation in the shallow reservoir is less pronounced that in the deep reservoir case, a 0.2K degrees difference between the maximum and the minimum value. This result is consistent for all running speeds considered and is due to the strong convective effects that are more pronounced in the shallower case.
Figure 5.6 Circumferential temperature for a deep reservoir; 5-lb load
Figure 5.7 Circumferential temperature for a shallow reservoir configuration; 5-lb load
In addition, from the study of the Figure 5.6 and Figure 5.7 one can conclude that the temperature increased with the increase of the shaft velocity and that the deep reservoir appears to have better cooling capability for all tested speeds. This qualitative result was obtained numerically for the case with closed outlets in section 3.2.4.

These findings are valid for all configurations tested and presented in the matrix of experimental runs (Table 5.1). All these experimental results are illustrated in APPENDIX E and APPENDIX F. It should be noted that the reference temperature in the room was not constant for all runs and its variation was illustrated on all graphs.

5.4 Eccentricities, attitude angles and orbits

The four proximity sensors have measured the displacements between the bearing housing and the shaft, at both ends of the shaft. The results presented in Figure 5.8 and Figure 5.9 illustrate the bearing locus obtained for a 5lb loading configuration and varying the shaft speed parametrically for a deep and a shallow reservoir configuration. During the running cycle, the shaft positioned itself to various locations at different rotating speeds; the time evolution between these positions (the full orbit) is illustrated in Figure 5.10 and Figure 5.11. The bearing locus was obtained performing a time average of the orbits for each rotating velocity. The vertical and horizontal components of the locus were used to estimate the attitude angles. These values are illustrated in Table 5.2 thought Table 5.4. Measurements for the minimum and maximum reading of the proximeters revealed a concentric clearance of 0.003 inches.
When comparing the orbits for deep (Figure 5.11) and shallow reservoir case (Figure 5.12) it can be concluded that the eccentricity did not vary considerably. The lip seals have an important effect on the values of the eccentricity. They are tightly fitted on the shaft and have the tendency to keep the shaft in a concentric position relative to the bearing housing, therefore the small eccentricity values of 0.5 to 0.7 are obtained for a deep reservoir (Table 5.2). In addition, it is clearly illustrated in Figure 5.8 and Figure 5.9 that the region of minimum clearance is situated in the upper region of the casing, since under the load applied the casing tend to move downward and the shaft is fixed.

Common for all configuration tested is the fact that the bearing housing was not perfectly aligned. This led to the ‘tilting’ of the housing resulting in different attitude angles for the motor side compared to open side. The values that are marked with (*) in the tables illustrated the case when the attitude angles shifted to a different quadrant compared to the initial position.

Figure 5.10 illustrates the case of a shallow reservoir for a running speed of 3 krpm when the load is parametrically increased. The housing is less tilted, with the exception of 0 and 3lb loading cases.

Qualitatively it can be observed that the bearing system was capable of holding load and running in a stable regime. The numerical model did not take into account the effects of the spiral groove seals, lip seals, thus making the comparison between the numerical results and the experimentally obtained data difficult.
Figure 5.8 Shaft locus for a deep reservoir configuration, 5-lb load, various speeds

Figure 5.9 Shaft locus for a shallow reservoir configuration, 5-lb load, various speeds

Figure 5.10 Shaft locus for a shallow reservoir configuration, 3krpm speed, various loads.
Table 5.2 Eccentricity and attitude angles for a deep reservoir, 5lb load

<table>
<thead>
<tr>
<th>Rotational speed [rpm]</th>
<th></th>
<th>eccentricity</th>
<th></th>
<th>attitude angle</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Motor side</td>
<td></td>
<td>Open side</td>
</tr>
<tr>
<td></td>
<td>1,000</td>
<td>0.702886</td>
<td>0.648777</td>
<td>17.28346</td>
</tr>
<tr>
<td></td>
<td>2,000</td>
<td>0.572868</td>
<td>0.657922</td>
<td>37.45757</td>
</tr>
<tr>
<td></td>
<td>3,000</td>
<td>0.549579</td>
<td>0.677525</td>
<td>54.65896</td>
</tr>
<tr>
<td></td>
<td>4,000</td>
<td>0.564954</td>
<td>0.731594</td>
<td>49.18621</td>
</tr>
<tr>
<td></td>
<td>5,000</td>
<td>0.504226</td>
<td>0.634693</td>
<td>64.10153</td>
</tr>
</tbody>
</table>

Table 5.3 Eccentricity and attitude angles for a shallow reservoir, 5lb load

<table>
<thead>
<tr>
<th>Rotational speed [rpm]</th>
<th></th>
<th>eccentricity</th>
<th></th>
<th>attitude angle</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Motor side</td>
<td></td>
<td>Open side</td>
</tr>
<tr>
<td></td>
<td>2,000</td>
<td>0.453826</td>
<td>0.589022</td>
<td>74.27977</td>
</tr>
<tr>
<td></td>
<td>3,000</td>
<td>0.444045</td>
<td>0.617646</td>
<td>36.34399</td>
</tr>
<tr>
<td></td>
<td>4,000</td>
<td>0.369669</td>
<td>0.593069</td>
<td>5.734575</td>
</tr>
<tr>
<td></td>
<td>5,000</td>
<td>0.58429</td>
<td>0.559087</td>
<td>12.331(*)</td>
</tr>
</tbody>
</table>

Table 5.4 Eccentricity and attitude angles for a shallow reservoir, 3krpm speed

<table>
<thead>
<tr>
<th>Load [lb]</th>
<th></th>
<th>eccentricity</th>
<th></th>
<th>attitude angle</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Motor side</td>
<td></td>
<td>Open side</td>
</tr>
<tr>
<td>0</td>
<td>0.601485</td>
<td>0.239153</td>
<td>38.829</td>
<td>83.2384</td>
</tr>
<tr>
<td>1</td>
<td>0.504378</td>
<td>0.288973</td>
<td>48.58883</td>
<td>69.12706</td>
</tr>
<tr>
<td>2</td>
<td>0.563871</td>
<td>0.387592</td>
<td>51.92891</td>
<td>59.64725</td>
</tr>
<tr>
<td>3</td>
<td>0.562095</td>
<td>0.632866</td>
<td>11.5639(*)</td>
<td>62.97386</td>
</tr>
<tr>
<td>5</td>
<td>0.444045</td>
<td>0.617646</td>
<td>36.34399</td>
<td>53.5171</td>
</tr>
</tbody>
</table>
Figure 5.11 Shaft orbits for deep reservoir configuration, 5 lb. load, and 1 to 5 krpm.
Figure 5.12 Shaft orbits for shallow reservoir configuration, 5 lb. load, and 2 to 5 krpm.
5.5 Total torque

The variation of the total torque with the increasing angular speed for a deep reservoir configuration is shown in Figure 5.13. The variation of friction torque for a shallow reservoir configuration is illustrated in Figure 5.14.

When the test journal is rotating, the friction torque on the bearing will try to rotate the bearing casing. The torque measured with the strain gages represents the overall effect due to: shaft rotation, spiral groove seal, and most important the influence of the lip seals which are fitted on the shaft hence producing resistance to movement (drag).

The starting torque, defined as the moment that must be overcome in order for the bearing to start rotating from the stationary condition, is captured for both reservoir

![Figure 5.13 Torque measurement for a deep reservoir configuration](image)
configurations (Figure 5.13, Figure 5.14). Starting bearing torque comprises of two elements which need to be overcome before rotation occurs: the friction generated between the lip seal and the shaft and the shear forces in the lubricant which are higher at lower temperatures. With the increase of angular speed the temperature of the lubricant increased. The increase in the temperature of the bearing decreased the coefficient of friction since the temperature caused the decrease in viscosity. The lubricant reached a viscosity point associated with smooth operation and the torque did not increase with the increase of angular speed from this point. For shallow reservoir from 2krpm to 5 krpm there is no variation in torque, while for the deep reservoir from 4krpm to 5krpm the torque is unchanged.

Figure 5.14 Torque measurement for a shallow reservoir configuration
One parameter that was not taken into consideration was the variation of the torque produced by the lip seals with running time. The seals have been changed periodically, however they were not changed after each run. This would have ensured the same torque conditions and represents an aspect that needs to be taken into consideration in future testing. Current results suggest that the torque for the shallow reservoir case is considerably higher than the torque produces for the deep reservoir case.

The total torque measurements for all loading configurations tested are presented in APPENDIX K and APPENDIX L indicating to an increase of the torque with the increase of load.

To conclude the findings in this section, the torque was clearly affected by the lip seals that were tightly fitted on the shaft. The torque increased with the increase of angular speed until it reached a threshold after which it remained constant.
CHAPTER VI

CONCLUSIONS AND FUTURE WORK

The present work proved, using numerical simulations and experimental validation, the feasibility of a self-lubricating tribological system where the fluid was continuously circulated as a result of continuity and momentum laws and in the absence of an external circulating system. The 3D numerical solution was obtained via the commercial code CFD-ACE+ and used a finite volume methodology to discretize the governing equations.

The numerical results concluded that the permeability of the porous media is controlling the pressure in the bearing clearance, as well as in the reservoir region. The increase in permeability translated into a decrease in pressure and hence in load carrying capability. For higher permeability ranges (1E-10\text{m}^2 \text{ and } 1E-11\text{m}^2), the depth of the wrap-around reservoir played an important role in obtaining a maximum pressure in the active space; this has been found to be caused by the fact that the active space was effectively enlarged by the addition of the reservoir and the shaft saw a ‘larger virtual clearance’. For low permeability ranges (1E-12\text{m}^2 \text{ and } 1E-13\text{m}^2) the pressure in the active zone was not affected by the depth of the attached reservoir since the porous medium introduced a high resistance to flow and eliminated the effect of the ‘virtual clearance’.
The increase in porosity had the same effect as the increase in permeability upon the pressure build-up. The increase of any of these two properties translated into a decrease of pressure magnitude in the active region. The thickness of the porous bushing increase also translated into an increase of the pressure magnitude. Furthermore, the numerical results have illustrated the inverse interdependence between the strength of the flow circulation (between the active region and the reservoir) and the load carrying capacity.

The thermo-hydrodynamic simulations proved that the system can be cooled efficiently by increased natural convection with the addition of an array of heat exchanging fins. Multiple fins configurations were tested and an optimal configuration was established. The thermal model predicted lower pressure magnitudes while the circulation concept was still valid. The bearing parameters had the same type on influence on the pressure build up as they did in the isothermal numerical model.

Furthermore, the static and dynamic characteristics of this new type of bearing have been investigated. Both shallow and deep configurations presented similar attitude angles for the eccentricities tested when keeping the permeability of the porous bushing constant, concluding that the reservoir depth had very small influence on these parameters. In addition, for low permeability ranges (1E-12m^2 and 1E-13m^2), the attitudes angles of the porous bearing were, as one expects, fairly similar to those of a solid bearing. For the higher eccentricity ratios this porous bearing exhibited the properties observed by Cameron et al.[21] which stated that the pressure did not trend towards infinite values for an eccentricity equal 1 therefore the attitude angles did not reach the 0 value. For the thermodynamic model, the system presented higher attitude
angles for the high permeabilities case when compared to the isothermal case, indicating a more precarious stability. The dynamic characteristics investigation revealed that for high permeabilities the system had very small elastic response, however when the permeability was increased the response was different then zero, as one expects. The reservoir depth did not influence these parameters, confirming the findings for the static coefficients. For the thermohydrodynamic model the dimensionless values for the dynamic coefficients were similar to the ones obtained for the isothermal model.

In the second part of the dissertation, the experimental tests have proved that the pressure distribution in the reservoir was qualitatively similar to the one predicted theoretically, and that the bearing was stable for all running speeds and static loading configurations tested. If was confirmed experimentally that the deep reservoir configuration had better cooling capabilities compared to the shallow configuration. In addition, both reservoir configurations did not overheat during the extensive testing periods, therefore proving the circulating concept and the associated thermal management concept.

To conclude, when comparing this porous bearing containing a wrapped around reservoir to a classic solid hydrodynamic bearing, the following differences are marked: the elimination of a circulatory system for the lubricant; the reduction in load carrying capability; the reduction in stability due to higher attitude angles; the capability to run at higher speeds without overheating due to the reduced shear in the clearance region.
Future work can be performed to continue the numerical investigations in the effects of the spiral groove seals and their contributions to the increase in pressure magnitude and stability of the bearing.

In addition, the experimental results for eccentricities and attitude angles have been greatly influenced by the axial lip seals. A test installation designed in such a way to collect the small oil leakage that might result after eliminating the lip seals represents an improvement and a step closer to comparing the experimental and the numerical results. Furthermore, the lip seals have greatly limited the running speed of the shaft due to the fact that they were tightly fitting on the shaft. The elimination of these seals will also allow testing at higher angular speeds. The loading table was a simple design that can be improved upon to allow the application of higher loads. Finally, the use of liquid metal lubricants such as gallium and indium represents a step further for the development of a low maintenance, high speed, and high temperature bearing.
BIBLIOGRAPHY


APPENDICES
APPENDIX A

PRESSURE TRANSDUCERS CALIBRATION

Figure A1 Pressure calibration curves for pressure transducers 1 to 6
Figure A2 Pressure calibration curves for pressure transducers 7 to 12
APPENDIX B

PROXIMITY SENSORS CALIBRATION

\[ y = -0.0046x - 0.003 \]

\[ y = -0.0047x - 0.002 \]

\[ y = -0.0046x - 0.0047 \]

\[ y = -0.0046x - 0.0043 \]
APPENDIX C

PRESSURE DISTRIBUTION IN THE RESERVOIR FOR A DEEP CONFIGURATION

Figure C1 Circumferential pressure for a deep reservoir configuration; baseline
Figure C2 Circumferential pressure for a deep reservoir configuration; 1-lb load
Figure C3 Circumferential pressure for a deep reservoir configuration; 2-lb load
Figure C4 Circumferential pressure for a deep reservoir configuration; 3-lb load
APPENDIX D

PRESSURE DISTRIBUTION IN THE RESERVOIR FOR A SHALLOW CONFIGURATION

Figure D1 Circumferential pressure for a shallow reservoir; baseline
Figure D2  Circumferential pressure for a shallow reservoir; 1-lb load
Figure D3 Circumferential pressure for a shallow reservoir; 2-lb load
Figure D4 Circumferential pressure for a shallow reservoir; 3-lb load
APPENDIX E

TEMPERATURE DISTRIBUTION IN THE RESERVOIR FOR A DEEP CONFIGURATION

Figure E1 Circumferential temperature for a deep reservoir configuration; baseline
Figure E2 Circumferential temperature for a deep reservoir configuration; 1-lb load
Figure E3 Circumferential temperature for a deep reservoir configuration; 2-lb load
Figure E4 Circumferential temperature for a deep reservoir configuration; 3-lb load
APPENDIX F

TEMPERATURE DISTRIBUTION IN THE RESERVOIR FOR A SHALLOW CONFIGURATION

Figure F1 Circumferential temperature for a shallow reservoir, baseline
Figure F2 Circumferential temperature for a shallow reservoir; 1-lb load
Figure F3  Circumferential temperature for a shallow reservoir; 2-lb load
Figure F4 Circumferential temperature for a shallow reservoir; 3-lb load
APPENDIX G

SHAFT ORBITS FOR A DEEP RESERVOIR CONFIGURATION

Figure G1 Shaft orbits for deep reservoir configuration, baseline.
Figure G2 Shaft orbits for deep reservoir configuration, 1-lb. load
Figure G3 Shaft orbits for deep reservoir configuration, 2-lb. load
Figure G4 Shaft orbits for deep reservoir configuration, 3-lb. load
APPENDIX H

SHAFT ORBITS FOR A SHALLOW RESERVOIR CONFIGURATION

Figure H1 Shaft orbits for shallow reservoir configuration, baseline.
Figure H2 Shaft orbits for shallow reservoir configuration, 1-lb. load
Figure H3 Shaft orbits for shallow reservoir configuration, 2-lb. load
Figure H4 Shaft orbits for shallow reservoir configuration, 3-lb. load
APPENDIX I

LOCUS FOR A DEEP RESERVOIR CONFIGURATION

![Graphs showing locus for a deep reservoir configuration with baseline, 1lb load, 2lb load, and 3lb load at different RPMs.](image-url)
APPENDIX J

LOCUS FOR A SHALLOW RESERVOIR CONFIGURATION

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2lb load

---

3lb load
APPENDIX K

TORQUE MEASUREMENTS FOR A DEEP RESERVOIR

1lb load

2lb load

3lb load
APPENDIX L

TORQUE MEASUREMENTS FOR A SHALLOW RESERVOIR

1lb load

2lb load

3lb load
Figure M1 Temperature reading of thermocouple 1 when the speed is varied parametrically and the loading configuration is baseline and 1 lb.
Figure M2 Temperature reading of thermocouple 1 when the speed is varied parametrically and the loading configuration is 2lbs and 3 lbs.
Figure N1 Temperature reading of thermocouple 1 when the speed is varied parametrically and the loading configuration is baseline and 1 lb.
Figure N2 Temperature reading of thermocouple 1 when the speed is varied parametrically and the loading configuration is 2 lbs and 3 lbs.
APPENDIX O

EXECUTION DRAWINGS FOR BEARING COMPONENTS