Dynamic Measurements of Lubrication Film Thickness of UHMWPE Contacts for Total Joint Replacements

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DYNAMIC MEASUREMENTS OF LUBRICATION FILM THICKNESS OF UHMWPE CONTACTS FOR TOTAL JOINT REPLACEMENTS

A Dissertation
Presented to
the Graduate School of
Clemson University

In Partial Fulfillment
of the Requirements for the Degree
Doctor of Philosophy
Bioengineering

by
Andrew Chapman Clark
May 2007

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Thomas Pace, M.D.
ABSTRACT

A new contact sensing technology previously developed in the Biotribology Laboratory at Clemson University was further studied, evaluated, and characterized to extend its use to the measurement of lubricating film thickness. First, the laboratory’s force-controlled knee joint simulator was used while dynamic contact pressure measurements under both dry and lubricated conditions were made using the sensor technology employed in two different artificial knee implant geometries. Each implant was machined by the manufacturer from custom blocks of ultra high molecular weight polyethylene (UHMWPE) containing a grid of discrete sensing regions. The difference between the dry and lubricated contact areas measured at different phases of the gait cycle for each implant suggested that the dynamic lubrication thickness might also be able to be quantified by the sensing technology. To gain insight on this, simplified contacts of metal on UHMWPE were studied with the sensing technology being employed in the UHMWPE side of the contacts. First, the UHMWPE sensor’s outputs were studied under static, lubricated conditions while the surface separation was directly controlled. The insights gained during the static testing were used to develop a more representative contact that was then characterized under hydrodynamic conditions. The experimental contact model was designed to mimic a single sensing point of the knee sensors used earlier in this study. It
consisted of a UHMWPE sensor pin with a spherical tip sliding on a flat stainless steel counterpart with an implant-grade finish. Hydrodynamic motions were applied to the contact with the laboratory’s custom-designed multi-axis pin on disk wear testing machine with friction measuring capabilities. To relate the sensor pin’s output to the mode of lubrication, a Strubeck curve was experimentally developed and was used to determine the lambda (\( \lambda \)) values specific to the UHMWPE on metal sliding point contact. It was found that the boundary lubrication regime existed for \( \lambda < 1 \), mixed lubrication was present for \( 1 < \lambda < 3.5 \), and fluid film lubrication existed for \( \lambda > 3.5 \). Calibration equations relating the sensor’s output to the film thickness were obtained using simple linear reciprocating motion, and it was found that in the boundary lubricated regime, the sensor’s output was linearly related to the film thickness. It was also determined that for mixed and fluid film lubrication, the sensor’s output was linear on a log-log scale to the film thickness; thus, there was a power-law relationship. Finally, the calibration equations were used to measure the lubricating film thickness of the UHMWPE contact in a clinically relevant, cross-path motion complete with sliding speeds relevant to the phases of gait where lubricating films can potentially exist for artificial knee joints. Two different loads were applied to the contact for these measurements. For the lightest load, mixed lubrication and HL were measured, and the film thickness varied from 2\( \mu \text{m} \) to over 10\( \mu \text{m} \). With the higher load, the film thickness was seen to fall to 1\( \mu \text{m} \) for a small portion of the cycle,
showing that the contact experienced the full range of lubrication modes from boundary to full hydrodynamic lubrication.
ACKNOWLEDGMENTS

I would like to thank my advisor, Dr. Martine LaBerge, for giving me the opportunity to conduct this research and for her patience and guidance throughout. I would also like to thank my committee members for their time and for their input towards the completion of this work. Finally, I would like to thank Dr. John DesJardins for his input, guidance, and assistance during my time in the biotribology lab at Clemson.
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1.0 - INTRODUCTION

Osteoarthritis accounts for more than $65 billion in economic losses and more than one million surgeries each year in the United States [AAOS, 2001]. The leading type of surgery due to arthritis is total joint arthroplasty (TJA), with 326,000 knees and 165,000 hips being replaced in the United States during 2001 [Hall et al., 2003]. Approximately one-third of these surgeries were performed on patients under the age of 65, meaning that the average life expectancy of the patient was greater than the 10-year average clinical life of the implants [Hall et al., 2003]. By year 2030, the number of hip and knee replacement surgeries is expected to exceed 700,000 [NIH, 2002].

More than four percent of total knee arthroplasties (TKA) fail due to wear of the polymer component, ultrahigh molecular weight polyethylene (UHMWPE) [AAOS, 1996, Hall et al., 2003]. Artificial knee joint failure causes pain and suffering for the patient, as well as increased risks associated with a second surgery to replace the failed joint. Because a replacement surgery is at least 33 percent more costly than the initial surgery, total joint replacement (TJR) failures account for hundreds of millions of dollars in economic losses each year [AAOS, 1996].

The main reason for the 10 year average life span of an artificial knee joint is wear debris from the UHMWPE tibial insert. As the femoral component articulates on the softer tibial component, sub-micron wear particles are generated [AAOS, 1996]. The body’s immune system can adversely react to
these wear particles resulting in bone loss surrounding the implant. This condition is known as osteolysis, and it severely weakens the bone, which can lead to loosening of the implant and ultimately failure [Collier et al., 1990, Harris et al., 1999]. One way to increase the life expectancy of artificial joints would be to decrease the amount of UHMWPE wear debris produced. In order to reduce the wear generated in TJR prostheses, all of the contributing factors, such as component geometry, loading conditions, the resulting contact stresses, and the lubrication conditions must first be understood [Harris et al., 1999, Matsuda et al., 1999].

A new technology has recently been developed that takes advantage of one of the unique electrical properties of a conductive composite of UHMWPE to directly measure the interfacial contact force between it and a metallic component. The electrical properties of the polymer bearing are modified without affecting its mechanical properties. The modified electrical properties give the polymer a variable contact resistance that enables it to conduct electricity through it at a rate proportional to the contact force applied to its surface by another conductive material [Clark et al., 2006, Clark and LaBerge, US Patent Application 2006/0184067, Clark, MS Thesis, 2003]. By manipulating the geometry of discrete sections of the modified sensing polymer within a bulk sample of the unmodified polymer, the distribution of contact pressure over the surface of a bearing can be obtained. The results of the previous work, in conjunction with the preliminary studies conducted to date, indicate that the
technology has the potential to quantify the mode of lubrication and the surface separation of the UHMWPE artificial joint.

It will be shown in this research that this sensor technology can be used under dynamic conditions to quantify the lubricating film thickness and distinguish the mode of lubrication under which it is operating. This research is presented in three main thrusts of experimentation: thrust 1) investigation of the effects of lubrication on dynamic readings obtained from instrumented prostheses loaded in a knee joint simulator, thrust 2) investigation of the sensor's output under hydrostatic conditions of controlled surface separation, and thrust 3) characterization of the sensor's output under controlled hydrodynamic conditions to obtain calibration equations, and using these equations for measurement of lubricant film thickness during clinically relevant, complex motions.

**Background and Significance - Sensor Material**

There has long been a need to experimentally measure the dynamic contact conditions of important engineering tribological systems, especially those with polymeric bearing surfaces that prove difficult to model. In order to experimentally quantify the dynamic contact conditions of geometrically complex polymeric bearing surfaces, a composite sensor material of UHMWPE and carbon black (CB) was developed [Clark et al. 2006]. Carbon black is an economical, conductive filler that can be added to UHMWPE to create a
conductive composite material [Clark et al. 2006, Chan et al. 1997, Bin et al. 1999, Xu et al. 1998, Breuer et al. 1999]. Additionally, a composite that is partially conductive under no strain will become more conductive if it is placed under compressive strain [Sherman et al. 1986, Tang et al. 1996]. It has been shown that the main method of conduction in a CB-filled polymer is electron tunneling from one conductive particle to the next, known as “percolation”. Percolation requires inter-conductive particle distances to be 10 nm or less within a polymer. As conductive particles are added to a composite such that the inter-particle distance approaches 10 nm, a small increase in the amount of conductive filler will cause a sharp increase in composite conductivity, as the inter-particle distance becomes less than 10 nm. This point is known as the “percolation threshold” [Sherman et al. 1986].

The terminology used to describe CB is also important to distinguish. A primary particle of CB is the “particle size” as listed by the manufacturer and is a generally spherical nano-sized particle. However, the smallest “base unit” of CB that can be obtained in a dispersion is called an aggregate – as shown in Figure 1.1. Aggregates are generally considered indivisible and are made up of many CB primary particles. The typical size of a CB aggregate is usually 50 to 500 nm in size and varies according to the manufacturer of the CB. While aggregates are the smallest unit of CB that can be obtained in a CB-filled polymer, CB is most often found in agglomerates. Agglomerates are dense configurations of many aggregates held together by van der Waals forces [Accorsi, 1999]. The best possible dispersions of CB are those consisting of small CB aggregates with
no large agglomerates. Therefore, the method of mixing the CB with the polymer must apply enough energy to overcome the van der Waals forces that hold together the CB agglomerates.

Figure 1.1 - Carbon black terminology [Accorsi, 1999]

Because UHMWPE has such a high melt viscosity, it is difficult to disperse CB using traditional methods. However, there are three main ways that CB can be dispersed in UHMWPE, including the solvent solution method [Bin et al., 1999, Xu et al., 1998], the sintering method [Chan et al., 1997], and the multi-polymer blend method [Breuer et al., 1999, Feng et al., 2000]. The polymer-blend method is less desirable for engineering applications because superior mechanical and wear properties of UHMWPE are reduced due to the presence of another polymer [Bin et al., 1999]. Considerable research has been focused on using solvent solution methods to dissolve the UHMWPE followed by mixing in
CB [Bin et al., 1999, Xu et al., 1998]. However, the solvent solution method requires large percentages of CB to obtain conductivity, which could lead to inferior mechanical properties of the resulting composite [Breuer et al., 2000].

The sintering method is a form of compression molding, a method used to convert UHMWPE powder into a solid form. In comparison to other ways of processing a CB/UHMWPE composite, sintered UHMWPE composites require less conductive filler because they form a segregated network. Segregated network conductive composites have an internal morphology defined by small volumes of polymer that contain the conductive filler surrounded by larger volumes of polymer that contain no conductive filler [Chan et al., 1997, Bouchet et al., 2000].

In previous work conducted by the author, the UHMWPE/CB sensor material being used in this research was shown to have a segregated network structure where nano-sized CB primary particles were well dispersed within the structure, as seen in Figure 1.2.
To obtain the composite sensor material, the UHMWPE and CB were mixed together in powder form. Close examination of the mixed powder with field emission scanning electron microscopy (FESEM) showed how the excellent dispersion of CB was able to be achieved. When the powders were combined, the much smaller CB particles were able to coat the surface of the UHMWPE particles. Seen in Figure 1.3, the CB particles became enmeshed within the spheroid and fibril network of the UHMWPE particles. Upon compression molding, the CB coated UHMWPE particles fused together, locking the CB within the 3-D structure of the composite.
The results of the previous work also showed that the composite exhibited a force-dependant conductance at different weight percentages of CB from 0.5wt% to 8wt%. The same composites showed no change in mechanical properties from virgin UHMWPE [Clark et al. 2006, Clark AC, MS Thesis, 2003]. Readers are encouraged to see the references for a more detailed and thorough discussion on the bulk morphology and properties of the sensor material. Additionally, the mechanical properties and design principle, as well as
calibration and validation studies of the style of sensor used during aim 1 of this research are also presented in previous work [Clark AC, MS Thesis, 2003].

Experimental methods of measuring lubricant film thickness

Optical interferometry is the most widely used method for studying fluid film thickness. In order for optical interferometry (Figure 1.4) to be used to observe fluid film thickness, one of the contacting surfaces must be optically clear. Typically this will be a flat sheet of glass or sapphire. The other contacting surface must be reflective, such as polished steel. For this reason, optical interferometry is normally only useful in model contacts.

In optical interferometry, the glass sheet is coated with a semi-reflective layer so that when light is passed through it, some of the light will reflect back to the source, and some of the light will continue through the lubricating film and reflect off the steel surface. Thus, light has been reflected off 2 surfaces that are separated by the thickness of the oil film, and the two reflected light rays will interfere with each other, sometimes constructively, and sometimes destructively, depending on how thick the oil film is. A video camera is used to capture the reflected light, and the fringe patterns of light interference seen in the video indicate the film thickness.
Figure 1.4 - Diagram of optical interferometry using a) the standard method and b) the spacer layer method [Johnston et al. 1991]

It is most commonly used in fairly common tribology test rigs such as ball-on-disc testers. In general, the actual thickness of the film is a little bit difficult to determine using optical interferometry, since the film thickness at some point usually must be known. However, the interference patterns easily reveal the relative difference in film thickness as well as show the distribution of that thickness over the entire contact. The contributions made during the 60’s and 70’s using optical interferometry to measure EHL point contact were so good that experiment was ahead of theory until powerful enough computers came along to solve the problems numerically [Dowson 1995].
Figure 1.5 shows a good example of a classic point contact under EHL as shown by optical interferometry. The typical features visible are the side lobes, the near constant film thickness in the center regions, the constriction at the exit, and the cavitated wake [Dowson 1995]. Bassani and coworkers, in 1997, used image analysis software to automatically interpret the results and calculate the actual film thickness [Bassani et al. 1997] (Figure 1.6). They also took pictures of static fringes such that the system could use a calibration curve to calculate the actual film thicknesses that are presented in the 3-D graphs.
There have been many advances in this method since it was first developed in the late 1960’s, and the method can now be used to measure films down to 1 nm, making it a useful method, even for boundary lubrication measurements [Anghel et al. 1997]. The disadvantage of this method is that it can only be employed in model contacts, since an optically clear surface is required.

However, not all of the model contacts using optical interferometry have consisted of metal on glass. Particularly in the area of cushion bearing research, optical interferometry has been employed to measure the film thickness. Using
two different elastomers contacting glass, McClure, Jin, Fisher, and colleagues used optical interferometry to study the lubrication of both entraining and squeeze film motions with water and 40\% glycerol solutions [McClure et al. 1996].

In addition to optical interferometry, other optical methods have been employed to measure lubrication film thicknesses. Johnson and colleagues used a laser displacement transducer to measure the isoviscous elastohydrodynamic film thicknesses in an elastomer – glass contact and high speeds with a high viscosity silicone oil lubricant [Johnson et al. 1997]. With this method they measured thick films from $1\mu m$ to $100\mu m$ with the laser displacement transducer with a vertical accuracy of 1nm and a special accuracy of a $1.5\mu m$ diameter spot size. Unlike optical interferometry, the laser displacement transducer only provides one thickness value instead of a picture of the distribution and shape of the film. In order to use optical displacement transducers, one of the surfaces must be optically clear, so many of the same disadvantages apply to this method as well.

The capacitance method for measuring lubricating film thickness, especially for line EHL contacts, was made popular in the late 50’s and early 60’s [Dowson 1995]. Many of the experimental measurements made were models of metal gear contacts and they confirmed the theory for that type of contact [Dowson 1995]. The capacitance method, unlike the optical methods, does not require an optically clear material. As long as both surfaces are conductive (as in metallic contacts of most studies), the capacitance method can be used. Although the
method can provide numeric results for film thickness, there is no information about the shape of the film and the type of lubricants being used must be carefully selected to avoid problems with the precision required for the measurements [Dowson 1995]. The capacitance method has continued to provide film thickness measurements, and is the best electrical method for measuring lubricant films of greater thickness. Lucca and Wright used the method to measure hydrodynamically lubricated slider bearing film thicknesses of 10\(\mu\)m to 20\(\mu\)m [Lucca et al. 1991]. Hahm and coworkers also used the capacitance technique to examine the solid lubricant film coating magnetic disk drives [Hahm et al. 1998]. While this is not a fluid lubricant, the principles are just the same for the capacitance method. However, because the film they were measuring was a solid coating, they used ellipsometry to calibrate their capacitance measurements, which would not be possible in fluid lubrication experiments.

The other main electrical technique used to measure lubricant film thickness is the resistive technique. While this technique is also sometimes referred to as the voltage method or the voltage drop technique, they all use the same principle that a lubricating film separating two metal surfaces will have a high electrical resistance. Conversely when the lubricant film breaks down and the metallic surfaces touch, the resistance between them will be low. The voltage techniques were the first methods attempted to measure lubricant films in the 1950’s, but most of the techniques proved to be unsuitable for quantitative film thickness measurements [Dowson 1995].
This technique is useful for measuring lubrication films in actual conditions, such as contact between the piston rings and cylinder wall in internal combustion engines. An example of the voltage drop method used to gather film information in an engine is shown in Figure 1.7. When the voltage across the contact was high, the lubricant film was separating the two surfaces, and when the voltage dropped, the lubricant film was getting thinner [Taylor 1992]. It can be seen in the curve of Figure 1.7 that the data does not correspond to a quantifiable film thickness, but rather it gives some more information about the dynamics of the lubricant film and how it changes with the cam angle [Taylor 1992].

![Figure 1.7](image-url)

**Figure 1.7** - Comparison of theoretically calculated film thickness and film behavior measured by the voltage technique between the cam and follower in an internal combustion engine [Taylor 1992]
Kawamura and colleagues used the voltage drop method to study the lubrication of metal on metal contact in a common four ball testing apparatus [Kawamura et al. 1975]. They were observing the effects of lubricant additives on the lubricant film with a hydrocarbon oil base stock. As the circuit diagram below indicates, the voltage applied to the contact was $100\text{mV}_{\text{DC}}$, by means of a voltage divider circuit. The additional current-limiting resistor in series with the contact also makes the relationship between current flow and recorded voltage non-linear. The voltage vs time curve shows the qualitative view of the lubricant film behavior that is provided by this method.

![Figure 1.8 - Electrical circuit used for voltage drop method (also referred to as electrical resistance method) [Kawamura et al. 1975]](image)
Figure 1.9 - Voltage vs. Time curve - showing lubricant film behavior where higher voltage represents lubricant film separating the surfaces [Kawamura et al. 1975]

Figure 1.10 - Voltage vs. Time curves for different oils, loads, and speeds (black represents fluid film lubrication and while represents metallic contact) [Kawamura et al. 1975]
The grouping of voltage vs. time curves shown in Figure 1.10 is shaded such that the black areas are a high voltage, where fluid was insulating the contact, indicating fluid film separation of the surfaces. The white areas show low voltage, which means the surfaces were contacting, and thus no lubricant film completely separated the surfaces. The grouping of curves shows the lubricant film’s time response to different loads and speeds, as well as different oil viscosities [Kawamura et al. 1975]. For example, oil K with a 47.7 kg load showed complete contact for all speeds tested. Conversely, oil A at 6.8 kg load showed full fluid film lubrication for all speeds, with the metallic surfaces being completely separated. Some of the other oils showed films that varied with time during the test. As noted by Kawamura, the areas shown in white can represent full metallic contact, or could be boundary lubrication over just a small part of the contact patch, with the rest of the contact experiencing fluid separation. Kawamura also noted that in previous studies with similar oils and 0.1V\textsubscript{DC}, that the voltage would not drop until the oil film decreased to only several tens of Angstroms thick. In other words, the oils used in the study would not ionize until being several nanometers thin.

There has also been use of the electrical resistance technique of lubrication monitoring in investigating cushion bearings. In a study published by Ohtsuki, Murakami, and colleagues, a 316 stainless steel sphere was articulated against three different simple geometries of cushion bearing implant designs with a 3mm layer of conductive silicone as the compliant layer. Standard knee simulator flexion angle and axial loads were applied to the three simplified joint geometries
while the lubrication of silicone oil was monitored using the electrical resistance method [Ohtsuki et al. 1997]. A sample of data from this study, that was also re-published as qualitative experimental data in support of a transient EHL model by Jin, Dowson, Fisher, and co-authors is seen in Figure 1.11. Degree of separation is defined as the ratio of measured voltage to applied voltage [Ohtsuki 1997]. Therefore, a degree of separation of 1 corresponds to complete separation of the two contacting surfaces by the lubricant film, and a degree of separation of 0 corresponds to contact between the two surfaces, with boundary lubrication or perhaps some mixed lubrication dominating the bearing. In Figure 1.11, the frictional torque curves were noted as agreeing with the degree of separation curves, with the lowest friction seen in the joint geometry with the highest degree of separation [Jin et al. 1998].
The biggest reason for the lack of success of the electrical resistivity methods for measuring film thickness has been due to the fact that it has always been employed between two highly conductive surfaces, such as metal-metal contacts or metal-conductive polymer contacts. Because both surfaces are so electrically conductive, as soon as any material contact is made between the two surfaces, the resistance across the contact basically falls to zero and does not get much lower even if the surfaces achieve a greater degree of intimate contact. For most of the voltage drop experiments that have been conducted, the experimental data will not differentiate between full boundary lubrication with a
very high degree of contact between two surfaces or two surfaces barely touching asperities just as mixed lubrication is transitioning into full fluid film lubrication. Again, the reason for this is that metals and highly conductive composites are both very highly conductive, and their resistance does not vary.

However, the material being used in this research is different. It has been previously shown that for partially conductive composites of CB and UHMWPE, the material’s conductivity is force-dependant [Clark et al. 2006]. For this reason it is hypothesized that it is possible to not only obtain qualitative results such as those discussed above, but it should also be possible to obtain quantitative measures of the lubricant film thickness.
2.0 – DYNAMIC CONTACT STRESS DISTRIBUTION IN A CONFORMING AND A STANDARD TIBIAL INSERT USING A NEW CONTACT SENSOR TECHNOLOGY

2.1 - Introduction

Wear particles from ultra high molecular weight polyethylene (UHMWPE) bearing surfaces of total joint replacement (TJR) prostheses have been shown to cause adverse biological reactions which can lead to loosening of the prosthesis, and ultimately to failure [Collier et al., 1990]. In order to reduce the wear generated in TJR prostheses, contributing factors, such as component geometry, loading conditions, and the resulting contact stresses, must first be understood [Collier et al., 1990, Collier et al., 1991, Sathasivam et al., 1994].

The literature suggests that there is a correlation between the contact stress distribution on the tibial component of the artificial knee joint and polyethylene wear [Rostoker et al., 1979, Wright et al., 1986, Collier et al., 1991]. The relative geometry between the femoral and the tibial component surfaces is known to affect the contact stresses [Bartel et al., 1986]. Less conforming, flatter femoral geometries have less contact area, leading to higher contact stresses. In contrast, fully conforming designs generally have lower contact stresses but restrict normal anteroposterior motion, resulting in high shear stresses being transmitted to the bone-implant interface during high flexion angles [Matsuda et al., 1999, Sathasivam et al., 1994]. This is thought to be the biggest cause of the higher degree of loosening reported for fully conforming designs [Ilstrup et al., 1976].
It is well established that a contact stress exceeding a polymer's yield stress will lead to abrasive wear [Postak et al. 1996, Matsuda et al. 1999, Sathasivam et al. 1994]. However, the amount of damage to the implant depends on how much of the surface area experiences this high contact stress. If the surfaces of two different implants experience contact stresses exceeding the yield stress, the implant with the least amount of surface area experiencing this condition would be expected to show less abrasive wear damage, even if this implant contained a higher peak contact stress value within that area. The surface area that experiences a contact stress exceeding the yield stress is known as an “overloaded area” [Postak et al. 1996], and it is often less than the total contact area. Therefore, the single measurements of contact stress and contact area alone are not sufficient to predict abrasive wear damage. The magnitude of the contact stress as it varies over the entire surface of the implant (the contact stress distribution) is the information that can be used to predict abrasive wear damage. It is difficult to quantify the contact stress distribution since it is a term containing two variables; therefore, the quantity “overloaded area” is typically used to qualitatively assess expected abrasive wear damage.

Many experimental methods have been used to measure both the contact stress and the contact area of TJR prostheses including Fuji pressure-sensitive film [Postak et al., 1996, Ateshian et al., 1994, Harris et al., 1999, Matsuda et al., 1999, Liau et al., 2001], stereophotogrammetry [Ateshian et al., 1994], dye injection methods [Black 1981], silicon casting methods [Fukubayashi et al., 1980], piezoelectric transducers [Manoul et al., 1992], micro-indentation
transducers [Ahmed et al., 1983], and commercial electronic pressure transducers [Harris et al., 1999]. While some of these methods have been more successful than others, Fuji pressure-sensitive film has traditionally been the most common method of measuring the contact area and stresses of both TJR prostheses and natural joints. While it has long been known that the stiffness of Fuji pressure-sensitive film prohibits it from conforming to the complex curvatures associated with diarthrodial joints, this can be overcome by specially cutting the film to conform to the joint surfaces [Ateshian et al., 1994]. However, it has been shown that the use of Fuji-film will overestimate the contact areas, and will also change the contact characteristics of the joint [Ateshian et al., 1994, Liau et al., 2001]. Moreover, Fuji-film will not allow for dynamic, real-time measurements to be made. Electronic transducer methods can provide dynamic measurements of contact stress and contact area, and some can even provide dynamic contact stress distribution plots. However, even the state-of-the-art electronic transducer systems that provide dynamic measurement of contact stress distributions have proven to be quite difficult to implement in more conforming TJR prostheses because of calibration issues and problems with fixation and durability, especially when using in joint simulators in-vitro [Harris et al., 1999, Liau et al. 2001]. All of the methods available still introduce a different material of at least 75-100 µm thickness, which will alter the contact characteristics of the joint.

While simple materials testing machines can be used to apply loads to TJR prostheses, the accepted standard for wear testing is the joint simulator. A joint simulator is used in-vitro to apply physiologically relevant loading patterns to an
artificial joint implant. While it would be ideal to measure the contact stress distribution dynamically during physiological loading of an implant on a joint simulator, the current state of the art will only allow for contact stress distributions to be measured under less complex conditions such as various static loads applied to the implant with simpler materials testing machines [Beaule et al. 2002]. Therefore, it remains necessary to use joint simulators to carry out time consuming, expensive lifetime wear testing of implants to quantify the amount of wear damage that will occur using a given set of clinically relevant conditions. There is a need to be able to dynamically measure the contact stress distribution while testing in a joint simulator to allow qualitative prediction of lifetime wear performance without the expense and time consumption of a lifetime wear test.

In order to address the need to experimentally measure the dynamic contact stress distribution on the UHMWPE tibial insert without altering the mechanical properties of the articular surface, a unique sensor has been developed that is integrated into the surface of the tibial insert, has identical mechanical properties to UHMWPE, and can provide dynamic measurements of the contact stress distribution throughout the entire gait cycle. To achieve this, a UHMWPE composite with modified electrical properties was developed and integrated into the implant. To validate this new sensor technology and to address the hypothesis that unconstrained TKR geometries have less contact area than conforming designs, a multi-axis, force-controlled knee joint simulator was used to apply a standard walking cycle load pattern to both a highly conforming, PCL sacrificing tibial insert, and a less conforming, PCL retaining tibial insert. Both
implants contained the new sensor technology so dynamic measurements of the contact stress distribution could be collected.

2.2 - Materials and Methods

Two rectangular blocks (82.55 mm L x 57.15 mm W x 19.05 mm H) were formed by compression molding GUR 4150 UHMWPE resin powder in a stainless steel mold at 210 °C at a pressure of 10 MPa for 20 minutes, then 44 MPa for 40 minutes [Chan et al., 1997]. The same processing conditions were used to mold many 1.59 mm diameter pegs of the UHMWPE-composite sensor material. Then 28 columns by 18 rows of 1.59 mm diameter holes on a 2.54 mm grid were machined into the blocks, as demonstrated in Figure 2.1. After inserting the UHMWPE-composite pegs into the holes, the blocks were compression molded again under the same processing conditions to fuse the composite pegs with the virgin UHMWPE blocks. This resulted in two completely fused, solid blocks of mainly virgin UHMWPE with localized areas of the UHMWPE-composite sensor material. Because the bulk mechanical properties of the composite sensor material are not significantly different from those of the virgin UHMWPE [Clark et al. 2006], the instrumented blocks were able to be machined into tibial insert geometries by the implant manufacturer using standard milling operations on their production units. This ensured that the instrumented tibial inserts used in this study had contact geometries identical to the corresponding production tibial inserts implanted in patients. The two geometries machined were the Zimmer Natural Knee II Standard-Congruent
insert (standard), and the Zimmer Natural Knee II Ultra-Congruent insert (congruent). Both tibial inserts were size 3, right side knee implants.

Figure 2.1. 3-D view of block with enlarged grid pattern to illustrate how blocks were fabricated.

Figure 2.2. Top view of instrumented and machined standard insert
After the tibial insert contacting surfaces were machined, wires were attached to the un-machined backside of each tibial component. The wires connected each sensing point to its corresponding channel of a custom data acquisition system. The data acquisition system was used to filter and collect the raw sensor outputs into the computer. The system consisted of dual 256 channel custom-designed analog multiplexer arrays and the necessary filtering amplifiers to output a single-channel data stream of voltage values that corresponded to the instantaneous contact stress measured at each sensing point on each condyle of the tibial insert’s surface. This single-channel data stream was clocked into a high-speed A/D converter card on the computer. An additional wire was attached to the femoral component of the implant so that an excitation voltage could be applied to it. Once the data was in the computer, a graphically intensive software interface (Labview 7.1, National Instruments Corporation, Austin, Texas) was
used to store, visualize, and analyze the dynamic contact stress and its
distribution.

A four-station, force-controlled TKR wear-testing simulator (Instron/
Stanmore, Model KC Knee Simulator) was used to apply physiological loads to
both of the instrumented knee implants. Unlike displacement controlled
simulators, this force-controlled, multi-axis simulator allows six degrees of
freedom so that the individual implant will respond to the input loads based on its
specific design geometry. Because wires were attached to the un-machined
backside of the tibial inserts, the tibial tray components were not used in this
study. Instead, the tibial inserts were potted directly into the tibial cup fixtures of
the simulator using polymethyl methacrylate (PMMA) bone cement. The
recommended surgical alignment guidelines were followed to ensure that the
alignment of each tibial insert matched the alignment of the femoral component.
The tibial inserts were aligned with 0° A/P tilt, 0° varus/valgus tilt, and 0° axial
rotation. For kinematic evaluation, the “home” position for each implant was
found by applying light axial force with no A/P or M/L restraints. This home
position represents where the implant would rest in vivo at 0° of flexion. Then
the A/P actuators and the A/P buffer springs were adjusted and connected such
that each implant had the necessary A/P position and I/E rotation to rest in the
home position at 0° flexion under light axial loads only. The A/P buffer springs,
which are designed to simulate soft tissue restraints, were set-up to simulate a
PCL retaining implant. This same spring set-up was used with both the standard
congruent (PCL retaining) and the ultra congruent (PCL sacrificing) inserts. The
femoral component was mounted to the simulator’s femoral fixture such that it was allowed to rotate about a single axis that provided the least amount of proximal/distal translation between 0° and 60° of flexion [DesJardins et al. 2000]. No lubrication was used during this testing since the presence of lubricant under dynamic conditions adds another level of complexity to the contact mechanics, a situation that was not investigated in this study.

Static testing of the implants was first performed to compare the sensor readings to the literature. At flexion angles of 0°, 30°, 60°, and 80°, the static axial load applied was 2.9 kN (4 x B.W.).

A fully dynamic standard walking cycle pattern (ISO 14243 waveform verified) was then applied to both tibial inserts at 1Hz (Figure 2.4). The kinematic data of femoral flexion angle, A/P tibial displacement, and I/E tibial axial rotation, as well as the imposed actuation magnitudes of axial compressive force, A/P force, and I/E torque were collected by a computer at a rate of 50 Hz and averaged over 15 successive cycles.
Measurements were performed under both lubricated and un-lubricated conditions. Most of the measurements were performed under un-lubricated conditions in order to rule out the possibility of a conductive lubricant interfering with the sensor’s readings. In order to investigate the effects of a lubricant, olive oil was used since it is a non-conductive lubricant. It should be noted that the viscosity of olive oil (∼80 cP) is higher than the viscosity of the bovine serum lubricant (10 – 20 cP) that is normally used during simulator testing.

2.3 - Results

Static loading of the implants at four different degrees of flexion (Figure 2.5) revealed that the congruent insert, with its more conforming geometry, had more contact area than the standard insert at all the flexion angles. Additionally, the
data show that the contact area of both implants decreased as the flexion angle increased.

![Figure 2.5 - Total contact area under static loading at different flexion angles](image)

The kinematic reactions of the implants, as well as the soft tissue loads and the implant reaction loads measured by the simulator are seen in Figure 2.6. It can be noted that the maximum posterior displacement of 3mm and the maximum internal rotation of 3 degrees occur just at the end of stance phase. Both implants then experienced a rapid anterior displacement and external rotation at the beginning of the swing phase.
The average contact pressure distribution at four different phases of the gait cycle can be seen in Figures 2.7 and 3.9. The contact pressure intensity scale at the right side of the figures indicates that any regions with a red color experience pressures greater than 15 MPa, which is the yield point of the polymer, and indicates an overloaded region. The average contact pressure distribution for the
standard insert (Figure 2.7) shows that the lateral condyle experiences some regions of overload during all phases of gait, with the larger overloaded areas being seen from mid stance through terminal stance. Intracondylar contact can also be seen on both condyles throughout the gait cycle. During early and mid stance phase, the contact on both condyles is aligned with the central axis of the insert. During terminal stance, the medial condyle contact is shifted in the anterior direction approximately 7mm anterior of the central axis, while the lateral condyle contact intensifies but does not shift. During swing phase, the contact on the lateral condyle shifted about 5mm posterior of the central axis.
Figure 2.7 - Average contact pressure distribution over key phases of gait for Standard Insert

Figure 2.8 shows the average contact pressure distribution over the congruent insert throughout the 4 phases of gait. During the early and mid stance phase, the axis of contact across both condyles is parallel to the central axis of the implant, but is posterior. The contact on the medial condyle was less intense than that seen with the standard insert, not exceeding 15 MPa during any of the phases of the gait cycle. The contact on the lateral condyle was more intense and remained focused over the same area of the implant throughout the gait cycle.
Figure 2.8 - Average contact pressure distribution over key phases of gait for Congruent Insert

Figure 2.9 - Standard insert - total contact area and percentage of total contact area that is under more than 15 MPa contact pressure
The total contact area over the surface of the tibial insert as measured by the sensor as a function of time is shown in Figures 2.9 and 2.10. Seen on the second axis of each plot is the percentage of the total contact area that experienced a stress of more than 15 MPa. This is what’s referred to as the overloaded area, and it is plotted as a percentage of the total contact area at each point in time. As indicated by the blue curve in each plot, the total contact area was much higher during stance than during swing phase. The amount of the contact area that was overloaded was observed to be higher during the swing phase and lower during the stance phase.

Figure 2.10 - Congruent insert - total contact area and percentage of total contact area that is under more than 15 MPa contact pressure
Figures 2.11 and 2.12 show the total instantaneous contact area plotted against the percentage of the gait cycle for the standard insert and the congruent insert for both dry and lubricated conditions. The lubricant used to obtain the contact areas measured under lubricated conditions was olive oil because of its dielectric properties.
2.4 - Discussion

The sensor technology presented in this study is based on a composite of UHMWPE with modified electrical properties that allow it to quantify the contact pressure being applied to its surface. By applying an excitation voltage to the metallic counterface, upon contact between the femoral component and each sensing point on the tibial insert, an electrical current proportional to the contact pressure will flow through the sensing point. The electronics measure this current at a high rate at each sensing point, which gives the contact pressure distribution of the tibial insert surface. By continually measuring these signals, the dynamic contact stress distribution on the tibial insert surface can be measured over the duration of the gait cycle. Because the exact location of each
sensing point on the surface of the instrumented tibial insert is known, and because the geometry of the grid pattern of sensing points dictates a given area covered by each individual point, the dynamic contact area and its exact location on the surface of the tibial insert can be determined by monitoring which sensing points show contact. The spatial accuracy of the sensor is therefore determined by the size of each sensing point and can be tailored to the specific application. The spatial accuracy observed in this study was 6.5 mm$^2$.

The contact pressure distribution plots (Figures 2.7 and 2.8) illustrate the laterally pivoting design of both implants. On the standard insert during terminal stance, the contact on the lateral condyle was shown to remain centered around the central axis with a higher intensity while the contact on the medial condyle was shown to shift in the anterior direction. The pivot point on the lateral condyle can be clearly seen during terminal stance, with a relatively large overloaded area. From the contact distribution plots, the key areas where wear will occur can be readily identified.

During early and mid stance phase, the axis of contact of the congruent insert was parallel to the central axis of the implant (low point of implant), but was posterior to the central axis. In contrast to the standard implant, the contact on the congruent implant during early and mid stance phase was not at the lowest point on the insert. As with the standard insert, the pivot point on the lateral condyle can clearly be identified.

During stance phase, there were three distinct peaks seen in the contact area for both the standard and the congruent insert. These peaks directly
correspond with the local minima and maxima of the flexion angle input curve. When the flexion angle was changing directions (with the associated acceleration in sliding velocity), the contact area was seen to momentarily peak. The magnitude of the peaks was seen to correspond with the axial load input, with the largest contact area for both implants occurring during terminal stance at the same time that the flexion angle reached a minimum.

It can also be observed from Figures 2.9 and 2.10 that the total contact area measured in each insert as a function of time was very similar. However, the congruent insert showed a slightly higher amount of its contact area being overloaded, especially during the stance phase.

A large contact area can be beneficial, if it evenly distributes the load, causing less of the implant’s surface to be overloaded. However, if the load is not evenly distributed over a large contact area, then it may not be as beneficial. With the congruent insert, the amount of the contact area that was overloaded during the stance phase was seen to be higher than for the standard insert. It can be seen from Figure 2.7 that the load was not distributed through the congruent insert as evenly as it was through the standard insert. The congruent insert was seen to have much higher contact pressures on the lateral side than on the medial side.

The distribution plots together with the contact area graphs show how the congruent insert geometry contact conditions compared to the standard insert geometry contact conditions. The congruent insert experienced a larger overloaded area during stance because the lateral condyle experienced a more
intense contact pressure over a larger area. This was due to the contact pressures on the medial condyle of the congruent insert being fairly low, ranging from 0 to 5 MPa. Conversely, the standard insert geometry allowed for both the lateral and the medial condyles to experience a slightly more distributed load, with a larger portion of the area experiencing moderate contact pressures of around 7 MPa to 10 MPa.

It should be noted that the contact areas of each insert were also measured statically at differing flexion angles. For all degrees of flexion, the contact area measured for the congruent insert was about 50% greater than that measured for the standard insert. However, under dynamic loading conditions, it was found that the contact areas were about the same and that the distribution of load through each implant was different. From the static data, it might be concluded that the congruent implant would experience less wear due to the higher contact area. However, under the dynamic conditions that the implants might experience in vivo, the congruent insert experiences contact on the posterior slope of the lateral side, yielding a smaller lateral side contact area and a more defined lateral side pivot point. This resulted in a higher percentage of the contact area during stance being overloaded, suggesting that the congruent insert may experience more wear than the standard insert.

The goal of this study was to investigate the effects of implant geometry on the dynamic contact stress distribution, therefore, the same simulator settings of alignment and soft tissue restraint springs were used for each implant. Because the more conforming design is usually indicated for PCL sacrificing procedures,
the contact pressure distributions could also be examined with simulator spring settings that more closely approximate these soft tissue characteristics. During set-up of the simulator it was noted that the contact stress distribution was extremely sensitive to even slight alignment differences or soft tissue spring settings, suggesting that the new sensor technology could also be ideal for performing studies on a particular implant’s sensitivity to misalignment.

To determine the contact area under lubricated conditions, olive oil was used as the lubricant, and because olive oil is very non-conductive, it follows that the contact area measured was due to direct material contact. Thus, no reading was measured for any of the sensor points over which pressurized lubricant transferred the load without allowing the femoral surface to contact the tibial surface. Figure 2.11 shows that during stance, there is not much difference in the measured contact area between dry and lubricated conditions. This indicates that during stance, the lubricant does not provide much surface separation, which would indicate that boundary or mixed mode lubrication prevails during stance. During swing phase, the lubricated measurements indicated no discernable contact, whereas the dry measurements indicate that there is still a good degree of contact. This suggests that full fluid film lubrication may indeed exist during the swing phase for the congruent insert with olive oil as a lubricant. It should be noted that the viscosity of olive oil is higher than that of bovine serum, which is the standard lubricant used for *in-vitro* simulator testing.

The contact area plotted against the gait cycle for the congruent insert for both dry and lubricated conditions is seen in Figure 2.12. The data for the
congruent insert show that during stance, the contact area measured under lubricated conditions was less than under dry conditions. This indicates some degree of surface separation. Interestingly, there appears to be little difference between dry and lubricated during swing phase. When the contact pressure distribution plots of the congruent insert during dry testing are also taken into account, it becomes apparent that the lateral (right) side contact patch’s location on the posterior slope of the implant is the cause for contact readings during swing phase under lubricated conditions. The relatively high intensity of this contact patch, and its stationary location (due to the fact that it is the pivot point) attribute to the lack of fluid film lubrication over that portion of the tibial surface during swing phase. The pressure distribution plots for the congruent insert during stance phase showed that the medial (left) condyle had a U-shaped contact patch of relatively low intensity, which would seem to suggest close tolerances between the femoral and tibial surfaces in the points immediately surrounding the points that measured contact. The contact area graph (Figure 2.12) would seem to confirm that the medial (left) condyle may very likely have experienced elastohydrodynamic lubrication during much of stance, since the contact area measured during stance for the lubricated condition was about one-half that measured for the dry condition. Again, it is important to note that these results were obtained using a lubricant with higher viscosity than what would be expected during operation of artificial knee joints. Therefore, this data does not necessarily suggest that the congruent insert’s medial side will operate under EHL during simulator testing or in vivo conditions. However, the data does
suggest that the sensor technology can provide insight into the lubrication mechanisms of the joint.

2.5 - Conclusion

The static measurements performed in this study, as expected, showed that the congruent insert had a higher contact area at all flexion angles when compared to the standard insert. This might lead to the conclusion that the stress levels would therefore be lower in the congruent insert indicating it would experience less wear. However, the results of dynamic measurements unexpectedly showed that the contact area as it varied over the gait cycle was quite similar for both implant geometries. Additionally, the contact stress distribution was shown to be much more intense on the lateral side of the congruent insert, causing the congruent insert to show the larger amount of overloaded area, suggesting that the congruent, rather than the standard insert, has the higher wear potential in simulator testing. The fact that the dynamic results lead to such a different conclusion than static measurements alone shows the importance of measuring contact stress distributions under dynamic simulator conditions.
3.0 – CORRELATION OF SENSOR OUTPUT TO SURFACE SEPARATION

3.1 - INTRODUCTION

One of the key aspects of tribology is surface topography, or surface texture, which explains the 3-D geometry of the bearing surface. Larger, macro-scale features, including visible machining marks, are important and they can be optimized to allow for reduced friction, good lubrication, and reduced wear. The micro-scale features are just as important as the macro-scale features, and an understanding of the terminology used to describe these features is necessary. Surface roughness is the main term used to describe the micro-scale roughness of the bearing surface. Many surfaces that look and feel perfectly smooth and flat may be fairly rough at the micro-scale. The UHMWPE component of artificial joints falls into this category. Most tribological theory relates to metal-on-metal contact, since the majority of mechanical bearings are still composed of metals. However, the same terminology applies for all materials. There are many different measures of surface roughness, but some of the more common measures are average roughness ($R_a$), peak-to-valley roughness ($R_t$), and root-mean-square (rms) roughness ($R_q$). Each of these values describes the height of the peaks and valleys on a surface. These features, also known as asperities, are common to all surfaces, and their magnitude depends on the manufacturing processes.

The Co-Cr metallic component of an artificial joint is typically polished to a mirror finish during manufacturing. The typical rms roughness of a Co-Cr
component is usually around 20nm, rarely exceeding 50nm. The UHMWPE component in artificial joints is considerably rougher, with typical Rq values ranging from 700 nm to 1,200 nm.

Because of the large difference in surface roughness between the two components, the metallic component’s roughness can be considered negligible. Therefore, from a micro-scale schematic view, the metallic component can be visualized as a flat surface contacting a rough surface that represents the polymer. As the flat metallic surface approaches the rough polymer surface, it first makes contact with the highest asperity peaks of the polymer. As the compressive force pushing the flat metallic surface into the polymer surface increases, the asperity peaks of the polymer surface will deform, both elastically and plastically, leaving micro-scale gaps between the two surfaces where the asperity valleys are.

With lighter loads pushing the two surfaces together, the real contact area (RCA or $A_R$) is the area of each surface that is in direct contact. The RCA is often a small percentage of the apparent contact area, which is defined by the macro-scale geometry of the contacting surfaces. Therefore, the RCA is heavily dependant upon the surface roughness of the surfaces and the mechanical properties of the materials. Because the metallic component of the artificial joint has a modulus that is two orders of magnitude higher than the polymer component, one can conceptualize that only the polymer surface deforms when the metallic surface compresses against it. The larger the compressive force pushing the two surfaces together, the larger the RCA becomes. As the RCA
becomes larger, the resistance to horizontal translation between the surfaces increases due to higher interaction between the surfaces. This is one of the main contributing factors to friction, and explains how frictional force is proportional to normal force.

In order to conceptualize what happens at the micro scale when the metallic surface approaches and contacts the polymer surface, the surface roughness values must be considered. Because the metallic surface is much harder and much smoother than the polymer surface, it can be considered smooth and flat over the small diameter of the sensor point. For UHMWPE, the \( R_t \) value is 10 µm and the \( R_q \) value is 1 µm. Another useful measure when considering the total peak-to-valley height of the asperities is the \( R_z \) value. The \( R_z \) value is the average of the 10 largest peak-to-valley readings measured during a scan. The \( R_z \) value for UHMWPE is about 6 µm.

In order to describe the mode of lubrication in terms of surface roughness values, the specific film parameter, lambda (\( \lambda \)), is often used. Although not perfect, \( \lambda \) is a common design parameter used for many bearing surfaces. The \( \lambda \) value is usually defined as the ratio of the minimum film thickness (\( h_{min} \)) to the rms average surface roughness (\( R_o \)). Because the majority of lubrication engineering applications relate to metal on metal contact, the commonly cited \( \lambda \) values relating to the mode of lubrication have usually been obtained for metal on metal contacts.

When conditions of lubricant viscosity, load, and entrainment speed are appropriate to completely separate two contacting surfaces, such that no
asperities collide, the mode of lubrication is either elastohydrodynamic lubrication (EHL), or full hydrodynamic lubrication (HL). The $\lambda$ value usually associated with EHL is usually between 3 and 10. For HL, $\lambda$ values can range from 5 to 20 or more. As $\lambda$ values decrease, mixed lubrication begins to occur. During mixed lubrication, asperities will collide, but there is also much of the two surfaces still separated by a lubricant film. During mixed lubrication, pressurized fluid under portions of the contact area will support some of the load, preventing some direct contact between the surfaces. When $\lambda$ decreases even more still, usually below 1, boundary lubrication becomes prevalent. In boundary lubrication, most of the two surfaces are in nearly direct contact with each other. Little to no pressurized lubricant is found under the contact area, but very thin layers of lubricant can still be found on the surfaces due in large part to absorption or chemical reactions. Under the same loading conditions, the coefficient of friction will usually be lower in boundary lubrication than in dry contact because the thin layers of lubricant still provide some degree of protection to the surfaces. The friction in boundary lubrication is a combination of shear of asperities on the two surfaces and shear of the lubricant.

Boundary lubrication is thought to be common in artificial joints; however, it is likely that the two different surfaces (metal and polymer) have entirely different modes of establishing boundary layers. Additionally, many polymers, including UHMWPE exhibit what is commonly referred to as self-lubrication. In general terms, this refers to the fact that these polymers exhibit equally low coefficients of friction whether operated under dry conditions or lubricated conditions. However,
in technical terms, lubrication is beneficial for many other reasons besides just lowering the coefficient of friction, including heat dissipation, so this is not to suggest that lubrication does not play a very important role in UHMWPE joints.

The amount to which the surface roughness of the UHMWPE will “flatten” when heavily loaded against a highly polished metal is somewhat of an unclear issue. While increasingly complex theory can be applied to this subject, one notable reference proposed that during normal operation of artificial joints containing UHMWPE, the pressures required to totally flatten all of the asperities would need to be on the order of 100 MPa [Auger PhD Thesis 1992, Dowson et al. 1991). Therefore, over the majority, if not all, of the polymer’s surface, asperity valleys will always be present to retain pressurized lubricant and the real area of contact will always be less than the apparent area of contact.

The goal of the research presented in this chapter was to deliberately control the separation between the two surfaces while performing measurements with the contact being immersed in the lubricant. This was accomplished in two main ways: a small materials testing machine, and a custom lever rig system. Many different sensor configurations were examined during the process of this dissertation research. Although most of the configurations investigated were not included in this dissertation, all of their advantages and disadvantages led to the development of the most notable sensor configurations, which are presented in this dissertation.
3.2 – MATERIALS AND METHODS

3.2.1 – Fabrication of Sensor Block #1

In order to construct sensor block #1, seen in Figure 3.1, virgin UHMWPE powder (GUR 1150, Ticona Engineering Polymers, Florence, KY) was compression molded into a solid rectangular block of approximately 0.4” thickness. A second block of the same dimensions was formed of the composite UHMWPE sensor material. These blocks were then placed in the same mold and the compression molding cycle was repeated to fuse the blocks together. The mold used during this procedure was a stainless steel rectangular mold with 0.25” thick walls, 0.25” radius inside corners, and inside dimensions of 3.25” length x 2.25” width x 1” depth. A laboratory press (Model C Laboratory Press, Fred S. Carver Inc., Wabash, IN) equipped with electric heaters was used to apply temperature and pressure to the mold.

Sensor block #1, seen in Figure 3.1, is approximately 3.25” in length, 2.25” in width, and 0.6” in height. In the center of the rectangular block, a 1” diameter hole extends 0.4” through the top polymer layer to expose the composite sensor material at the bottom of the hole.

The block was machined with a desktop 3-axis milling machine (Roland Modela MDX-15, Roland DGA Corporation, Irvine, CA) with a 1/8” straight-shaft spindle and a modified drive motor to allow added control over milling speed. Surfacing operations were performed on the sensor block to ensure both the top
and bottom sides were parallel. This was accomplished using a 1/8” diameter, 2-flute, center-cutting, solid carbide square end mill with a 0.75” length of cut and a 1.5” overall length (MSC Industrial Supply Company, Melville, NY). The surfacing toolpath that produced best results was a linear path, cutting in both the x and y directions with an xy step-over of 0.03125”, and a 0.03” depth of cut. The feed rate was 15 mm/s, the machine’s maximum, and the tool speed was 12,000 rpm. The hole in the center of the sensor block was machined to tight tolerances to accept a 1” diameter stainless steel plunger with a 4.1” radius spherical tip, thus ensuring vertical alignment of the plunger. To create the toolpath for the hole, a basic CAD/CAM program was used (VisualMill 5.0 Basic, MecSoft Corporation, Irvine, CA), with the same basic speed and feed used for the surfacing toolpath.

Figure 3.1 – Sensor Block # 1 shown with 4.1” radius of curvature spherical plunger
3.2.2 – Measurements with Sensor Block #1

In order to control the surface separation and normal load of the contact, a bench top materials testing machine (Vitrodyne V-1000, LiveCo Inc., Burlington, VT) was used. In conjunction with the materials testing machine, load cells (Transducer Techniques, Temecula, CA) ranging from 10g to 1kg capacity were also used. The most common loading pattern applied was a displacement-controlled trapezoidal compression ramp at the machine’s minimum speed of 10μm/s.

Sensor Block #1 was used in three main configurations during this research. As mentioned previously, the block was machined to accept a 1” diameter spherical-tipped plunger, as seen in Figure 3.1. Two other stainless steel indenters were also used: a small-radius spherical indenter, seen to the left of Figure 3.2, and a flat indenter, seen on the right in Figure 3.2. The flat indenter was a diameter of 0.438”, which is smaller than the 1” hole diameter of the sensor block, therefore, great care had to be exercised to ensure parallel alignment of the indenter and the sensor’s surface. This was accomplished by letting the indenter rest on the sensor’s surface with the cross-head of the materials testing machine positioned approximately 1mm above the top surface of the indenter. While taking care not to move any of the components, cyanoacrylate instant adhesive (Loctite 414, Loctite Corporation, Rocky Hill, CT) was applied to the small gap between the cross head and the top surface of the indenter. This procedure had to be repeated whenever there was any movement.
of the sensor block to attempt to maintain perfect parallelism between the flat indenter’s bottom surface and the sensor’s surface.

![Image of sensor block](image)

Figure 3.2 - Small radius spherical (left) and small radius flat (right)

The well-shaped hole in the center of the sensor block not only allowed for vertical alignment of the large radius spherical plunger, but it also allowed the use of lubricant in a controlled manner. During preliminary testing, many different types of lubricants were used with Sensor Block #1, including deionized water, tap water, 50% bovine serum, olive oil, a mineral oil viscosity standard, and a fluorocarbon specialty fluid used for cooling of electronics. Extensive testing during the preliminary phases of this research led to the decision to use the fluorocarbon liquid (Fluorinert FC-70, 3M Electronics Markets Materials Division, St. Paul, MN) due to its superior electrical properties, shown in Table 3.1.
<table>
<thead>
<tr>
<th>Viscosity (mPa*s)</th>
<th>Volume Resistivity ($\Omega\text{-cm}$)</th>
<th>Dielectric Strength (kV, 0.1” gap)</th>
<th>Dielectric Constant (at 1kHz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>24</td>
<td>$10^{15}$</td>
<td>40</td>
<td>1.98</td>
</tr>
</tbody>
</table>

Table 3.1 – Physical properties of fluorocarbon liquid (from manufacturer supplied data sheet)

In order to collect and record the data from the sensor, a 1 MHz, 16-bit analog I/O data acquisition system (Personal Daq 3000, Iotech Inc., Cleveland, OH) was used. The system interfaced with a laptop computer via high speed USB 2.0 interface. The system was controlled using custom software programs written in a graphical, high-level language designed for data acquisition applications (LabView 8.0, National Instruments, Austin, TX). The software allowed control of both the analog output and analog input features of the data acquisition system. The analog output was used as the excitation for the sensor, applying a controlled voltage – most commonly a 10V peak-to-peak (3.5 $V_{\text{RMS}}$) sinusoidal signal at a frequency of 20 Hz. Three analog input channels were used, typically collecting data at a rate of 60 kHz. The three inputs that were measured and stored were the excitation voltage (or reference voltage, $V_{\text{ref}}$), the analog output from a signal conditioning box connected to the load cell, and the sensor’s output voltage ($V_O$).

The electronics interface with the sensor consisted of a fairly simple electronic circuit designed to accurately measure the current flowing through the
sensor and output a voltage proportional to that current. The circuit, commonly referred to as a current-to-voltage converter, is based on an operational amplifier (op-amp) with its non-inverting input grounded. The sensor was connected to the inverting input of the op-amp, thus allowing the current flowing through the sensor to see a “virtual ground”, allowing for the most accurate measurement of current that varies over a large range. Therefore, the output voltage from the sensor ($V_O$) is a direct measure of the current flowing through the sensor due to the applied reference voltage.

### 3.2.3 – Fabrication of Sensor Block #2

The combined results of the experimental data collected with sensor block #1 led to the development of a second sensor block for testing of different contact configurations. Sensor block #2 consisted of four individual wells, each containing a discrete sensing point, as seen in Figure 3.3. In order to construct sensor block #2, the same basic procedure used during the fabrication of sensor block #1 was followed. However, a larger stainless steel mold was used for sensor block #2. The inside dimensions of the larger mold were 3.75” in length, 3.25” in width, and 1.5” in depth, with 0.25” thick walls and 0.25” radius corners.

The four independent wells of sensor block #2 were designed to test four different contact configurations involving flat contact where the sensor is overlapped by the metal. There were two different well diameters of 0.438” and 0.623” designed to accept plungers of the same diameters. The wells were
machined with tolerances that ensured the stainless steel plungers remained perpendicular to the sensor’s surface, maintaining parallelism of the contact geometry. There were also two different sensing point diameters at the bottom of each well. Thus, a 0.14” diameter sensing point and a 0.29” diameter sensing point were located at the bottom of both the small and large diameter well.

Figure 3.3 – Sensor Block #2 with smaller flat plunger

3.2.4 – Measurements with Sensor Block #2

The measurements for sensor block #2 were obtained following the same basic procedures that were followed for sensor block #1. A materials testing machine (Vitrodyne V-1000, LiveCo Inc., Burlington, VT) was used to control the
displacement of the plungers. For most of the testing performed on sensor block #2, a 50g load cell was used on the materials testing machine. As with the sensor block #1 measurements, the plunger was mechanically connected to the load attachment screw of the load cell by means of an instant adhesive. This ensured that the force applied to the plunger was perpendicular to the surface of the sensor. The main loading pattern used with sensor block #2 was a trapezoidal compression ramp with a hold time of approximately 10s at the lowest point of the ramp, which was determined at the beginning of each set of experiments to be a vertical displacement that achieved approximately 60% - 80% of the full-scale reading for the load cell. Care was taken to avoid applying more than the full-scale load to the load cell to avoid permanent damage to the load cell. Therefore, a typical maximum applied load when using the 50g load cell was between 30g and 40g.

As with sensor #1, many different lubricants were used in the preliminary rounds of experiments, but a fluorocarbon electronic cooling liquid, FC-70, was the main lubricant used with sensor block #2. The same procedure for acquiring sensor readings was also followed for sensor block #2, including the application of a 10V_{p-p}, 20Hz sinusoidal excitation voltage. For most of the experiments with sensor block #2, the data acquisition was started at the bottom of the compression ramp, during the hold time when the maximum load was applied. Data collection continued as the plunger load decreased and as the plunger lifted away from the sensor’s surface. Data acquisition was ended after complete surface separation had occurred. The materials testing machine’s control
computer also recorded and stored the cross-head position. This meant that the
data needed was collected on two separate computers: the materials testing
machine’s control computer, and the sensor data acquisition system’s control
computer. Because two different computers were used to collect all of the
desired information, it was necessary to merge the position data collected on the
separate computer with the load and sensor information. A separate LabView
program was written to accomplish this merging and alignment of the two data
sets. Therefore, the end result from each experimental run was a graph
containing 4 time-domain curves showing the deflection-corrected sensor
position, the applied load, the 3-cycle DC sensor output voltage, and the 3-cycle
rms sensor output voltage.

3.2.5 - Fabrication of Sensor Pin #1

The combined data collected from the sensor blocks led to the development
of a UHMWPE pin containing the sensor material. This contact configuration still
allowed for the metal to overlap the sensor point, but by inverting the contact, the
experimental procedures were simplified. Sensor Pin #1, as seen in Figure 3.4,
had a spherical tip with a radius of curvature of 0.1m, which allowed for very
slight experimental misalignment. The pin was constructed such that it consisted
mostly of virgin UHMWPE, with the composite sensor material exposed on the tip
of the pin as a 0.0625” diameter sensing point. The sensor material extended
internally up the shaft of the pin, connecting with a 0.2” band of the sensor
material that transected the pin 0.4" up from the tip. The total length of the pin was 1" and the diameter was 0.5".

Because of the more complicated shape of sensor pin #1, the final machining parameters required were different than those used for the sensor blocks. For the final machining of the sensor pin, a 1/8" diameter ball mill was used. The solid CAD model for the pin seen in Figure 3.5 was used in CNC machining software (VisualMill 5.0 Basic, MecSoft Corporation, Irvine, CA). Multiple roughing steps were taken to allow the finishing step to remove the optimal amount of material. For the milling of sensor pin #1, a 0.01" margin of material was left for removal during the final finishing operation. A 3-axis parallel tool path, as seen in Figure 3.6, was used for milling the tip of the pin, thus, the tool traveled in a linear path, removing material with each pass, while the z axis
followed the vertical contour of the pin with each pass. The input parameters for the parallel finishing toolpath were a linear feed rate of 15mm/s with a speed of 15,000 rpm and a step-over distance of 0.0025” per pass.

Figure 3.5 - CAD image of sensor pin with 0.1m radius tip
3.2.6 - Measurements with Sensor Pin #1

Another method for more accurately controlling the vertical displacement was used for the measurements obtained with sensor pin #1. To accurately control the separation of the two surfaces, a custom system making use of a vertical comparator and rigid bar was devised based on the basic mechanical principles of the lever and mechanical advantage. The lever rig, seen in Figure 3.7, consisted of a fixed pivot point, a load cell attachment point, and a vertical displacement control point. The operating principle is simple – a rigid member is firmly anchored to a pivot point on one end that allows only one degree of freedom. The pivot allows free, unrestricted rotation in the x-z plane, but
prevents all other rotation and displacements. A digital-readout vertical comparator (Mitutoyo Model 192-631, Mitutoyo Corporation, Japan) with a resolution of $\pm 10\mu m$ was used to control the vertical height of the other end of the rigid bar by a ball-on-flat interface. This type of joint ensured that the vertical force vector acted through the same point on the bar, regardless of the bar’s angle. This point was located a distance $L_T$ from the pivot point. Finally, the load cell was rigidly attached to the arm in between the pivot point and the vertical lift point at a distance $L_L$ from the pivot point. To determine the vertical displacement of the load cell, the readout on the vertical comparator was scaled by the $L_L:L_T$ ratio. During this research, the $L_L:L_T$ ratio was 1.65” : 16.5”, or 1:10, such that a change of 10$\mu m$ on the digital readout corresponded to a vertical displacement of 1$\mu m$ for the load cell. To determine the vertical displacement of the sensor itself, the deflection of the load cell also had to be accounted for. The whole lever rig system resided on a large, flat marble mass, which allowed for remarkable repeatability of manually controlled vertical displacements of less than 1$\mu m$. With a 50 gram load cell, the vertical point of initial contact could easily be determined. During typical use, the displacement was manually recorded and the load cell output and sensor output were either collected and saved on the computer, or monitored on the computer and recorded manually. The disadvantage with this system was that the viscoelastic properties of the UHMWPE sensor material could theoretically limit the accuracy when applying load due to the step-wise nature of the manually dialed-in displacements.
Figure 3.7 - Lever Rig for accurate (±1μm) displacement control

The metal counter face used with sensor pin #1 was a mirror-polished stainless steel bar, and the contact was immersed in FC-70. All of the measurements were performed on the way down to the maximum displacement for each experimental run, and 15 minutes were allowed between each of 3 runs. The starting point for each experiment was set as 0μm by using the zero function of the vertical comparator, and all values below that point were negative. The starting point was chosen at the beginning of the experiment to allow for at least 10μm of surface separation, and the starting point was not changed from one run to the next. The 1kg capacity load cell’s vertical position was determined from
the 1:10 ratio of load cell movement to vertical comparator movement. A custom LabView program was used to take measurements at each manually dialed-in control position. Measurements were taken for about 6 seconds at each control point at a rate of 10 kHz. The program output the 6-second average rms sensor output, DC sensor output, and load cell reading.

A spreadsheet program was used to analyze the data. The position of the sensor was determined by accounting for the load cell deflection. The three run average and standard deviation sensor position and rms sensor output was then plotted and a piece-wise regression analysis was performed.

3.3 - RESULTS

3.3.1 – Measurements with Sensor Block #1

The first notable contact configuration used with sensor block #1 was the large radius of curvature spherical plunger. A representative example of the results from this configuration can be seen in Figure 3.8, where the white waveform is the sensor’s output, the red waveform is the reference voltage, and the green waveform is the strain gage box’s analog output of the load cell reading. The vertical axis is voltage and the horizontal axis represents time. The most notable feature is the shape of the sensor’s output waveform. The fact that it is 180 degrees out of phase with the reference voltage waveform is due to the inverting electronics, but the notable feature is that it is not completely sinusoidal.
The results from the small radius spherical contact on sensor block #1, as seen in Figure 3.9, were very good during conditions of contact – loads of 0.1g or more. In Figure 3.9, the white waveform is the sensor’s output and the red waveform is the reference voltage, with the x-axis representing time, and the y-axes being voltage with different scales shown on left and right. The frequency of the waveforms is 20Hz. The data was collected at 60kHz, and the waveforms were smoothed by averaging 5 points of data for every one point plotted. It should also be noted that the phase of the reference voltage was offset by 180 degrees to correct for the inverting electronics and allow for better visualization of the non-sinusoidal nature of the sensor output.
A magnified view showing one cycle of the reference voltage and sensor output are seen in Figure 3.10. The non-sinusoidal shape of the sensor's output (white waveform) is clearly visible compared to the pure sinusoidal shape of the reference voltage (red waveform). It should be emphasized again that although the white waveform as shown is in units of volts, it is a direct measure of the current flowing through the circuit, where the current is 0.001 times the output voltage. Therefore, Figure 3.10 represents the time-domain plots of the same data that would be seen in an I-V curve, as is used, for example, to characterize semiconductors. An I-V curve shows the current flowing through a material as a function of the applied voltage, and although one could be constructed from one-quarter of a cycle of this data, doing so might be misleading since I-V plots are usually taken over much larger timescales than 0.05s. Nonetheless, it is
interesting to note this non-linear relationship between the current and the voltage.

![Figure 3.10 - One cycle of applied voltage (red) and resulting current (white)](image)

**3.3.2 - Measurements with Sensor Block #2**

Figure 3.11 shows an example of the results obtained from a flat contact geometry on sensor block #2. The data was collected as described above, such that the plunger was in contact with the sensor material at the start of data collection, and the plunger was lifted upwards. In Figure 3.11, the plunger ended at a height 28 μm above the polymer’s surface. The x-axis represents time and the y-axis is the voltage inputs for load, \( V_{o\text{RMS}} \), and \( V_{o\text{DC}} \). For the plunger height (green line), the y-axis is in units of μm, as shown on the far right.
Figure 3.11 - Flat indenter on 4-well sensor block showing load cell output (red), rms sensor output (white), DC sensor output (blue), and plunger position (green)

There are many points to note about Figure 3.11. The load cell output (red line) went from compression into tension as the plunger started to lift away from the surface. Because the load cell’s output was offset to correct for the plunger’s weight, this indicates that the lubricant (olive oil in this example) caused a suction effect (negative pressure) in the opposite mechanism of squeeze-film lubrication. Just as the data collection was ended for this test, the load started to decrease back to zero, but was still in tension, indicating that in the 28μm gap, there was still negative pressure due to the viscosity of the lubricant. This result was quite common for all of the tests involving the flat contact geometry and could not be corrected using the Vitrodyne materials tester due to the minimum speed setting.
This is one of the main reasons supporting the next contact geometry discussed in the next section.

Another important point about Figure 3.11 is that the plunger’s position (green line) started at about –6 μm. The plunger position was offset so that a reading of 0μm corresponded with the same point in time that the load cell’s output was 0V (approximately point 80 in Figure 3.11). The maximum load during this specific test was 33g, which was the case from point 0 (left side) until about point 25. At 33 grams of load, the plunger’s reading was -6μm. Because the deflection constant of the load cell was checked and confirmed before and after this test, this is a correct reading, which indicates that at 33g load, the flat surface of the metal is 6μm below the horizontal plane that defines the vertical position of 0g of load (will call this the “zero load plane”). In other testing, with a 50g load cell at rated output, the maximum penetration seen was about 8μm below the “zero load plane”. This is an important distinction, because when reporting fluid film thickness, the value is typically referenced from the mean center line (MCL). Thus, before the green line of the figure above can correspond to hₙ, it must be offset by the distance between the MCL and the “zero load plane”.

The main importance of the data shown in Figure 3.11 is the sensor’s readings. The white line is the 3 cycle average rms value of the sensor output, and the blue line is the 3 cycle average DC value of the sensor output (which is the equivalent to the mathematical average of every data point contained in 3 cycles). Seen in this data and many other tests, the DC value of the sensor
output remains negative until right about the same time that the plunger reaches the “zero load plane”. The rms value corresponds very well to the plunger height for values below the “zero load axis”.

3.3.3 - Hydrostatic Sensor Readings (Measurements with Sensor Pin #1)

Table 3.2 shows the sensor’s average rms output voltage as a function of the vertical height of the sensor. For control positions below approximately –6 μm, there was contact, and because of the upwards deflection of the load cell, the sensor’s position did not decrease as much as the lever arm’s control position. Additionally, it should be noted that the load cell registered an average load of about 200g between the control positions of –10 μm and –15 μm. As the data in the table indicates, each run of the experiment was begun at a control position of 0 μm.
Table 3.2 – Hydrostatic sensor data showing average and standard deviation rms output voltage

<table>
<thead>
<tr>
<th>Control Pos'n (10^8 m)</th>
<th>Sensor Pos'n (10^8 m)</th>
<th>$V_s$ RMS (V) ± std. dev. (V)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>8.8</td>
<td>0.0019 ± 0.00004</td>
</tr>
<tr>
<td>-2</td>
<td>6.8</td>
<td>0.0023 ± 0.000086</td>
</tr>
<tr>
<td>-4</td>
<td>4.9</td>
<td>0.0034 ± 0.000063</td>
</tr>
<tr>
<td>-6</td>
<td>3.4</td>
<td>0.0161 ± 0.000309</td>
</tr>
<tr>
<td>-8</td>
<td>2.0</td>
<td>0.0323 ± 0.000964</td>
</tr>
<tr>
<td>-10</td>
<td>1.0</td>
<td>0.0581 ± 0.00789</td>
</tr>
<tr>
<td>-15</td>
<td>-0.8</td>
<td>0.1217 ± 0.00641</td>
</tr>
<tr>
<td>-20</td>
<td>-2.4</td>
<td>0.2045 ± 0.01445</td>
</tr>
<tr>
<td>-30</td>
<td>-5.6</td>
<td>0.3758 ± 0.01861</td>
</tr>
<tr>
<td>-40</td>
<td>-8.8</td>
<td>0.5259 ± 0.01518</td>
</tr>
</tbody>
</table>

The sensor position and average rms output voltage seen in Table 3.2 are plotted in Figure 3.12. The error bars for each data point in Figure 3.12 represent one standard deviation of data for each point.
Figure 3.12 – Average sensor rms output voltage plotted against the sensor position for hydrostatic conditions

A regression analysis of the data plotted in Figure 3.12 was performed in a piecewise manner, with the results shown in Figure 3.13. It can be seen in Figure 3.13 that the output for sensor positions of less than +1\(\mu\)m was a linear fit with an \(R^2\) value of 0.9975, indicating a good linear fit. This linear region of data occurred for load cell readings above approximately 50g. It can also be seen that the data to the right of 1\(\mu\)m was non-linear. A power law fit of the data to the right of 1\(\mu\)m yielded good \(R^2\) values that can be seen in Figure 3.13, and are further shown in Figures 3.14 and 3.15.
Figure 3.13 – Curve fit information of average sensor rms output voltage vs. sensor position under hydrostatic conditions

Figure 3.14 shows the sensor's average rms output voltage as a function of the sensor's position for sensor position values greater than 0 \( \mu \)m. Figure 3.14 is shown on a linear scale, along with the equations and \( R^2 \) values of a power law fit of the data. Again, the error bars seen in the figure represent one standard deviation of the data.
Figure 3.14 – Power-fit information of sensor’s average rms output voltage vs. sensor position under hydrostatic conditions

Figure 3.15 shows the sensor’s average rms output voltage as a function of the sensor’s position for sensor position values greater than +1 \( \mu \text{m} \). Figure 3.15 is shown on a logarithmic scale for both the x and y axes. The equations and \( R^2 \) values of a power-fit of the data are shown in the figure, and it is important to note that on a log-log scale, power-fit data should appear linear. Again, the error bars seen in the figure represent one standard deviation of the data.

It is clear in Figure 3.15 that there is a good fit of the data from +1\( \mu \text{m} \) to +3.5 \( \mu \text{m} \) (with an \( R^2 \) of 0.9816) and from +5\( \mu \text{m} \) to +9\( \mu \text{m} \) (with an \( R^2 \) of 0.9869). However, between +3.5\( \mu \text{m} \) and +5\( \mu \text{m} \), there is not a good fit for the data. The load cell indicated contact at a sensor position value of approximately +5\( \mu \text{m} \)
Figure 3.15 – Power-fit information of sensor’s average rms output voltage vs. sensor position under hydrostatic conditions plotted on a log-log scale

The sensor’s RMS output voltage plotted against the load cell’s readings in grams is seen in Figure 3.16. The data shows a very good linear fit with a coefficient of determination of 0.9994. The error bars seen on the graph indicate one standard deviation of the RMS output voltage from the sensor. It can be seen that the linear fit line falls within one standard deviation of each data point.
3.4 – DISCUSSION

3.4.1 – Measurements with Sensor Block #1

The first notable contact configuration used during this phase of the research was a stainless steel large radius of curvature spherical plunger in on sensor block #1. The sensor block was machined to provide close tolerances to keep the plunger vertical and always making contact in the same spatial location on the sensor’s surface. The first notable feature of the design of sensor block #1 was that the mass and size of the sensor block, as well as the flatness of its bottom surface, kept it from moving as loads were applied. The second important feature is that the electrical connection to the sensor material was made in a manner that was found to be reliable and reproducible. The well
shape of the hole in sensor block #1 also allowed the use of lubricant in a controlled manner. The disadvantage of the design that became apparent after use was that with smaller load cells, the metal plunger was too heavy to accurately lift it away from the sensor's surface without risking damage to the load cell.

In order to use light load cells and still separate the surfaces, a small radius spherical indenter was used with sensor block #1. While surface separation was possible while using small load cells, the contact geometry was not necessarily as relevant to the knee sensor described during chapter 1 of this dissertation. Nevertheless, many tests were performed with this configuration to gain insight into construction of the next sensor block.

It was recognized that a flat-on-flat contact geometry was perhaps more relevant to represent a single sensor point of 1/16” diameter as seen in the knee sensor of chapter 1 of this dissertation. For this reason, sensor block #1 was also used with a flat indenter to obtain readings. The flat indenter was able to provide measurements for a certain distance after material contact was lost, but it was possible that the edge effects of pressure magnification caused problems when the materials were in contact. This information was used to design the next sensor block. The most notable feature of Figure 3.9 is the sensor’s output waveform, shown in white. Because of the circuitry used, the output voltage of the sensor was linearly proportional to the current flowing through the sensor. Therefore, the white waveform was the current flowing through the contact, and
the red waveform was the voltage applied across the contact. The current waveform was not sinusoidal as it crossed the zero voltage axis.

One cycle of the waveforms of Figure 3.9 is shown in Figure 3.10. This shows the relationship between the applied voltage and the current flowing through the sensor material. It is important to note that in the curves of Figure 3.9 and 3.10, the cycle time was 0.05s, which means that the load over this time was essentially constant. Therefore, over this small time period, the current was only affected by the applied voltage, and not by the load. In Figure 3.10, the current waveform was not sinusoidal near values of zero voltage. It was observed that this was only the case when there was force pushing the surfaces together. When the surfaces were separated by a few microns, as determined using a 10g load cell, a small amount of current would still flow through the junction, but in this situation, the current waveform was sinusoidal as it crossed the zero axis.

3.4.2 - Measurements with Sensor Block #2

There were several notable features of the design of sensor block #2. There were wells to contain the fluid, and the tolerance of the walls ensured vertical alignment of the two different radius flat indenters. Two different sized sensing points were also used, to determine the effect of the sensing point’s area. Most importantly, the contact configuration was changed, such that the metal surface
completely overlapped the sensor’s surface. By preventing the edges of the metal plunger from contacting the sensor material, any inaccuracies due to point loading from slight misalignments were eliminated.

As a final topic of discussion, the following two paragraphs deal with the technical issue of load cell deflection and its importance in this research. All load cells are built upon strain gage technology. The electrical signal output by a load cell is produced by a set of strain gages mounted inside the device to a strain member. As the external load changes, it strains the internal components to which the strain gages are mounted, thus producing the output signal. For this reason, a load cell has a property called deflection (or compliance). As the external load increases, the free-end of the load cell (to which the plunger is attached) deflects a small amount, by design, to produce the internal strain necessary to measure the load. For most applications, the deflection is more of a formality when post-processing the data. However, for this research, load cell deflection is a very important issue. The 50 gram load cell is advertised to deflect 0.004” at it’s rated output of 50g – although this must be calibrated often for ultimate precision. The addition of a metallic plunger or polymer pen, and the means by which they are attached, also changes the overall deflection constant of the load cell-attachment system. Over the 50 gram range of compressive load, the cross-head of the materials testing machine will move over 100μm further in the compression direction after initial contact is made, whereas the contact surface will only move a distance of 5-8μm – due to compressive strain of the polymer surface features. Although many of the experimental technicalities
such as this have not been noted, this topic bears consideration because it shows that the UHMWPE contact sensor technology is inherently accurate and sensitive enough to show when the load cell deflection needs to be re-calibrated again. In early testing, the load cell outputs were corrected using the documented deflection constants from the manufacturer. However, the data would often exhibit a change in plunger displacement rate without the contact sensor’s output showing the same. It was found that manually calibrating the deflection constant fixed the inconsistencies. Many more occurrences of the data initially indicating erratic plunger movement have happened during this research. Each time this was noted, the load cell’s deflection constant would be re-calibrated, and each time it was found that it had drifted since the last calibration. Upon the application of the updated deflection constant to the data, it was found each time that the contact sensor’s output was correct all along, having never been calibrated. When an incorrect deflection constant is used, it can skew the calculated plunger displacement by many microns, which is quite significant when a ±1\(\mu\)m mechanical control accuracy was generally able to be attained.

3.4.3 – Hydrostatic Sensor Readings (Measurements with Sensor Pin #1)

The piece-wise regression analysis of the data obtained during static measurements with the sensor showed that the linear fit region of the data occurred at sensor positions of less than +1\(\mu\)m. The load cell indicated that this
linear fit region of the data occurred for loads greater than about 50g. The load cell also indicated that the sensor’s RMS output voltage was linear with respect to load for all sensor measurements. For the contact geometry of this specific set of experiments, a load of 50g correlated to a Hertzian point contact stress of about 2 MPa, with the maximum Hertzian point contact stress being slightly less than 5 MPa. Thus, the linear portion of the sensor’s output seen in Figure 3.13 occurred at Hertzian point contact stress levels that would be expected to be within the linear elastic region of the material’s compressive stress-strain curve.

The sensor’s RMS output voltage as a function of load as determined from the load cell can be seen in Figure 3.16. An $R^2$ value of 0.9994 shows a good linear fit of the data. In fact, it was found during this research and during previous research that the sensor’s output always has a linear relationship to the applied load. The linear fit of the data in Figure 3.16 shows a slope of 1.5 mV/g. This constant was observed many times during this research when the excitation to the sensor was 10Vp-p at 20Hz. The linear relation of the sensor’s output to the applied load suggests that during the linear elastic portion of the material’s stress-strain curve there should also be a linear relationship between the sensor’s output and the sensor’s vertical position – i.e. strain. Indeed, this was found to be the case, as seen in Figure 3.13. It should be noted that the compressive stress-strain curve for UHMWPE is linear within the elastic region of the material at stress levels above a minimum threshold level. In fact, this is the portion of the stress-strain curve from which the compressive modulus can be determined. In UHMWPE, there is a non-linear relation between the
compressive stress and strain at very low levels of stress. This commonly known behavior of UHMWPE was confirmed for the composite sensor material as well during previous research [Clark, MS Thesis, 2003]. Therefore, the nonlinear portions of the sensor’s rms output data may be, in part, related to this. For sensor positions between +1\( \mu \text{m} \) and +3.5\( \mu \text{m} \), the data seen in Figure 3.15 also followed a power law fit. The data within this range of sensor positions occurred when the Hertzian contact stress values ranged from a high of slightly less than 2MPa to a low of about 1MPa. This is in the low level stress range of the nonlinear portion of the compressive stress-strain curve for UHMWPE.

For sensor positions above +5\( \mu \text{m} \), there was no contact registered by the load cell. Therefore, in Figure 3.15, the log-log linear region seen for sensor position values above +5\( \mu \text{m} \) represents the sensor’s output when there is no physical contact between it and the metal counter-face. Thus, it should be expected that when measuring the film thickness under hydrodynamic conditions, the sensor’s rms output voltage should follow a power law.

When the cumulative results seen in Figures 3.13, 3.14, and 3.15 are considered, the data suggest that the sensor’s output should be linear under boundary lubrication conditions, and should show a power law relationship under mixed lubrication and full fluid film lubrication (HL) conditions. Moreover, the slope of the log-log curve, which is called the proportionality constant for a power law fit, would be expected to differ between mixed lubrication and full film lubrication.
For sensor positions between +5µm and +3.5µm of Figure 3.15, there were not enough data points to determine a fit. Because this occurred just as the load cell was registering contact, the deflection of the load cell might have caused some inaccuracies in the calculation of the sensor position. The very small values of load measured by the load cell within this region of the data means the deflection of the load cell would be expected to be outside of its linear range. The larger standard deviations seen in the data within the lighter range of load measured could likely be due to the non-linear nature of the load cell’s deflection at these very small values of load.

3.5 – CONCLUSIONS

Under hydrostatic conditions, the curve of the sensor’s output voltage plotted against the sensor’s vertical position showed a piecewise relationship. There were 3 main regions seen within this data. A linear region was noted for the higher values of the sensor’s output voltage where the sensor’s position was less than +1µm. The other two regions of the data were linear on a log-log scale, showing a power-law fit. The power-fit region of the data for higher output voltages was different than for the lower output voltages. The load cell indicated that the lower voltage power-fit data was obtained when there was no physical contact between the two surfaces. It can be concluded that the relationship between the sensor’s output voltage and the surface separation exhibits a power-law relationship when the surfaces are fully separated, and exhibits a linear relationship at higher contact loads.
4.0 - HYDRODYNAMIC SENSOR READINGS

4.1 – INTRODUCTION

During the previous chapter of this dissertation, the hydrostatic measurements obtained showed that there were three distinct regions of the data, which would seem to correspond to the different modes of lubrication. However, because the lubricant was not pressurized for any of the measurements obtained in the previous chapter, any surface separation attributed to elastic deformation of the polymer would not be accounted for. Therefore, this chapter addresses hydrodynamically generated lubricant films.

4.2 – MATERIALS AND METHODS

4.2.1 – Experimental Friction Data

The purpose of this section was to generate a Stribeck curve for the specific materials of the tribological contacts used during this dissertation, those being the UHMWPE composite sensor material sliding against mirror-polished 316L stainless steel (SS). As the modulus of elasticity of 316L SS is similar to that of Co-Cr alloy commonly used in total joint arthroplasty (193 GPa and 195-219 GPa respectively), contact properties against UHMWPE are assumed to be similar as well. In order to accomplish this, the dynamic coefficient of friction was measured at different sliding speeds that were calculated to be sufficient to cover
the full range of lubricating regimes, from boundary lubrication, through mixed lubrication, into full-film hydrodynamic lubrication (HL).

4.2.1.1 – Fabrication of Sensor Pin #1

Sensor Pin #1, as seen in Figure 4.1, had a spherical tip with a radius of curvature of 0.1m. The pin was constructed such that it consisted mostly of virgin UHMWPE, with the composite sensor material exposed on the tip of the pin as a 0.0625” diameter sensing point. The sensor material extended internally up the shaft of the pin, connecting with a 0.2” band of the sensor material that transected the pin 0.4” up from the tip. The total length of the pin was 1” and the diameter was 0.5”. The tip of the pin was also tapered to allow for a better view of the contact. The tip’s radius of curvature of 0.1m also gave a small allowance for vertical misalignments while still ensuring a point contact.

![Figure 4.1 - Sensor Pin #1](image)

For the final machining of sensor pin #1, a 1/8” diameter ball mill was used. A solid model of the pin was used in a CNC machining software package
(VisualMill 5.0 Basic, MecSoft Corporation, Irvine, CA) to generate a tool path to mill the pin from a solid block that was prepared by a multi-step compression molding technique described in chapter 2 of this dissertation. Multiple roughing steps were taken to allow the finishing step to remove the optimal amount of material. For the milling of sensor pin #1, a 0.01" margin of material was left for removal during the final finishing operation. A 3-axis parallel-finishing tool path was used for milling the tip of the pin. For this tool path, the tool traveled in a linear x-y motion, while the z axis followed the vertical contour of the pin. Material was removed on each pass, thus, both conventional cutting (up-cut), and climb-cutting (down-cut) were performed on the part. The input parameters for the parallel-finishing tool path were a linear feed rate of 15mm/s (35.4 in/min), with a tool speed of 15,000 rpm, and a step-over distance of 0.0025” per pass.

4.2.1.2 – Cross-shear Machine Settings

To achieve sliding speeds necessary to generate lubricating films, this research used a multi-axis wear testing machine developed in the biotribology laboratory at Clemson University [DesJardins, PhD Dissertation, 2006]. This machine, referred to in this dissertation as the cross-shear machine, employs a table connected to two servo-motors by precision-ground worm gears, each at right angles, forming an x and y axis allowing for any motion desired in the x-y plane.

Sensor pin #1 was connected to the upper shaft with a short piece of #6-32 NC threaded rod and the upper shaft adapter collar. The threaded rod was
secured to the adapter collar for the upper shaft using 4 copper-tipped locking screws for which the adapter collar of the upper shaft was designed to accept. The adapter collar was then secured to the upper shaft with a short section of threaded rod and some locknuts. The whole assembly of the upper shaft, adapter collar, and sensor pin #1, was weighed and found to be 184g. Thus, the normal force applied during this experiment was 1.8N. The final length of the assembly was such that the moment arm calibration ratio used by the machine’s control software was a value of 3.025.

During the friction experiment, a mineral oil lubricant with a fairly high viscosity was used. The lubricant was a certified viscosity standard (N350, Cannon Instrument Company, State College, PA) with a room-temperature viscosity of 1 Pa*s (1,000 cP). To put this in perspective, this is roughly 100 times more viscous than the 50% bovine serum used as the standard lubricant for orthopedic bearing wear testing. Because the viscosity standard used here was a Newtonian fluid, and bovine serum is a non-Newtonian fluid, this difference in viscosities is even higher at very high shear rates. The metal counter face used during this experiment was a 2.5” diameter disk of stainless steel that was polished to a mirror finish. The disk was attached to the table of the cross-shear machine to form the lower bearing surface. Paper shims were used to level the disk in conjunction with a dial micrometer. The disk was mechanically fastened and locked into place being horizontally level to within 0.001”.

The motion pathway chosen for this experiment was a continuous circular pathway such that the entraining speed would remain constant while
measurements were made. The diameter of the circular pathway was 30,000 encoder counts, referred to from this point forward as 30k cts, which corresponds to a diameter of 38.1mm. The input to the control software was in units of motor rpm, therefore, the speeds chosen for this experiment, in units of rpm, were 2, 5, 10, 25, 50, 75, 100, 150, 200, and 300 rpm. This corresponds to a sliding speed that varied from 4.2 mm/s to 170 mm/s in 10 increments. The total time of each test varied, but six revolutions of the pathway were completed. The cross-shear machine’s control software collected friction measurements from the machine at a rate of 1kHz, and averaged the data and stored to disk with 100 data points per second.

4.2.1.3 – Analysis of Friction Data

The raw data output by the cross-shear machine’s control program was analyzed using Microsoft Excel spreadsheet software. Within the spreadsheet an algorithm was developed to locate friction readings taken at the same area of the circular pathway during different cycles. The 6-cycle average and standard deviation of this friction data was then calculated. Next, a Striebeck curve was generated from the friction data. The sliding speed, fluid viscosity, applied load, and point contact radius were used to calculate the dimensionless parameter called the Sommerfeld number (S), also sometimes referred to as the bearing characteristic number. The Sommerfeld number for sliding point contacts is defined as the inverse of the applied load multiplied by the product of the viscosity, sliding speed, and point contact radius [Auger PhD Dissertation 1992].
Then, the specific film thickness, known as $\lambda$, was calculated with the surface roughness values and the minimum film thickness as determined by the Hamrock-Dowson equation [Hamrock and Dowson 1978].

4.2.2 – Hydrodynamic Calibration Data

4.2.2.1 – Fabrication of Sensor Pin #2

Sensor pin #2 was fabricated in a similar manner as sensor pin #1 described above. However, the tip of sensor pin #2 had a radius of curvature of 10m. Because of the larger radius of curvature of the tip, it was also not tapered. The larger radius of curvature was necessary to be able to generate the full range of lubrication modes with a less viscous fluid that was determined to produce the most reliable electrical readings. Like sensor pin #1, the diameter of sensor pin #2 was also 0.5”.

For the final machining of sensor pin #2, a 1/8” diameter ball mill was used. Multiple roughing steps were taken to allow the finishing step to remove the optimal amount of material. For the milling of sensor pin #2, a 0.01” margin of material was left for removal during the final finishing operation. A 3-axis parallel-finishing tool path was used for milling the tip of the pin. For this tool path, the tool traveled in a linear x-y motion, while the z axis followed the vertical contour of the pin. Material was removed in only one cutting direction so as to provide for a higher degree of accuracy during the machining step. The type of cut was a conventional cut, also known as an up-cut. The input parameters for
the parallel-finishing tool path were a linear feed rate of 15mm/s (35.4 in/min), with a tool speed of 15,000 rpm, and a step-over distance of 0.00125” per pass.

4.2.2.2 – Surface Roughness Measurements

In order to calculate the specific film thickness (\(\lambda\)), surface roughness values were needed for the mirror-polished stainless steel surface and for the UHMWPE composite sensor material. The 316L stainless steel bars 1 inch wide by 4 inches long by 0.375 inches thick had been previously polished to a mirror finish using silicon carbide paper with a decreasing grit of 200, 400, 600, 800, 1000, 1200, and 2000 on an oscillating grinder (Exakt Vertriebs GMBH, Germany). The UHMWPE composite was prepared as described elsewhere in this dissertation. A non-contact profilometer (NT2000, Veeco, Tucson, AZ) was used to measure the surface roughness of unworn samples. Measurements were made at magnifications ranging from 25X to 3X, and the average roughness (\(R_a\)), root mean square roughness (\(R_q\)), maximum profile height (\(R_T\)), and 6 point maximum profile height (\(R_z\)) were calculated for each scan.

4.2.2.3 – Cross-shear Machine Settings

For this experiment, the cross-shear machine was used to produce a linear reciprocating sliding motion. Thus, the entraining speed was not constant, and at each end of the pathway there was a start-stop motion. The pathway was a distance of 50k cts (63.5 mm) with \(x = 0\) cts in the middle of the pathway, such that \(x\) varied from –25k cts to +25k cts. The acceleration was set at 100 rotations
per second per second, allowing the control speed of each test to be reached well within the first 1mm of each end of the pathway. The whole assembly of the upper shaft, adapter collar, and sensor pin #1, was weighed and found to be 184g. Thus, the normal force applied during this experiment was 1.8N. Like the previous section, the speed was the only variable for this experiment, and it was input to the control software in units of motor rpm. The speeds used during this experiment, in units of motor rpm, were 2, 5, 8, 10, 25, 50, 75, 100, 150, 200, and 300 rpm. The lubricant used was a fluorocarbon specialty fluid used for cooling of electronics (Fluorinert FC-70, 3M Electronics Markets Materials Division, St. Paul, MN), with the specifications as listed in Table 4.1.

<table>
<thead>
<tr>
<th>3M Fluoroinert FC-70 Electronic Liquid</th>
</tr>
</thead>
<tbody>
<tr>
<td>Viscosity (mPa*s)</td>
</tr>
<tr>
<td>-------------------</td>
</tr>
<tr>
<td>24</td>
</tr>
</tbody>
</table>

Table 4.1 – Physical properties of fluorocarbon liquid (from manufacturer supplied data sheet)

4.2.2.4 – Sensor Specific Settings

In order to collect and record the data from the sensor, a 1 MHz, 16-bit analog I/O data acquisition system (Personal Daq 3000, Iotech Inc., Cleveland, OH) was used. The system interfaced with a laptop computer via high-speed USB 2.0 interface. The system was controlled using custom software programs written in a graphical, high-level language designed for data acquisition
applications (LabView 8.0, National Instruments, Austin, TX). The software allowed control of both the analog output and analog input features of the data acquisition system. The analog output was used as the excitation for the sensor, applying a 10V peak-to-peak (3.5 V\text{RMS}) sinusoidal signal at a frequency of 20 Hz. Two of the analog input channels were used to collect both the sensor’s output voltage ($V_o$) and the excitation voltage (or reference voltage, $V_{ref}$). These signals were collected at a rate of 10kHz and stored to the hard drive to allow for further analysis.

The electronics interface with the sensor consisted of a fairly simple electronic circuit designed to accurately measure the current flowing through the sensor and output a voltage proportional to that current. The circuit, commonly referred to as a current-to-voltage converter, is based on an operational amplifier (op-amp) with its non-inverting input grounded. The sensor was connected to the inverting input of the op-amp, thus allowing the current flowing through the sensor to see a “virtual ground”, allowing for the most accurate measurement of current that varies over a large range. Therefore, the output voltage from the sensor ($V_o$) is a direct measure of the current flowing through the sensor due to the applied reference voltage.

The sensor data acquisition software was configured for this experiment to output time, rms output voltage ($V_{o\ RMS}$), and DC output voltage ($V_{o\ DC}$). These values were obtained by first averaging the raw data array by a factor of 5, and then calculating the 3-cycle rms and DC (arithmetic average) values of the $V_o$.
waveform. The 3-cycle average in this case refers to three cycles of the 20 cycle per second excitation voltage applied to the sensor.

The procedure used for collecting the data from the sensor was to first start the linear reciprocating motion of the cross-shear machine at the specific speed of each test. Once the reciprocating motion was occurring, the sensor's data acquisition was started so that collection always began when the pin was moving from right to left. Therefore, the first spike seen in the \( V_o \) waveform was due to the stop-start motion at the left side of the pathway.

4.2.2.5 – Analysis of data

The data that was output from the sensor data acquisition software was imported to a spreadsheet program for further analysis. The algorithm used in the spreadsheet program used the time data to determine the spatial location on the motion pathway of the sensor's output data. The point on the pathway that was chosen was 12.7 mm from the left end of the pathway. This corresponded to an encoder count of +15k, where 0 is the center, +25k cts is the left side of the pathway, and -25k cts is the right side of the pathway. Therefore, the pin has traveled 12.7mm before crossing the measurement point when traveling from left to right, and has traveled 50.8mm before crossing the measurement point when traveling from right to left. In both cases, there has been ample time and sliding distance for the lubricant film to become established. The spreadsheet program calculated the average and standard deviation of \( V_o \) \( \text{RMS} \) at each crossing of the described measurement point giving at least 6 readings for each speed setting.
Because of the simple linear motion, the Hamrock and Dowson formula of point contact EHL [Hamrock and Dowson, 1978] can be used to calculate the minimum film thickness. This equation is not valid close to each end of the pathway where the direction reverses, but it is valid for the rest of the pathway, including the measurement point at the x-axis value of +15k cts. The Hamrock and Dowson equation for minimum film thickness for a point contact in EHL is:

\[ h_{\text{min}} = \frac{2.8 U^{0.65} R}{W^{0.21}} \]

where \( R \) is the radius (m), and \( U \) is the speed parameter and is defined as follows:

\[ U = \frac{\eta u}{E' R} \]

where and the load parameter, \( W \), is defined as:

\[ W = \frac{w}{E'R^2} \]

and \( E' \) is defined as:
where $E_1$ and $E_2$ are the modulus of elasticity of each counterpart, in units of Pa.

Using these equations, and the other experimental parameters, $h_{min}$ was calculated for each speed setting for which measurements were taken during this experiment. This allowed the $V_o \text{RMS}$ readings for each speed to be equated to a film thickness value as calculated by the above equations.

The average and standard deviation of $V_o \text{RMS}$ for each speed setting were then plotted against this film thickness value, which was assigned the title $h_s$, which means the sensor-measured film thickness. Error bars on each data point were used to graphically represent the standard deviation. A piece-wise regression analysis of the data was performed, and the best-fit equations and $R^2$ values for each segment were determined.

The hydrodynamic data was also compared to previously collected static data from chapter 2 of this dissertation. Both sets of data were plotted on the same graph, with the x-axis representing $H_s$. This allowed further insight into the validity of the calibration equations obtained from the hydrodynamic data.

### 4.2.3 – Relevant Motion Pathway Hydrodynamic Data
In order to use the sensor to actually measure a lubricating film in a clinically relevant motion pathway, the sensor data had to be collected at the same time as the cross-shear control computer was controlling the motion pathway and collecting the friction and position data. Then, during later analysis, the data from both computers had to be combined and aligned in the same data set. Special attention was paid to the procedure to allow this to happen.

The friction data that was collected consisted of the coefficient of friction, the X axis position, and the Y axis position. These were collected at a rate of 1 kHz. The cross-shear machine’s software then averaged this data to 100 data points per second for the output file. The sensor data was collected at a rate of 10 kHz. The reference voltage applied to the sensor was a sinusoidal, 10V peak-to-peak, 20Hz signal. The 3-cycle rms and DC sensor output voltage were then calculated for this data.

The procedure for running the experiments consisted of first starting the sensor computer’s data acquisition while the sensor pin was at the starting point of the motion profile, but before motion was started. Next, the motion profile was started, with a pre-set number of cycles to perform (20 cycles). Then, the friction data collection on the cross-shear computer was started. The number of friction data cycles collected was such that the friction data was still acquiring when the motion profile stopped after the completion of 20 cycles. Last, the sensor computer’s data acquisition was stopped. The end result of this procedure was that the total sensor data showed the start and end points of the motion profile. The friction data showed the end point of the motion profile, and also contained
the X and Y positions of each friction measurement. The sensor data and the
friction data were able to be aligned by matching the time-points from each data
set where the motion profile started. Then each data set was re-sampled in
another custom LabView program and re-combined such that the final output
was the six-cycle average and standard deviation of the sensor’s rms output
voltage, the coefficient of friction, the X position, and the Y position, all as a
function of the percentage of one cycle of the motion pathway.

The combined and six-cycle averaged data was then plotted against the
percentage of the motion pathway cycle. The sensor’s rms output voltage was
converted into a measure of film thickness in units of μm using the calibration
equations obtained in this study.

4.3 - RESULTS

4.3.1 – Experimental Friction Data

The surface roughness measurements of the stainless steel bar and the
UHMWPE composite sensor material are shown in Table 4.2. The units of
measure were nm. Because of the high roughness of the UHMWPE sensor
material, its roughness will be referred to in units of μm. It is also important to
note that the polished stainless steel’s roughness was nearly 3 orders of
magnitude smoother than that of the UHMWPE sensor material.
Table 4.2 – Average roughness values for polished stainless steel and UHMWPE composite sensor material

Table 4.3 shows the experimentally measured friction data. During this experiment, the control input for speed was motor rpm. Based on the cross-shear machine’s specific parameters, the sliding speed was calculated from the motor rpm.

Table 4.3 – Experimental friction data of motor speed and average coefficient of friction ± standard deviation (n=6)

The experimentally gathered data shown in Figure 4.2 is the average dynamic coefficient of friction of 6 cycles as collected at the same position on the circular pathway, as indicated by the red box in the figure. The sliding speed was
the only variable for this set of experiments. The error bars on the data represent one standard deviation. At low sliding speeds the friction was about 0.15. At moderate sliding speeds, the friction reached a low value of about 0.06. After this low value, the friction increased with speed, eventually reaching a higher value than the friction at low speed.

Figure 4.2 – Average dynamic coefficient of friction as a function of sliding speed for 6 cycles where red box indicates measurement position on the circular pathway

The friction data was then plotted against the dimensionless parameter known as the Sommerfeld number. This plot of the coefficient of friction as a function of the Sommerfeld number is called a Stribeck curve, as seen in Figure 4.3.
Figure 4.3 – Stribeck curve for experimental friction data

The data shown in Figure 4.4 is the same average and standard deviation dynamic coefficient of friction shown in the previous two figures. The independent axis in Figure 4.4 was converted to specific film thickness ($\lambda$). Additionally, Figure 4.4 was marked to indicate the different regions of lubrication evident in the data. Boundary lubrication was seen for $\lambda$ values less than about 1.2. For values of $\lambda$ between 1.2 and about 3.5, mixed lubrication is indicated by the data. For values of $\lambda$ greater than about 3.5, elastohydrodynamic lubrication (EHL) is possible, and either it or hydrodynamic lubrication (HL) are the dominant lubrication modes. For values of $\lambda$ greater than about 5.5, the data indicate that HL, also called full fluid film lubrication, is the dominant mode of lubrication.
4.3.2 – Hydrodynamic Sensor Readings

The speed, $h_s$, and average rms output voltage ± one standard deviation data collected during the hydrodynamic linear reciprocating experiments are shown in Table 4.4. The independent variable for the experiment was the motor speed, which directly yielded the sliding speed. The value $h_s$ was calculated based on the motor speed and other parameters of the experiment. The sensor’s rms output voltage was measured for each speed as shown in Table 4.4.
Table 4.4 – Data collected during controlled hydrodynamic sensor experiments

<table>
<thead>
<tr>
<th>Speed (RPM)</th>
<th>( h_s ) ((10^{-6} \text{ m}))</th>
<th>( V_o \text{ RMS} ) ((\text{V}))</th>
<th>( \pm \text{ std. dev.} ) ((\text{V}))</th>
</tr>
</thead>
<tbody>
<tr>
<td>300</td>
<td>8.70</td>
<td>0.0073 ± 0.00047</td>
<td></td>
</tr>
<tr>
<td>200</td>
<td>6.68</td>
<td>0.0079 ± 0.00067</td>
<td></td>
</tr>
<tr>
<td>150</td>
<td>5.54</td>
<td>0.0083 ± 0.00048</td>
<td></td>
</tr>
<tr>
<td>100</td>
<td>4.26</td>
<td>0.0118 ± 0.00093</td>
<td></td>
</tr>
<tr>
<td>75</td>
<td>3.53</td>
<td>0.0140 ± 0.00061</td>
<td></td>
</tr>
<tr>
<td>50</td>
<td>2.71</td>
<td>0.0208 ± 0.00036</td>
<td></td>
</tr>
<tr>
<td>25</td>
<td>1.73</td>
<td>0.0372 ± 0.00548</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>0.95</td>
<td>0.0614 ± 0.00618</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>0.61</td>
<td>0.2694 ± 0.02216</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>0.33</td>
<td>0.3834 ± 0.02848</td>
<td></td>
</tr>
</tbody>
</table>

The data from Table 4.4 can be seen plotted in Figure 4.5. The standard deviation of each data point is represented by the error bars on the plot. As seen in Figure 4.5, the rms output voltage was highest when \( h_s \) was \(1\mu\text{m} \) or less.
Figure 4.5 – Sensor readings obtained during linear reciprocating hydrodynamic experiments

Figure 4.6 shows a piece-wise regression analysis of the plot of Figure 4.5. The vertical lines on the figure represent the divisions between the lubrication modes. The red vertical lines are the divisions as determined from the experimental friction data of Figure ch3.4. The green vertical lines are the divisions that can be inferred from this data. Both divisions are between the same data points. As indicated in the figure, the data was linear in the region where boundary lubrication is expected. Within the region where mixed lubrication is expected, a power law fit of the data resulted in a high $R^2$ value. The coefficient of determination for the power law fit of the data within the region
where full HL is expected was even higher than that of the mixed lubrication region.

Figure 4.6 – Hydrodynamic sensor data with lubricating mode divisions indicated and piece-wise curve fit information displayed

Figure 4.7 shows just the linear portion of the data of Figure 4.6. The error bars on the graph indicate one standard deviation of the data. As shown in the figure, the linear portion of the data has a good $R^2$ value and occurs in the region of boundary lubrication.
Figure 4.7 – Linear portion of hydrodynamic sensor data

The non-linear portion of the hydrodynamic data is seen in Figure 4.8, which has a linear scale on both axes. It can be seen from the figure that the values of $h_s$ for which this data occurred are in the mixed lubrication regime and the HL regime.
Figure 4.8 – Power fit portion of hydrodynamic sensor data on a linear scale

The data contained in Figure 4.9 is the same as that of Figure 4.8 except that there is a logarithmic scale on both axes (log-log scale). As seen in the figure, the power-fit data is linear when plotted on a log-log scale. As indicated by the vertical lines in the figure, the film thickness expected for the division between mixed lubrication and full HL is about 5μm, which is the same film thickness value that the change in the power law trend of the data is seen.
In order to compare the hydrodynamic data with the static data collected in chapter 2 of this dissertation, the results of each were plotted on the same scale. Figure 4.10 shows the linear portion of each data set. The average data points, along with error bars representing one standard deviation of the average, as well as the lines of best fit can be seen in the figure.
Figure 4.10 – Comparison of linear portion of static and hydrodynamic data

Figure 4.11 shows the power fit portion of both the static and the hydrodynamic data on a log-log scale. The averaged data points, along with error bars equivalent to one standard deviation and the lines of the power-law fit are seen in the figure. Additionally, the equations of best fit and the $R^2$ values are also shown in the figure. Figure 4.11 shows the similarities and differences between the static data and the hydrodynamic data, to be discussed below.
Figure 4.11 – Comparison of power fit portion of static and hydrodynamic data on a log-log scale

The equations of best fit for the hydrodynamic data are calibration equations that can be used to determine the film thickness from the measured sensor rms output voltage. As seen best in Figure 4.7, the linear portion of the hydrodynamic data was valid for $V_{o \text{ RMS}}$ values greater than 0.06136 V, or 61.36 mV. To determine $h_s$ for $V_{o \text{ RMS}}$ values greater than 61.36 mV, the calibration equation

$$h_s = \left[ \frac{(V_0 - 0.5693)}{-0.524} \right]$$

(4.1)
was used, where $h_s$ is in units of $\mu$m and $V_o$ is in units of V. Equation 3.1 is valid within the boundary lubrication regime, which is measured by the sensor when $V_{o \text{ RMS}}$ is greater than 61.36 mV. When the sensor’s rms output voltage is less than 61.36 mV, but greater than 8.32 mV, the calibration equation used to determine the film thickness is

$$h_s = \left[ \left( \frac{V_o}{0.0629} \right)^{-0.8628} \right]$$

(4.2)

where $h_s$ is in units of $\mu$m and $V_o$ is in units of volts. Equation 3.2 is valid for mixed lubrication, which is measured by the sensor when $V_{o \text{ RMS}}$ is between 61.36 mV and 8.32 mV. To determine $h_s$ for $V_{o \text{ RMS}}$ values less than 8.32 mV, the calibration equation

$$h_s = \left[ \left( \frac{V_o}{0.0135} \right)^{-3.5511} \right]$$

(4.3)

was used, where $h_s$ is in units of $\mu$m and $V_o$ is in units of volts. Equation 3.3 is valid for the HL regime, which is measured by the sensor when $V_{o \text{ RMS}}$ is less than 8.32 mV.
4.3.3 – Relevant Pathway – Figure 8 Pattern

The data in Figure 4.12 shows the motion profile and the points of interest. The points of interest are labeled A through H, and are the basic control points of the motion pathway. The pathway started and ended at point A, although there was no stoppage or pause of motion whatsoever, as one cycle blended into the next in a continuous fashion. The order of the lettered points is the same order in which they were encountered throughout the motion pathway, and the table in the bottom left of Figure 4.12 shows the % of cycle location of each point. Arrows are shown on the pathway to indicate the sliding direction of the pin. It is also worth noting that in reality, the pin was stationary and the flat metal slid underneath the pin. However, for simplification of analysis, the pathway is viewed and represented here from the perspective of the pin moving.
Figure 4.13 shows the 6 cycle average of the sensor’s rms output voltage as a function of the % of one cycle of the figure 8 motion pathway. The error bars in the figure indicate one standard deviation of the average data. The vertical division lines represent the locations of the points of interest in the motion pathway where the acceleration is largest or the motion pathway crosses itself. The vertical lines are labeled on the bottom with the letter corresponding to the point and on the top with a description of the point. It can be seen in Figure 4.13 that there are two peaks in the curve, both occurring just after the x-axis ends of the motion pathway seen in Figure 4.12. The peaks also subside before the next points of interest are reached. Thus, the peaks occur after the X ends of the pathway and before the Y maximum values of the pathway.
Figure 4.13 – Average (n=6) sensor rms output voltage for one cycle for a normal load of 184g

Figure 4.14 shows the 6 cycle average coefficient of friction that was measured by the cross-shear machine. The error bars in the figure indicate one standard deviation of the data, and the vertical lines show the location of each point of interest on the motion pathway. The curve has a clearly defined peak that occurs at 35% of the cycle, which is the same position as point D of the motion pathway, which corresponds to the +X maximum. Thus, the friction decreases right after point D is passed. Similarly, the friction decreases right after point H is passed, which is the –X minimum, although the overall friction is much less on the –X side of the pathway.
Figure 4.14 – Average (n=6) friction coefficient for one cycle for a normal load of 184g

The speed of the motion pathway as it varies over one cycle for the 184g data can be seen in Figure 4.15. Again, the vertical lines representing the points of interest of the motion pathway are shown in the figure. The speed has a maximum value of about 50mm/s, with this value occurring at both pathway crossing points. The minimum speed during the cycle is slightly more than 10 mm/s, occurring just before the X ends of the motion pathway, which makes sense because that is where the direction changes most abruptly. It is important to note that there is never any stopping motion occurring on the pathway, with one cycle blending continuously into the next. Therefore, the absolute minimum speed affecting the friction measurements and the sensor measurements is slightly higher than 10 mm/s.
The film thickness that was measured by the sensor is seen in Figure 4.16. Again, vertical lines depicting the points of interest of the motion pathway are shown in the figure. Additionally, horizontal lines are shown, giving the approximate divisions between lubrication modes as determined by both the experimentally obtained friction data and the hydrodynamic sensor calibration data. The mode of lubrication is also shown as a transparent background in the region of the figure that it occurred. The first thing noticed about the data set is that boundary lubrication was never measured for a normal load of 184g. Also, 2 minimum film thickness peaks are seen in the data. These minimum values occurred just after the X ends of the motion pathway were crossed. Thus, the minimum peaks occurred as the pin was accelerating just after changing X directions. The lowest minimum peak dips well within the mixed lubrication regime, occurring just after the +X end of the motion pathway. The maximum
film thickness values measured occurred just after the cross-path point of the motion pathway, just after the speed had reached its maximum values of the cycle.

Figure 4.16 – Film thickness measured by the sensor for a normal load of 184g

Figure 4.17 shows the 6 cycle average of the sensor’s rms output voltage as a function of the % of one cycle of the figure 8 motion pathway for a normal load of 684g. The error bars in the figure indicate one standard deviation of the average data. The vertical division lines represent the locations of the points of interest in the motion pathway where the acceleration is largest or the motion pathway crosses itself. The vertical lines are labeled on the bottom with the letter corresponding to the point and on the top with a description of the point. It can be seen in Figure 4.17 that there are two peaks in the curve, both occurring just after the x-axis ends of the motion pathway seen in Figure 4.12. The peaks also
subside before the next points of interest are reached. Thus, the peaks occur after the X ends of the pathway and before the Y maximum values of the pathway.

![Graph showing sensor rms output voltage](image)

Figure 4.17 – Average (n=6) sensor rms output voltage for one cycle for a normal load of 684g

Figure 4.18 shows the 6 cycle average coefficient of friction that was measured by the cross-shear machine. The error bars in the figure indicate one standard deviation of the data, and the vertical lines show the location of each point of interest on the motion pathway. The coefficient of friction rose sharply starting just before the +X end of the pathway. The friction decreased from the +X point to the crossing path point, and then increased again until reaching the –X end of the pathway. At the –X end of the pathway, the friction dramatically fell to a very low value. Before reaching the Y max value at point A, the friction was...
increasing again, where it continued to increase until just before the cross-path point.

Figure 4.18 – Average (n=6) friction coefficient for one cycle for a normal load of 684g

The speed of the motion pathway as it varied over one cycle for the 684g data can be seen in Figure 4.19. Again, the vertical lines representing the location of the points of interest of the motion pathway are shown in the figure. The speed had a maximum value of slightly more than 50mm/s, with this value occurring just before both pathway crossing points. The minimum speed during the cycle was slightly more than 10 mm/s, occurring just before the X ends of the motion pathway, which makes sense because that was where the direction changed most abruptly. It is important to note that there is never any stoppage motion occurring on the pathway, with one cycle blending continuously into the next. Therefore, the absolute minimum sliding speed that affected the friction
measurements and the sensor measurements was slightly faster than 10 mm/s, so, unlike many linear reciprocating motions, there was never any opportunity for the lubricating film to completely break down.

Figure 4.19 – Speed as a function of one cycle for a normal load of 684g

The film thickness that was measured by the sensor for a normal load of 684g is seen in Figure 4.20. Again, vertical lines depicting the points of interest of the motion pathway are shown in the figure. Additionally, horizontal lines are shown, giving the approximate divisions between lubrication modes as determined by both the experimentally obtained friction data and the hydrodynamic sensor calibration data. The mode of lubrication is also indicated by arrows and labeling within the figure. The first thing noticed about the data set is that boundary lubrication was just barely reached for less than 5% of the cycle, occurring just after the + X end of the motion pathway. A second minimum film
thickness peak was also seen just after the – X end of the motion pathway, and while the value was under 2µm, the film thickness was not quite into the boundary lubrication regime. Thus, the minimum peaks occurred as the pin was accelerating just after changing X directions. As seen in Figure 4.20, with a normal load of 684g, the film thickness of the contact was in the mixed lubrication regime for about 40% of the cycle, as opposed to less than 10% for the 184g data. The maximum film thickness values measured occurred just after the cross-path points of the motion pathway, just after the speed had reached its maximum values of the cycle. The film thickness quickly decreased between the minimum Y points of the pathway and the X ends. The value of the film thickness was similar for each point’s mirror image, except for points A and E, which were the maximum Y values for the – X and + X ends, respectively. The mode of lubrication occurring at point A was mixed lubrication, while the contact was fully separated at point E, experiencing full film hydrodynamic lubrication.

Figure 4.20 – Film thickness measured by the sensor for a normal load of 684g
The coefficient of friction measured for both normal loads can be seen directly compared to each other in Figure 4.21. Contrary to what might be expected, the coefficient of friction measured for the normal load of 184 g had a much higher peak value than that measured for heavier normal load. In fact, for over 40% of the cycle, the lighter normal load produced a higher coefficient of friction.

Figure 4.21 – Comparison of coefficient of friction for both normal loads

In order to further understand the unexpected coefficient of friction measurements, the frictional force measured for both normal loads can be seen in Figure 4.22. The frictional force data shows that for virtually the entire cycle the frictional force for the higher normal load was indeed higher than that of the lighter normal load.
4.4 - DISCUSSION

4.4.1 – Experimental Friction Data

A circular motion pathway was used to collect the experimental friction data to make sure that constant entraining velocity was maintained for long periods of time to ensure a stable lubricant film. The higher viscosity lubricant was used to ensure that large enough film thickness values could be generated to ensure complete separation of the surfaces. It should also be noted that the higher viscosity lubricant might have attributed to the fact that the friction coefficient values obtained in the HL region were higher than those during boundary lubrication. This frictional behavior of UHMWPE is well cited in the literature, and it is furthermore commonly observed in joints with so called “self-lubricating”
polymers – polymers that can have a lower coefficient of friction in boundary or un-lubricated conditions than in full film lubrication.

A Strubeck curve describes the frictional behavior of a certain type of bearing based on the geometry and materials. Therefore, the same Strubeck curve applies for a bearing joint even when the contact radius, fluid viscosity, sliding speed, and applied load of the bearing joint are varied. The experimentally obtained Strubeck curve seen in Figure 4.3 characterizes the UHMWPE sensor material pin – on – stainless steel bearing configuration for sliding, point contact. The shape of the Strubeck curve is used to determine the range of Sommerfeld values for the different modes of lubrication. Thus, a Strubeck curve is a useful design tool when a certain mode of lubrication is desired for different operational conditions of bearings.

The experimentally measured Strubeck curve was then used to generate a plot of friction coefficient vs $\lambda$. The shape of the friction vs $\lambda$ curve is the same as the Strubeck curve, only the x-axis units are scaled from the dimensionless Sommerfeld number to the dimensionless $\lambda$ parameter. The friction vs $\lambda$ curve holds true for the generalized bearing condition of a point contact, sliding joint comprised of stainless steel and the UHMWPE sensor material. Therefore, the values of $\lambda$ for the mode of lubrication divisions hold true for any contact radius sensor pin, any fluid viscosity, any applied load, and any sliding speed. It is for this reason that the values of $\lambda$ measured with the 0.1m radius sensor pin #1 in 1 Pa*s fluid are valid for the 10m radius sensor pin #2 in 24 mPa*s fluid.
It is also suggested that because the UHMWPE is the softer, rougher material of the contact, these curves should also hold true for implant grade Co-Cr substituted for stainless steel. When the specific values of these three materials are used in the different equations, the significantly higher roughness values of the polymer cause the calculated composite roughness values to be virtually the same as the polymer’s roughness, regardless of small changes in the roughness values of the metal. Therefore, it is suggested that the overall results of the research can be easily transferred to measuring films in artificial joints, just as contact pressure was measured in chapter 1 of this dissertation.

It is also worth noting that in Figure 4.4, the sharp division noted at $\lambda = 3.5$ is the division where surface separation occurred, thus, it is the division between mixed lubrication and hydrodynamic lubrication. As noted in the figure, it is not clear whether the mode of lubrication right after $\lambda = 3.5$ is full HL or whether it’s in the EHL regime. For the purposes of measuring film thickness, there is not a big distinction between HL and EHL, since the surfaces are separated either way. The contact pressure on the sensor pin during the experimental friction data was approximately 3.8 MPa and the sliding speed at the $\lambda = 3.5$ division was approximately 50 mm/s. Because of the relatively high contact pressure at this point, the surface separation (film thickness) was certainly due, in part, to elastic deformation of the UHMWPE surface. As the speed increased further and film thickness increased, the degree of elastic deformation would likely decrease, and the lubrication mode would not be referred to as much as elasto-hydrodynamic as just hydrodynamic. It is well cited in the literature that the values of $\lambda$ for the
transition from mixed lubrication to HL are between $3 < \lambda < 5$. Generally, when $\lambda$ values of 3 are used, the transition is more often referred to as EHL, and when $\lambda$ values of 5 are used, the transition is more often referred to as HL. The fact is, the Hamrock-Dowson equation [Hamrock and Dowson 1978] most often used, as in this research, is based on EHL theory, which was developed from the study of metal on metal joints and bearings. Therefore, the correct terminology for all $\lambda$ values higher than 3.5 should perhaps be EHL. Even though it is well cited that EHL theory is valid for UHMWPE on metal joints, this is mainly only in the orthopedic and artificial joints research fields. In most other fields, the terminology elasto-hydrodynamic lubrication is generally thought of as involving very high pressures (in the GPa range) and very high speeds, since these are necessary for metal-on-metal joints and bearings. Therefore, outside of the orthopedic research arena, referring to EHL for the relatively low contact pressures and speeds encountered in this research might cause confusion, and it is very common to see general references stating that EHL will only occur with pressures much higher than are ever encountered during UHMWPE bearing research. While it is recognized that elastic deformation of the softer polymer will certainly occur when the lubricant becomes pressurized enough, the distinction between elasto-hydrodynamic and “hydrodynamic” lubrication, for the purposes of this research, is not the issue. The important aspect of this is whether the lubricant film is fully separating the surfaces or not. Therefore, the three modes of lubrication that will be referred to for the sensor’s measurements are boundary lubrication, mixed lubrication, and hydrodynamic lubrication (HL).
notation, HL is not meant to literally distinguish whether or not elastic deformation is occurring; rather, it is meant to simply state that for the given load, viscosity, and contact radius, the dynamic entraining velocity is high enough to generate lubricant pressure that completely separates the contact surfaces.

4.4.2 – Hydrodynamic Sensor Readings

As seen with the static data from chapter 2 of this dissertation, the hydrodynamic readings generated by the sensor from simple linear reciprocating motion have 3 distinct regions when analyzed in a piecewise manner. The highest rms sensor output readings occurred at $h_s$ values of less than $1 \mu m$. As seen in Figure 4.7, this data fits a linear trend to a high degree of accuracy, with an $R^2$ value of 0.9903. Based on the $\lambda$ value of the division between boundary and mixed lubrication that was determined from the friction data, the film thickness where boundary lubrication would change into mixed lubrication is around $1 \mu m$. Therefore, the linear portion of the data seen in Figure 4.7 is clearly within the boundary lubricated regime. The standard deviation of the linear portion of the hydrodynamic data seen by the error bars in Figure 4.7 should also be noted. The standard deviation error bars clearly show that each data point is significantly different from the next, but the relative magnitude of the standard deviation within this linear region is much greater than that seen within any of the other regions of the hydrodynamic data. Furthermore, the standard deviation of each data point increases as the film thickness decreases. Thus, the standard deviation increased as the sliding speed decreased. While there is not enough
evidence to point to any one clear cause of this, there could be many factors contributing to this trend, including such phenomena as stick-slip action or spatial deviations in surface roughness and asperity size distribution. The last data point represents a film thickness of less than 0.4\(\mu m\). For the roughness values of UHMWPE, this indicates a very high degree of direct contact between the two surfaces with a large degree of the real contact area likely experiencing no appreciable lubricant film between the metal surface and the polymer surface. At this scale of boundary lubrication, the surface chemistry of the materials and the lubricant become more dominant and the actual separation of the two surfaces varies by a relatively large amount depending upon the size and distribution of the surface asperities.

When plotted on a log-log scale, it is clearly visible that the power fit data seen in Figure 4.9 has a distinctive change in its scaling factor (slope of log-log data) at about 5\(\mu m\). The standard deviation error bars of Figure 4.9 should also be noted. The relatively small standard deviation obtained for this portion of the data, along with the relatively high values of \(R^2\) suggest that the sensor readings obtained for the simple linear reciprocating hydrodynamic experiments were very repeatable. No load cell was used during the hydrodynamic experiments, and the data obtained during these experiments seems much more repeatable than the data obtained during the static experiments. While it cannot be concluded that the load cell was the sole cause for the larger standard deviation of the data during the static experiments, it is interesting to note that the data obtained without the use of a load cell had higher coefficients of determination.
It is also important to distinguish between the different values of $h$ used during this research. In chapter 2 of this research, the control position of the lever rig was the experimental variable. Because of the load cell attached between the lever arm and the sensor, when the system was loaded, the lever arm moved a much larger distance than the sensor, due to the deflection of the load cell. Thus, the sensor position of chapter 2 was the deflection-corrected vertical height of the sensor. The value $h_s$ in chapter 3 was the film thickness as measured by the sensor. Hydrodynamic forces are needed to generate this film thickness, thus, it could not be generated during the static conditions of chapter 2. Because $h_s$ refers to the film thickness, it can never be less than $0 \mu \text{m}$.

However, for the static data, the linear portion of the curve contained negative values. Thus, the static data showed that the linear portion of the curve occurred at higher loads. The hydrodynamic data showed that the linear portion of the curve begins in the boundary lubrication regime.

When the friction-predicted boundaries between the lubrication modes are transposed on the curves generated from the linear reciprocating hydrostatic sensor data, it becomes clear that, as suggested in chapter 2 of this dissertation, the sensor’s output has three distinct regions where the data can be fit to a high degree of accuracy that coincide with the mode of lubrication occurring in the contact. Figure 4.6 shows the friction-derived divisions between boundary and mixed lubrication and between mixed and full film lubrication. These boundaries fall within one data point’s accuracy of the boundaries that would be predicted based on the piecewise fit of the data. Thus, with the number of data points
obtained, there’s no clear difference between the friction-derived boundaries and the sensor-predicted boundaries. When the very small standard deviation of the data points is also taken into account, the friction data and the hydrodynamic data combined strongly indicate that based on the rms output of the sensor, the mode of lubrication can be determined, and the actual thickness of the lubricant film can then be determined by applying the applicable calibration equation.

When the results of the static data, presented in chapter 2 of this dissertation, are plotted along with the results of the linear reciprocating hydrodynamic data, the similarities and differences between the data sets can be observed. While the linear portions of each data set do not appear to correspond well at first, there are some important key similarities. Most importantly, the linear portion of each data set has a maximum film thickness of approximately $+1\mu m$. This is a very important key point that further validates the linear region of the data as corresponding to boundary lubrication. Figure 4.10, which shows the linear portion of each data set, has the x-axis labeled $h_s$. However, as discussed in chapter 2 of this dissertation, the displacement that was measured for the static data was a measure of the sensor’s global vertical movement. Therefore, as the compressive force on the sensor increased, the contacting face of the sensor experienced compressive strain that was not able to be accounted for in the data. For this reason, the negative values of film thickness for the static data shown in Figure 4.10 are obviously not really a measure of film thickness. When this is taken into consideration, one can view how the linear data sets might align, which they indeed do, once the load cell readings of the static data are
considered. The fact that both data sets are linear in this region, along with the fact that the linear portions both start at a film thickness value of +1 μm, further validates what the hydrodynamic friction data shows – that boundary lubrication occurs with this specific contact configuration at film thickness values of less than +1 μm.

The non-linear portions of the static and hydrodynamic data seen in Figure 4.11 show that the data sets align very well with each other for film thickness values between 1 μm and 3.5 μm. As previously discussed, this falls within the mixed lubrication regime. As shown in chapter 2, because of a lack of data points, the static data could not be fitted for sensor position values between 3.5 μm and 5.5 μm. The log-log scale “slope” of the static data is also seen to be nearly the same as that of the hydrodynamic data within the mixed lubrication regime. The parts of each data set that correspond to full separation of the surfaces do not match up as well as the rest of the data. There are many possible reasons that this could have occurred, many of which were discussed in chapter 2 of this research. The important aspect of the combined data is that both the static and hydrodynamic data show that during mixed lubrication and during HL, the sensor’s rms output voltage is related to the film thickness by a power law relationship. Furthermore, both the static and the hydrodynamic data show that the log-log scale “slope” is different between the mixed lubrication portion and the HL portion.

The experimental friction results for the specific type of contact being investigated in this dissertation, as well as the overall good correlation between
the static and the hydrodynamic data, provide clear evidence that the sensor’s rms output voltage is related to the hydrodynamic film thickness in a piece-wise manner. Equations 3.1, 3.2, and 3.3, seen in the results section, are the calibration equations for sensor pin #2. These equations can be used to determine the film thickness from the sensor’s measured rms output voltage. Each equation is valid for a certain range of $V_{o \, RMS}$ values. When using the sensor to measure film thickness, the sensor’s rms output voltage is first used to determine the mode of lubrication. The cut-off values for the lubrication mode are the same values used to select which film thickness equation to be used.

The calibration equations were obtained by matching the sensor’s rms output voltage to the theoretically calculated film thickness value for that speed during linear reciprocating motion. The points on the linear pathway from where the sensor readings were taken were located far enough away from the ends of the pathway such that adequate time was allowed for the theoretically predicted film thickness to establish itself. Although this measure of film thickness was based on theory, it is widely accepted that the theory is a good predictor for simple motion where there is constant entraining velocity. Nonetheless, it is uncertain whether the film thickness measured by the sensor corresponds exactly to $h_{min}$, or whether it’s a better measure of $h_{cen}$ (central film thickness), or some other value. Because of the relatively large area of the sensor’s tip, the film thickness would indeed vary over this spatial region, and since the sensor only gives one output value for this spatial region, the output is presumably some type of average or composite film thickness measure. For these reasons, the film
thickness that is measured by the sensor has been termed $h_s$. There are two main points that are most important about the sensor’s ability to measure lubrication.

1. The first point is that the sensor’s output clearly indicates the mode of lubrication. The lubrication mode is generally considered more important to know than the actual value of the film’s thickness, since the actual film thickness value does not by itself indicate the type of lubrication, as surface roughness values must also be taken into account, hence the reason for the $\lambda$ parameter.

2. The second important point is that the $h_s$ values measured by the sensor over a complex pathway with constantly varying entraining velocity are all comparable to each other. Therefore, whether $h_s$ is the minimum thickness or a central thickness is not nearly as important as how the thickness changes from one moment to the next over the course of a complex motion pathway.

### 4.4.3 – Relevant Pathway Sensor Readings

When the plots of coefficient of friction over the figure 8 pathway measured for each normal load were compared, it was somewhat unexpected that the 184g data produced a higher coefficient of friction than the 684g data. The plot of frictional force measured for each normal load, as seen in Figure 4.22, shows that the resistance to sliding was actually higher for the higher normal load, as
would be expected. However, the frictional force was not very much larger for the 684g data, especially since the normal load was 3.7 times greater. Therefore, the coefficient of friction was actually lower for the 684g data. For this reason, it is not at all impossible that the data is exactly correct. However, when such results are obtained, it is wise to carefully consider any possible reasons that the data could be incorrect. In this case, even though the 684g normal load was 3.7 times greater than the 184g normal load, both loads produced very small contact pressures because of the very large radius of curvature of the tip of sensor pin #2. This meant that fluid films were fairly easily generated during these tests, and as the sensor data shows, there was a fairly high degree of surface separation achieved during much of the testing cycle. Therefore, the measured frictional force was likely due in large part to the drag created by the lubricant as much as it was to the direct contact between the surfaces. The frictional resistance created by just the lubricant is independent of normal load and much more dependent on speed. To know what proportion of frictional force was attributed to each mechanism cannot be determined, but the recognition of this possibility gives the friction data some degree of credibility. However, there is another explanation that seems just as likely, if not more so.

Although not presented in this dissertation, friction and sensor data were collected over many repeated runs for both normal loads, using the exact same experimental inputs for each run. The sensor’s data was very comparable from one run to the next, as was the motion pathway’s position and speed data. However, the friction data was not nearly so comparable. Although the shape of
the friction plots were generally the same, showing the same trends during the cycle, there was a considerable difference in the y-axis offset of the plots. The sensor-terminology for this observation is termed drift, and such differences seen in the friction data were almost certainly due to the manufacturer’s specified errors that are so inherent in load cells. To a much lesser degree, it is also possible that the offset errors could have resulted from external EMI and RF noise, processing hardware limitations, or control and acquisition software issues with the cross-shear machine. However, because there was virtually no drift associated with the sensor’s measurements from one experimental run to the next, the drift in the friction data was almost certainly due to load cell issues. The load cell array that measures friction on the cross-shear machine is comprised of 25-pound (110 N) load cells that have a stated nonlinearity of ±0.03%, a hysteresis of ±0.02%, and nonrepeatability of ±0.01%, and a 20min creep of 0.025%. These are standard force transducer specifications that tell the maximum error due to each category as a percentage of the rated output, which is 25 lbf in this case. This means that the maximum error due to the load cell’s nonlinearity is ±0.033N (±3.4 gf). Thus, at best, the load cells are accurate to within 3.4gf, however, creep and hysteresis can add to nonlinearity errors, especially over the time frame of multiple experimental runs. To put these errors into perspective, Figure 4.22 shows that the frictional load measured during 60% of the 184gf data was less than 10gf, and this was measured with a device that has an accuracy of ±3.4gf.
The above calculations show the mechanical accuracy of the load cells used to collect the friction data. Load cells can also experience errors due to the aforementioned term “drift”. Load cell manufacturers specify this as an electrical specification of the maximum error due to zero balance. For the load cells used to collect the friction data, the zero balance error is ±1% of the rated output of 110N. This makes the zero balance error ±1.1N, or ±112 gf. In fact, from one experimental run to the next, the zero balance error was readily observed, where the frictional force curves were offset from each other by as much as 10g.

The point of this in-depth discussion of load cell basics is that the potential offset-error associated with the friction data is much larger than the magnitude of the data itself. Therefore, when making a direct comparison of the 184g and 684g friction coefficients, the accuracy range of the equipment must be kept in mind.

It is interesting to note that due to the fundamental nature of the contact sensing technology of the lubrication sensor (sensor pin #2), drift due to zero offset is virtually eliminated. This was illustrated by the sensor data obtained over multiple experimental runs. While the friction data showed relatively large drift from one run to the next, the sensor data showed very little. The largest difference seen between one run and the next in the sensor data was not in the magnitude of the output voltage, but rather in the x-axis offset, which was due to the methodology by which the sensor data that was collected on a separate computer was aligned with the rest of the friction, position, and speed data collected on the cross-shear computer. The only other noticeable difference in
the sensor’s output from one run to the next was the magnitude of the average peak output voltage values. However, this difference in peak values from one run to the next was less than the standard deviation of the peak values measured during each of the 6 cycles used to compute the average of each experimental run. In other words, there was no statistically significant difference seen in the sensor data from one run to the next.

As seen in the film thickness plots measured by the sensor, for a normal load of 684g, the contact experienced mixed lubrication for about 40% of the cycle. This is in contrast to the film thickness measured with a normal load of 184g, where the contact experienced mixed lubrication for less than 10% of the cycle. Boundary lubrication was hardly seen during the figure 8 motion pathway because for a pin with such a large contact radius of 10m, both normal loads were relatively small, with the Hertzian contact pressure of the 684g contact being about 270 kPa. Thus, at the sliding speeds encountered over the pathway and the 24 cP viscosity of the lubricant, one would not expect a high degree of contact.

To say that these film thickness measurements are relevant to many other pin-on-disk studies would be incorrect. Most pin-on-disk studies, especially in the bioengineering arena, are performed as wear tests. For many of these studies, an accelerated wear rate is even desirable such that the effects of different materials or formulations on the wear performance can be observed. In such studies, lubricating films are not normally considered. In the area of artificial joint research, more complex simulators are used instead of simpler pin-
on-disk machines when the overall performance of different devices is intended to be studied. It is only in these studies that the beneficial effects of a lubricant film is desired. Therefore, measuring the film thickness seen during many pin-on-disk studies relevant to artificial joint research would be missing the point.

Indeed, this was not the intention of this part of the study. The figure 8 motion pathway chosen is relevant to advanced pin-on-disk testing, but moreover, it is clinically relevant to the motion of the central contact seen in artificial knee joints [DesJardins PhD Dissertation, 2006]. The overall goal of this research was to be able to expand upon the contact pressure measuring capabilities of the knee sensor described in chapter 1. In other words, during the swing phase, when axial loads to the implant are low, it would be desirable to know the film thickness distribution over the implant’s surface. Thus, to show that the calibration equations obtained in this chapter can allow such measurements, the low pressure, high sliding speed, variable motion conditions of the swing phase of a knee joint were best approximated with a figure 8 motion pathway with very low normal loads.

The Vo rms measurements of the sensor data were used in conjunction with the calibration equations from the linear reciprocating pathway to calculate the film thickness occurring during the figure 8 motion pathway. Because the calibration data was only determined for a maximum film thickness of 9μm, the portions of Figure 4.16 and Figure 4.20 with a film thickness of more than ~9μm are shown to depict the relative change in the thick HL regime during these portions of the motion pathway. It is important to keep in mind the significance of
the film thickness values when interpreting the data. As previously shown, the experimentally determined $\lambda$ values dictate that film thicknesses of greater than 5\(\mu\)m mean the contact is completely within the hydrodynamic lubrication regime, with the surfaces being completely separated. Therefore, the actual magnitude of film thicknesses of greater than 9\(\mu\)m starts to become insignificant, compared to simply knowing the contact is well within the HL regime.

It is also important to note that the friction data and sensor data were collected on two different computers. Because of this, the data had to be aligned for analysis, as described in the materials and methods section of this document. Therefore, small errors in the alignment between the two data sets are to be expected, so when comparing the friction data to the sensor's readings this must be kept in mind. It must be understood that the possible error in alignment can only affect the relative position of the curve with respect to the x-axis. The magnitude and duration of the curves were unaffected by the alignment procedure. Thus, events that are separated by a certain distance on the x-axis are unaffected by the alignment, whereas the actual % of the cycle of an event may not be the same for both the friction and sensor data. The most obvious example of this can be seen with the data collected at each end of the figure 8 pattern, which occur 50% apart in the cycle. The peaks in both the friction data and the sensor data corresponding to the ends of the pathway are also separated by 50% of the cycle, even when the actual value of the % cycle of the sensor data and the friction data are about 5% apart. Whether the peak in the sensor data actually occurred 5% after the peak in friction data and speed, or
whether the issue is due to small alignment errors would be impossible to determine without collecting all of the friction and sensor data on the same computer.

4.5 - CONCLUSIONS

The experimentally obtained Strubeck curve obtained for the UHMWPE-on-metal sliding point contact showed that the boundary lubrication regime existed for $\lambda < 1$, mixed lubrication was present for $1 < \lambda < 3.5$, and fluid film lubrication existed for $\lambda > 3.5$. The piecewise nature of the simple-motion hydrodynamic data and the calculated film thickness of the divisions between the different fits showed that the sensor’s output reveals the mode of lubrication and yielded calibration equations. The agreement between the $\lambda$ values from the Strubeck curve and the calculated film thickness divisions of the simple hydrodynamic data further proved the sensor’s ability to determine the mode of lubrication.

With a clinically relevant sliding pathway and velocity too complex for theory, the sensor measured the film thickness as it changed over the motion cycle for two different applied loads that allowed the full range of lubricating regimes. For the lighter load, HL was measured for the majority of the cycle, except for about 10% of the cycle that occurred just after the $+X$ end of the pathway, where mixed lubrication was briefly measured. With the heavier applied load, the film thickness was seen to significantly decrease after both the $+X$ and $-X$ ends of the pathway, dropping into boundary lubrication for a brief period just after the $+X$ end of the pathway. These combined results show the sensor’s ability to
measure all modes of lubrication and to quantify the film thickness as it changes over the course of a clinically relevant motion pathway.
5.0 – OVERALL CONCLUSIONS AND RECOMMENDATIONS

The overall conclusions that can be drawn from the three main thrusts of this dissertation are that the UHMWPE sensor material that had previously been shown to measure dry contact pressure can now also be used to measure the lubrication film thickness of a UHMWPE-metal sliding point contact. The film thickness measurements that were performed in this research were made throughout a complex cross-path motion pathway that is clinically relevant because of both the shape and the varying velocity of the pathway. Moreover, the shape and style of contact with which the measurements were made was specifically designed to mimic a single sensing point of an instrumented tibial insert for use in a knee joint simulator.

Future recommendations include using the results of this research to fully characterize the dynamic contact conditions that occur on the surface of artificial knee joints being tested in a knee joint simulator. By applying a sinusoidal, low frequency excitation to each sensing point, the magnitude of the output will show the mode of lubrication and will allow the appropriate calibration equation to be applied. It is recommended that a final cut-off voltage be determined to distinguish between using the boundary lubrication mode equation to relate the voltage to film thickness, or using the load calibration equation to relate the voltage to load applied to each sensor point. As discussed within this dissertation, the sensor’s output is a direct measure of the applied load (which is solely responsible for the real contact area that the sensor seems to measure),
and during boundary lubrication, the film thickness is due to compressive strain of the polymer. Therefore, it is suggested that the sensor’s output is related to boundary lubrication film thickness by the compressive stress-strain response of the UHMWPE sensor material. When the compressive stress is within the linear-elastic region, there is a linear relationship between stress and strain, so, there is also a linear relation between the sensor’s output and strain. For this reason, it is thought that the linear relation between the sensor’s output and the boundary lubrication film thickness will only remain linear while the contact is linear-elastic, so there will be an upper voltage limit for this calibration equation. However, the terminology of “boundary lubrication film thicknesses” at higher contact stress levels becomes somewhat convoluted. At these levels, it is suggested that a measure of contact stress would prove far more useful than a measure of the boundary layer's thickness. The importance is that the sensing technology gives the ability to measure both these interrelated quantities without negatively affecting them, which is quite unique when compared to any other similar alternatives.

It is also suggested that although obtaining film thickness measurements with the knee sensor would be difficult and expensive with current hardware and technology, obtaining such measurements in future years may be much easier. The data acquisition system used during the final 2 years of this dissertation research was a 1MHz analog I/O system with USB 2.0 connectivity. When the sensor material was first being developed some 5 years prior to this dissertation’s date, such a system was not available – in fact, the interface
protocol did not even exist in commercial systems. For these reasons, implementing these measuring capabilities to the knee sensor to obtain spatial distributions of film thickness is more related to electronic hardware and control software issues than it is to materials and biomechanics.
APPENDICES
APPENDIX A – NOMENCLATURE

\( h_{\text{min}} \) – minimum film thickness under a contact

\( h_{\text{cen}} \) – central film thickness under a contact

\( h_{\text{max}} \) – maximum film thickness under a contact

\( h_{\text{sensor}} \) or \( h_s \) – sensor’s measure of film thickness (\( h_{\text{min}} < h_s < h_{\text{max}} \))

\( \lambda \) - specific film thickness parameter (defined as \( h_{\text{min}} / R_q \))

\( \text{MCL} \) – mean center line for surface roughness values

\( R_a \) – numerical average of surface heights that lie above the MCL

\( R_q \) – rms average of surface heights that lie above the MCL

\( R_t \) – Highest peak to lowest valley distance measured during one scan of a profilometer

\( R_z \) – numerical average of the 10 largest peak to valley distances measured during one scan of a profilometer
APPENDIX B – UN-SUBMITTED ABSTRACT OF A VALIDATION STUDY OF A PROPOSED VOLUMETRIC WEAR ANALYSIS PROCEDURE

Introduction: Wear particles from ultra-high molecular weight polyethylene (UHMWPE) bearing surfaces of total joint replacement prostheses have been shown to cause adverse biological reactions which can lead to loosening of the prosthesis, and ultimately to failure [1]. The most accepted method of evaluating total joint replacement implants is to perform a lifetime wear test using a joint simulator. The most common method of measuring the wear of the UHMWPE bearing is by weighing the amount of material lost during the testing – referred to as gravimetric analysis. A less common method of quantifying wear is volumetric analysis, where the volume of material lost is directly measured by digitizing the implant with a coordinate measuring machine (CMM). While the gravimetric method is the easiest to perform, its accuracy comes into question since the amount of fluid absorbed by the implant during testing is unknown. Although this is corrected for by using controls that are soaked in the same fluid environment as the implant, there is debate over how much the dynamic loading of the implants affects their fluid absorption. The volumetric analysis technique also has drawbacks. Over the course of testing, the UHMWPE bearing can experience creep and plastic deformation leading to an overestimation of wear. Additionally, performing accurate CMM measurements can be time consuming. Ideally, both techniques would be used during a lifetime wear test to provide the most complete assessment of implant performance. The authors are proposing a method by which both gravimetric and volumetric analysis can be performed without unnecessary down time due to accurate digitizing of the implant. The proposed method also has the advantage of preserving replicas of the implant at set intervals during the testing such that the progression of the damage can be more easily recorded.

Materials and Methods: A complex geometry was machined using standard milling operations into a rectangular puck of 4.50 mm UHMWPE with dimensions of 7.72 mm x 7.72 mm x 7.72 mm (Figure 1). The UHMWPE puck was weighed (n=5) before and after the machining operation to determine the mass of material removed. A negative mold of the puck was then made using a silicone rubber (Silastic® E RTV, Dow Corning Corporation) both before and after machining. A high vacuum of 28 mmHg was used to evacuate any air trapped in the silicone during mixing and pouring. In order to form a replica of the puck, a high-strength epoxy and filler system (AdTech EC-410, Case Polymers, Charlotte, NC) was poured into the negative mold, and again a high vacuum was used to remove any air entrapped in the epoxy. A coordinate measuring machine (MDX 15, Roland DGA Corporation, Irvine, CA) with an accuracy of 0.005 mm in the x and y-direction and 0.0025 mm in the z-direction with a stylus tip radius of 0.04 mm was used to digitize the puck both before and after the machining with an x-y scan pitch of 0.13 mm. The volume of material removed during the machining was calculated using advanced 3-D scanner analysis software (Rapidform 2004, INUS Technology, San Jose, CA). This volume measurement was then compared to the gravimetric measurement using the material’s reported density.

Figure 1 – Digitized image of puck replica with artificial wear

Results and Discussion: The weight of material removed during the machining process was measured to be 169.9 ± 0.3 mg. The volume of material removed was measured to be 136.9 mm³. The density of UHMWPE as reported by the manufacturer is 0.93 g/cm³. Using the density to convert the measured volume into weight of material removed, it was calculated that 173.7 mg of material was removed. This represents a 2.2% error or discrepancy between the two methods, which could likely be explained by the reported density which was used to compare the two methods. It is well accepted that UHMWPE has material properties that can vary depending on many factors, including the specific grade of material used.

<table>
<thead>
<tr>
<th>Method</th>
<th>Measured (mg)</th>
<th>Volumetric (mm³)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measured</td>
<td>169.9 ± 0.3</td>
<td>136.9</td>
</tr>
<tr>
<td>Calculated</td>
<td>173.7</td>
<td>182.7</td>
</tr>
</tbody>
</table>

Table 1 – Measured wear using both methods compared to value calculated from other method

Figure 2 – Artificial wear depth (mm) measured

Because the wear was created artificially under dry conditions, the possibility of fluid absorption was eliminated, leaving no uncertainties with the gravimetric analysis. For the same reasons, the volumetric analysis was also complete since there was no possibility of material loss outside of the machined area.

APPENDIX C – LABVIEW PROGRAMS

The following programs were written in LabView, a high-level, graphical programming “language” tailored to data acquisition and analysis. A LabView “program” is called a “virtual instrument”, which has the file extension “.vi”. The “front panel” is the LabView terminology for the graphical user interface and it is how an individual program is run and controlled. The “wiring diagram” is the programming interface where the individual programs are “written”. Each object on the front panel is linked to the wiring diagram where it is programmatically controlled. The “wiring diagram” contains each step of the program represented by graphical “icons” that are connected by wires, with the programming logic flowing from the top left to the bottom right.

The programs used during this research all fall into two categories: 1) “acquisition vi’s” that control the data acquisition hardware and display the data in real time while storing it for later analysis, and 2) “analysis vi’s” that access previously stored raw data and are used for analysis of the data.

There were 3 main programs used for the knee sensor. The first program collected, stored, and displayed the basic pressure data from the knee sensor electronic hardware “MUX box” in real time. The raw data stream was saved to disk as an ASCII file of single precision values of the four voltage channels and 3 digital input channels, with an average file size being 6MB. The other two programs used for analysis of the data were continually updated throughout the research, but the structure of their input files continued to remain unchanged.
throughout the research to ensure continuing raw data file compatibility. The
second program converted the 1-channel multiplexed data stream into a 2-D
matrix format representing each data point’s spatial location on the knee sensor,
and then averaged and aligned the multiple-cycle raw data frames of the spatially
oriented data into the temporal domain, such that there was one frame of the 2-D
sensing point matrix for each percentage point of the gait cycle. This allowed the
knee sensor data to be aligned with the simulator’s data, and formed a 3-
dimensional array data structure representing both the spatial and temporal
“location” of each raw data value. This 3-dimensional array structure was the
main data format used in the third program, which was the main data analysis
program. This final program was designed to allow interactive analysis of the
data, where multiple representations of the data could be observed. The only
“complete” representation of the data is a “moving” 3-dimensional graph, where
each frame of the “movie” represents a snapshot in time, and the 3-D graph
represents the spatial location and the magnitude of the contact pressure
measured by the sensor. Because of the shear size of the raw data files, and the
extreme complexity of the data analysis, the main analysis program contained
many interactive features allowing both spatial and temporal analysis of any
desired subset of the main 3-dimensional raw data array.

The lubrication sensor used 4 different programs. Again, one of the
programs was for the real-time acquisition, display, and storage of the data.
There were two different alignment VI’s for the lubrication sensor, one for
alignment of the sensor data with the materials testing machine’s output data,
and one for alignment of the sensor data with the “cross-shear” machine’s output data. There was also a main data analysis program for the lubrication sensor.
Figure C.1 - Front Panel Interface for Knee Sensor Data Acquisition VI
Figure C.2 – Wire diagram for Knee Sensor Data Acquisition VI with case 3 shown
Figure C.3 – Wire diagram for Knee Sensor Data Acquisition VI - case 0
Figure C.4 – Wire diagram for Knee Sensor Data Acquisition VI - case 1
Figure C.5 – Wire diagram for Knee Sensor Data Acquisition VI - case 2
Figure C.6 - Front panel interface for knee sensor cycle alignment VI
Figure C.7 – Wire diagram for knee sensor cycle alignment VI
The flexion trigger counter sensor is high when the flexion is below about 40 and low when the flexion is above about 40. Therefore, triggering on the rising edge looks for the point where the flexion

For these bottom 2 FOR LOOPS:  
Because there is a hardware page size of 256, this is the value that the array is subset to (each iteration of the loop, making the array smaller and smaller). Then, the reshape array takes the 256 long 1-D array and shapes it into 14 rows by 18 rows. This is 252 rows, and only 250 are wired, so what's shown as the last 2 rows on the graph aren't hooked up. This drops the last 4 pts of the 1-D 256 long array to match the number of points on the sensor. This lets the next iteration start in the right place. The next iteration, it goes to

Figure C.8 – Sequence structure (1 of 4) for knee sensor cycle alignment VI
Figure C.9 – Sequence structure (2 of 4) for knee sensor cycle alignment VI
Figure C.10 – Sequence structure (3 of 4) for knee sensor cycle alignment VI
Figure C.11 – Sequence structure (4 of 4) for knee sensor cycle alignment VI
Figure C.12 - Front panel interface for knee sensor main data analysis VI
Figure C.13 – Wire diagram for knee sensor main data analysis VI
Figure C.14 – Sequence 1 of 6 for knee sensor main data analysis VI
Figure C.15 – Sequence 2 of 6 for knee sensor main data analysis VI
Figure C.16 – Sequence 3 of 6 (true case) for knee sensor main data analysis VI
Figure C.17 – Sequence 4 of 6 for knee sensor main data analysis VI
Figure C.18 – Sequence 5 of 6 for knee sensor main data analysis VI
Figure C.19 – Sequence 6 of 6 for knee sensor main data analysis VI
Figure C.20 - Front panel interface for lubrication sensor data acquisition VI
Figure C.21 – Wire diagram for lubrication sensor data acquisition VI
Figure C.22 - Front panel interface for lubrication sensor data analysis VI
Figure C.23 – Wire diagram for lubrication sensor data analysis VI
Figure C.24 - Front panel interface for “data file merger for friction and sensor data.vi”
Figure C.25 – Wire diagram for “data file merger for friction and sensor data.vi”
Figure C.26 - Front panel interface for “data file merger – graphic analysis.vi”
Figure C.27 – Wire diagram for “data file merger – graphic analysis.vi”
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