Fluorescence Imaging of Lubricants in Microtextured Bearings

by

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Abstract

The work described here concerns the possibility that textured surfaces may reduce coefficients of friction and increase load support within hydrodynamic bearings. This topic is of particular interest in the context of the potential of texturing to reduce friction in internal combustion engines using techniques adapted to suit high volume production, such as Laser Honing. Despite the existence of a number of theoretical explanations (which include shear rate reduction, inlet suction, effect of the presence of cavitation and the mitigation of starvation) and numerous experimental reports as to the effect of texture on friction, there has been no simultaneous study of friction, cavitation and oil film thickness in textured hydrodynamic bearings. This is because such studies are difficult. One must address certain barriers which include the large range of oil film thicknesses encountered and the need for accurately known feature dimensions on the surfaces in question.

A new test rig has been designed and constructed that enables the simultaneous measurement of friction, load and oil film thickness. Novel features include a rotating glass disc on a silicon pad, a high-sensitivity fluorescence imaging microscope system to determine the oil film thickness and a non-contact displacement sensor for friction measurement. The process of photolithography was used to accurately produce predetermined textures on silicon surfaces. Tests were conducted on convergent plain and textured silicon pads whilst concurrently monitoring friction, load, cavitation (if present) and oil film thicknesses.

The results were compared to theoretical predictions based upon the solution of Reynolds equation, with cavitation, in two dimensions. Results show broad agreement with the theoretical predictions and suggest that textured surfaces may be either beneficial or detrimental according to both their geometry and the operating conditions.
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Nomenclature

\( \eta \) Dynamic viscosity
\( h \) Film thickness
\( U_1, U_2 \) Velocities of surface 1 and 2 in the \( x \) direction
\( U \) Glass disc speed
\( V_1, V_2 \) Velocities of surface 1 and 2 in the \( y \) direction
\( w_1, w_2 \) Velocities of surface 1 and 2 in the \( z \) direction
\( x, y, z \) Spatial co-ordinates
\( u, v, w \) Fluid co-ordinates
\( h_p \) Pocket depth
\( K \) Convergence ratio
\( K_f \) Fringe order
\( P \) Fluid film pressure
\( l_{arm} \) Length of the load arm
\( u \) Load arm displacement
\( F \) Friction
\( k_{sm} \) Stiffness of load arm elastic hinge
\( l \) Length of pad bearing
\( b \) Breadth of pad nearing
\( W^* \) Non-dimensional load support
\( W \) Load
\( dh_y \) Increment of pad bearing in the \( z \) plane
\( h_i \) Inlet film thickness
\( h_o \) Minimum outlet film thickness
\( h_{oL} \) Minimum outlet film thickness at the left hand side of pad
\( h_{oR} \) Minimum outlet film thickness at the right hand side of pad
\( h_{oCtr} \) Minimum outlet film thickness at the centre of the pad
\( \mu \) Coefficient of friction
\( b_s \) Length of elastic shim
\( d_s \) Thickness of elastic shim
\( E \) Young’s Modulus
\( I \) Moment of inertia
\( \theta \) Convergence angle of calibration piece
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Chapter 1

Introduction
1.1 Background

This chapter introduces the current need for the work being undertaken in the field of hydrodynamically flooded films in textured conformal surfaces. It aims to give a brief overview of the areas of focus that are of particular interest when considering such circumstances. Machined surfaces (such as piston liners situated within IC engines) have a certain degree of roughness as a result of the manufacturing process they undergo. Plateau honing is a prime example. This is in contrast to textured surfaces which have been subjected to a deliberate surface finish technique.

1.1.1 Textured Surfaces

There has been a great deal of interest with regards to the recent discovery of using textured surfaces when designing closely rubbing conformal contacts. Photo-etching and focused laser surface texturing (L.S.T.) are two of the more popular choices for texturing surfaces. Positive asperities are termed as ‘protuberances’ while negative asperities are known as ‘recesses’. It is thought that the presence of these asperities may lead to the onset of cavitation occurring within the lubricant film, and this is thought to give rise to several phenomena; evidence has shown that these textured surfaces have resulted in significant levels of improvement to not only the maximum load capacity within the contact region, but also to favourable coefficient of friction reductions. This will be dealt with in more detail later in Chapter 2.
These textured surfaces are currently being utilised in many applications. Examples include Journal bearings, Piston/Liner interfaces and Rotary/Lip seals. However, the mechanisms of friction reduction are not yet fully understood either theoretically or by means of validated experimental evidence and so further research is necessary.

1.1.2 Lubricant Film Thickness

If rubbing conformal contacts are to maintain their optimum working characteristics, certain constraints must be adhered to at all times. Of these, the existence of a lubricant film is vital as to prevent asperity-asperity contact which may lead to premature bearing failure. These lubricant films can be of either aqueous or pseudo-solid form, and suffer as a result of the harsh environments they have to rigorously endure. The film can vary from tens of micrometers down to nanometres in thickness, depending upon the regime of lubrication that the contact region is undergoing. Two fundamental fluid film regimes of lubrication are Hydrodynamic (isoviscous rigid) and Elastohydrodynamic (piezo-viscous elastic) lubrication. These will be discussed in more detail in Chapter 2.

1.1.3 Cavitation

Cavitation is the formation of a vapour phase (in a liquid) in response to the liquid being subjected to sub-ambient or negative pressure.

It has been shown in previous work, discussed later in Chapter 2, that cavitation can modify the antisymmetric pressure distribution which occurs near the surface asperities.
It is thought that the balance of higher pressures over lower, often sub-ambient pressures is the primary explanation for the generation of load support in parallel or near-parallel surfaces.

1.2 Aims and Objectives

This particular study aims to improve the understanding (of the underlying principles) of how microtextured surfaces give rise to reductions of coefficients of friction and increased load support. A related issue is the evaluation of film thickness measurements of the lubricant film, and the understanding of what precise role cavitation plays, if any, in all of the above.

One of the primary aims of this study is to see how exactly the coefficient of friction is reduced and load support is increased. It is hoped that the mechanisms by which this occurs can be comprehensively understood.

Another important aim of the project is to evaluate the lubricant film thickness within rubbing contacts. As cavitation occurs within the film, the film thickness will also be measured simultaneously with friction. At present, the nature of the cavitation geometry envisaged is thought to be highly influential within these mechanisms.
The overall aim of the current research is to carry out simultaneous measurements of friction and film thickness in textured bearings in order to distinguish the possible mechanisms of friction reduction.

1.3 Collaboration

This research program includes a comparative analysis between the experimental results gained from this specific programme and computational results being researched by a collaborator concurrently. A Computational Fluid Dynamics (CFD) approach is closely related to this project in terms of determining the influence of microtextured surfaces upon reducing friction coefficients and increasing load support. The current work follows on from computational analyses of hydrodynamically lubricating films in plain and pocketed surfaces being undertaken at Imperial College; a two-dimensional code capable of solving the separation of two non-conformal surfaces loaded against each other and in relative motion, by a lubricating film.

1.4 Thesis Structure

The work described here in this thesis can effectively be partitioned into three main parts. The first part (Chapters 1-2) is an introduction to, and how previous efforts have attempted to deal with, the question of textured surfaces. A literature review will highlight the areas of research which have been crucial in determining the knowledge base thus far. But more importantly, it will identify those areas which have not yet been
explored. A complete and thorough understanding of the exact nature of textured surfaces and the impact that they have upon friction is key to the research being carried out by the automotive industry at present.

The second part (Chapters 3-4) introduces the concept of the test rig and the techniques used to develop the test method as well as the calibration applied to certain components on the test rig. Each individual sub-system of the test rig is introduced and the reasoning behind the specific design is given. The test pad specimens used in the current research are described and the manner in which they are tested is explained.

The results produced using the hydrodynamic test rig are presented and commented upon in the final portion of this thesis (Chapters 5-6). The results are compared against theoretical predictions. Discussion of the significance of these results and comments are offered where differences from analyses occur, and the method in which they may be eliminated in future work proposals. A final summary and conclusions (Chapter 7) are made as to the validity of the current findings.
Chapter 2

Hydrodynamic Study – A Literature Review
2.1 Classical Lubrication

This section deals with the fundamentals of lubrication theory. It aims to give an overview of the main types of lubricating films present whenever there are two or more opposing moving surfaces. It will deal briefly with how these films are formed and what factors affect their relative performance under specific conditions. The issue of friction and its relationship to the nature of film formed is also introduced.

2.1.1 Introduction

The majority of modern day machinery inevitably contains moving parts that are in constant contact with each other and thus, continual rubbing and sliding occurs. These contact surfaces are frequently situated within harsh environments and extreme working conditions. As a result, damage and wear is usually associated with the presence of friction due to surface interactions. Over the years, attempts have been made to alleviate these problems by means of changing the working conditions (load, temperature & speed) of the contacts, modifying the materials in contact, and introducing lubricants to physically separate the contacting surfaces to name but a few solutions.

This has led to the gradual development of what we today call ‘Tribology’, a term originating from the Greek “tribos” which translates to attrition/rubbing. This is effectively study of the ‘science and technology of interacting surfaces in relative motion’ [1] or in simple terms, the effect of friction and wear between rubbing contacts and the
problems this causes. Continual demands upon tribological advances have ensured that these disciplines regularly collaborate and extend on areas of interest.

2.1.2 Mechanisms of Fluid Film Formation

By separating the rubbing surfaces with a low shear strength film, the friction and surface damage imposed upon the surfaces is greatly reduced. Inevitably, the surfaces of most materials have a certain roughness value. This implies that there are asperities on the surface layer of the material. For a lubricant film to be effective, its thickness should be greater than the combined height of two such asperities on the opposing surfaces. This would ensure that the two surfaces are physically separated during any movement. ‘Dry’ contacts occur in the absence of any lubrication film [1].

The lubricant film can occur in a number of ways, more commonly known as the ‘fluid film regimes’ of lubrication. These are as follows;

- **Hydrodynamic Lubrication** (HD) – Where the typical film thickness lies between 1 and 100 µm. The oil is swept into the conformal contacts operating at moderate to high speeds, and is classed as isoviscous.

- **Elastohydrodynamic Lubrication** (EHL) – The film thickness are generally smaller, ranging from 5 down to 0.1 µm. Again, the oil is dragged into the non-
conformal contact by the relative motion of the opposing surfaces, generally at high pressure (piezo-viscous).

These fluid film regimes consist of a lubricant which is most commonly a hydrocarbon based oil, gas or solid. Not only does the lubricant act to separate the surfaces, it also serves to cool the surfaces, transport debris away from the rubbing areas, protect the surfaces from water and to prevent wear. Lubricants are chosen so that they are effective for long periods of time without the need for constant maintenance.

As Dowson [2] discussed in a recent review paper, the ‘birth’ of fluid film lubrication studies could be indebted to two leading figures in the nineteenth century, namely Beauchamp Tower and Nikolai Petrov. Tower [3] commented on his experimental work that a ‘coherent film of lubricant separated the bearing surfaces in well behaved bearings’. This was the result of the measurement of pressures within the bearing well beyond that of the mean bearing pressure. This was discovered purely by chance. Petrov’s [4] findings were of the same conclusion however, he states that ‘the friction torque in rolling stock bearings could be attributed to the shearing of a viscous lubricant in the annular gap formed between a rotating shaft and concentric bearing’. Leading on from this, Osborne Reynolds [5] in 1886 suggested a theoretical approach to the fluid film formation mechanism by effectively reducing the Navier-Stokes equations (explained in detail in [6]) to conditions suited to that of creeping flow. These were then combined with fundamental continuity equations resulting in second order partial differential equations.
To this day, these equations are still used to describe the pressure generation and support mechanisms in fluid-film bearings.

**Hydrodynamic Lubrication – Action of Converging Wedge**

All lubricated pairs effectively contain a convergent wedge, be they steep or shallow. It is this convergence, coupled with the moving surface speed and oil viscosity, which generates the oil pressure film. A liberal translation of the German descriptive phrase *Druckberg* is given as the ‘pressure hill’ [7]. Figure 2.1 depicts a simple oil wedge formation.

![Pressure profile](image)

*Figure 2.1 A converging oil wedge*

The bottom surface is moving from left to right in the above depiction. This invokes a viscous fluid (i.e. the oil in this case) to be entrained into the gap formed between it and the stationary surface. Once the oil starts to move into the gap, it is essentially trying to fill a fixed volume with less vacant space available for it to move into. Further oil cannot
be accommodated within the gap at this instant. Oil is generally classed as an incompressible fluid and thus, the only way to resolve the issue of the oil moving into a reducing space is for the oil itself to generate a pressure. This has the consequence of preventing further oil being forced into the converging wedge due to it encountering the rising pressure (i.e. the pressure differential which is classed as the pressure hill [7]). Once the oil already within the gap has travelled beyond the maximum pressure zone, it is ejected through the reduced space the end of the convergent pad by means of the elevated pressure gradient present.

The narrow angle created by the two opposing surfaces is often referred to as the shallow wedge [7]. This design feature is intentionally created to entrain lubricant in between the surfaces during relative motion. The lubricant is forced to withstand high pressures and thus support external loading. As mentioned, the lubricant film has to be ideally thicker than the combined thickness of the high points of the surface asperities. Neither deformation of the surfaces nor increases in lubricant viscosity commonly occur in hydrodynamic lubrication due to the relatively low pressures encountered within the contacts. Plain journal bearings, thrust pad bearings and piston/liner interfaces are all examples of hydrodynamic lubrication. By analysing the fluid flow inside the region of interest mathematically, Reynolds equation can be derived [7]. This details two solid surfaces sliding or squeezing past each other and the resultant pressure distribution within the contact area. Consider the situation as shown below in Figure 2.2;
The pressure of the fluid is a function of the dynamic viscosity $\eta$, the fluid film $h$ separating the surfaces and the velocities ($U_1, U_2, V_1, V_2, w_1, w_2$) of the surfaces in the $x$, $y$ and $z$ direction. Both the fluid film and dynamic viscosity may vary in the $x$ and $y$ directions. The fluid coordinates are denoted by $u$, $v$ and $w$.

The Navier Stokes equations can be simplified to give Reynold’s equation, or it can be derived by the combination of fluid element equilibrium with the continuity of flow. This derivation of Reynold’s equation entails certain assumptions;

1. Body forces are negligible
2. The pressure is constant throughout the fluid film (in $z$ direction)
3. No slip occurs at the boundaries
4. A Newtonian fluid
5. Laminar flow

6. Negligible Inertia

7. Constant fluid film density

8. Constant velocity throughout the fluid film in x and y directions only, but not z

The first three assumptions are unequivocal for all cases, whilst 4, 5 and 6 are normally valid and the last two are questionable (mainly due to possible thermal effects present in the fluid film) [1].

The velocity profiles through the fluid film are first calculated using ‘equilibrium of an element’. These are in turn integrated to produce the fluid film flows. Finally, ‘continuity of volume flow’ is applied to produce Reynolds equation;

\[
\frac{\partial}{\partial x} \left( \frac{h^3}{\eta} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left( \frac{h^3}{\eta} \frac{\partial p}{\partial y} \right) = 6 \left( U_1 \frac{\partial h}{\partial x} + V_1 \frac{\partial h}{\partial y} + 2(w_2 - w_1) \right) \quad (1)
\]

This is Reynolds equation in 2-dimensions (which assumes \( U_2 \) and \( V_2 \) to be zero) as derived fully in [1] and [7]. The terms on the left hand side of equation 1 above relate to the Poiseuille flow; describing the nature of flow induced by the pressure gradient present in the oil film (oil is driven from regions of high to low pressure). Pressure gradients in the x and y direction effectively govern these terms.
Both the terms with $U_1$ and $V_1$ present refer to the Couette flow between the surfaces; the flow induced by the sliding of one surface relative to the other. Oil is dragged into and through the contact zone by the moving surface. The two variables that control this are the surface sliding speed and the (changing) film thickness profile of the film.

The final term concerns the vertical oil flow present due to any vertical movement of the surfaces, if present. This is referred to as the ‘squeeze film’. In many applications of Reynolds equation, this term is commonly equated to zero and removed from the above equation.

Reynolds’ equation represents the continuity of mass (or of flow if constant density is assumed) within the oil film whilst balancing the multiple driving forces behind oil transport.

### 2.1.3 Friction

The term ‘mixed’ lubrication is widely used when describing the lubrication regime of gears and spinning disc experiments. The load is carried by a combination of hydrodynamic forces (involving small coefficient of friction values) and ‘boundary’ lubrication (coefficients of friction of up to one hundred orders of magnitude higher). This boundary lubrication regime occurs when there is not enough hydrodynamic or elastohydrodynamic lubrication film present to sufficiently separate the surfaces in
contact. As slower moving contacts rarely allow the formation of thicker lubricating films to occur, this type of lubrication is considered crucial (the same applies in high temperature contact situations where lubricant viscosity is low).

In thin fluid film bearings, friction results due to the presence of relatively high levels of shear within the film. The shear stress (and consequently the strain rate) is elevated in the thin film regions. The addition of pockets on the surfaces within these regions may ultimately act to serve as a shear reducing mechanism [14].

Only the very ends or tips of the surface asperities of the material come into contact with each other in the mixed lubrication regime. It is widely accepted that the mixed lubrication regime is very important since a lot of everyday components and machinery operate for part of their cycles in the mixed lubrication phase. Wear frequently occurs in this regime, as is demonstrated practically by engine components such as cams and piston/liner interfaces. Experimentation has shown that as the speed of a typical spinning steel disc/pin configuration increases the friction reduces significantly, but is unaffected by any loading differential. Lower coefficients of friction within high load bearings and higher coefficients of friction within low load applications are often experienced. Low coefficients of friction for low load contacts are in great demand.
2.2 Textured Rubbing Surfaces

A brief history of the concept of textured surfaces and their impact upon lubricated surfaces is dealt with in this section. It can be separated into two main categories; the theoretical approach followed by the contribution of experiment. Both are discussed with their relative impacts upon the effect of texture within moving lubricated contacts.

2.2.1 Introduction

In recent years, there has been much interest in the growing field of surface texturing of rubbing materials. All materials have a roughness, and processes such as cutting, polishing and hardening can affect this roughness value. This can be thought of as a texture naturally imposed upon the surface, even though man made. In contrast, surfaces can also be textured intentionally, where textures of a certain specified geometry are fabricated. This will be discussed in more detail later. One such process of texturing the surfaces is by Laser Surface Texturing (L.S.T.), often referred to as laser honing [11]. This creates small ‘dimples’ or pockets on the material surface, with some having depths and widths of up to 40 µm. These pocketed surfaces are being introduced more readily into high friction applications, as they tend to show a favourable decrease in friction coefficients as well as higher load bearing capabilities [11].

Honing (the process whereby material asperities are removed leading to an improved surface finish) has been used to finish piston liners within IC engines for a number of years. This process is often attributed to prevent scuffing, retain hot oil on the piston
liner for adequate hydrodynamic lubrication and to accommodate wear debris, if and when present [11 & 12]. A typical honed liner as measured on a 3D optical profiler is depicted in Figure 2.3.

![Honed Cylinder Liner](image)

**Figure 2.3 A honed cylinder liner**

The use of L.S.T. or laser honing, typically achieves the same end result (i.e. reduced friction and an equivalent surface roughness) but with the added bonus of also being influential on load support. As opposed to plateau honing, the nature of the laser honing distribution is usually well defined and bounded. This has the effect of allowing a pressure differential to build up within the textures [13]. A laser honed sample is shown below in Figure 2.4.

![Laser Honed Steel Sample](image)

**Figure 2.4 A laser honed steel sample**
2.2.2 A Theoretical Approach

It was shown in the mid 1960’s that textured surfaces had an advantageous impact on friction between moving surfaces, e.g. Hamilton et al [8] who described the mechanisms based on surface ‘micro-irregularities’ (asperities and depressions) and subsequent film cavities present within a rotor-stator set-up. They concentrated efforts upon face seals containing micro-asperities. The primary explanation given for the advantages is by the ‘anti-symmetric pressure distribution which would normally occur about these irregularities is modified by film cavitation, which appears to permit the high film pressures required to overbalance the low film pressures which results in a significant load capacity capability’. As the cavitation streamers appeared to originate and terminate where the irregularities were situated, it was hypothesised that the surface roughness was an influence on cavitation. It was further shown that depressions (pockets) located on the surface had a similar effect as the asperities. Cavities were shown to be formed at the leading edge while high pressures were generated above the trailing edge. It was concluded that the predicted cavity size interactions, with pressure distribution arising from the asperities and pockets, were in agreement with experimental findings. There was some difference in the load support predicted by the theory and the values achieved experimentally, ranging from 2 to 5 times the prediction. A typical load support versus disc speed correlation for three different texture patterns they obtained is shown by Figure 2.5.
Figure 2.5 Load Support vs. Speed for various texture patterns (from [8])

Key points to note in this research are that the lubrication is controllable (achievable by managing the asperity/pocket pattern and frequency) and that the arrays of microasperities increase load support and film thickness concurrently. This is ascribed to the occurrence of cavitation located at the individual asperity points whilst a negative pressure being maintained over the macro contact area. The presence of micropockets behaving in a similar fashion is also hypothesised. Later work by Anno et al in [9] & [10] further make the cases for improved lubrication and load support whilst decreasing coefficients of friction in textured surfaces.

Etsion [11] points out in his recent review of laser surface texturing methodology and implementations that the use of L.S.T. is commonly known for its ability to overcome adhesion and stiction during start up in Microelectromechanical Systems (MEMS).
Modern day magnetic storage devices often have their surfaces textured for these reasons. There have been many contributions from Etsion’s research group with regards to microtextured analysis over the past decade or so. Varying kinds of bearing geometries have been modelled, sometimes with accompanying experimental data. Parallel bearings under pure sliding conditions moving in a reciprocating manner have shown friction reductions of up to 40% when the use of ‘micropockets’ have been implemented [12]. A comparison of theoretical versus experimental friction traces obtained for the texturing of piston rings can be seen in Figure 2.6.

![Figure 2.6 Friction vs. Crank angle obtained for a textured piston ring (from [12])]()
Optimum pocket depths in the region of 10 µm were also investigated, which led to the
general consensus that deep pockets (as depicted by a pocket height, \( h_p \) of 19 µm in
Figure 2.7 above) perform considerably worse due to the onset of starvation (as indicated
by the increasing elapsed time between successive drops of oil on the x-axis in Figure 2.7)
as discussed by Fowell [14]. It is thought that to maintain a hydrodynamic film in the
presence of deep pockets, a larger oil supply mechanism is required.

However, the conservation of flow across the inlet of the texture pockets is not
maintained by the Reynold’s boundary condition, as implemented by Etsion [11, 12 &
21]. An improved method would be to use a mass-conserving model.

The use of textured surfaces upon mechanical seals has also been reported [15]. The
findings showed the ability of microtextures to delay the eventual collapse of lubricant
films under normal loading. The optimal area of the surfaces microtextured has also been
investigated with the use of ‘pin on disk’ methods [16] as shown in Figure 2.8. The
distance along the bearing in the sliding direction is indicated by the axis ‘Dimensionless
coordinate x’ and the ‘dimensionless pressure p’ is a function of the ambient and local
pressures as well as the dynamic viscosity and sliding velocity.
It has been shown by Olver et al [17] that a ‘parallel pad bearing with a closed pocket situated near the inlet can support load if operating at ambient pressures well in excess of the cavitating pressures of the lubricating fluid’. This is mainly due to sub-ambient pressures driving fluid into the inlet region of the pad, referred to as ‘inlet suction’. Also, it was revealed that the maximum load bearing capability is achieved when the location of the pocket is near the bearing inlet and the onset of cavitation occurs.

Further investigations into this phenomenon of inlet suction, specifically aimed at low convergence micropocketed bearings are thoroughly discussed by Fowell et al [14]. A thorough argument for sub-ambient pressures caused by one of the sliding surfaces is again presented. It is shown that as these pressures are less than the external atmospheric pressure, lubricant is ‘sucked into the bearing through the inlet land’ where it later arrives at the diverging pocket entrance. Sub-ambient pressures within the bearing being the main

\[ \text{Figure 2.8 Effect of textured area upon load support (from [16])} \]
driving force is the primary explanation offered for this phenomenon. A one dimensional analytical solution is employed to derive the hydrodynamic load support and friction for simply pocketed bearings at various convergence ratios. Reynold’s boundary condition does conserve the mass for the cavitation boundary in a smooth bearing (i.e. non-textured) but does not for a reformation boundary or for textured surfaces. In one dimensional analysis, the mass flow is conserved by ensuring the flow rate is constant throughout the bearing. In the two dimensional approach, the problem is much more difficult and requires the Elrod boundary condition [14]. This is why many two dimensional analyses are deemed incorrect. Figures 2.9 and 2.10 highlight some of the findings with regards to the effect of pocketed surfaces upon load support and friction as given by [14]

![Figure 2.9 Effect of textured surfaces upon load support (from [14])](image)
Parallel surfaces are also shown to rely on inlet suction as the only means of hydrodynamic load support as there is no net entrainment. Beneficial outcomes for traditionally high ($K>1$) convergence ratios are also reported. The convergence ratio of a tilted plane is defined as;

$$K = \frac{h_i}{h_o} - 1$$

Where $h_i$ and $h_o$ are the inlet and outlet film thicknesses respectively.

Recommendations that the cavitation pressure of the lubricant should ideally be as low as possible and that the ratio of inlet to outlet pad length should be small, are made.

If one was to consider a flat bearing that possesses little or no convergence, it is classed as parallel. This is thought to reduce the resistance faced by the fluid flow as it enters the bearing as opposed to leaving the bearing as discussed by Tonder [18]. He states that

\[\text{Figure 2.10 Effect of textured surfaces upon coefficient of friction (from [14])}\]
small features located within the bearing negate the effects of lubrication as opposed to the larger structure of the bearing itself; in other words the ‘micro-geometry controls the frictional properties rather than the macro-geometry’. The load capacity, friction and coefficient of friction were all theoretically calculated. Improvements in both load support and coefficients of friction across three different samples can be seen in Figures 2.11 and 2.12 below.

![Figure 2.11](image1.png)

**Figure 2.11** Effect of textured surfaces upon load support (from [18])

![Figure 2.12](image2.png)

**Figure 2.12** Effect of textured surfaces upon coefficient of friction (from [18])

The x-axis denotes the ‘k-values’ (as given by the pocket step height divided by the film thickness) whilst the y-axes represent the non-dimensional load capacity and coefficient of friction. It was seen that in some textured cases, load support was more pronounced for thinner films. Two mechanisms are presented with regards to these potential benefits
shown by theoretical means; the method by which the microtextures are ascribed to inlet steps as shown in a typical Rayleigh step bearing (i.e. each pocket behaving like that of step bearings) and secondly, the presence of these pockets restricts the attempted flow of lubricant out away from the lubricated zone by means of acting very similar to miniature reservoirs of oil.

### 2.2.3 The Contribution of Experiment

The issue of controllability as well as repeatability was discussed by Stephens et al [15]. The case for showing ‘potential enhancements of lubrication in conformal contacts using micro asperities’ was put forward. Again, positive and negative asperities were produced, but this time using laser texturing techniques. Results showing reduction of friction coefficients of up to 22% were presented, as well as highlighting the possibility of further reductions in friction coefficient by optimizing the asperity geometry and layout.

One of the few topics that have rarely been dealt with when considering the effect of textured surfaces is that of lubrication starvation. Ryk et al [19] & [20] have shown the effectiveness of reducing friction in ‘reciprocating automotive components’, in this case piston rings. They report that approximately 50% of the friction losses occur within the piston/liner interface, of which 70-80% comes from the piston rings themselves. It was concluded that a low viscosity lubricant coupled with an optimum pocket depth gave rise to beneficial friction reduction with the use of textured surfaces. He goes on to state that ‘the laser textured surfaces may provide oil retention capability that will protect the
sliding surfaces against seizure’. His experiments show that optimum dimple depths coupled with low viscosity lubricants were beneficial over a wide range of flow rates, whilst texturing with deeper pockets and/or high viscosity lubricants was often ‘detrimental under certain operating conditions’. Extensive work has also been carried out on piston ring samples by the same author. Reciprocating test rig experiments have shown friction reductions of up to 30% [21] and examples of surface texturing implemented into the piston-liner mechanism have given reductions in fuel consumptions of up to 4%. It is stated that the mechanism by which the friction reduction occurs is the ‘collective effect of the dimples that provide an equivalent converging clearance between nominally parallel mating surfaces’. Similarities to work conducted by Tonder [22] carried out earlier are mentioned.

Work carried out by Tateishi [23] also claims reductions in fuel consumption by ‘several’ percent, and also improvements in wear rates and scuffing of the piston rings. He attributes this to several factors that do not involve the use of surface texturing; reduction of piston ring tension, using two ring set-ups instead of the typical three rings and finally, using surface composite plating techniques.

Experimental texture work was carried out on internal combustion engines is described by Rahnejat et al [24]. A single cylinder engine with a laser-etched textured liner was used. Torque improvements of approximately 4.5% were shown to exist when using the textured liner when compared to the standard plain liner configuration throughout the
engine speed range. This improvement is accredited to ‘lubricant retention in the laser etched grooves’. The percentage torque gains can be seen clearly in Figure 2.13, plotted against engine speed;

![Figure 2.13 Effect of texturing upon friction torque for a laser-etched liner (from [24])](image)

The use of textured surfaces also extends to use on other rotating engine components, as described by Gangopadahyay [25]. Tappet shim arrangements were textured by means of ‘parallel line V-grooves and square grooves’. Friction reductions of up to 35% were reported over standard production shims. Again, references are made to work carried out previously by Wang [26] implying that ‘the liquid trapped in the dimple (i.e. the texture) can be considered as a secondary source of lubricant which is drawn in to the area surrounding the dimple by the relative movement of the surfaces’. In essence, the presence of the texture acts as oil reservoirs. A graph of friction torque versus camshaft
speed for both a V-groove and parallel square groove tappet shim is presented in Figure 2.14;

![Figure 2.14](image)

**Figure 2.14** Effect of textured surfaces upon friction torque for V-grooved and parallel grooved tappet shims (from [25])

An alternative method to determine the effects of textured friction is to measure the resultant film thickness, as well as the friction. Research carried out on ‘reciprocating textured steel surfaces under hydrodynamic conditions’ is described [27]. Textured plane steel surfaces were loaded against cylindrical counter bodies under hydrodynamic lubricating conditions. A capacitance technique was used to measure the minimum film thickness present between the two surfaces (a technique based upon the theory that the capacitance present between two parallel plates is inversely proportional to the separation gap between the plates). It is stated that the film thickness readings using this method had an accuracy of 10% of the reading. Various textures comprising several pattern types (circular, grooves and chevrons) were tested. Selections of film thickness graphs versus load are presented in Figure 2.15;
Figure 2.15 Effect of textured surfaces upon minimum film thickness for reciprocating textured surfaces (from [27])

It can be seen that the minimum film thickness increases (for a given load) for a number of textured samples. Samples T1-T4, T12-T15 are circular in nature, T5, T6, T8 and T9 are grooved textures whilst T10 is a chevron design. It was concluded that chevron patterned textures pointing along the sliding direction gave the highest film thicknesses as
opposed to those pointing across, and that grooved textures were the least effective when considering the fluid film thickness.

It can be noted that the majority of the research reviewed above is primarily in the low film thickness regime (mixed lubrication) generally with high loads. The current research aims to extend the current lack of knowledge in the fully flooded hydrodynamic fluid film regime by exploring low load contacts.

2.3 Film Thickness Measurements

The various techniques used to measure film thickness in fluid film lubrication are described. The progression from laboratory experiments to engine measurements is introduced.

2.3.1 Classical methods – Optical Interferometry

It has been discussed by Cameron and Gohar [28] that earlier work regarding white light optical interference carried out by Kirk [29] was not suitable due to inadequate loading conditions upon a weak Perspex-on-Perspex contact, the combined effect of which was insufficient test parameters to ‘permit the onset of true elastohydrodynamic lubrication’ which presupposes a large increase of oil viscosity with pressure (i.e. piezo-viscous ehd could not be achieved). Cameron and Gohar then proceeded to employ flint glass with a higher refractive index and a steel ball to give detailed contact area mapping and oil film thickness, the results of which are presented in [30] and [28].
The work of Hardy [31] using ‘interference colours to measure lubricating films on glass’ has been acknowledged by Wedeven [32] in his research involving optical measurements in elastohydrodynamic rolling contact bearings. He stated that the use of light was favoured considerably as the wavelength of light is a ‘convenient unit of measure’ when one considers the scale of the film to be measured. The fact that an interferometry image highlights more sensitivity in the direction of depth, when comparing it to the direction of extent, was a great advantage, as is the case when considering EHL film mapping. The method of interferometry works on the principle of ‘divisions of amplitude’ of the incident light source wave, and the relative position of the partial reflected light source wave from the semi-reflective coating layer on the bottom of the glass surface. Interference fringes arise from the phase difference between the incident and refractive light rays, and this depends upon the ‘optical path difference between the two rays’. The Fringe order, $K_f$, is related to the optical path difference, and is an integer starting from zero. Wedeven went on to state that the use of a duo-chromatic light source was far more beneficial than either that of a white light or monochromatic system. These focused primarily upon the area of fringes. The issue of fringe visibility, spacing, sensitivity and fringe order were discussed in detail. Compromises were made in each case, but the eventual case for a Red-Green duo-chromatic light source was put forward [32].

It was originally envisaged that the use of optical interferometry in this current work would be employed for the determination of the convergence angle of the linear pad bearing set-up. As some test cases call for a near parallel surface set-up, one may argue
that the only viable solution is to use this technique of ‘fringe counting’ as this can measure very thin films to an unprecedented level (accurate to 1 nm). However, certain difficulties encountered called for a change of approach, which will be discussed in Chapter 3.

### 2.3.2 Laser Induced Fluorescence Methods

Laser Induced Fluorescence (L.I.F) is a term coined in the 1970’s when it was observed [33] that some oils exhibited a ‘natural fluorescence’ phenomenon when light of a certain wavelength was imposed upon them. It was shown that a range of oil film thicknesses could be measured using calibration plots of known film depths and comparing them quantitatively against observed images and output signal traces.

When light of a certain wavelength is shone onto a fluorescent oil (or dye), the quantum energy level of the individual dye molecules momentarily increases and the molecules are elevated to a higher energy state. As soon as this occurs, energy (in the form of light that is of a higher wavelength than the incoming light wavelength) is expelled instantaneously from the molecules in their elevated state and the quantum energy level immediately drops to that of its original ‘resting’ state. This process continually occurs very rapidly across the entire film/dye thickness.
Thin films will produce low levels of fluoresced light (i.e. light intensity) and vice versa. This can be used to measure the thickness of a fluorescing fluid film by calibrating known film thicknesses against varying levels of light intensity. The controlling factors of the fluorescence process include the concentration of dye mixed in with the fluid film (if the fluid film has no natural fluorescence) as well as the intensity and wavelength of the incoming (exciter) light source.

The types of experiment which involve L.I.F. can be approximately categorised into two groups; those experiments which measure film thickness in engine components (i.e. piston/liner mechanisms, piston skirt region and various engine bearings) and those which do not.

One of the pioneers of the L.I.F. technique [33] discussed the advantages of using a blue laser system as an exciter light source for use in doped oil fluorescence. These included better stability and power output at blue wavelengths, the use of ordinary optical glass rather than quartz due to the absorbance of UV in a glass medium and the ability to physically being able to see the blue beam of light as opposed to the ‘invisible’ UV rays. A Photomultiplier tube (P.M.T.) coupled with a digital voltmeter would provide the necessary output signal required for a signal trace, a direct relation to the measured film profile as shown in Figure 2.16.
The authors cite a number of reasons as to the benefits of an induced fluorescence technique; these include the option to direct a beam of light to a specific region without the need to cater for scatter, the long working distance of the illumination system allowing film thickness measurements in awkward locations and the ability of certain oil/dye mixtures to fluoresce more than one wavelength to cater for a range of film thicknesses. Similar comments regarding laser suitability in L.I.F. work are mentioned in [34] and later by Hidrovo et al [35, 36, 37] These raise the issue of ‘variations in intensity of a single laser system’ and the need for a dual laser/camera system (dual emission laser induced fluorescence).

Most relevant work to the current study is that of Sugimura et al [38] in which point contacts observed using an 8bit monochrome camera under fluorescence derived from a ball on disk configuration were used. He goes on to state that an oil film thicknesses

![Figure 2.16](image-url)
down to 30 nm was detectable, albeit with two consistent problems; the ‘light interference within thin films and the abundant background effect caused by cavities formed at the film exit’. Special mention is made to the ability of fluorescent imaging to capture cavities present at the contact exit due to no film being present within a cavity and thus a dark region results. Camera sensitivity is also commented upon; ‘A camera with ten times higher sensitivity would provide film thickness data at a resolution comparable with ultra thin film interferometry’ as postulated by Johnston et al [39].

Varying the fluorescent intensity of a specific oil/dye is carried out generally by varying the concentration of dye added to the lubricant film. Concentrations ranging from 0.01-2% of total fluid mass are common in this field [40, 41 and 42]

The use of L.I.F. methods to determine the oil film thicknesses present within the piston/liner mechanism within an IC engine was a natural progression for the technique. Again, many systems use Lasers (Helium-Cadmium or Argon-Ion) as the light source [43, 44, 45 and 46]. Combinations of both high sensitive cameras and photomultiplier tubes connected to the engine block by means of fibre optics are discussed. These works are primarily concerned with reciprocating surfaces where the associated film thickness are typically low (in the region of 10 µm and lower at times [44, 45]). The use of a high powered laser coupled to a sensitive photomultiplier by means of fibre optics allows for the adequate detection of thin films.
Calibrations from 30 to 80 µm were carried out by Hoult et al [43]. This consisted of placing varying thin shims between optically flat microscope slides and taking P.M.T. traces of the subsequent thickness film. They then extrapolate calibration coefficients to complete the calibration response for thinner films. A film thickness deviation of 5% was found to exist when larger films (approximately 120 µm) were encountered.

Richardson and Borman [44] reported a highly linear response when referring to the fluorescent signal versus film thickness calibration. They take advantage of the natural fluorescence phenomena of oils. However, they also state that the repeatability in their calibration coefficients varied by as much as 10%. They attribute this to the ‘laser power supply fluctuations’ ultimately causing output variations. They make the case for an in-situ calibration to be carried out prior to each test and also suggest doping the oil with a fluorescent dye (on the condition that temperature fluctuations do not affect the fluorescent intensity signal).

It has been possible to calibrate to a much thicker film thickness using the above methodology; oil film calibrations of up to 500 µm were presented by Arcoumanis et al [45] by means of an ‘enhanced resolution’ micrometer. The optical fibre was mounted flush with on the micrometers anvil and the oil film thickness continuously varied.
The use of thickness gauges (i.e. feeler gauges) and surface plating was implemented by Nakayama et al [46]. Film thicknesses from 0 to 100 µm were calibrated ‘arbitrarily’, although the film thicknesses present during testing were in the region of 10-30 µm.

2.4 Cavitation

The precise nature of which form of cavitation that occurs within plain bearings and its role in lubrication has been dealt with in detail only by a select few in the past. The two types of cavitation that are widely accepted in lubricating film scenarios are ‘Gaseous’ and ‘Vapour’. However, the former is of great interest at present due to theories of increased load support and inlet suction. There has been research conducted ranging from looking at bearing lubrication cavitation in numerical analysis [47, 48 and 49] to cavitation effects in oils under operating conditions [50] and finally the issue of cavitation occurring in elastohydrodynamic contacts [38].

The majority of studies have explored the implication of cavitation in reciprocating contacts. Findings from Priest et al [49] conclude that ‘Reynolds boundary condition and fluid film reformation may be applicable at high piston ring loads whilst fluid film separation may be applicable at low loads. It is stated that the cavity pressure is equated to the saturation pressure, which is in turn approximated to the atmospheric pressure. Findings regarding cavitation are made by Broman [47] in which the presence of sub-atmospheric pressure within spiral groove bearings is analysed. It is suggested that sub-
atmospheric pressures are not necessarily required for lubricant to enter the bearing and that cavitation may not occur for flat spiral groove thrust bearings. This is postulated for a proposed ‘modified’ bearing design. This is further concluded by remarking that ‘the velocity of the rotating bearing surface’ would ensure adequate lubrication of fluid film into the pressurised region effectively increasing the performance of the bearing.

Experimental contributions in reciprocating piston ring assemblies have been made from Arcoumanis et al [51 and 52]. The onset and development of cavitation in the lubricant film were visualised by means of a charge coupled device (CCD) imaging system illuminated by means of a flash lamp. Coupled to this, the film thickness was measured using a capacitance technique. Later work by the same authors [52] implemented the use of L.I.F. techniques to determine the film thickness. The presence of cavitation in the ring-liner pairing in the diverging region was documented. However, efforts were concentrated upon the evolution of the cavitation into ‘ferns, fissures, strings and bubbles’.

There is an apparent lack of experimental investigation to aid analytical solutions when examining cavitation. It is something that has always been regarded as a phenomenon that is present in lubricated contacts, but has only ever been considered at the exit plane of most contacts. The current research considers the reported effects of cavitation near the bearing inlet.
2.5 Summary of review

An overview of the current state of knowledge of textured hydrodynamic bearings has been presented. Mechanisms of fluid film formation, postulated texture theories, film thickness developments and the implication of cavitation have all been introduced and a brief overview of each given.

However, it appears that there is still a lack of experimental research within textured bearings that attempts to explore the relationship between film thickness measurements and friction simultaneously, in both textured and non-textured bearings. In addition to this, very little work has been carried out with regards to optical (i.e. visual) film thickness measurements in the hope of advancing the precise role of cavitation in hydrodynamic bearings.

The current contribution is needed in order to see whether the observed differences in friction are indeed due to the hydrodynamic action of the modified topography (or due to other explanations such as starvation or in-situ oil reservoirs).
Chapter 3

Experimental Development
3.1 Introduction

The current work focuses upon lubricant films present in a convergent pad loaded against a flat surface (a flat surface was chosen for ease with regards to setting the pad attitude). The pad in this case, can be thought of as a stationary surface with microtextured pockets on one side, and this will be loaded against a rotating glass disc, which represents the flat surface. The primary objective that was kept in mind when designing the experimental rig was the ability of the apparatus to accurately measure and record the film thickness, friction and possibly cavitation, all in situ and all simultaneously.

To date, there have been no published works on the experimental findings of the simultaneous study of friction, cavitation and film thickness in textured hydrodynamic bearings, primarily because such studies are difficult. Potential problems included the dynamic stability of the overall system (especially as the bearing surfaces tend towards near parallel), the large range of film thicknesses encountered and the need for surface features of accurately known dimensions to be located on the bearing surfaces.

3.2 The Design and Commissioning of the Test Rig

The proceeding section deals with the three main systems and the methodology in which they were originally conceptualised.
3.3 The Hydrodynamic Test Rig – General Outline

The rig incorporates three main areas of interest; The L.I.F. System, the Load Arm Mechanism and the Friction Analysis System. The following criteria were kept in mind when originally designing the test rig;

The rig would measure light intensity fluctuations in the lubricant film containing a dissolved fluorescent dye. The film will be illuminated with a corresponding light source, matched to the fluorescent dye, causing the dye to fluoresce. This will give information with regards to the film thickness, as there is a relationship between excitation light intensity and film thickness.

It will also provide measurements of friction between the two bearing surfaces. This will be achieved by accurately recording the small displacements of a load arm attached to one of the surfaces in contact. A displacement sensor located directly next to the load arm will be employed for this. The stiffness of the arm will be of a known value, and the displacement of the arm can be translated into a friction reading. The main loading arm concept and friction analysis system of the rig was designed by Dr Simon Medina of Imperial College, who supervised initial experiments conducted. An outline of these can be seen in Figure 3.1.
Both of these areas of interest were designed with many factors in mind; their ability to accurately measure and record data, their stability during operation, and the financial implications incurred.

**Figure 3.1 Main system components (Friction, Load, Disc and Oil supply)**
3.3.1 Film Thickness Measurements – L.I.F. System

The specific L.I.F system relevant to this research was chosen for a number of reasons. These include the low costs of implementing such a system, the reliability of equipment used and the overall non-intrusive technique that it entails. The basic outline of any L.I.F set-up comprises three main areas:

- Light Source
- Microscope
- Camera

The general consensus of a Laser having to be used as a light source is a wide misconception. Many L.I.F systems use an ordinary lamp/bulb configuration which emits light at the desired wavelength, most commonly Ultraviolet (U.V) or Blue light. These types of light sources include Black Light-Blue bulbs, U.V lamps and Mercury-Xenon bulbs. All of these various light sources emit a strong U.V wavelength spectrum. This is ideal for many hydrocarbon oils which exhibit a natural fluorescence phenomenon. Lasers tend to be used due to their precisely known wavelength and very high intensity. However, financial implications associated with lasers ruled them out.

The microscope set-up contains one of the most crucial aspects of the rig, the beam splitter, which contains the ‘fluorescent cube’. This allows the incoming light to be
reflected down towards the test specimen and the reflected light up through towards the camera. As L.I.F generally involves very low light levels, efficiency is important.

It can be argued that the final component in the L.I.F system is the most important of the three, the camera. A monochromatic CCD camera is widely regarded as the most suitable for the job, primarily for its high sensitivity when operating in low light conditions. This will be discussed later in Section 3.7.

The Microscope System

The microscope set-up consists of three main components; the main tube, the beam splitter and the Objective (the lens). A typical microscope can be seen depicted in Figure 3.2;

The main tube is, as its name suggests the piece of tubing which supports the camera on top and connects to the beam splitter below. It is a hollow piece of tubing which can
either be connected in series to a zoom lens, which allows the user to magnify the image beneath by up to a factor of 2. This magnification is on top of the magnified image as seen by the objective. For example, an objective with a 10x magnification with a main tube zoom of 3x would result in an image with a final 30x magnification projected onto the camera.

The beam splitter can be considered one of the most crucial parts of the microscope set-up. This piece of equipment contains the dichroic mirror (a semi-transparent piece of glass which allows light from the light source to be reflected downwards towards the specimen). It also allows fluorescent light to be transmitted from the specimen upwards towards the camera. The incoming light from the light source is first directed through an Exciter filter. This filter allows light between two known wavelengths through (a band pass filter). The filter has a lower and an upper cut-off point, and transmits light at varying transmission between the two. Directly situated below the main tube is another filter; the Emission filter. This is generally a long pass filter which allows light above a certain known wavelength through. This is specified so that any light that is produced as a result of fluorescence in the dye/lubricant mixture is able to pass through the emission filter, but light that is reflected off of the dichroic mirror directly upwards from the light source will be prevented from entering the camera. Figure 3.3 shows the beam splitter and all its components:
This highlights the roles of the exciter and emission filters quite clearly. The dichroic mirror is contained within a fluorescent cube which has its walls lined with black felt; this is to absorb any stray scattering of light within the cube.

Originally, a simple microscope was also incorporated into the rig for the interferometry measurements needed primarily to calculate the convergence angle of the pad sample loaded against the horizontally flat disc. It has been shown previously [32] that a duo-chromatic light source is ideal for this due to a ‘compromise between the monochromatic and white light sources, while the summation of the individual monochromatic fringes’ gives rise to the
coloured fringes produced. This has the advantage of having the capability of measuring a relatively thin film thickness while maintaining good fringe visibility and clarity. A Xenon arc lamp with a suitable filter placed in between the lamp and the light guide is all that is required to achieve this duo-chromatic light source. A mono-chromatic filter can just as easily be placed in front of the light source for the purposes of ‘fringe-counting’, a technique used to determine the height between two surfaces at a known distance from a common juncture, and thus the film thickness is given. The original two microscope design can be seen in Figure 3.4;

For reasons discussed later in section 3.6, the microscope system was eventually modified to only incorporate the L.I.F. microscope set-up.

Figure 3.4 The original dual microscope set-up
Light Source and Fluorescent Dye

It was decided that a range of light sources would be considered when designing the LIF system. As earlier work [33, 40 and 34] all involved using a high pressure Mercury lamp of some kind, these were investigated first. They tend to produce relatively high output intensity coupled with low costs. Filters must be employed however, to specifically target and match the excitation wavelength of the fluorescent dyes. The use of lasers was ruled out at this early stage even though they are generally considered to be the optimal technique in producing intense beams (and at a single known wavelength) of light. The beam of light is non-divergent and thus has the advantage of easy focussing on the specimen [34]. Financial constraints as well as numerous health and safety restrictions meant that lasers could not be considered for the current research.

Examples of high pressure gas discharge lamps include Mercury lamps, Xenon lamps, Mercury-Xenon lamps, Sodium lamps and Black light Blue bulbs. These types of bulbs usually consist of either a filament coil encased within a small glass envelope or two electrodes separated by a small gap (this forms the ‘arc’) and this envelope is located within a larger bulb. The atmosphere surrounding the two bulbs and the envelope itself is filled with a high pressure gas, dependent on the type of bulb. These types of lamps require large amounts of start-up voltages to ignite the arc, referred to as the ‘burner’. These lamps do suffer from the drawback of limited service life, and consequently the efficiency losses associated with this must be considered.
One of the most versatile fluorescent dyes used in fluorescent applications today is Pyrene. The advantages attached to this dye are that it ‘dissolves well in oil and is readily available’ as discussed by Sugimura et al [38]. It was originally decided for these reasons that it would be employed as the fluorescent dye for the current tests. This dye has an absorption peak of 336 nm and an emission peak of 480 nm at high solute concentrations according to the *Handbook of Fluorescence* [53]. High concentrations are generally those which involve a dye concentration of 1% mass and above. The emission and exciter filters as well as the dichroic mirror, as shown in Figure 3.4 were chosen to match these figures accordingly. Mercury lamps are a very common light source paired to Pyrene for use in fluorescence work, and thus it was decided that the use of a black light blue high pressure mercury lamp would suit. However, trials of a lab manufactured light source proved to be unsuccessful due to the inability of being able to focus the majority of the light onto the liquid light guide (10mm across in diameter). It was for this reason that a Hamamatsu UV Spot Light Source LIGHTNINGCURE L8222 was later purchased for use as a suitably powerful source of UV light. It contains a 150 watt Mercury-Xenon arc lamp and has an electronically controllable shutter. Coupled with this, the use of Pyrene has been demonstrated successfully previously, but in relatively low volumes. Due to the sheer volumes of oil being considered in this research (in the region of 1 litre per test), it was deemed best not to pursue the use of Pyrene due to high costs. A commercially available oil tracer dye, ‘Dye-Lite’ was selected. It is a blend of two separate component dyes (Perylene and Napthalimide) each with their own characteristic absorption profiles and is
recommended to be used with all UV and Blue light systems. The absorption profile for \textit{Dye-Lite} can be seen in Figure 3.5.

\begin{figure}[h]
\centering
\includegraphics[width=0.7\textwidth]{absorption_spectrum.png}
\caption{The absorption spectrum for ‘Dye-Lite’}
\end{figure}

As it is designed and formulated as to not interfere with the working properties of the host oil (for example the viscosity) it is to be mixed into, it was decided that it would be suitable for use in the current work.

\textbf{Camera}

This particular area of interest can be considered as one of the most crucial, as it involves meeting certain scientific criteria. As was the importance of choosing the right camera, a dedicated light box with a Mercury Black light Blue bulb was fabricated specifically for
the purpose of selecting a suitable camera as the Hamamatsu light source had not yet been purchased at this stage.

The specification required by the primary work objectives must all be adhered to, as they are often inter-dependent. Such points include:

- **Dynamic Range** – The ability of the camera to simultaneously detect low and high light intensities at one instance in time.

- **Sensitivity** – The minimum level of light that the cameras sensor can detect at any given time.

- **Shutter Speed** – The shortest amount of time the cameras shutter is open, crucial when trying to snap very small features.

- **Cooling** – Cooler sensors induce a lower noise level, which is recommended for high sensitivity applications such as L.I.F.

- **Electron Multiplying (EM) Gain** – A specific type of CCD which amplifies the stored charge before the ‘read-out’ stage. This has the advantage of signal amplification minus noise amplification. Again, this is fundamental in a high sensitivity (i.e. low light) setup.

High sensitivity cameras (EM gain cameras in particular) use some of the most advanced CCD technology available and are as such, expensive.
Various cameras were demonstrated to the author by various companies, the most successful being the Rolera MGi, a ‘high sensitivity Firewire digital EMCCD camera’. Positive features such as high quantum efficiency, high-speed readout and low noise electronics were desirable in such an application such as this current work. Specifications include a ‘Linear Full Well Capacity’ (the largest charge a sensor particle can hold before saturation of that particle, leading to eventual degradation of the signal) of 800,000e- coupled with a ‘Read Noise’ (the inherent output amplifier noise) of <1 e- rms. To insure that thermal noise is kept to a minimum, the ‘Dark Current’ rating is 0.5 e- per pixel. Also, to ensure that thermal noise under low light conditions would not affect the cameras CCD; the entire camera is cooled using ‘Three Stage Peltier Cooling’, attainable to -25 degrees Celsius. Low exposure times and a 14bit sensor (resulting in 16384 levels of distinguishable difference) ensure that the maximum amount of ‘information’ is attainable in the least amount of time available.

3.3.2 Test Specimen Loading Principle – The ‘Load Arm’

Hydrodynamic fluid films are in the region of tens of micrometers often leading to issues such as ‘squeeze’ term effects. This is opposed to Elastohydrodynamic films which are far thinner (typically sub-micron) and are consequently, stiff. This is important to consider when designing the loading mechanism. The current work involves the use of rectangular pads (for which selection reasons are discussed in Section 3.6). The load application arrangement constitutes a ‘load arm’ attached to a vertical stage with a fine pitched screw mechanism. In essence, the load is a resultant of a fixed displacement by the stage. The
load arm is bolted securely to the vertical stage by means of a flexible hinge arrangement (for friction sensing purposes, see Section 3.3.3 for details). Figure 3.6 depicts the original design proposal for the load arm mechanism.

*Figure 3.6 Schematic of the Friction Analysis System*

The original working specifications for the load arm system were that it is capable of loading up to 100 N, accommodate a rectangular 10mm by 20mm test pad specimen, be suitable for disc speeds of up to 2 m/s and that it accurately detect low friction forces (<5 N) for relatively large loads (~100 N). Figure 3.7 depicts the load arm arrangement in both side and plan view.
Figure 3.7 Side and plan views of the Load Arm arrangement
3.3.3 Friction Analysis System

This mechanism relies upon measuring the horizontal deflection of the load arm at a known length from the pivot, and relating this to the change in friction at the test pad contact zone (it is in effect, a ‘Tribometer’). This deflection is brought about by the converging test pad being raised up towards the glass disc (which is stationary in the z plane) It is the convergence of the test pad that induces the horizontal deflection, by movement in the vertical (z) plane of the load arm.

The sensitivity of the system is ultimately determined by the thickness of the thin steel plate. The flexibility and elasticity of the plate vary accordingly thus. The friction measurements themselves are given by using a high resolution non-contact displacement sensor (which operates on an eddy-current principle) permanently placed beside the load arm. As the arm displaces in the horizontal plane due to the induced friction at the test pad, the gap between a fixed plate permanently attached to the load arm and the sensor varies. This relationship between distance and load is directly proportional. The resolution of the friction system is governed by the accuracy and sensitivity of the non contact displacement sensor located besides the fixed plate. The fixed position of the sensor can be seen in Figure 3.6 above.

It can be argued that one issue that may have arisen with this ‘steel plate hinge mechanism’ is the possibility of the plate being plastically deformed as opposed to the envisaged elastic deformation taking place while under duress. This will have a negative
effect upon the reliability of the non contact displacement sensor technique for measuring the friction. A simple beam theory calculation was carried out to determine the most suitable thickness for this hinge. As a consequence of the hinge thickness, the ability of the load arm to pick up small changes in friction will be limited with the use of a thicker plate. The length of the load arm was calculated by considering the moment exerted by the elastic hinge. The free body diagram is presented below in Figure 3.8;

\[ F = k_m \frac{u}{l_{arm}} \]

And;

\[ F = k_m \frac{u}{l_{arm}^2} \]

\( F \) is the friction force (present due the fluid film between the spinning glass disc and test pad). The moment of stiffness of the elastic hinge is defined as \( k_m \). Hence;

\[ Fl_{arm} = k_m \frac{u}{l_{arm}} \]
Where $k_m$ is defined as;

$$k_m = \frac{3EI}{l}$$

Where $I$ is the moment of inertia of the shim (of length $b_s$ and thickness $d_s$) as given by;

$$I = \frac{b_s d_s^3}{12}$$

And $E$ is Young’s modulus of the steel shim

### 3.4 Peripheral Lubrication – Oil Supply

EHL test rigs consist of a circular stainless steel oil bath, in which the entire ball holder can either be partially or fully submerged into the oil filled bath. This eliminates the need for a recirculation type of set up as the oil is continually being dragged in to the contact by means of entrainment. However, due to the nature of the current design, this was not possible. A polypropylene oil bath was designed and constructed. All interfaces were sealed with a suitable liquid gasket type sealer. An automotive oil pump was attached to an electronically variable speed motor, coupled by a toothed belt. The original design incorporated a diverging nozzle pointing up towards the bottom of the oil disc near the pad inlet area. Initial preliminary tests highlighted a couple of serious flaws in this design, the most serious of which being the spread of oil film was not adequate enough to cover the entire area of the pad inlet.
This was resolved by removing the nozzle from the end of the high pressure tubing and sealing it with a grub screw. A series of small holes (1 mm in diameter) were then drilled in a line over a distance of approximately 20 mm. This section of tubing then had a small semi-rigid piece of stainless steel wire inserted into it to allow for it to be bent into shape and positioned steadily. This also had the added effect of distributing the oil uniformly on the bottom of the glass disc. Oil that has run-off into the bottom of the oil bath was recirculated by means of a return feed pipe. Figure 3.9 shows the original oil bath design located around the disc/pad interface.

*Figure 3.9 Original layout of the Test Rig*
3.5 Rotating Disc – The Drive Train Mechanism

As shown in the overall schematic of the system (see earlier, Figure 3.1) the glass disc is bolted down to the main shaft by a single lock nut arrangement. This shaft is held in the test rig platform by two ball bearings. The bottom of the main shaft is coupled to a twin V-pulley by means of a simple locating keyway. This pulley is attached to a second V-pulley via two V-belts. This second pulley is driven by a high torque electric motor, controlled by means of control loop feedback system. With regards to the disc speed, it was envisaged that in accordance with previous computational research that velocities of up to 2 metres per second would be attained. This was later changed to a maximum disc speed of 1 metre per second for reasons relating to the spread and fling of the oil at higher speeds.

3.6 The Test Pad Specimen

The current research involves the use of a 10 mm by 20 mm rectangular pad. Previous theoretical work undertaken within Imperial College has typically modelled pads with an \( l/b \) ratio of 0.5 (where \( l \) is the length of the bearing and \( b \) is the breadth)

Several laser honed steel samples had been supplied externally via the industrial sponsor, a typical example which can be seen in Figure 3.10;
These steel pads were ground, the laser honing technique applied to produce requested pocket depths of 20 micrometres and then finally polished flat. After initial experimentation, it was found that these steel pads would not be ideal for either testing or calibration purposes. Surface analysis inspections by means of a Zygo were conducted, and it was shown that differences in height (over the length of the pad) of up to 20 µm were present in some cases. For the laser honed steel pad shown in Figure 3.10, a two-dimensional surface profile can be seen below in Figure 3.11, whilst Figure 3.12 depicts the three-dimensional surface oblique plots.

Figure 3.10 An example of a laser honed steel pad specimen
It is stipulated that the most likely causes for these differences is due to the post-processing polishing carried out on the laser honed surface (as a mirror finish was originally requested). The pads can quite clearly be seen to exhibit a very specific chamfering effect towards the ends of both their breadth and length, as well as at the edges. The fact that the pocketed zones of the pad do not seem to accentuate this effect.
can largely be attributed to local hardening occurring due to the sheer heat produced by the energy-rich laser honing process. Consequentially, the pads cannot be utilised to employ optical interferometry to deduce the convergence angle of the pad, either during the calibration phase or during testing. A truly ‘flat’ surface would produce a set of equally spaced parallel fringes when exposed to a monochromatic light source as opposed to a crowned surface.

Other issues which presented difficulties included the quality of the laser honed pocket itself. The process of firing a high powered laser over a small concentrated area (a typical pocket measuring 100 µm wide and 1 mm long) often results in a poor pocket roughness. Surface roughness issues become apparent when the bottom layer and edges of the pocket are observed in finer detail, as seen in Figures 3.13, 3.14 and 3.15;

*Figures 3.13 & 3.14 Pocketed specimens highlighting roughness (magnification at 10x and 50x respectively)*
The final polishing process that takes place after the laser honing may also have a detrimental effect in that it may cause surface debris and potential high-spots present on the specimen to be deposited back into the pocketed zone.

Recent work carried out on the photo-lithography of Silicon wafers [54] has proven to be a successful technique to introduce precisely ‘machined’ features upon a surface as Figure 3.16 for depicts;

**Figure 3.15** Surface profiles of the pocketed region within a laser honed pad
It was decided to use Silicon pad specimens for the current research. A series of pad texture geometries were drawn up (with guidance from the aforementioned collaborators carrying out the computational study) and submitted to a dedicated mask manufacturing company. The results can be seen in Figures 3.17 and 3.18.

**Figure 3.16** Textured surfaces on Silicon (from [54])

**Figures 3.17 & 3.18** Chromium 5 inch square masks, pocket geometries shown on the left, pad cut-out on the right
Figure 3.17 depicts the actual pocket patterns to be etched on to the silicon wafers, whilst figure 3.18 shows the 20 mm by 10 mm rectangular pad cut-outs to be applied later to the pocketed silicon wafer. The five inch square Chromium masks were placed over a four inch diameter silicon wafer, and the pockets etched to a pre-determined depth by the process of photolithography. Once complete, the pad cut-out mask was strategically placed over the silicon ‘pocketed’ wafer and the process again applied (through the entire thickness of the 500 µm deep silicon wafer) to produce the individual pads. Examples of the finished pocketed silicon wafer are shown below in Figures 3.19 and 3.20.

**Figures 3.19 & 3.20 Examples of pocketed Silicon wafer pads produced**

The overall finish and quality of the silicon etched wafers were far superior to that of the original laser honed steel samples. Figure 3.21 shows a comparison of the two samples as given by their optical profiles (as given by the Veeco Wyko NT9100 optical profiler).
Figure 3.21 Optical profiles of the textured silicon (left) and textured steel (right) pad samples

Figure 3.22 shows an oil film fluorescing under U.V light for various pocket depths in the silicon wafer. The pocketed areas (bright pixels) can quite clearly be distinguished from the top surface (flats) of the wafer where the film is much thinner.

Figure 3.22 Fluorescent imaging of doped oil within various stated pocket depths

Criteria for the design of the test pad specimen holder (the gimbal) include its ability to self align (in the direction perpendicular to sliding) under test conditions and to allow the
user to perform precise adjustments when setting the initial convergence angle. The initial design proposal for the gimbal is shown in Figure 3.23;

![Diagram showing the initial design proposal for the test pad holder (bottom views shown)](image)

*Figure 3.23 Initial proposed design for the test pad holder (bottom views shown)*

The ‘centre-ring’ (labelled A in Figure 3.23) is held in position to an ‘outer-ring’ (labelled B in Figure 3.23) by means of freely rotating support pins. This outer-ring is then attached to the main gimbal housing (labelled C in Figure 3.23), again, by freely rotating pins. The direction of sliding (for the plan view in Figure 3.23) would be from left to right or vice versa. Thus, the pad is free to rotate in the axis 90 degrees to the direction of sliding. The angle of the pad, i.e. the convergence, is set using the small support bar (labelled D in Figure 3.23) permanently fixed across the main gimbal housing. This bar consists of an M3 grub screw located directly in the centre, which in turn allows for the adjustment of the outer-ring piece. Initial tests highlighted some issues with this initial proposal. It was found that excess play was present in the support pins, even during the loaded phase, and consequently the initial angle set was not maintained. Small bushes were produced and added to the outside of the support pins, their tolerance remaining small enough for a transitional fit, but not too tight as not to allow the self
aligning of the structure. Also, it soon became evident that the single M3 grub screw mechanism to adjust the convergence angle in minute increments would not suffice. A secondary support bar was manufactured and attached on the opposite side to the original. An M2 fine threaded grub screw was added as this would allow for finer adjustments with each complete turn of the screw thread.

3.7 Instrumentation

A brief outline of all the system sensors and their subsequent interfaces are described here. A simple approach would be to consider the main system components as given previously in Figure 3.1, and describe each of the five primary sub-systems separately. These being; Imaging, Friction, Disc Speed, Test Pad Load and Temperature monitoring. Each of these areas consists of various methods of detection, monitoring and recording. Again, the best method to visualise and configure all of these sub-systems is to represent them pictorially, as done so by Figure 3.24;
Figure 3.24 Instrumentation of the main system components

The Fluorescent imaging sub-system is not shown on the above figure for display purposes.
3.7.1 Imaging

The camera (model Rolera MGi) can be considered the main sensory device when referring to the imaging instrumentation. As it is a bought-in part, little needs to be said in regards to its operation and specific working details. It is connected to a stand alone dedicated computer due to its memory intensive nature (sequences of images up to 100MB in size were recorded during trial runs). It is thus connected to the computer by means of a IEEE 1394 FireWire cable (capable of large amounts of data transfer at high speeds). The computer was fitted with the highest possible physical memory (RAM) available (4GB) coupled with a dual core high speed processor and large 500GB hard disc drive.

3.7.2 Friction

As explained in Section 3.3.3 earlier, the friction sensing is achieved by means of non-contact displacement sensor situated besides the load arm. This has the ability to detect minute deflections of a specified target area situated on the load arm. The equipment employed in the current set-up is a Kaman Instrumentation Multi-VIT displacement measuring system, specifically the KD-2300. It is defined as a ‘linear proximity measuring system’ that makes use of an eddy-current sensing method. As depicted in Figure 3.25, the 12 mm diameter sensor tip is permanently attached besides the load arm via an adjustable platform.
This is strategically aimed at a bracket that is attached to the load arm. As the load arm moves away (or towards) from the sensor, a voltage differential is the result. The output voltage is proportional to the distance between the target material (aluminium in this case) and the sensor face. A resolution of 0.3 µm over a measuring distance of 2.5 mm is the specification given. The unit is simply connected to a data acquisition unit (the DAQ, explained later in Section 3.8).

### 3.7.3 Disc Speed

An 8 bit rotary position encoder (giving a 1.41 degree resolution) fixed securely to the bottom of the main shaft pulley, was used to determine the angular position of the disc/shaft coupling as well as the angular velocity (later translated to give the relative speed at the disc/pad interface). This converts the encoder pulse to an analogue speed voltage which is fed into the DAQ (a high-speed data acquisition device). The DAQ then
outputs a signal to the speed control circuit for a requested speed, and consequently outputs a known (but variable) voltage to the motor control unit, using a feedback circuit from the original analogue speed signal from the encoder. The control box unit (as well as the power supply unit powering it) and DAQ can be seen in Figure 3.26;

![The DAQ system](image)

**Figure 3.26 The DAQ system**

### 3.7.4 Test Pad Load

The applied load is given by four foil strain gauges located on upper steel bridge piece that forms part of the coupling between the pad specimen holder and the load arm. This is shown in Figure 3.25 above. Two strain gauges are located on the top side of the bridging piece, whilst two are located below. In accordance with standard recommended procedure, the strain gauges are fixed securely (using Cyanoacrylate) and then siliconed over to provide adequate liquid-sealing properties. These strain gauges were calibrated with known loads to accurately determine the load applied to the arm during a test. This
is explained further in Chapter 4. The amount of strain gauge deflection directly affects the resistance, which in turn is directly proportional to the voltage output. A simple electronic set-up consisting of a Wheatstone bridge amplifier is used to convert the strain gauge outputs into an output voltage. Again, the continually variable voltage outputs are fed into a data acquisition board (DAQ).

### 3.7.5 Temperature Monitoring

A type ‘T’ thermocouple was fixed securely within the tubing at the delivery end of the oil nozzle as seen below in Figure 3.27.

*Figure 3.27 The oil delivery nozzle containing the type ‘T’ thermocouple*
The thermocouple was bought in as part of a unit (TME MM200), specifically chosen due to the type ‘T’ thermocouple that it uses. Type ‘K’ thermocouples generally have a greater temperature range but at the cost of resolution and accuracy. Type ‘T’ thermocouples have a reduced measuring range (up to 400°C) but a significantly improved accuracy and resolution to suit. This specific model has an accuracy of ±0.15% of the reading at 23°C (approximately translated to ±0.2°C at room temperature), coupled with a resolution of 0.1°C over the entire temperature range. As no heating equipment is used in the current set-up, it was originally envisaged that there would be little or no temperature increases during testing. A handheld RS instruments ‘Mini Infra-red thermometer’ was used to monitor the temperature of the oil pump as well as the motor driving the oil pump. The accuracy and resolution of this unit were significantly of a lower specification than that of thermocouples used (±2.5°C at 0.1°C respectively).

3.8 Experimental Data Capture and Storage

As previously mentioned the friction, load and speed components all output data via a fluctuating voltage signal coupled to a data acquisition board. For simplicity, a USB powered ‘National Instruments USB-6008’ 12bit analog input DAQ was purchased and connected to a second computer. For ease of communication and control, LabVIEW was the preferred choice of software due it being an easy to use graphical programming tool. Interfaces relating to each of the inputs were constructed and saved collectively. Real-time monitoring and data acquisition was the primary goal when designing the software
package. For this reason, a one-click virtual record button was incorporated into the software which allowed for mass data to be written in raw format (.txt file) to a specific target location for any desired length of time. An external 500GB hard disc drive was connected to this second computer to simultaneously record data as back-up to the main drive of the computer.

The ability of the camera to accept an external trigger enabled the use of the software to send a trigger signal (produced by manipulating data from the ‘sync-pulse’ received from the 8bit rotary encoder attached to the main shaft driving the disc) to the camera. The triggering circuit supplies a TTL signal to the camera when the main shaft passes a set position, controlled by an 8bit digital signal from the DAQ. The circuit is subsequently triggered when this value matches the encoder output.
3.9 Summary of Chapter

A concise overview of the main experimental techniques and the developments made to any design features has been presented in this chapter. The reader is referred to the photographs below for the final design features selected for the rig.

**Figure 3.28** The steel hinge ‘shim’ assembly

**Figure 3.29** The pad holder (gimbal) assembly
**Figure 3.30** The oil pump assembly

**Figure 3.31** The oil delivery nozzle in its operating position
Figure 3.32 An assembly view depicting the load arm, non-contact displacement sensor and the oil bath/nozzle arrangement

Figure 3.33 The completed test rig apparatus (LIF camera not shown)
Chapter 4

Experimental Method
4.1 Introduction

This chapter describes the techniques and methods used to calibrate each of the specific areas of the test rig. These are dealt with thoroughly and suitable justifications and reasoning given where necessary. The final test procedure used to produce data from the rig is then explained, outlining all the test variables and their respective controllability.

The primary focus that was kept in mind when designing the experimental rig was the ability of the rig to accurately measure and record the film thickness, the friction/load conditions and possibly cavitation, all \textit{in-situ} and all simultaneously.

4.2 System Calibration

The literature has placed caution on the repeatable achievement of accurate calibrations of both the L.I.F. and friction measurements. The three crucial areas of focus in the proceeding section deals with the L.I.F. and camera system, the friction and load calibrations.

4.2.1 L.I.F. Calibration

A certain degree of emphasis can be placed upon the importance of the L.I.F. calibration as the L.I.F. images taken during the experimental phase were to be compared directly to calibration images taken immediately before testing. Historically, this has been the most accurate method of determining each individual film thickness [44]. The first procedure
performed in this work to achieve this was to set up a pad of fixed convergence against a glass disc (optically flat and semi-reflective Chromium coating applied on the lower side) with an oil film present (by means of placing a small drop of oil in the centre of the pad) between the two surfaces. As the pad is brought into contact with the static glass disc, one side will touch the glass disc with no film present in between, while the other side of the pad will have the maximum possible film thickness. With regards to the current work, it was envisaged that optical interferometry would be employed to determine the convergence angle of the system. This can be determined by counting the number of fringes visible over a known length of pad section. The approximate wavelength of light (green 532 nm) is known with the use of a narrow band pass filter and consequently the depth of a certain point along the pad can be calculated using trigonometry, as the pad is presumed to be rigid and optically flat.

The L.I.F. camera system was positioned at the inlet region of the pad (referred to as ‘zero-point’ in this case), and a fluorescent image was captured. The whole camera set-up was displaced a known distance along the length of the pad and another image taken. The process continued along the entire length of the pad until a complete profile had been recorded. Simple trigonometry gave the theoretical film depth at any given length. Figure 4.1 depicts this motion of the fluorescent camera over the convergent pad.
However, it soon became apparent that this technique was severely flawed; preliminary tests consistently over-estimated the film thickness readings as predicted by theoretical analysis, regardless of the test conditions. The following section presents several arguments put forward and the manner in which they were investigated.

The first port of call was to thoroughly investigate the initial calibration technique used. A typical wedge film thickness comparison for experimental data obtained versus theoretical values calculated is shown in the graph in Figure 4.2;
Clearly, the experimental film thickness is consistently higher than the predicted theoretical film thickness (obtained by extrapolating the convergence angle as given by the initial optical interferometry method, over the entire length of the pad).

The newly fabricated silicon wafer pad samples were then measured using both optical (Veeco Wyko NT9100) and surface profiling (Taylor-Hobson Talysurf) instruments. Figures 4.3 and 4.4 show the centre line profile for the plain silicon pad across both the length and breadth, as given by the Talysurf;
Figure 4.3 The profile of the silicon pad

Figure 4.4 The profile of the silicon pad
The same measurement as given by the Veeco Wyko NT9100 is shown below in Figure 4.5;

![Figure 4.5 The profile of the silicon pad](image)

The original steel pads supplied exhibited extreme cases of chamfered edges and an overall ‘warped’ surface profile, an issue which was thought to have been assumed eradicated with the introduction of optically flat silicon wafers. This was not the case. This non-linear surface profile is thought to have detrimental consequences upon the calibration curve produced as a result.

These surface profiles quite clearly show a small lip on the outer edges of the silicon pad, both along the length and breadth of the entire pad. Although relatively small in comparison to the overall dimensions of the pad (approximately 1.5 µm over a 20 mm long by 10 mm wide pad), it was concluded that this non-intentional feature of the pad was enough to produce an incorrect initial pad geometry (and consequently deeming the
calibration curves as incorrect). As the convergence angle was determined by fringe-counting over the first millimetre of the pad, any angle present on this section of the pad leads to serious errors in the extrapolated spacing for the remainder of the pad. Consequent film thicknesses derived from image intensity values calibrated against this spacing would also be false. This lip issue can be attributed to the process by which the epoxy weld sets and cures during its drying process. This two-part epoxy adhesive was used as early attempts to adhere the silicon pad to the steel pad using Cyanoacrylate adhesive proved to be unsuccessful due to a rippled surface resulting due to uneven curing rates over the entire surface area of the pad, resulting in a ‘warped’ surface.

At this stage, efforts were concentrated upon the production of a more viable means of calibration, albeit with one crucial condition in mind; The calibration step must take into account the same glass disc and test pad material (silicon in this case) due to both transmission and reflective phenomena. A calibration curve produced from implementing a thinner, darker or even different material (Quartz as opposed to fused Silica) in conjunction with a material of differing reflectivity from that of silicon wafer may produce a degree of variance for light-transmission purposes. For this reason it was decided to use a modified calibration technique, described as follows.

A small section of a #1 Chance and Propper glass cover slip was attached (by means of epoxy adhesive) to the top surface of a plain silicon pad (fixed securely to a steel pad with epoxy adhesive). This arrangement can be seen in Figure 4.6;
This is effectively, again, a calibration wedge produced by introducing a small step of known height at a known distance from the edge of the silicon. However, the fundamental difference from the previous attempt was that the convergence angle ($\theta$) is now known unequivocally. The calibration wedge loaded up against the glass disc is shown in Figure 4.7;
Even though the lip is still present at the edge of the silicon (at the location referred to earlier as ‘zero-point’) its impact upon the overall angle is minimal. As before, surface measurements of this calibration piece (primarily to determine the height of the cover slip step) were undertaken. The results of which can be seen in Figure 4.8.

![Figure 4.8 The revised calibration piece optical profile](image)

The average height of the cover slip over the leading edge was found to be approximately 145 µm. Both the 2D and 3D surface profiles shown in Figure 4.8 above are over the
bottom right hand corner of the cover slip arrangement. The thickness range (as quoted by the manufacturer) for a #1 cover slip is between 130 and 170 µm. Measurements conducted using both Mitutoyo vernier micrometer and callipers gave an average thickness of 140 µm ± 10 µm. The various surface profiles as measured by the Veeco surface profiler seem to support these readings. It must be noted that any inaccuracies in the adhesion process would ultimately lead to a step height greater than ~145 µm as an air-gap would be present under the cover slip possibly giving an incorrect light intensity reading (leading to a false film thickness reading). With regards to any misalignment of the cover slip during adhesion to the silicon pad, it was presumed that the flexibility of the cover slip would be sufficient to overcome any gaps present (between the cover slip and silicon pad), and would consequently ‘flatten’ when loaded against the glass disc. The revised calibration wedge proved to be a much improved technique, as can be seen in Figure 4.9;

**Figure 4.9** The revised film thickness profile along the length of the pad
The effect of temperature upon fluorescence has been previously studied [36] and subsequently concluded that small variations in temperature of an oil/dye mixture had significant effect. To test this phenomenon in the current work, a controllable heating mechanism was constructed. This consisted of a pad heater ‘sandwiched’ between two plates of aluminium. A piece of silicon wafer (same thickness as the silicon pads, approximately 0.5 mm) was then placed on top of the heater, with two #1 cover slips laid either side of a drop of oil/dye mixture. Finally, a 10 mm thick float glass disc (the same as to be used in testing) was then placed on top of the two supports, trapping the oil/dye mixture between the silicon and glass, providing a constant film thickness as seen from above. Figure 4.10 depicts the arrangement;

![Figure 4.10 The pad heater arrangement](image)

The L.I.F. microscope was set up directly above the oil/dye droplet, where it was to remain for the remainder of this particular test. An initial reading was taken after five minutes, once the oil/dye mixture had adequately settled whilst maintaining a full film
between the silicon wafer and glass disc. Readings were taken at five minute intervals over a period of approximately three hours. The heater, set initially to 40°C, was switched on ten minutes after the start of the test, and raised to 50°C thirty minutes later. Two and a quarter hours later, the heater was permanently switched off, and readings were continually taken until a little over three hours had elapsed since the beginning of the test.

The images taken were then compared to calibration image files (obtained earlier in the day) to give film thickness data for the test, the results of which can be seen in Figure 4.11;

*Figure 4.11 Oil film temperature as given by the pad heater arrangement*
At no point during the above test did the film thickness increase due to a temperature increase. It was therefore concluded that the fluorescent intensity of this specific dye does not increase with a rise in temperature. The fluorescent intensity (and consequently the film thickness) starts to decrease from the very beginning of the test, a result typical of the effects of photobleaching. If a dye is continually or over-exposed to a light source (typically matched to its excitation wavelength) it eventually degrades the fluorescent component present within the dye.

During the test explained above, the oil/dye mixture was only exposed to the light source at the instant the image was taken by controlling the shutter built into the light source; over 40 images were taken over the entirety of the test. With each image taking approximately ten seconds to complete, it is estimated that the mixture was exposed to the light source for seven minutes. The light source was operated with a completely open shutter mechanism, and thus the oil/dye mixture was exposed to an intense source of U.V long enough to degrade the fluorescent component. As expected, any subsequent images taken showed marginal differences in pixel intensity from an image taken previously. Repeats conducted of the above test with the shutter left continually open and the heater switched off throughout, exhibited the same photobleaching phenomenon occurring at a much faster rate. The effects of photobleaching were determined to be of the utmost importance when considering image intensity data, and the issue of controlling the shutter time during the calibration process was duly noted (i.e. kept at a minimum at all times).
Following on from this issue of external factors affecting the fluorescent characteristics of the dye, the effect of elapsed time upon the EM camera was investigated. As has been previously mentioned in Chapter 3, an inevitable draw back of the Rolera MGi EM gain camera is its trait of drift in linearity. Calibration curves obtained from ten consecutive days of testing are presented in Figure 4.12 below;

![Graph showing calibration curves for high and low gains](image)

**Figure 4.12 Calibration curves produced for high and low gains**

The two distinct curves are for the two gains that the calibration was conducted at (camera set at a high gain for detecting thin films and lower gains for thicker film). Efforts were made to ensure that the calibration procedure (i.e. continuity of EM gain camera and light source warming-up period, adequate mixing of the oil in the bath, cleaning of the calibration piece) remained identical in terms of both preparation and execution;
however, external factors such as room temperature and humidity (ambient factors which affect the working characteristics of the EM gain camera) could not be controlled but were noted. Clearly, as can be seen from Figure 4.12, small variances in the image intensity for the same apparent film thickness can be seen to exist.

This finding highlights two important factors that were evident in the current research; the importance of carrying out the calibration at the beginning of each day’s testing, and secondly, to appreciate the difficulty in determining the film thickness when considering images taken at lower gains. For a small difference in the image intensity (Y-axis) in Figure 4.12 a wider distribution of film thickness (X-axis) is a consequence, especially when one considers the curves produced from the high gain setting. It is estimated that for film thickness below 80 µm, the accuracy of the repeatability is approximately 5%. For film thicknesses above this value, this accuracy decreases to around 10% of the film thickness reading.

The graph above in Figure 4.12 considers the ‘long-term’ effect of time upon the intensity variation. Following on from this, the shorter term effects were also investigated. The exact same procedure as described earlier to attain the film thickness relationship for the calibration wedge was set up. Film thickness readings over the length of the pad were taken over a period of eight hours. This was carried out for both low and high gain settings on the camera, as presented in Figures 4.13 and 4.14 respectively;
Figure 4.13 A comparison of the film thickness variation at low gain

Figure 4.14 A comparison of the film thickness variation at high gain
An obvious difference present in both graphs is the gradual decrease of perceived film thickness, regardless of the absolute film thickness value and the gain at which the imaging was undertaken. This effect is more pronounced as the film thickness increases. Interestingly, the film thickness at the lowest distance from zero point (0.5 mm) as given by the low gain setting is perceived as near zero. At the high gain setting, the film is in the region of 1 µm. Consider the two curves as presented in Figure 4.12; small fluctuations in the image intensity are easily detectable by the camera at the higher gain setting, and accordingly there is a narrower range of possible film thickness to suit. As well as the possibility of photobleaching, it appeared that there was a tendency of the camera’s EM gain property to drift over time.

An unexpected issue that presented itself in the midst of the above tests was that of focussing. Every camera system has a certain depth of focus within its optics, and the current work is no exception. This is often defined as the vertical range that the microscope objective lens can focus to within the target area at any one time, the optimum being when the target is in perfect focus during imaging (determined by the sharpness of the image). Consider a film of thickness \( h \) deposited between a glass disc and silicon surface as depicted in Figure 4.15.
Figure 4.15 Variation of focusing depths through a known film thickness

A film of known constant thickness (ascertained by using a #1 cover slip) was imaged for a few millimetres along the horizontal plane. Firstly, the focus of the microscope objective was set in a position where the focal plane was close to that of the underside of the glass disc (position 1 in Figure 4.15) and not adjusted during which time a number of images were taken. The focal plane was then adjusted to be close to the silicon base layer of the set-up, as depicted by position 2 in Figure 4.15.

Embedded within the QcapturePro camera software employed to capture the images, is a real-time spectral distribution graph. This encapsulates the given image in terms of both gray scale and intensity plotted against each other during the time the live image is displayed on screen.

It is presumed theoretically that if a fluorescing film were to be imaged from above in Figure 4.15, the image intensity reading would be at its greatest directly in the centre of
The measured thickness of the specific cover slip used in this test was found to be 150 µm, as depicted by the red base line in Figure 4.16. The two anomalous points in both sets of readings (at position 5 and 11 mm) are due to the presence of two small separate pockets located in the silicon wafer used for this test, each of approximately 21 µm depth. Interestingly, the average film thickness value of all 24 points obtained from both
sets of curves equates to be 157 µm, an error of 7 µm from the baseline. However, more concerning is the relatively large range of film thicknesses possible. Readings taken at focal position 1 had an average reading of 176 µm (17% above baseline) whilst readings taken at focal position 2 averaged 138 µm (8% below baseline), but it is the peaks and troughs present within these sets of data that pose the most concern. One anomalous reading would suffice to distort an acceptable level of tolerance.

As the current research is concerned with the measurement of an unusually large range of film thicknesses, this issue posed a degree of difficulty. During film thickness measurements, no easily identifiable areas of interest (i.e. features or marks on the either the glass or silicon) were present to enable easy focusing. A level of compromise to combat this was achieved in the way of utilising the spectral distribution graph as mentioned earlier. The graph plots a peak intensity level whilst focusing; it reaches a peak midway through the film thickness (presumed to be the centre depth as discussed, highlighted by x in Figure 4.15) and then proceeds to decline as the focal plane approaches the bottom of the film. It was decided that once this intensity reaches a peak, that would be the midway distance of that specific film thickness and consequently an image would only be taken at that focal position. This same method of focusing was used in the calibration process.
4.2.2 Friction Calibration

Calibration of the friction sensor was as follows; the load arm was displaced in the tangential direction by loads in 0.098 N increments up to a maximum of 1.667 N. To achieve this, an aluminium pad with a simple loop of fine cotton passed through the centre of the pad was manufactured. The cotton line was fed through the side of the oil bath and over a low friction pulley rigidly fixed to the base of the test rig. Voltages were monitored for an entire loading cycle (loading up and down) and the average values used to determine the equation of the relationship. Using this relationship between the frictional force and voltage output attained by means of calibration, it is possible to determine the respective friction for a given voltage during testing simply by applying the equation of the line to the given voltage. A typical friction calibration graph is presented in Figure 4.17.

![Figure 4.17 A typical friction calibration relationship](image-url)
4.2.3 Load Calibration

The load arm strain gauges were loaded to give a calibration plot of load (in Newton’s) versus voltage output. The load arm was loaded incrementally to approximately 40 N, and then unloaded (achieved by fabricating a test pad holder capable of supporting large laboratory weights). Again, the average of the two voltage readings was used as a basis for producing the equation of the line, and thus the calibration plot. A typical load calibration graph is presented in Figure 4.18.

![Figure 4.18 A typical load calibration relationship](image)

4.3 Test Variables and their Controllability

The four textured pads selected for this research are detailed by Figures 4.19 and 4.20. These were selected on the basis of the predicted increase in load support, decrease in friction and the possible occurrence of cavitation due to the pocket geometry. A pocket depth of 21 micrometres was chosen (this was a result of advice received from the computational collaborators).
Figure 4.19 Pocketed pad specimens (A05 on the left A07 on the right, all dimensions in mm, diagram not to scale)

Figure 4.20 Pocketed pad specimens (C07 on the left C06 on the right, all dimensions in mm, diagram not to scale)
The black rectangular areas in the above figures depict the pocketed zones. The plain pad (not shown above) has the same outer dimensions as the pocketed pads (20mm by 10mm). Once permanently adhered to the steel pads, the direction of each pad can be flipped by 180° effectively giving two possible configurations for each pad. Pockets were placed either at the front or rear of the pad, relative to the oil inlet. Therefore the current research investigates a total of nine pad samples.

Textured pad A05 has a single large square pocket (dimensions of 7 mm by 7 mm) located 3 mm from the edge of the pad. Pad A05 consists of a single rectangular (dimensions 7 mm by 3 mm) pocket, again, situated 3 mm from the edge of the pad. The two remaining pads, C07 and C06, differ from these single large pocketed pads in that they are made up of six smaller distributed rectangular pockets. These smaller pockets are 1.2 mm by 3 mm in size, and are located either perpendicular (pad C07) or inline (pad C06) to the direction of the oil flow.

4.3.1 Oil Temperature

As this study is not concerned with the performance of specific oils, little attention was paid to temperature effects upon the oil as a result. However, temperature rises within the bulk oil reservoir were noted over prolonged periods of time; attributed directly to the work done by the oil pump on the oil itself during pumping. As a result, the viscosity of the oil will vary also.
As the oil pump employed in the current research is one that is capable of attaining a large \textit{head of flow} (typically over ten metres for an automobile oil pump such as this), naturally work done on the fluid results. The oil and motor temperatures were both monitored over a length of time, as seen in Figure 4.21 below;

\begin{quote}
\textbf{Figure 4.21} Film thickness comparisons at high gain
\end{quote}

In accordance with normal test procedure (discussed in the proceeding section), the above trial was carried out before any prior testing on that specific day to ensure ambient oil (and motor) temperatures at the beginning of the trial.

As can be seen from Figure 4.21, the rate of temperature increase of the motor temperature is slightly ahead than that of the oil, and it does seem that the correlation...
between the two is quite evenly matched. It was the general consensus that neither the oil nor motor temperature would plateau to a sustained level after any given length of time, and that the temperature would continue rising. The addition of a series of cooling fans across both the motor and the oil pump proved to be successful in that the temperature rise of the oil (as taken at the oil delivery/nozzle end) remained to within $2^\circ$C after two hours of continuous running. It was deemed unlikely to limit this temperature differential any further as fluctuations in the room temperature (i.e. ambient temperature) were of similar magnitude. Coupled with this, the result of temperature variations upon the test oil would be dealt with by means of correcting for the viscosity by reading off the viscosity index chart at the relevant temperatures, which would be noted for every test. An oil viscosity chart for the oil used (a Group 1 base oil) throughout the test sequence is presented in Figure 4.22 below.

![Oil viscosity versus temperature chart](image)

**Figure 4.22 Oil viscosity versus temperature chart**
4.3.2 Variation of Convergence with load

Static loading of the test pad (i.e. the load arm) was carried out. A small drop of oil/dye mixture was placed on a plain pad and loaded up towards the stationary glass disc. A series of L.I.F. images were taken at known locations, both towards the ‘zero-point’ and thick film regions. Using simple trigonometry, this enabled the user to calculate both the inlet and outlet film thicknesses $h_i$ and $h_o$ respectively. A graph of load versus the difference between the inlet and outlet film thickness, is shown in Figure 4.23;

![Graph showing variation of convergence with load](image)

*Figure 4.23 The effect of static loading upon the attitude of the pad*

A series of measurements were taken whilst loading up to 90 N. As is rather telling from the above figure, the initial convergence was not maintained and the continual variation of the effective convergence of the perceived test pad specimen is significant. The original confines of the research were inclined to set the initial convergence angle, and assume it to be maintained throughout the entirety of the test. Based upon the outcome of the
above trial, this was concluded not to be the case, and amendments were made accordingly;

Initially, the original loading stipulations were such that the test pad specimen was envisaged to be loaded up to 100 N in 10 N increments. This was changed to a maximum of 20 N; with three loading steps of 5, 10 and 20 N. The vertical z-Stage used to displace the load arm was turned until the desired load was achieved (as shown on screen by LabVIEW)

Secondly, the convergence of the pad specimen was calculated during the actual test procedure. It was clear that the initial inclination of the test pad is not maintained as the load is changed, most likely as a result of elastic displacements of the suspension system. However, since the film thickness was measured at several locations within the bearing primarily for other reasons (see Chapter 5), the actual inclination (and the consequent convergence ratio) was determined explicitly for each experimental condition examined.

4.3.3 Disc Speed

As mentioned, the disc speed was controlled by adjustment of the request voltage within LabVIEW. The required speed would typically be attained after a few seconds, due to the type of feedback employed. The two disc speeds used in this current work are 0.5 and 1.0 metres per second.
4.4 The Test Procedure

A brief description of the actual test procedure used is now presented. Prior to testing, calibrations of load, friction and L.I.F. sub-systems were all carried out. As described previously, a series of masses (gradually increasing in total mass) were placed in a specialized pad carrier located at the end of the load arm. The relationship constants were then substituted into LabVIEW. This process was repeated for the friction calibration, a technique of hanging a small series of known masses from the end of the load arm to precisely determine the relationship between the distance moved by the load arm and the equivalent friction. Again, constants in LabVIEW were adjusted to suit. Calibrations of the three systems were carried out prior to the start of testing.

For consistency, the camera and light source were switched on for a period of no less than thirty minutes before the start of any testing. This fulfilled two important criteria; the camera reaching its peak performance operating temperature by cooling the CCD sensor below ambient (stated as -25 degrees Celsius) and that the Mercury-Xenon lamp used in the light source had adequately reached its optimum steady light output.

As well as the above criteria, the cleaning of the test samples and pad specimens were also carried out. Normal procedure to ensure a satisfactory cleanliness was achieved by rinsing the sample in toluene first, and then in acetone to ensure effective removal of any solvent residue. The sample is then left to dry naturally. This entire process was performed upon the glass disc, the calibration wedge sample and the test pad specimen itself.
The L.I.F. calibration wedge piece was placed in the load arm gimbal, and both adjustment grub screws were set in the fully open positions, enabling maximum uninhibited movement of the pad with regards to its convergence angle, so that once loaded up against the glass disc, the calibration piece would find its natural convergence angle (always a constant), as determined by the position and height of the cover slip (described in detail in Section 4.2). After application of a drop of oil/dye mixture onto the centre of calibration wedge, the load arm was loaded up towards the bottom of the glass disc to approximately 2 N, a load not likely to unnecessarily cause any elastic deformation of either the glass disc, the edge of the silicon pad or the thin glass cover slip. A series of images were taken in accordance with the calibration procedure described in detail in the previous chapter.

Following a successful calibration, the calibration wedge piece was replaced with the test pad specimen to be used (either plain or textured). The convergence adjustment grub screws were consequently tightened to exhibit a small degree of convergence on the test pad specimen (this was initially set by eye, and then later adjusted if necessary by placing a few drops of the oil/dye mixture on to the test pad, loading to low loads and shining a hand-held U.V. light source over the glass disc on to the contact film).

To ensure a thorough circulation of the oil/dye mixture, the oil pump motor was left to run for approximately ten minutes at a predetermined speed, after which the oil pump cooling fans were switched on. This had two consequences; to minimise any 'settling'
effects of oil and dye that may have occurred over time, and to bring the oil/dye mixture up to 'operating temperature', typically 2-3 degrees above the ambient temperature (both the laboratory room temperature and humidity level were noted at the beginning of each days testing). Great fluctuations between oil temperatures from a day-to-day basis were not experienced and small temperature differences were accounted for by use of a measured viscosity-temperature relationship.

The disc speed was selected (a velocity of 0.5 metres per second was set initially) and the load arm loaded up on to the glass disc by means of inducing a vertical displacement to the vertical z-stage (to which the load arm is attached). Once the desired load had been achieved (5 N to begin with) the rotating mechanism situated on the z-stage was locked in position as to not allow any undesired further movement, thus eliminating any possibilities of temperamental loading occurring. Once loaded, a series of L.I.F. images were taken of the hydrodynamic film in four known locations over the area of the test pad specimen, shown below in Figure 4.24.

![Diagram of L.I.F. image locations](image)

**Figure 4.24 The L.I.F. image locations as shown from above (Not to scale)**
Positions A, B and C are situated on the right hand side of the pad 0.8 mm from the edge of the pad, whilst position D is situated 0.8 mm from the left hand edge, as depicted by the plan view in Figure 4.24 above. The two points A and D were chosen to be 1.0 mm from the top edge of the pad (i.e. the outlet of the pad). The distance between positions A and B and between B and C were known during testing by noting the position of the micrometer vernier attached to the L.I.F. camera platform stage. A sequence of twenty images were taken at each location, each image being triggered by the position encoder as discussed. The higher gain setting on the camera was used for when taking images at low film thicknesses (typically at positions A and D) whilst the lower gain employed for the larger film thicknesses (positions B and C).

Upon completion of the imaging, the friction present between the pad and spinning glass disc was recorded for a ten second burst. A feature within LabVIEW allows real-time data capture for specified periods of time. A ten second burst of data was recorded onto a specified destination, during which time the temperature of the oil/dye mixture at the nozzle end was noted, and embedded in the file name of the friction file. It was at this stage that the entire hydrodynamic film (i.e. covering the whole pad area) was monitored for any signs of cavitation present.

The load arm was then loaded to the next progressive step, 10 N in this case, and the imaging and friction-monitoring procedure repeated. Once data for a final load of 20 N had also been captured, the load arm was completely unloaded away from the bottom of
the glass disc. The speed of the glass disc was increased to 1.0 metre per second, and the loading procedure repeated allowing for image and friction capture.

Following completion of two disc speeds, both the glass disc motor and oil pump motor were shut down for a brief period to allow for the convergence angle of the pad to be altered. Normal testing procedure would typically begin with relatively high convergences, which were progressively lowered throughout the days testing. At times, a very low convergence would subsequently prove to be too low as upon loading of the test pad up towards the spinning disc, the initial convergent angle on the pad would deviate to a divergent angle. Support of any load would ultimately prove to be too great a burden upon the film and triggered the collapse of said film. A number of repeats for various convergence angles were conducted throughout any given day of testing. Whilst the absolute convergence value would not be known until image analysis, if by chance two similar convergences were set for any two or more tests, this was judged to be a good indicator of the repeatability of the rig if applicable.

The entire test procedure was repeated for the four textured pad samples, both with the pockets orientated towards the front of the pad inlet (A05, A07, C07 and C06) and situated at the rear of the pad inlet (A05r, A07r, C07r and C06r).
4.5 Summary

This chapter has not only described the techniques used to calibrate the three main sub-systems employed on the rig, but also documented the developments of the method. Most notably, the L.I.F. calibration has been reported. The possibilities of certain reported factors affecting fluorescence intensity have been investigated and eliminated or accounted for. The resultant novel technique to determine the film thickness (and subsequently the convergence ratio) of a test pad specimen during testing, has also been described. Finally, the four unique textured and one plain test pad specimens examined within this study have been introduced and the manner in which they have been tested has been presented.
Chapter 5

Hydrodynamic Test Rig Results
5.1 Introduction

Corresponding film thickness and friction data for all eight textured configurations and one plain pad sample is presented in this chapter. The results are graphically depicted together with data obtained from analysing a plain pad arrangement theoretically, an approach that utilises Reynolds equations. Alongside this, the percentage difference in friction between the theoretically derived plain pad analysis and the experimentally derived data is shown. The relationship between the theoretically derived load support ($W^*$) versus convergence ($K$) for all test pad cases is also introduced, where $W^*$ is defined from [1];

$$W^* = \frac{h_o^2}{6U\eta B^2L}W$$

And $K$ as;

$$K = \frac{h_i - h_o}{h_o}$$

Finally, theoretical $h_o$ versus experimental $h_o$ values are compared.

5.2 Variation of Minimum Film Thickness, $h_o$

Prior to the start of the main programme of testing, a plain pad specimen loaded up towards the spinning glass test was set-up. This test incorporated all of the previous criteria with regards to both calibration and procedure; however, the only fundamental difference was the number and location of fluorescence images taken. Consider the plain pad case depicted in Figure 5.1;
Up to this point, the mention of convergence in all of the previous cases has been in the $x$ direction as portrayed in Figure 5.1. An early assumption was made that due to the self-aligning design feature used in the gimbal mechanism, that no convergence would be present in $y$ direction. As mentioned, a typical plain pad set-up was conducted, but with images taken in 120 locations as opposed to the standard four locations as introduced in Chapter 4. A disc speed of 2 metres per second and a constant load of 60 N were used for this test. Figure 5.2 highlights the approximate grid reference used to locate these 120 positions;
Figure 5.2 The 120 point grid referencing layout

The location of all 120 of these points is known by using the edge of the pad as a guide, and this is accomplished by traversing the L.I.F. microscope on the vernier micrometer XY platform accordingly. The film thickness data obtained for this test is presented in Figure 5.3.
As can be seen from Figure 5.3, there is a certain element of attitude in the $y$ direction (due to the expected relationship with the entrainment speed profile) which will be referred to as $d_{hy}$. A positive $d_{hy}$ term indicates that the film thickness at grid reference A1 is greater than the film present at grid reference F1, and vice versa. The film thickness values in and around region F20 were too large to be accurately determined by the fluorescence microscope, resulting in saturated images on screen during testing, and thus the data is partially missing in this region. However, one can imagine the film thickness trends by extrapolating the immediate data around the affected region.

**Figure 5.3 Film Thickness data for the 120 point grid test case**
During experimental testing, fluorescent images were taken at two locations near the rear (outlet) of the pad. Consider the pad configuration as given in Figure 5.2; an image was taken at grid reference A1. This film thickness shall be labelled as $h_{oL}$. A second image was taken at grid reference F1, which shall be referred to as $h_{oR}$. The average value of these two values would theoretically provide the film thickness at the central point along the length of the test pad. This film thickness shall be nominated $h_{oCt}$ and is the primary reading used to give the minimum film thickness, $h_o$ in the following section.

**5.3 Minimum film thickness and Friction versus Load for all Convergences**

The graphs presented within this section depict the relationships between the minimum film thickness at the outlet ($h_o$) versus the applied load and the actual friction readings versus applied load for all experimental pad configurations (i.e. pad convergence, load and speed accounted for).

A comparison of all the plain pad experimental data against the theoretical plain pad model (as given by two-dimensional Reynolds) is presented in Figure 5.4. Both minimum film thickness and friction obtained during testing are plotted against the actual load, for all convergences.
With regards to minimum film thickness, the general theoretical trend is replicated well by experiment. The minimum film thickness gradually decreases with an increasing load, for both the low and high speed conditions. The graph presents all the experimental data points over a large range of convergences, and inevitably there is a relatively large spread of minimum film thickness to suit. The range of minimum film thickness is marginally larger for those tests conducted at lower loads (5 N) when comparing against those exhibited at high (20 N) loads. If a perfect theoretical model is to be believed, the minimum film thickness data points at five Newtons are slightly over-predicted by experiment. The ranges of both low and high speed clusters are relatively similar, but absolute values are in the region of 10-15% too high when referring to equivalent experimental data points. The correlation improves, but still over-predicts, for mid-load (10 N) tests and further still for high loads (20 N). It must be noted at this stage that the range of film thicknesses for all plain pad tests is between approximately 15 and 70 µm, within the range calibrated for.

Good correlation is seen for friction values for all tests whilst comparing experimental and theoretical data. Again, the range of values is relatively large due to a large range of convergences seen during testing. The friction values increase steadily as the load increases, both for low and high speed tests. Coupled with this, the spread of friction results is larger for higher loads, where typically lower minimum film thickness values are encountered.
Even with this slight apparent 'over-prediction' in film thickness in mind, the minimum film thickness data as given by Figure 5.5 clearly displays a higher separation between both textured pad A05 (experimental) and theoretical plain pad data cluster points. This effect is exhibited for all test conditions; over all three loads and two speeds. Again, good agreement exists when considering the general trend of minimum film thickness decreasing with an increasing load, for both low and high speeds conditions. The friction data is marginally more advantageous for pad A05. All experimental data clusters are placed below the equivalent theoretical plain pad analyses. This holds true for pad A05 also. The experimental minimum film thicknesses are generally lower than their theoretical counterparts as Figure 5.6 suggests.

Similar trends are exhibited by pad A07 as shown by Figure 5.7. Both the film thickness and friction values appear to suggest beneficial gains in experimental cases over equivalent theoretical plain pad analyses. Figure 5.7 presents a couple of anomalous data points, situated at a load of approximately 3 Newtons. This will be discussed later in Section 5.7 as the ‘cavitating case’. Pad A07r shows a lesser extent of improvement with regards to both minimum film thickness and friction data. As data for all convergences is presented in Figure 5.8, it is difficult to determine exactly what condition would be preferable for use with this specific pad.

Pads C07 and C07r (Figures 5.9 and 5.10 respectively) show higher friction data for experimental test cases than predicted by their equivalent plain pad analyses, suggesting a
disadvantageous effect of smaller, horizontally placed pockets compared to a larger, single pocket. The minimum film thickness results in Figure 5.10 also seem to suggest this. The minimum film thickness data for both the 10 and 20 N load conditions are more conservatively grouped together than previous pads. This may be due to lower range of convergences tested for pad C07r.

Friction data for pad C06 appears to be closely matched to its equivalent theoretical plain pad analysis as shown in Figure 5.11. The measured values of the minimum film thickness are greater than those predicted for the plain pad (for the same test conditions). The error is of a very similar degree to that given by the plain pad analysis as presented previously (Figure 5.4).

Pad C06r shows a slightly higher overall coefficient of friction. The overall spread in friction for the entire convergence ratio values, $K$, is considerably larger in Figure 5.12 than in Figure 5.11; In essence, the value of $K$ is more influential when discussing coefficients of friction. It is expected that the reduction of friction over the plain pad case will be very dependent on $K$. This is further analysed in discussion chapter seven. The range of absolute minimum film thicknesses are on par with pad C06.

Finally in this section, Figure 5.13 shows the difference in minimum film thickness between the experimental plain and experimental pad A05. Pad A05 exhibits a marginal advantage in a thicker film for a given load when compared to the plain test pad case.
Figure 5.4 Minimum film thickness and Friction vs. Load for Plain Pad
Figure 5.5 Minimum film thickness and Friction vs. Load for A05
Figure 5.6 Minimum film thickness and Friction vs. Load for A05r
Figure 5.7 Minimum film thickness and Friction vs. Load for A07
Figure 5.8 Minimum film thickness and Friction vs. Load for A07r
Figure 5.9 Minimum film thickness and Friction vs. Load for C07
Figure 5.10 Minimum film thickness and Friction vs. Load for C07r
Figure 5.11 Minimum film thickness and Friction vs. Load for CO6
Figure 5.12 Minimum film thickness and Friction vs. Load for C06r
Figure 5.13 Minimum film thickness vs. Load comparison between experimental plain and experimental A05 pads (0.5 m/s speed above and 1.0 m/s below)
5.4 Friction variation between experimental pads and equivalent theoretical plain pad for all convergences

This section presents the percentage difference in friction between the experimental test pad and its equivalent plain pad (derived theoretically). The x-axis shows the number of days (repeats) on which a specific pad was tested, and consequently these graphs are more of a ‘repeatability’ indicator more than anything. Positive values (i.e. readings above the x-axis) represent higher and thus less beneficial, friction conditions. Negative values (i.e. readings below the x-axis) indicate a frictional benefit for that specific case.

![Graph showing percentage difference in friction from equivalent theoretical plain pad for all experimental plain pad test cases for the full range of convergence ratios](image)

*Figure 5.14 Percentage difference in friction from equivalent theoretical plain pad for all experimental plain pad test cases for the full range of convergence ratios*
Figure 5.15 Percentage difference in friction from equivalent theoretical plain pad for all experimental C07 test cases for the full range of convergence ratios.

Figure 5.16 Percentage difference in friction from equivalent theoretical plain pad for all experimental C07r test cases for the full range of convergence ratios.
Figure 5.17 Percentage difference in friction from equivalent theoretical plain pad for all experimental C06 test cases for the full range of convergence ratios

Figure 5.18 Percentage difference in friction from equivalent theoretical plain pad for all experimental C06r test cases for the full range of convergence ratios
Figure 5.19 Percentage difference in friction from equivalent theoretical plain pad for all experimental A07 test cases for the full range of convergence ratios.

Figure 5.20 Percentage difference in friction from equivalent theoretical plain pad for all experimental A07r test cases for the full range of convergence ratios.
Figure 5.21 Percentage difference in friction from equivalent theoretical plain pad for all experimental A05 test cases for the full range of convergence ratios.

Figure 5.22 Percentage difference in friction from equivalent theoretical plain pad for all experimental A05r test cases for the full range of convergence ratios.
**Figure 5.23** Percentage difference in friction from equivalent theoretical plain pad for all experimental test pad configurations and convergence ratios

Figures 5.14 to 5.22 present the percentage difference in friction when compared to its equivalent plain pad analyses. Every data point is calculated by taking convergence, load and speed into account. The plain pad repeats show good repeatability with regards to the overall average of percentage difference in friction, as presented in Figure 5.14. This repeatability holds true for most of the test pads barring one; experimental test pad C06r. Figure 5.18 clearly shows a shift in overall average of the friction readings over the course of three separate days of testing. The average friction readings progressively increase with no change in testing procedure.
Figure 5.23 is of all the test pad cases for percentage change in friction when compared to equivalent theoretical plain pad cases. It shows that the average change in friction for the experimental plain pad test cases is a near zero percent change, when compared to its theoretical plain pad equivalent. Pad C07r has an overall increase in average friction of 3 percent, whilst pad A05 has an overall reduction in friction of approximately 6 percent.

5.5 Load Support versus Convergence for all pads

The following graphs present the dimensionless load support $W^*$ versus the convergence ratio, $K$, for all the test pads.
Figure 5.24 Load support vs. convergence for Plain and A05 pads
Figure 5.25 Load support vs. convergence for Plain and A05r pads
Figure 5.26 Load support vs. convergence for Plain and A07 pads
Figure 5.27 Load support vs. convergence for Plain and A07r pads
Figure 5.28 Load support vs. convergence for Plain and C07 pads
Figure 5.29 Load support vs. convergence for Plain and C07r pads
Figure 5.30 Load support vs. convergence for Plain and C06 pads
Figure 5.31 Load support vs. convergence for Plain and C06r pads
Figure 5.24 presents the load support versus convergence data for experimental pad A05 compared to the experimental plain pad. Also on these graphs, the plain pad theoretical analyses are shown. There is obvious improvement in load support with pad A05 when comparing with the plain pad data. This effect is more apparent in the lower convergences, more specifically a convergence of 1.2 which is often deemed as the optimum for hydrodynamic bearings. Again, there is an absolute difference when comparing the experimental plain pad data to its theoretical equivalent as shown by the solid pink line and blue diamond data points.

Similar results exist for pads A05r, A07 and A07r as given by Figures 5.25, 5.26 and 5.27. Again, most benefit in load support is witnessed around the 1.5 convergence ratio area for these three test cases. The load support is fairly similar at all other convergence ratios.

From the load support data given by Pad C07 in Figure 5.28, at first glance one may be inclined to suggest that the textured pad load support is less beneficial that that offered by a plain pad, taking into account similar conditions. Reductions of approximately 30-40 percent in load support are experienced at the ‘useful’ convergence ratios. Similar load support is evident in Figure 5.29 where data for pad C07r is presented; suggesting that smaller pockets near the outlet of a hydrodynamic bearing may be of more benefit than those located near the inlet.
Although load support data presented in Figure 5.30 is fairly evenly matched for both experimental textured (pad C06) and plain pads, it would seem that more data points are required for a fairer conclusion to be drawn. This is apparent at the lower convergence ratios in general. An anomalous reading at a convergence ratio of approximately 7.1 exists for pad C06. The data for pad C06r (smaller longitudinal pockets situated at the rear of the pad) points to an ever so slightly less beneficial outcome in load support as depicted in Figure 5.31. Again, more data is needed for the experimental textured pad in the useful convergence region.

*Figure 5.32 Load support vs. convergence for plain, A05, A05r and C07 experimental pads*

Figure 5.32 presents four experimental case results; Plain pad, pads A05, A05r and C07. From the results obtained during testing, these four cases display the control, least/most
beneficial cases from all the results in the most desirable convergence ratio range. Even with scatter present in all the results, it is clear that pads A05 and A05r (large pocket situated near the front and rear) offer greater load support than a plain pad operating in similar conditions, whilst pad C07 is less beneficial (on average) than the plain pad, A05 and A05r test cases. Differences in load support of up to approximately 50 percent exist between the three textured test pad cases.

5.6 Minimum film thickness comparison

This section portrays the relationship between the experimental minimum film thicknesses, $h_o$, versus the equivalent theoretical plain pad analyses for the experimental pad configuration (i.e. convergence, load and speed).

![Comparison of various minimum film thicknesses between theoretical plain pad values and the experimental plain pad](image)

*Figure 5.33* Comparison of various minimum film thicknesses between theoretical plain pad values and the experimental plain pad
Figure 5.34 Comparison of minimum film thicknesses between theoretical plain pad values and experimental pad A05.

Figure 5.35 Comparison of minimum film thicknesses between theoretical plain pad values and experimental pad A05r.
Figure 5.36 Comparison of minimum film thicknesses between theoretical plain pad values and experimental pad A07.

Figure 5.37 Comparison of minimum film thicknesses between theoretical plain pad values and experimental pad A07.
Figure 5.38 Comparison of minimum film thickness values between theoretical plain pad values and experimental pad C07.

Figure 5.39 Comparison of minimum film thicknesses between theoretical plain pad values and experimental pad C07r.
Figure 5.40 Comparison of minimum film thicknesses between theoretical plain pad values and experimental pad C06

Figure 5.41 Comparison of minimum film thicknesses between theoretical plain pad values and experimental pad C06r
A direct comparison of the minimum film thicknesses have been presented in Figures 5.33 to 5.41. The actual minimum film thickness present during testing are plotted against their theoretical plain pad equivalents. Figure 5.33 would ideally replicate a perfect correlation if all the data points were to lie on the dashed green line. In practice, this is not the case. As mentioned previously, the experimental results for the plain pad exhibited an over-prediction in the most part when compared to theory. Note that depending upon which set of minimum film thickness results one uses, the degree of over-prediction can vary significantly. For all experimental data sets, the minimum film thickness results were taken from the $h_{oCt}$ readings. If one were to, however, use the data as given by the $h_{oR}$ readings, the correlation would be closer to ideal and consequently further from the perfect correlation if the $h_{oL}$ film thickness data readings were to be employed. This apparent phenomenon is exaggerated at higher film thickness (in the region of approximately 40 µm and higher) than at lower film thicknesses. This offset in film thickness data will be discussed in more detail in the following chapter.

The issue of which minimum film thickness data to use is continued in all the textured pad film thickness comparisons. Larger scatter exists in some of the data sets (Figures 5.34, 5.37, 5.38 and 5.40 showing data for pads A05, A07r, C07 and C06) than others, where more tightly grouped clusters transpire.
5.7 The Cavitating Case

Throughout the duration of all the aforementioned tests, no cavitation was seen in any of the test runs bar one. Section 5.3 introduced a special case of results referred to as the cavitating case, more specifically test pad A07 (small pocket near the inlet). A particular test run of disc speed of 1.5 metres per second was conducted. Whilst loading up to 5 N, a cavitating film was observed, the cavitation zone occurring over the first millimetre of the pocket whilst spreading across the entire width and depth of the pocket. Figure 5.42 shows the extent of the cavitated zone across the pocket. The cavitated oil film region is located entirely towards the front of the pocket, whilst the oil film meniscus situated within the pocketed area exhibits a slight angle. This is most likely due to the resultant velocity profile of the incoming oil flow (centripetal due to the nature of the spinning disc).
Figure 5.42 The cavitated zone within the pocketed area in test pad A07
5.8 Summary

This chapter has introduced all of the minimum film thickness, friction and load support data obtained from the hydrodynamic test rig. Comments have been made with regards to their performance and ranking, where necessary. Specific mention has been made to favourable test pads under certain conditions. In addition, the concept of a two-dimensional convergence phenomenon occurring on the test pad during testing has been noted. The significance and impact of these results will be discussed in the following chapter.
Chapter 6

Discussion
6.1 Introduction

The current work has introduced a novel and unique method to simultaneously measure friction, load support, minimum film thickness and cavitation within a hydrodynamic pad bearing.

This chapter reviews all the results presented in this current work. Comments are made with regards to their validity and significance in hydrodynamic bearing applications. All test case performances are analysed and arguments presented where unexpected findings arise, with both the possible consequences and implications of the findings reviewed. The experimental findings are analysed with respect to a limited theoretical study, and the relevant significance of such a task is reviewed. Some of the key points of interest that this chapter (and ultimately this current work) aims to address include the following;

- Do any of the textures significantly increase load capacity and/or reduce friction?
- If so, under what particular conditions do they occur and why? (i.e. the likely mechanism of increased load support and/or reduced friction)
- Can any particular texture be said to be preferable than a plain pad over a wide range of conditions, and why?

The overall experimental approach (taking into account both technique and methodology) is then discussed. Opinions and thoughts regarding their suitability are offered, and suggestions made where improvements may be required. The chapter is
finally concluded by a short summary. This entails the contribution of the current research and its role and position within the textured hydrodynamic bearing field. A parting comment regarding the originality of the current work is made.

6.2 Review of Results

This section discusses the experimental results achieved, as shown in Chapter 5. Load support graphs were produced using the experimental film thickness and coefficient of friction results. Their impact upon the use of textured surfaces in certain operating conditions is commented upon.

6.2.1 Film Thickness and Friction data

This section refers to the minimum film thickness and friction graphs presented in Chapter 5, Section 5.3. Generally, all the test cases exhibit similar trends to that predicted by the theory. The manner in which the plain pad film thickness and friction results subsequently decrease and increase with respect to load, is replicated well by experiment. However, if one considers absolute values of the minimum film thickness, an underlying difference is clearly present (discussed later in Section 6.4). This is true for all the ‘minimum film thickness versus load’ comparisons.

It is very difficult to judge when considering the friction results as most data points are overlapping across the entire load range. This is true for most test cases over the entire load range, barring a possible anomalous data point at 5 N for test pad C07 as shown in
Figure 5.9. The experimental friction reading is considerably (approximately 20 percent) lower than the theoretical plain pad equivalent.

The two anomalous data points representing test pad A07, as shown in Figure 5.7, are as result of the ‘cavitating case’ as introduced in Chapter 5, Section 5.7. The margin of difference within this cavitating case is very similar to the majority of the data points at the 5 N test load condition for all other test cases (where no cavitation occurred within the lubricating film). This applies for both the minimum film thickness and friction results.

In summary, well replicated trends are exhibited by the experimental cases, albeit with an absolute difference in minimum film thickness readings. As the load increases, the minimum film thickness reduces whilst the friction increases due to the uncontrollable change in the test pad convergence ratio (and coupled with the fact that the minimum film thickness cannot be set during a test run). This said, the friction results are more useful when considering the possible tribological gains typically associated with textured surfaces. The closely spaced friction result data points make it difficult to determine a ranking order for the entire test pad cases. Also, the minimum film thickness and friction graphs presented in section 5.3 do not give an indication as to the convergence ratio, $K$, for each individual data point and consequently, these graphs are not suitable to determine any such ranking order. Alongside this, each individual graph presents experimental data for a single test pad against its theoretical plain pad equivalent, as such
does not compare two experimental test pad cases. This is to be expected as each test pad case contains a great number of data points, and having two or more closely-related test pad case results on the same graph would be difficult to assess graphically. The only exception to this is Figure 5.13 which compares both the plain and A05 textured experimental pads, highlighting a very small difference between the two. This difference is more apparent at the low load test cases for both high and low disc speeds.

6.2.2 Percentage difference in Friction data

The majority of the data presented within the friction percentage difference graphs (Section 5.4, Chapter 5) shows good repeatability with regards to the percentage difference in friction from the plain pad theoretical data. Small differences in the vertical axis (percentage difference in friction between the experimental pad and its plain pad theoretical equivalent) may be explained by a large range of test pad convergence ratios. For example, this may be the case as shown by Figures 5.19 and 5.21, data for textured pads A07 and A05 respectively. However, Figure 5.18 exhibits a possible shift in percentage difference in friction from theoretical plain pad over the three consecutive testing days. This, again, may also be due to a large range of convergence ratios experienced during testing, or it may be due to noise/interference present within the friction results.

Figure 5.23 collates the percentage difference in friction results for all the test cases. The graph is presented in ranking order with regards to most effect upon the percentage
difference in friction over the plain pad analyses. Whilst the plain pad experimental results are pretty much on par with the theoretical analysis, all bar one of the test pads show an improvement in the average percentage difference in friction compared to the experimental plain pad case. Test pad C07r is the only case showing a disadvantage when referring to the average friction data. Typical friction percentage reductions are in the range of single figures, ranging from approximately -3 percent to +6 percent.

As with the previous film thickness and friction results, the results discussed in this section refers to the full range of convergence ratios. Further analysis would be required with regards to the performance of each textured bearing under specific operating conditions.

### 6.2.3 Load support versus Convergence ratio

As this study is attempting to establish the potential of textured surfaces to reduce friction, it is coupled with a search for increased load support (the non-dimensional representation being $W^*$). The load support graphs (introduced in Section 5.5, Chapter 5) show the relationship between the dimensionless load support and the convergence ratio. As the experimental plain pad data set is presented along with each textured test pad case, a cross-comparison is easily achieved.

The optimum convergence ratio for an infinitely long linear wedge is approximately 1.2, whilst within other prominent wedge shape bearings (i.e. step, tapered land and
exponential) it is close to $K = 1$. The experimental results show convergence ratio values ranging from 0.5 to 11. The plain pad experimental data points show good agreement with the plain pad theoretical analysis, albeit with an offset as the plain pad load support results presented in Figure 5.24 are clearly higher than what the theoretical plain pad analysis predicts but it is thought that the theory does not precisely model the experimental test rig conditions exactly (discussed later in Section 6.4).

If one focuses one’s attention on the lower convergence ratio tests (where $K \sim 0.5$ to 2), distinguishable differences exist between textured pad A05, A05r and C07. Pad A05 (large single pocket situated at the front of the pad) exhibits what can be considered the most beneficial or ‘improved’ load support in this region, as shown by Figure 5.24. Improved load support in this case means that a greater normal load is generated for a particular film thickness (or, a higher film thickness exists for a given load). It does not mean that the bearing will necessarily support a greater maximum load.

A similar outcome is evident for pad A05r (large single pocket situated at the rear of the pad) as depicted by Figure 5.25. In contrast to this in Figure 5.28, pad C07 which comprises several smaller pockets located near the front of the pad, clearly demonstrates its reduced load support capability across similar convergence ratios when compared against both the plain and other textured experimental test pads.
6.2.4 Minimum Film Thickness data

The minimum film thickness comparison graphs presented in Chapter 5 Section 5.6 can be thought of as a ‘scale’ to see the potential benefit in minimum film thicknesses when using textured pads. For any given theoretical $h_o$ value, an experimental film thickness value which is placed on the right hand side of the dashed green marker line would indicate a thicker minimum film thickness. Even taking into account the potential constant offset between theoretical and experimental data, film thickness results for pads A05 and A05r are marginally higher (a desired benefit), whilst pad C07 results are fractionally closer to the dashed line, indicating that negligible minimum film thickness improvements would result. As with the film thickness and friction graphs presented in Sections 5.3 and 5.4, these graphs are heavily clustered with closely related data points and as such, make it difficult to draw any real conclusions with the introduction of textured surfaces within these bearings.

6.2.5 Cavitation Implications

Interestingly, even with the apparent ‘benefit’ in minimum film thickness for the solitary data point obtained during cavitating case in pad A07 (as seen in Figure 5.7), the friction value obtained is slightly higher than the plain pad theoretical analysis, for the same given test conditions. Judgement is reserved for now as it is impossible to comment upon a scenario where only a single data point result exists. One may be inclined to suggest that the cavitating case leads to a higher friction reading than that of a plain pad as the maximum load capacity occurs at the onset of cavitation (as discussed by Olver et al [17])
and not during a cavitated test run. Coupled to this, any inaccuracies in the friction measurement (as the absolute friction value itself was relatively high, with a coefficient of friction, $\mu$, of approximately 0.13) may have led to any perceived differences.

However, the fact remains that cavitation occurred within the lubricating oil film and was consequently captured simultaneously alongside film thickness and friction data collation remains a novel aspect of the current work.

### 6.2.6 Summary of experimental results discussion

From the range of tests conducted, the most obvious choice of result to base any judgment would be the load support graphs as they are, visually, the easiest to configure; in which case pad A05 seems to possess the most beneficial texture whilst pad C07 would be the least effective. However, on the basis of ‘highest percentage difference in average friction’ from the plain pad theoretical analysis would be pad C07r.

### 6.3 Theoretical comparisons

The results presented thus far have only been compared to theoretical plain pad analyses. This section introduces one dimensional textured and plain pad analyses, as modelled by an infinitely long linear wedge pad bearing (these theoretical results have been produced by the computational collaborators working alongside this research). These are compared against the two dimensional plain pad analyses and select textured pad cases.
Figure 6.1 depicts the theoretical load support offered by a one dimensional plain pad, a one dimensional textured pad at two different values of $h_0$ (5 and 30 µm) and experimental plain pad and test pad A05 cases. These load supports are presented against convergence ratio as per the previous chapter;

![Graph](image)

**Figure 6.1** Load support vs. Convergence ratio comparison between one dimensional theoretical plain pad and experimental plain and A05 pads

Note that for a plain pad, $W^*$ is identical for all values of film thickness; for a textured pad $W^*$ depends on the ratio of pocket depth to film thickness (as indicated by computational collaborators) and so the value of $W^*$ depends on $h_o$. $W^*$ alone is not a suitable parameter for comparing different bearings and operating conditions. The
minimum film thickness was extremely difficult to adjust and control during testing as this is a function of the attitude (or convergence angle) of the pad. Additionally, all pocket depths within the textured pad samples were fixed at 21 micrometres. Thus, the ratio of \( h_o/h_p \) is only known post-testing.

The graph in Figure 6.1 is not ideally a valid theoretical/experimental comparison for one primary reason; the issue of side leakage cannot be addressed by the 1 dimensional model, and thus the additional load support generated by having an infinitely long bearing approximation leads to a false comparison. However, at this stage no 2-dimensional textured model results are available to the author.

As a final comment, the load support graphs do not take into account the friction, and ultimately the friction coefficient, associated with each test case. To combat this, a variable utilising both the friction coefficient \( \mu \), and minimum film thickness \( h_o \) across the breadth of the bearing \( b \), can be formulated, giving \( \mu b/h_o \); this is a function only of \( K \) for a non-textured bearing as defined below (from [1]);

\[
\frac{\mu B}{h_o} = K \left[ \frac{3K - 2(K + 2)\log_e(K + 1)}{6K - 3(K + 2)\log_e(K + 1)} \right]
\]
Figure 6.2 $\mu b/h_o$ vs. Convergence ratio comparison between both one & two dimensional plain pad theoretical and experimental plain pad

Figure 6.2 shows this term versus convergence ratio for both the one and two dimensional theoretical plain pad cases as well as the plain pad experimental test runs. Very good agreement exists between the two dimensional theoretical and experimental data. Note the behaviour of the one dimensional theoretical model at very low convergence ratios (0.2 and below) relative to the two dimensional model predictions. The one dimensional relationship has a sharp cut-off region at a convergence ratio of approximately 0.05 at which point the $\mu b/h_o$ curve decreases, whilst the two dimensional curve is more gradual over the entire range of convergence ratios. As expected for an infinitely long wedge plain pad bearing, the one dimensional case offers greater load support, and thus $\mu b/h_o$, for the same convergence ratios when compared to the two
dimensional case. Accounting for ‘side leakage’ of the oil film is the most probable cause for this.

A selection of four textured test pad case results for $\mu b/h_o$ that stand out with regards to their performance in both the load support and average friction results are plotted in Figure 6.3.

![Figure 6.3 $\mu b/h_o$ vs. Convergence ratio comparison for experimental pads A05, A05r, C07, C07r and the experimental plain pad](image)

**Figure 6.3** $\mu b/h_o$ vs. Convergence ratio comparison for experimental pads A05, A05r, C07, C07r and the experimental plain pad

It now becomes slightly more complex as to determining a ranking order between the experimental plain pad, pad C07 and C07r. However, pads A05 and A05r clearly
demonstrate their desirability compared to the other pads when an operating convergence ratio range of approximately 2 and under is called for.

6.3.1 Influence of textured surfaces

Regarding the mechanisms by which texture may affect lubrication in hydrodynamic bearings, it is proposed that large (and wide) pockets situated near inlet of bearing perform in a very similar manner to a Rayleigh step bearing. In this end, the increase in load support by using pad A05 is most likely due to the ‘entry-bridging’ effect upon lubricant entrainment (when pockets are situated near to the inlet of the bearing). The friction reduction associated with these types of textured bearings is postulated to arise from the shear stress reduction within the pocked region. However, one may be inclined to suggest that textured bearings with larger pocket densities would also act in a similar fashion. This was not the perceived case with the multiple pocketed pads used in this work. Concerning pocketed bearings with large pocketed areas near the rear of the bearing (as exhibited by pad A05r) where thinner films and higher pressure regions exist, shear rate reductions due to the presence of the large pocketed areas may be almost as effective as large pockets near the inlet. This was shown by the load support graphs (Section 5.5 Chapter 5) which highlighted similar load support trends for pad A05r when compared against pad A05. It is unclear at this stage as to why the presence of multiple pockets placed perpendicular to the flow of oil (as commonly present in a typical piston/liner interface with a honed surface finish) may have a detrimental effect upon both load support (as exhibited by pad C07) and friction (as exhibited by pad C07r). As
analyses have shown that inlet suction may be more pronounced at low convergence ratios ($K<0.0001$), it is near impossible to comment (based upon experimental findings) as to whether inlet suction may play an underlying role in reducing the coefficient of friction and increasing the load support for a given film thickness in hydrodynamic bearings.

6.3.2 Summary

Based upon the test conditions observed, it may be said that the texturing geometries within pads A05 and A05r would be the wiser choice when compared to an experimental plain pad. These textures specifically exhibit favourable load support and film thickness values over a wide range of convergence ratios, as well as a lower overall percentage reduction in friction compared to a plain experimental pad. Pads C07 and C07r may not be advisable to use in applications where similar test conditions are experienced as they tend to deliver poorer performance both in terms of load support (pad C07) and friction reduction (pad C07r).

6.4 Discussion of Experimental Technique and Future Proposals

This section deals with the validity of the techniques chosen for this current work, opportunities for improvement and proposed further work. These include such areas as the experimental methodology and calibration techniques applied.
6.4.1 Friction calibration

One of the biggest points of interest that can be considered when referring to the friction calibration is that of the possibility of deformation of the steel shim used within the loading arm system. It may be that a transition of the elastic-plastic deformation boundary occurs during a test, especially at higher friction readings (typically encountered at a high load/low convergence test condition), but more likely is a slight hysteresis effect occurring within the shim. This is extremely difficult to detect during a test run.

A possible solution would be to employ the use of a thicker shim. However, this in itself may have implications with regards to the resolution of the load arm to small changes in friction as effectively the sensitivity of the friction measurement system is ultimately reduced.

6.4.2 Load calibration

The current loading calibration process consists of loading the load arm with up to approximately 40 N prior to testing. As actual test conditions only call for the load arm to be loaded up to a maximum load of 20 N, this technique may be considered to be liable for change; the two steel bridging pieces used to couple the load arm to the test pad holder (i.e. the gimbal) may undergo slight plastic deformation during testing but more likely is the presence of hysteresis, an issue extremely difficult to detect throughout the duration of a single test run. This possible plastic deformation will most likely induce
errors in the strain gauge readings indicating the applied load, and consequently lead to load errors.

Again, the use of a thicker section of steel in place of the current pieces may reduce the chances (and/or the severity) of this error, but at a cost of resolution. Reduced loading during the calibration process would be advisable.

6.4.3 Film thickness calibration

The wedge technique used to calibrate for film thickness has been one of the major reworks of the current work. This has led to a drastic improvement over the previous obsolete interferometry method with regards to accuracy and repeatability of film thickness calibration. It is not without its potential shortcomings, however. Assumptions include the accurate positioning and step height measurement of the cover slip attached at one end of the flat silicon calibration piece. The tolerance of the x-y micrometer stage to which the camera itself is attached to is effectively 5 µm in both axes. This may have a small bearing upon the positioning of the camera over a specific film thickness region. A most viable solution to ensure an accurate calibration would be to introduce the possibility of using a known ‘stepped’ calibration piece in place of a wedge.

6.4.4 Image analysis

The number of images taken over a single location was set to 20 as previously mentioned. This could be considered as inadequate; particularly if there were pronounced misalignment issues still inherent within the spinning disc mechanism such as the run-out
experienced due to the limited manufacturing process (read accuracy) when producing the main shaft of the rig. This may ultimately have the potential to cause a difference in film thickness readings over the sequence of 20 images as ‘squeeze’ phenomena may occur within the film (i.e. those thick films with a low stiffness). No reported issues caused by vibrations in the camera system were experienced. All z-stage mounts were adequately dampened between rigid couplings, and efforts were made to remove any components with moving parts (i.e. fans, motors etc.) away from the bench upon which the camera system was mounted.

Images were taken at four separate locations over the pad area, but only across three corners of the test pad. These were concentrated near the edges of the pad, as opposed to the centre of the pad. This was to ensure that any film thickness measurements did not include the additional pocket induced film thickness depth. Film thickness imaging may have been possible at the left hand side of the pad inlet for a few test cases, but frequent image saturation occurred owing to the uncontrollable tilting of the pad perpendicular to the direction of travel. It was decided for this reason to only image the oil film at three corners of the test pad, for all test pads and all test conditions. Increasing the film thickness during film thickness calibration process would cater for thicker film measurements, and consequently more frequent imaging at all four corners of the test pad should be introduced.
An important point to note is the theoretical derivation of the final $h_o$ value from two experimental film thickness values (i.e. the average of $h_{oL}$ and $h_{oR}$). Although strictly speaking, the value of $h_o$ is obtained by averaging two experimental values, it was deemed accurate in the context of the current work. Related to this, it was shown that relatively large differences exist between some values of paired $h_{oL}$ and $h_{oR}$ values due to the presence of the aforementioned tilt. For the sake of continuity, the averaged value of film thickness was used throughout the entire results analysis. One may be inclined to suggest that either appropriate modifications to the theoretical analyses are implemented (as the theoretical analysis does not take into account this tilt that exists experimentally) or that images are taken directly in the centre line of the test pad in future tests.

A 3x3 pixel matrix, located directly in the centre of the image, was used to give an average intensity reading from any given image across the 9 pixels. This value was then used to determine a film thickness value. It may be argued that a larger matrix would be called for; within reason as image aberration was thought to occur within the image due to the nature of the curved optics located within the fluorescent cube microscope system. A thorough review of the optics within the fluorescence imaging system is called for.

### 6.4.5 Oil entrainment velocity

It is feasible to presume that a velocity gradient may exist in the oil entrainment speed between the spinning glass disc and the stationary test pad underneath. This is most likely the reason why the tilt phenomenon exists on the test pad during a test run, as introduced
in Section 6.4.4. The radius between the centre line of the test pad and the centre of rotation for the glass disc is 40 mm. As the pad is 10 mm wide, it is presumed that theoretically, a difference in oil entrainment speed across the width of the pad in the region of 25% exists between the outer (45 mm) and inner (35 mm) radii. Again, this is not currently modelled in the computational analyses. This may address the issue of the slight differences in absolute values that currently exist between the experimental and theoretical analyses. A possible solution to alleviate this issue may be to eradicate any tilt in the perpendicular direction to oil flow by redesigning the pad gimbal entirely.

6.4.6 Oil temperature and viscosity

A presumption of the oil temperature measurements, and consequently the oil viscosity values associated with those readings, is that the oil temperature is uniform both within the oil nozzle and over the entire area of the test pad. This may be misleading to a degree as these experimentally derived oil temperatures are used within Reynolds equation calculations to ultimately produce both the theoretical minimum film thickness, $h_o$ and non-dimensional load support, $W^*$. Small temperature differentials present within the oil result in viscosity gradients, as oil viscosity varies at an inverse exponential rate with temperature. At present, this situation cannot be improved upon the experimental rig, however, such techniques to improve the oil temperature monitoring over a wider range of locations would be considered paramount. The possibility of Infra-Red mapping (using a thermal imaging camera instead
of a high sensitivity camera for instance) of the hydrodynamic oil film may be useful in providing a more suited approach when considering oil temperatures within the contact zone, and coupled to this would be the introduction of a greater range of oil viscosities to the test regime in light of the occurrence of cavitation and to mimic temperature effects.

6.4.7 Theoretical Analyses Correction

As stated, there is the utmost need to ensure that the theoretical analyses correctly model the experimental test rig. The issue of the tilting pad and differences in oil entrainment velocity need to be addressed swiftly before valid comparisons can be made between the two sets of results.

6.4.8 Summary

This sub-section has dealt with possible issues arising from the current experimental technique employed and how they may be dealt with when considering future work. The two most crucial areas for improvement lie in the accuracy of the current film thickness technique (both oil film calibration and imaging) and the temperature mapping of the oil film. Possible correctional techniques have been put forward where necessary. The lack of many cavitating cases during test runs may indicate the need to further explore the pad geometry, the design of the test ad gimbal and the need to test various grades (viscosities) of oils used within the tests.
6.5 Summary of discussions

This chapter has reviewed the hydrodynamic test rig results thoroughly. Where observed advantageous trends and patterns were noticed, possible explanations were provided. The same applied to where discrepancies and detrimental findings were reported. Preferred texture choices under certain working conditions were recommended. A number of issues arising from the current research were listed, and recommendations to combat these were provided where necessary.
Chapter 7

Conclusions
7.1 Summary of thesis

This thesis has presented experimental research concerning the reported effects of increased load support and reduction of coefficients of friction of textured bearings within hydrodynamic applications. This has been achieved by the means of designing and constructing a novel test rig capable of measuring film thickness, friction and load all simultaneously. The use of a high sensitivity optical fluorescence imaging system to allow for the capture of both the film thickness measurements and cavitation in-situ alongside a high resolution friction detection system has been implemented.

To begin with, the relevant literature was reviewed and recommendations made regarding the apparent gaps in current knowledge relating to textured bearings in hydrodynamic applications. This covered such areas as; the underlying theoretical explanations offered as to why these textured surfaces generate extra load support and reduction in friction, the current state of play in film thickness measurements and the precise role (if any) of cavitation occurring within the lubricant fluid film during the perceived performance improvements.

Secondly, the development of the experimental techniques to be used to determine the above was presented. Selection criteria for a number of the employed methods were justified. Accurately produced texture geometries were processed upon optically flat silicon wafers to replicate textured bearings, and tests were carried out on both plain and
textured pad bearings. From the film thickness results given by the test rig, a novel technique to determine the convergence ratio of the convergent pad bearing was introduced. Friction measurements were also monitored. These allowed for the calculation of load support and coefficients of friction for any given test condition.

Finally, comparisons between the test results for all experimental pads against a limited theoretical study were made; enabling both the performance of the experimental test pads and the repeatability of the test rig to be quantitatively evaluated.

### 7.2 Conclusions

A novel experimental test rig capable of measuring the friction, load and film thickness present between a spinning glass disc and a loaded flat pad bearing whilst simultaneously being able to monitor cavitation has been achieved.

It has the capability of being able to detect small changes in friction that occur as a result of testing various textured bearings as well as thick hydrodynamic lubricant films. It can be seen quite clearly from the evidence presented that the resultant effects of using textured bearings can be distinctly observed when comparing data obtained from plain pad experiments (under equivalent test conditions). Even taking into account scatter of data within the results, marked differences in load support and reduction of coefficient of friction were easily observed for those textures consisting of a single large pocket either
near the inlet or outlet of the pad bearing. In some cases, detrimental effects of testing multiple pocketed pads were measured and reported. At present, more comparisons between the experimental data (produced as a result of this current work) and a complete set of theoretical analyses (which are indicative of the test rig) are needed to elucidate the experimental trends observed. This is with the aim of further increasing the current limited knowledge regarding the underlying principles of how the introduction of textured surfaces affect friction and load support in hydrodynamic bearings.

The opinion that this research has been a most worthwhile contribution to the current state of knowledge regarding the use of microtextured bearings, especially relating to pad geometries containing a single large pocket, is put forward.
References


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