Title
Macro/micro-feature development for improved hydrodynamic performance at the mechanical seal interface via laser surface texturing

Permalink
https://escholarship.org/uc/item/67m1f2r0

Author
Benedict, Joshua King

Publication Date
2011

Peer reviewed|Thesis/dissertation
UNIVERSITY OF CALIFORNIA, SAN DIEGO

Macro/Micro-Feature Development for Improved Hydrodynamic Performance at the
Mechanical Seal Interface via Laser Surface Texturing

A Thesis submitted in partial satisfaction of the requirements for the degree Master of
Science

In

Mechanical Engineering

by

Joshua King Benedict

Committee in charge:

Professor Frank E Talke, Chair
Professor Sungho Jin
Professor Vlado Lubarda

2011
The Thesis of Joshua King Benedict is approved and it is acceptable in quality and form for publication on microfilm and electronically:

______________________________________________________________________
______________________________________________________________________
______________________________________________________________________

Chair

University of California, San Diego

2011
DEDICATION

It is my hope that this Thesis may provide additional insight to others who may use the principles or methods found herein to help further technology. May all such technology be used to improve the quality of life.

It is unfortunate that we seldom thank those who often provide the largest impact in our lives. I would be remiss to neglect such an opportunity here, for I have learned that all of the information in the world is of little consequence if you have no one to share it with.

Though impossible to thank everyone, I will attempt to thank a handful of individuals who have providing the intellectual, spiritual, and financial foundation and support required for completing this work.

First and foremost, I would like to thank God who has created all things for our benefit, and for his son, Jesus Christ, who’s sacrifice and perfect example have provided the means whereby we can truly learn from our mistakes and become a better people.

I would like to thank my family; my parents Stan and Brenda Benedict, who have been true mentors in all aspects of my life, having always encouraged me to seek wisdom from the best books, to find joy in hard work, and to look for beauty in all things. Thanks for teaching me about greatness. And to my brothers and sisters; Brooke, Jason, Travis, Amy, Elisa, and Ashley, thank you for your support, for your unconditional love, for being great aunts and uncles to my children, and most of all for not giving up on me.

Along with my biological parents, I would like to thank my additional parents Steve and Deborah Cohen. You’re willingness to provide support to my family in so many ways throughout the years has been truly remarkable. Thank you for being so selfless. To Steven, Michelle, Jack, and Erin, thank you for helping out, and being such great Aunts and Uncles to my children.

Last, and most importantly, I would like to thank my wife Natalie, You’re unconditional love has been the best catalyst of all, for I am truly a better person because of you. Thank you for your support and sacrifice. And to my children; Samuel, Mallory, and Audrey, you all make me so proud; you are the motivation for all things that are worth doing.
I would like to acknowledge Professor Talke for his support as chair of my committee. Professor Talke’s encouragement and willingness to provide me with an audience of professionals has greatly benefited my work. And I would also like to acknowledge Professor Lubarda, and Professor Sungho, both of whom have been an inspiration for me. Thank you all for providing a true quality of education.

I would also like to acknowledge Flowserve Corp. for allowing me to use company time and resources for the development and testing of these micro-features. I must give special thanks to Lionel Young, whose guidance, mentorship, and creative freedom have always made work enjoyable and productive. Thanks for providing a professional environment for growth and opportunity. I would also like to give special thanks to the Research and Development team consisting of: John Davis, Jim Saucerman, Berhanu Wondimu, Brent Collins, Casey Lloyd, Steve Roso, and Charlie Beier, for all of their support, and without whom nothing would get accomplished.

I would like to thank the different companies who have provided outstanding software, allowing even an amateur programmer such as myself the opportunity to analyze vast amounts of data efficiently. These programs include: Matlab, Ansys, Inventor, SolidWorks, Labview, Aerotech, Surfer 9, Comsol and other proprietary CFD and FEA/Fluid Mechanics software packages. Special thanks to Alan Lebeck.
TABLE OF CONTENTS

Signature page.................................................................................................................. iii
Dedication ......................................................................................................................... iv
Acknowledgements ......................................................................................................... v
List of Figures.................................................................................................................. vii
List of Tables................................................................................................................... ix
Abstract of the Thesis....................................................................................................... x
Introduction ..................................................................................................................... 1
Ch.1: Mechanical Seal Background .................................................................................. 3
Ch.2: Basic Theory of Seal Operation .............................................................................. 9
Ch.3: Operation at the seal interface ............................................................................. 15
Ch.4: Periodic Features used for Hydrodynamic Liftoff ................................................... 20
Ch.5: Creating Micro-Feature Geometry through Laser Ablation .................................. 24
Ch.6: Code Developed to Simulate Laser Manufacturing ................................................. 34
Ch.7: Macro/micro-feature Optimization using Computer Analysis ................................ 45
Ch.8: Macro/micro-feature Testing ................................................................................. 51
Ch.9: Discussion of Results and Conclusions .................................................................. 62
References: ....................................................................................................................... 64
Bibliography: ................................................................................................................... 66
LIST OF FIGURES

Figure 1: Basic concept of a Mechanical Seal [1] ............................................................. 3
Figure 2: Geometric variations in rigid model produce leakage (left), or contact (right) [1]   4
Figure 3: Quarter section of seal assembly, indicating basic components used in mechanical seals ............................................................................................................. 6
Figure 4: Close up quarter section view of the mechanical seal faces .................................. 9
Figure 5: Axis-symmetric free body diagram of external forces acting on sealing rings...10
Figure 6: Example of hydraulic closing parameters used to define balance ratio [1] ............11
Figure 7: Pressure drop factor configurations: linear (left), convex (middle) and concave (right) [1] ..........................................................................................................................................12
Figure 8: Example of an axis-symmetric fluid boundary defined by the gasket locations14
Figure 9: Cartesian geometry for general lubrication theory [2] ........................................16
Figure 10: Fluid control volume [2] ..............................................................................17
Figure 11: Control volume defined in Polar Coordinates [2] .............................................18
Figure 12: Example of a periodic geometry used to create hydrodynamic lift [2] ..........20
Figure 13: Spectrum of electromagnetic radiation, known as light [5] ............................25
Figure 14: Atomic excitation energy diagram [5] ..........................................................26
Figure 15: Light resonator used for light amplification [5] .............................................26
Figure 16: Laser impact across at the material interface [6] .........................................27
Figure 17: Shock wave propagation and material ejection due to single laser pulse after 190ns (left) and 410ns (right) [7] .................................................................28
Figure 18: Range of laser interaction mechanisms [5] ....................................................28
Figure 19: Photographs of blast wave (left) and ejected material (right), taken 500ns and 2.9µs after start of laser pulse [8] .............................................................29
Figure 20: Copper showing continuous photo-thermal interaction after receiving 50 pulses (left), and 500 pulses (right) [10] ............................................................29
Figure 21: Silicon micro-dimple formed by single pulse [11] .......................................30
Figure 22: Wavy seal with combined hydrostatic/hydrodynamic features (Flowserve Corp) .........................................................................................................................31
Figure 23: Micro-dimples produced using single laser pulse [19] ..................................31
Figure 24: Depth to Diameter Ratio ........................................................................31
Figure 25: Macro/micro feature called a Tapered Channel .........................................32
Figure 26: Different configuration of the tapered channel. Depth comes to point (left) or flat (right) ........................................................................................................32
Figure 27: Map showing relative direction of each laser and mask axis ...........................33
Figure 28: Finite difference point method used for defining mask shape from dense population of grid points .........................................................................................34
Figure 29: Simulated ablation points defining macro/micro feature missed prior to using search boundary ........................................................................................................35
Figure 30: User selects the predefined mask shape and beam shape ............................36
Figure 31: Relative motion between the beam shape and mask shape is simulated.....38
Figure 32: Intersection points are indexed as new line segment within each boundary..39
Figure 33: Assessment points created on either side of each mask located between intersection points ........................................................................................................39
Figure 34: Odd/even mapping of assessment line segments intersection ........................40
Figure 35: Overlapped shape defined by the assessment points ....................................41
Figure 36: Vertical nodes outside the overlapped shape boundary are defined ............................................................... 41
Figure 37: Overlapped shape redefined as horizontal line segments spaced according to master grid ........................................................................................................... 42
Figure 38: Overlap shape area defined with nodes that map directly do the master grid .................................................. 43
Figure 39: Tapered channel macro/micro-feature shown in thin segmented slice of the sealing interface (left) and isolated feature (right) ........................................................................ 44
Figure 40: Film thickness (left) and fluid state (right) example results using proprietary software .................................................. 45
Figure 41: Fluid pressure at the interface (left) and seal face deflection with temperature results (right) using proprietary software.......................................................................................... 46
Figure 42: FEA results of temperature as a function of depth ......................................................................................... 47
Figure 43: FEA results of torque as a function of depth ................................................................................................. 47
Figure 44: FEA results of leakage as a function of depth ............................................................................................... 48
Figure 45: Pictograph of fluid cavitation as function of depth in macro/micro-features closest to the seal interface OD ................................................................................................................. 49
Figure 46: FEA results of temperature as a function of percent area density coverage .................................................. 49
Figure 47: FEA results of torque as a function of percent area density coverage ............................................................ 50
Figure 48: FEA results of leakage as a function of percent area density coverage .......................................................... 50
Figure 49: Laser scanned image of tapered channel geometry laser ablated into the sealing interface .................................................................................................................................................. 51
Figure 50: Specialized friction tester designed to measure running seal face torque (Flowserve) ........................................... 52
Figure 51: Test assembly used to validate macro/micro-feature performance ........................................................................ 52
Figure 52: Carbon rotating face (left) and silicon carbide stationary face (right) .................................................................. 53
Figure 53: Silicon carbide face with macro/micro-features laser burned into the interface .......................................................... 53
Figure 54: Tangential (left) and radial (right) trace of a single macro/micro hydrodynamic feature 3µm in depth .................................................. 53
Figure 55: Labview GUI showing data acquisition acquired during each test ............................................................................ 54
Figure 56: Recorded interface temperature over 24 hours ................................................................................................. 55
Figure 57: Recorded torque data over 24 hours .............................................................................................................. 55
Figure 58: Test results showing temperature as a function of depth .................................................................................. 56
Figure 59: Test results showing torque as a function of depth ........................................................................................... 56
Figure 60: Graph showing one baseline test compared with the optimized tapered channel ........................................................................................................................................................................... 57
Figure 61: Post-test image of tapered channel seal face .................................................................................................... 58
Figure 62: Friction post-test radial trace of un-textured carbon face (left) and tapered channel carbon face (right) ................................................................................................................... 58
Figure 63: Ethane seal face with tapered channel macro/micro-features ........................................................................... 59
Figure 64: Comparison showing tapered groove performance (blue) under changing pressure profile vs. non-textured face running at 4.4 MPa ................................................................................. 59
Figure 65: Ethane test comparison ........................................................................................................................................... 60
Figure 66: Micro-dimple showing photo-thermal edge affects as depicted by the raised portion dimensioned as U [19] ......................................................................................................................................... 61
Figure 67: Comparison of FEA and actual test results of temperature and torque vs. depth ................................................... 62
**LIST OF TABLES**

Table 1: Ablation data for SiC[^5] ..................................................................................30
Table 2: Operating Conditions .............................................................................................46
ABSTRACT OF THE THESIS

Macro/Micro-Feature Development for Improved Hydrodynamic Performance at the Mechanical Seal Interface via Laser Surface Texturing

By

Joshua King Benedict

Master of Science in Mechanical Engineering

University of California, San Diego, 2011

Professor Frank E Talke, Chair

Mechanical Seals are used throughout the world as the principle method whereby fluid containment may be achieved between a rotating shaft and the shaft housing. As a key component in fluid transportation, storage, and containment, the reliability and performance of mechanical seals is very important.

This paper deals with the design, development, and optimization of unique macro/micro-features, used to improve the mechanical seal performance. This Thesis begins with an introductory background of mechanical seal component design, followed by an introduction of external forces and boundary conditions used to describe the seal as an axis-symmetric model. A derivation of the governing equation (Reynolds equation) used to describe fluid pressure between two surfaces is then summarized. Followed with a simplification of the Reynolds equation used to describe hydrodynamic lift under the influence of periodic features. The theory behind laser machining, and the
method of creating unique features using a laser is then described. The iterative logic structure of code that I developed is then described; A software tool which uses user defined operation (conducted on imported line segments defining the mask and beam shape boundaries) to define and simulate the laser process, which in turn generates the three dimensional geometry. One of the output files from this software (describing the ablated seal interface geometry) is then imported into a proprietary finite element analysis/ fluid mechanics package. The results of the analysis are used for an optimization study to predict the performance of one such unique macro/micro-feature called a tapered channel (patent pending). Following the FEA analysis, actual testing was performed at the Flowserve facility in Temecula, CA. From the test results, it is concluded that laser ablated macro/micro-feature on the sealing interface of a mechanical seal face can be optimized to improve the sealing performance. This is evident by the 65% reduction in torque, and 69% reduction in the temperature change measured at the sealing interface in comparison to an un-textured seal face.
INTRODUCTION

Mechanical Seals are used throughout the world as the principle method whereby fluid containment may be achieved between a rotating shaft and the shaft housing. As a key component in fluid transportation, storage, and containment, the reliability and performance of mechanical seals is very important. In order to increase the reliability of a mechanical seal while maintaining very low leakage, the opposing mechanical seal faces must run as close as possible (asperity contact) to one another while maintaining fluid at the interface. This type of operation is referred to as “mixed film lubrication” [2]. This fluid is essential, as it provides a method for positive fluid displacement at the sealing interface, without which, seal failure would occur due to excessive heat and wear.

Another form of operation is referred to as “full film lubrication” [2] where there is no asperity contact. This type of operation results in very low wear and heat generation but at the expense of higher leakage. In order to create full film lubrication, hydrodynamic features may be utilized at the sealing interface. Hydrodynamic features are geometric deviations, which force the localized fluid volume (film) to expand and contract. The expansion and contraction of the fluid volume gives rise to a change in localized fluid velocity (under cavitation) resulting in pressure changes. As a result, macro/micro-features produce a pressure gradient at the interface, causing the interface to push apart or “lift”.

This paper deals with the design, development, and optimization of a unique macro/micro-feature, called a tapered channel, and how this feature creates hydrodynamic lift-off but also minimizes leakage.
In addition to the micro-feature itself, the laser method by which such micro-feature is created is also considered. As part of the development process, a Matlab program was created to simulate the laser ablation process due to the endless number of micro-feature geometrical possibilities.

The methods used for laser simulation, along with the predicted performance based on finite element analysis (FEA) are presented. Following these predictions, the results of actual testing are presented and summarized.
Ch.1: Mechanical Seal Background

In order to understand the advantage associated with micro features at the mechanical seal interface, it is beneficial to review the purpose of a mechanical seal and look at the external forces that have shaped the current mechanical seal designs.

From a rudimentary perspective \(^1\) the mechanical seal is simply a device designed to contain a fluid between a rotating shaft and the shaft’s housing. Ideally, one could think of a ring attached to a spinning shaft that rides against the shaft housing as shown in figure 1.

![Figure 1: Basic concept of a Mechanical Seal](image)

In a world void of true physics, this system would be perfectly plausible; simply create a perfectly matched pair of parallel, non-distorting, wear resistant parts and you could hold the fluid by virtue of having the interface gap between the spinning disk and the shaft housing held so tightly, that the fluid molecules could not escape.

In the real world however, this concept is too rigid. Current manufacturing techniques (though precise and repeatable), produce slight variations in component geometry. For many applications, these variations would be of little consequence, but for a mechanical seal designed to contain molecules of virtually every fluid known to man, a slight variation is something that must be accounted for.
If we were to manufacture a seal as described, the combined variation of parts even within this simple system would create problems. If the variations in part tolerance stacked up such that the location of the rotating disk relative to the shaft housing increased, a gap would form between the two surfaces allowing fluid to escape. Tolerance stack-up in the other direction would eliminate the gap altogether resulting in contact pressure between the parts (resulting in heat and wear). These two modes are shown in Figure 2.

![Figure 2: Geometric variations in rigid model produce leakage (left), or contact (right) [1]](image)

This scenario is also too rigid to handle wear, vibration, pressure distortion, shaft run-out, concentricity misalignments, perpendicularity misalignments, or thermal expansion/contraction of the system (all of which are present in real applications). In addition to this, other parameters must be considered, including:

- Fluid properties (such as viscosity, specific gravity, and vapor pressure)
- Tribology at the sealing interface (wear resistance, galling resistance, friction coefficient)
- Size and percent density of particulate that may be present within the fluid
- Material compatibility of each component to the fluid
- Physical geometry of the equipment used to house the mechanical seal
• Operating conditions under which the mechanical seal will be performing (such as temperature, pressure, shaft speed, duration)

• Material properties such as electrical and thermal conductivity and residual magnetism within a system are important for certain applications.

• Manufacturability/Machinability

• Availability of materials and resources

• Safety/Environmental (Toxicity, Disposal)

• Cost associated with all materials, manufacturing, transporting, installation etc...

As a result of these considerations, each mechanical seal is actually a complex system of high precision custom components designed and manufactured specifically for a particular piece of equipment and a specified range of operating conditions. Though the types and styles of mechanical seals available are as vast as the number of applications in which they are used, the majority of mechanical seals today have the same basic components. These components are outlined in Figure 3.
Figure 3: Quarter section of seal assembly, indicating basic components used in mechanical seals

The main purpose of each basic seal component shown in Figure 3 (following the sequence of the ballooned item numbers) is:

1. Spring Element: This item pushes the seal faces together, providing the proper amount of closing force needed to contain fluid under static conditions, while allowing the attached face varying degrees of freedom (both axially and radially). In this manner, the seal interface can contain the fluid despite undergoing certain levels of wear, misalignment, run-out, distortion, vibration, thermal expansion etc... In the case of a bellows design, the spring element may also act as a part of the fluid containment device, eliminating the need for a dynamic O-ring (10).

2. Rotating Seal Face: The shape of this ring has been designed to accommodate the different forces (such as pressure distortion and thermal distortion) that are present during operation. Everything from the thin profile at the sealing interface, to the notch cut out of the back, is there to allow the seal to run parallel to the
stationary face for proper sealing performance, despite pressure and thermal affects. In the configuration shown, this ring is usually made of a carbon graphite material.

3. Stationary Seal Face: This ring sits within the stationary housing (gland) which is attached to the shaft housing. The stationary seal face is designed to be replaceable (as the sealing interface wears over time), it also allows the seal manufacturer to control all aspects of the sealing interface (rather than sealing against the customers housing). In the configuration shown, this ring is usually made of Silicon Carbide.

4. Shaft Sleeve: This item is designed to slip over the shaft, allowing the entire seal assembly to be installed quickly as a cartridge. The sleeve provides a controlled surface over which the dynamic portion of the mechanical seal may slide, and protects the shaft from seal induced wear.

5. Secondary Containment Bushing: This item is designed to help contain the fluid in the event that the mechanical seal faces fail. This will not provide full containment like the sealing interface, but will block the majority of the flow.

6. Seal Gland: This item bolts up to the shaft housing, and provides external fluid access ports used to help remove particulate from the seal interface. This gland acts as a secondary housing to hold the stationary sealing ring, the secondary containment bushing, and connect components together as a cartridge, while protecting the shaft housing.

7. Setting Clips: These clips are used to align the sealing components as a cartridge prior to installation, and protect the seal during shipment. Once installed, the clips are removed.
8. Drive Collar: The drive collar is the coupling between the shaft and the sleeve causing the rotating components of the mechanical seal to move with the shaft.

9. Drive Pins: Each seal face uses a device such as a pin to keep the face from rotating relative to components holding the face (Only the stationary drive pin is shown in Figure 3)

10. Dynamic gaskets: Gaskets (typically O-rings) are used in locations where relative motion between components is significant.

11. Spring Holder: This is designed to hold the rotating components together. The spring holder has drive pins (not shown) which engage the rotating face, while allowing the face to move axially to maintain the proper contact during operation.

12. Backing Ring: This device transmits the spring load evenly across the back of the rotating face (eliminating any localized spring distortion on the face).

13. Flush Port: Allows the user to help control the fluid environment of the seal by providing a means for positive displacement (for removing particulate and vapor trapped above the sealing interface) and for temperature and pressure control (for acceptable thermal and vapor pressure margins).

14. Static gaskets: Gaskets (typically O-rings) are used in locations where relative motion between components is insignificant.

All of these components have been designed to help overcome the sealing environment, allowing the seal interface to run as close and parallel as possible, while maintaining fluid within the interface. The mechanical seal is now able to maintain proper sealing contact despite being subject to an environment where certain amounts of vibration, miss-alignment, expansion, and component wear are expected.
Ch.2: Basic Theory of Seal Operation

Having provided accommodations for the sealing environment, we can now look at the physics that dictate mechanical seal performance. Due to the current design features (Outlined in Chapter 1) and symmetric nature of mechanical seals, the entire seal assembly performance can be predicted by looking at an axis-symmetric model of the sealing faces themselves. This is done by creating a free body diagram in which all external components and forces are replaced with an equivalent set of external forces and boundary conditions acting on the two sealing rings. For example, if we look closely at a cross section of the seal faces within the system described above, we would end up with a view similar to that shown in Figure 4.

![Figure 4: Close up quarter section view of the mechanical seal faces](image)

This system could then be reduced by first eliminating all physical components (including the fluid) outside of the sealing rings, replacing these components with the appropriate forces as shown in Figure 5.
A static summation of these forces in the axial direction results in the following closing force equation:

\[ P_{total} = \Delta P (B - K) + P_s - P_h \]  

(2.1)

where:  
\( P_{total} \) = Total face closing pressure  
\( \Delta P \) = Differential pressure across seal interface  
\( B \) = Balance ratio  
\( K \) = Pressure drop factor  
\( P_s \) = Spring pressure on seal face  
\( P_h \) = Hydrodynamic lift pressure

As we look at equation 2.1, we see that there are different factors that contribute to the closing pressure at the seal interface. Differential pressure across the seal interface is calculated as follows:

\[ \Delta P = |P_o - P_i| \]  

(2.2)

where:  
\( P_o \) = Pressure at the outer diameter of the sealing interface (Pa)  
\( P_i \) = Pressure at the inner diameter of the sealing interface (Pa)
Balance is simply a function of the seal face geometry, and is therefore easy to control. In this equation, balance ratio \( B \) is a ratio of hydraulic closing forces as outlined in Figure 6.

The balance ratio is calculated using the following equations (See Figure 6):

\[
B = \frac{OD^2 - BD^2}{OD^2 - ID^2} \quad \text{if} \quad P_o > P_i \quad (2.3)
\]
\[
B = \frac{BD^2 - ID^2}{OD^2 - ID^2} \quad \text{if} \quad P_o < P_i \quad (2.4)
\]

where: \( OD \) = Outer diameter of the sealing interface (m)  
\( ID \) = Inner diameter of the sealing interface (m)  
\( BD \) = Balance diameter of the sealing interface (m)

The pressure drop factor \( K \) is a function of the pressure profile at the sealing interface. The pressure drop factor is arguably the most unpredictable parameter in this equation. It is governed by the interface profile which is a function of the sealing interface geometry. Throughout the life of a seal, this parameter is likely to change, as the initially flat sealing interface undergoes different modes of wear (ranging from simple asperity contact, to 3rd body abrasion). Figure 7 shows different \( K \) values (For a linear pressure profile \( K=0.5 \), convex or converging film \( K>0.5 \), concave or diverging film \( K<0.5 \)).
It may be said that the K value of the seal will ultimately determine the seal performance. If the seal becomes subject to significant environmental changes such as a pressure spike, shaft motion, thermal changes, or increase in solid particulates, the magnitude of the environmental change is likely to produce a proportional response from the seal. This change can cause vibration, deflection, tilt, and nutation at the sealing interface. Uneven motion at the interface results in uneven wear which alters the profile at the interface and the K value of the system. Significant changes in the K value can cause the seal to wear out due to an increase in the hydraulic closing force (referred to as burning up the seal face), or it may lead to excessive leakage due to an increase in hydraulic pressure at the sealing interface (referred to as the faces blowing open).

Similar to the balance ratio, the spring pressure ($P_s$) is easy to control, as it is also a function of component geometry. Contact pressure at the seal interface due to spring load is calculated as:

$$P_s = \frac{R(FL - WL)N_s}{A}$$  \hspace{1cm} (2.5)

$$A = \frac{\pi(OD^2 - ID^2)}{4}$$  \hspace{1cm} (2.6)

where:  
$R =$ Spring rate (N/m)  
$FL =$ Free length or uncompressed length of each spring (m)  
$WL =$ Working length, or compressed length of each spring (m)  
$N_s =$ Total number of springs used to load the face
Contact surface area of the narrow face at sealing interface (m²)

The final component of this equation is the hydrodynamic lift pressure (Pₜ). Hydrodynamic lift pressure is governed by the geometry of the sealing interface (making this parameter controllable, though somewhat cumbersome to calculate). The benefit of hydrodynamic lift will be better understood in Chapter 3, where the calculations that characterize the pressure affects at the seal interface are outlined, however from a summary standpoint, micro-features provide the ability to reduce interface friction while maintaining load support. In other words, micro-features can help offset the unpredictable nature of the pressure drop factor K (which is the underlying source of seal performance). As a result, the following benefits can be obtained from applying hydrodynamic features properly:

- Reduces interface contact (leading to friction reduction)
- Reduces wear at interface (for longer seal life; resulting in less down time and repair costs)
- Reduces torque (resulting in less power consumption; saving money and energy)
- Helps prevent catastrophic failures (Due to more stable k value)
- Reduces heat generation

To solve this system, one would also have to define the surface boundary conditions. These include: fluid pressure, convection, temperature, heat flux, and fluid regions across the surface of each seal ring. Figure 6 shows an example of clearly defined surface boundary regions; where the dark outline towards the top and back surface of the rotating face represents the fluid region and the lighter outline mainly located towards the bottom and back side of the stationary face indicates the
atmospheric fluid boundary for this seal. These two regions are separated by the o-rings and sealing interface.

![Diagram of axis-symmetric fluid boundary](image)

**Figure 8: Example of an axis-symmetric fluid boundary defined by the gasket locations**

Changes in seal geometry would also have to be considered when calculating the true performance of a mechanical seal. As a result, predicting seal performance requires an iterative method of coupled multi-physics. Pressure and thermal loads cause distortion of the seal faces which in turn alter the fluid mechanics at the interface and resultant pressure distribution. The change in interface pressure distribution in turn alters the distortion of the seal rings. This process must be iterated on until equilibrium is reached. For the purpose of this Thesis, calculations will be limited to that of a summary of the fluid pressure distribution at the sealing interface as outlined in Ch.3 and 4. To fully describe mechanical seal behavior would require work far beyond the context of this thesis. Anyone desiring further information can turn to those who have already masterfully laid out such work, referenced in Ch.3 and 4.
The pressure balance equation (2.1) described above is a good indication of closing force. This information is valuable for designing the basic shape of a mechanical seal, and can provide a good estimation of seal performance for one with experience and knowledge of designing mechanical seals for known applications. However, this equation is but a summation of external forces affecting the seal interface, which is contingent on the interaction that occurs at the interface itself.

Therefore, in order to truly predict sealing behavior, one must understand what is going on at the interface itself. To do this, we must consider the Reynolds equation. The Reynolds equation is the governing equation (derived from the Navier-Stokes equation) which is used to model the fluid pressure distribution at the sealing interface. Many authors have derived the Reynolds equation from a mechanical seal context. Without re-writing the entire comprehensive derivation here, I would like to summarize the key equations and assumptions masterfully composed by Alan O. Lebeck\,[2]. Equations, figures, and general context that follow were taken from his work:

To define the pressure distribution at the sealing interface, you begin from a general lubrication problem characterized by a finite volume of fluid bounded within two arbitrary surfaces moving relative to one another at a close distance (as shown in Figure 9).
Using this geometry, one can describe the pressure and fluid motion by considering the effects of gravity, viscosity, inertia, and velocity gradients, as defined by the Navier-Stokes equation of motion:\[3\]:

\[
\begin{align*}
\rho g_x - \frac{\partial p}{\partial x} + \mu \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right) &= \rho \frac{du}{dt} \\
\rho g_y - \frac{\partial p}{\partial y} + \mu \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right) &= \rho \frac{dv}{dt} \\
\rho g_z - \frac{\partial p}{\partial z} + \mu \left( \frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right) &= \rho \frac{dw}{dt}
\end{align*}
\] (2.7) (2.8) (2.9)

where: $\rho$ = Fluid density
$x, y, z$ = Cartesian coordinates
$g$ = Gravitation force in direction (x,y,z) indicated (m/s$^2$)
$p$ = Pressure (pressure gradient in direction indicated) (Pa)
$\mu$ = Absolute viscosity (Pa · s)
$u, v, w$ = Components of velocity along x,y,z Cartesian coordinate directions (m/s)
Next, using a control volume of fluid (Figure 10), such that “The net volume rate of flow into the control volume must equal the time rate of change of the control volume” [2].

Valid under the appropriate boundary conditions and assumptions listed below:

- Gravitation force of fluid are neglected (insignificant compared to other forces within the system)
- Inertia effects of the fluid are neglected (insignificant compared to viscous forces present)
- Fluid consists of thin film laminar flow (void of turbulence and vortices)
- Localized pressure changes normal to the sealing interface is zero
- Only tangential component of velocity is considered at the sealing interface
- Condition of continuity (Fluid flow into control volume of Figure 10, must equal time rate of change of the control volume)
- Fluid considered is Newtonian (of constant viscosity and density)
- Squeeze effects are neglected
- Interface is comprised of flat parallel rings, one rotating, one stationary with averaged defined (constant) surface roughness

The Navier-Stokes equation is transformed into the following Reynolds equation:

\[
\frac{\partial}{\partial x} \left( \frac{h^3}{12\mu} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left( \frac{h^3}{12\mu} \frac{\partial p}{\partial y} \right) = \frac{U \partial h}{2 \partial x}
\]

(2.10)

where:  \( h \) = Nominal film thickness (m)

\( U \) = Sliding speed at mean radius (m/s)

Due to the cylindrical geometry of mechanical seals, it is more useful to use a polar coordinate system to define the Reynolds equation. This is done by redefining the control volume (Figure 11) with polar coordinates rather than the Cartesian coordinate system used in Figure 10.

Fluid flow through this polar control volume is now defined in the tangential and radial directions, transforming the Reynolds equation as shown:
where: \( r, \theta \) = Cylindrical Coordinates

The equation defined here is the general Reynolds equation which can be used to describe the fluid pressure distribution at the sealing interface. It should be noted that this equation is the general form of the Reynolds equation used for the analysis conducted in Ch. 7. Further simplification to this equation can be used when considering periodic hydrodynamic features as described in Ch. 4 (which is convenient for hand calculated estimates).
Simplification of the Reynolds equation can be derived by considering the periodic function of the hydrodynamic features, similar to the example shown in Figure 12.

![Figure 12: Example of a periodic geometry used to create hydrodynamic lift][2]

The periodic geometry of hydrodynamic features allows us to use the following assumptions (in conjunction with those already prescribed):

- Seal acts as a narrow bearing (width of the seal is much smaller than the length of each wave).
- Features are considered harmonic waves of a given amplitude ($h_a$).
- Minimum film thickness is greater than or equal to the contact film thickness.
- Any contact that occurs at the sealing interface will be the fraction of load support required for equilibrium, and this contact will occur at the wave peaks.
- The wave features do not deflect.
- Fluid is under cavitation (pressure equals zero) for the divergent portion of each wave feature.
- One face is wavy, the other is nominally flat.
These assumptions lead us to the following simplified Reynolds equation:

\[
\frac{\partial}{\partial r} \left( \frac{r h^3}{12 \mu} \frac{\partial p}{\partial r} \right) = \frac{r \omega \partial h}{2} \frac{\partial \theta}{\partial \theta} \tag{2.12}
\]

Now, as we neglect the variations in flow due to the change of circumference (assuming a thin interface), for a seal pressurized at the outer diameter, we are left with the following equations:

\[
p - p_i = \frac{3 \nu \mu d h}{h^3} (r - r_i)(r - r_o) + (p_o - p_i) \frac{r - r_i}{r_o - r_i} \quad \text{for: } 0 < \theta < \frac{2\pi}{2n} \tag{2.13}
\]

\[
p - p_i = 0 \quad \text{for: } 0 < \theta < \frac{2\pi}{2n} \tag{2.14}
\]

where: 
- \(r_i\) = Inner radius (m)
- \(r_o\) = Outer radius (m)
- \(p_i\) = Pressure at the inner radius (Pa)
- \(p_o\) = Pressure at the outer radius (Pa)
- \(r\) = Radial location being analyzed (m)
- \(p\) = Pressure at location \(r\) (Pa)

These equations define pressure at any location in the sealing interface, showing that pressure within the convergent portion of the wave is the sum of the hydrodynamic and the hydrostatic pressure at that location, and zero otherwise (assuming that there is no pressure at the inner diameter of the seal).

Integrating the pressure distribution from equation 2.13, (over the regions in which the specific wave geometry is applied) yields the load support across the entire sealing interface as defined:

\[
W_f = 2\pi r_m (r_o - r_i) \left[ \frac{\eta \mu \omega}{8\pi} \left( r_o - r_i \right)^2 \left( \frac{1}{h_{min}^2} - \frac{1}{(h_{min} + 2h_d)^2} \right) + \frac{p_o}{4} \right]
\]

if:

\[
\frac{\eta \mu \omega}{8\pi} (r_o - r_i)^2 \left[ \frac{1}{h_{min}^2} - \frac{1}{(h_{min} + 2h_d)^2} \right] > \frac{p_o}{4} \tag{2.15}
\]
\[ W_f = 2\pi r_m (r_o - r_i) \frac{P_o}{2} \]

\[ \text{if: } \frac{\eta \omega}{8\pi} (r_o - r_i)^2 \left[ \frac{1}{h_{min}^2} - \frac{1}{(h_{min} + 2h_a)^2} \right] < \frac{P_o}{4} \]  \hspace{1cm} (2.16)

where:  
\( W_f \) = Load support by fluid pressure (N)  
\( h_{min} \) = Minimum film thickness (m)  
\( h_a \) = Net waviness amplitude at the interface  
\( n \) = Number of waves around the seal face

This is an iterative process, in which the minimum film thickness \( h_{min} \) is initially approximated using the contact roughness height, (which is about 3 times the standard deviation of the combined roughness of the interface material), and later determined by converging on a film thickness that provides equilibrium of the system. In general, one desires to obtain a fluid film which is similar in size to the roughness of the seal face \(^1\).  

A fluid gap on this order helps to minimize the amount of asperity contact between the seals, while providing minute amounts of positive fluid displacement at the seal interface needed for lubricity \(^4\). For basic seals, this fluid film is on the order of 12 millionths of an inch (\( \approx 0.3 \) microns) thick.

Once the film thickness is found, the total seal leakage can be calculated as follows:

\[ Q = \int_0^{2\pi} - \frac{dP}{dr} \left| \frac{h^3}{12\mu} r_m d\theta \right. \]  \hspace{1cm} (2.17)

where:  
\( Q \) = Total leakage by volume (m\(^3\))  
\( r_m \) = Mean face radius (m)

The total friction force can be found by combining the viscous and contact friction forces as defined:

\[ F_f = \frac{\pi r_m^2 \mu \omega (r_o - r_i)}{h_{min} \sqrt{1 + 2h_a/h_{min}}} \]  \hspace{1cm} (2.18)
\[ F_m = f_c W_m \]  \hspace{1cm} (2.19)

where: \( F_f \) = Tangential viscous friction, 
   neglecting viscous shear in cavitated regions (N) 
\( F_m \) = Tangential contact friction (N) 
\( f_c \) = Contact friction coefficient 
\( W_m \) = Contact load (N)

Lebeck\(^{[2]}\) also provides more advanced analysis, allowing for taper and deflection at the sealing interface, along with a description of the boundary conditions used, and their iterative affects. Though such methods were used for predicting the seal performance associated with the micro-features developed in this thesis (Ch.7), further depth is left to the reader.
Ch.5: Creating Micro-Feature Geometry through Laser Ablation

In the past, many hydrodynamic features were produced using some method of sandblasting, or deformation grinding. The repeatability and accuracy of such methods were less than ideal, requiring frequent testing and reprocessing until proper performance parameters were achieved. As laser machining technology improved, and became readily available and more affordable, these old methods were quickly replaced by the more precise, accurate, and efficient method of producing features on a mechanical seal interface.

There are many types of lasers available. Most popular for micro machining are Excimer and YAG. Though the power source (gas, solid state, semiconductor, liquid, free-electron, X-ray etc...), wavelength, and energy range of each laser differs, the basic principle behind the laser is the same.

Lasers work by emitting a stream of photonic energy packets known as electromagnetic radiation (which we call light). The light produced in the laser is designed to be a single wavelength, called Monochromatic radiation. The electric and magnetic waves within this monochromatic light (as in all light) always travel perpendicular to one another and at right angles to the direction of propagation [5] as shown in Figure 13.
The light (or released photons) within a laser are created through atom excitation, in a process where atom collision is purposefully instigated. The induced collision process transfers energy to atoms such that electrons gathered within the various levels (shells) surrounding the nucleus of those atoms absorb the impact energy (called absorption). This energy propels the electrons into higher shell positions (called an excited state). The electrons do not remain in this excited state very long (about $10^{-8}$ seconds), quickly releasing the energy as packets of photons (called spontaneous emission) allowing the electrons to jump back down to their stable position. It was discovered that if an emitted photon collides with an excited atom traveling in the same direction, the excited atom will emit photons (called stimulated emission). This light amplification is the key to producing laser beams (where laser is an acronym for “light amplification by stimulated emission of radiation”).
Lasers are therefore created by taking advantage of this light amplification through optical feedback produced by bouncing photons back and forth between mirrors within the laser chamber. The excited photons multiply as they bounce back and forth until they are released through a transparent section of one of the mirrors (See Figure 15).

The formed beam typically passes through a shutter (for pulse control), and is often propelled and reflected down a series of tubes used to position the beam above its
intended target. As the light moves through the tubes, one can take advantage of the lower density photonic energy, by conditioning and shaping the beam using geometrical shapes called masks. Once formed, and prior to colliding with the intended target, the photons are sent through a final refraction lens, which focuses the laser beam, thereby consolidating the energy density of the beam. When designed properly, the photonic energy of the laser acts similar to an atomic bomb (on a small scale), as the photonic energy collision results in an explosion against the material surface. The point of impact turns into a ball of plasma, sending shock waves in all directions causing the surrounding material to undergo some state of melt or vaporization. Figure 13 shows a diagram illustrating such a reaction.

![Diagram of laser impact across the material interface](image)

**Figure 16: Laser impact across the material interface**

This reaction occurs very quickly, and on such a small scale, that the energy released is quickly absorbed by the surrounding atmosphere. Images of this reaction have been captured using extremely quick photography. Figure 17 shows a high speed picture of the interaction between the laser and substrate.
After the collision, the type of reaction and final state of the material depends on the energy density of the blast (called fluence), the laser wavelength, and the material being ablated. Laser induced interaction varies from fully photochemical to fully photothermal as shown in Figure 18.

Photochemical interaction occurs when the imparted energy from the photon collision exceeds the elemental bonding energy between material molecules (such as covalent bonds). When this occurs, the material is vaporized (process often referred to as cold ablation), causing the complete removal of material no longer attached to itself.
The majority of particles under this regime are ejected from the surface during the explosion as shown in Figure 19.

![Photographs of blast wave (left) and ejected material (right), taken 500ns and 2.9µs after start of laser pulse][8]

Photo-thermal interaction occurs when the energy imparted is sufficient to create a phase change of the material (from solid to liquid) within the immediate vicinity without vaporizing the material. When this occurs, the material moves in fluidic motion within the immediate vicinity without removal. Figure 20 [9] shows an example of a photo-thermal interaction:

![Copper showing continuous photo-thermal interaction after receiving 50 pulses (left), and 500 pulses (right)][10]
The majority of materials exhibit some combination of photo-chemical/photo-thermal interaction under laser ablation. An example of this is shown in Figure 21.

![Figure 21: silicon micro-dimple formed by single pulse](image)

Perhaps the most common material pairs used for mechanical seals are either silicon carbide running against carbon, or silicon carbide running against itself. Though there are different types and grades of these materials, in general, silicon carbide is an excellent material for laser machining. Silicon carbide is a type of ceramic, and when the photons collide against this material, the resultant interaction is almost completely photochemical. As a result, the material ablates, forming relatively smooth, clean, accurate and precise features. When considering the type of laser to use the substrate should be considered. Lists are available giving information regarding the wavelength, fluence, and ablation depth/pulse that one should expect to remove as shown in Table 1.

<table>
<thead>
<tr>
<th>Wavelength (nm)</th>
<th>Threshold fluence (J cm⁻²)</th>
<th>Etching rate (nm/pulse/ J cm⁻²)</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>193</td>
<td>1.5</td>
<td>5</td>
<td>Gower (1993)</td>
</tr>
<tr>
<td>248</td>
<td>2.0</td>
<td>4.7</td>
<td></td>
</tr>
<tr>
<td>308</td>
<td>2.5</td>
<td>7.6</td>
<td></td>
</tr>
<tr>
<td>193</td>
<td>1.5</td>
<td>4.4</td>
<td>Gower (1993)</td>
</tr>
<tr>
<td>248</td>
<td>2.3</td>
<td>4.1</td>
<td></td>
</tr>
<tr>
<td>308</td>
<td>4.0</td>
<td>6.4</td>
<td></td>
</tr>
<tr>
<td>308</td>
<td>4</td>
<td>10</td>
<td>Geiger et al. (1991)</td>
</tr>
<tr>
<td>308</td>
<td>4</td>
<td>4</td>
<td>Tönshoff and Gedrat (1989)</td>
</tr>
<tr>
<td>308</td>
<td>3</td>
<td>12</td>
<td>Schmatkjo et al. (1988)</td>
</tr>
</tbody>
</table>
Currently, there are many different types of micro depth features and other periodic laser ablated designs used in the sealing industry. Most of these features fall well within the narrow bearing assumptions, having features with lengths much larger than the width of the sealing interface (Flowserve, and others). One of these patterns uses waves which extend out past the sealing interface to provide additional hydrostatic load support, while promoting circulated flow for particulate expulsion (See Figure 22).

![Figure 22: Wavy seal with combined hydrostatic/hydrodynamic features (Flowserve Corp)](image)

Others have provided micro-dimple features across the sealing interface (Figure 23). Numerous studies have been conducted to evaluate the effects of micro-dimple surface texturing on friction reduction in mechanical seals.[12-18]

![Figure 23: Micro-dimples produced using single laser pulse](image)

Optimum ratios for a dimple configuration, depth (h) to diameter (D) (Fig. 1), range from 0.02 to 0.5, and area density ratio (ratio of dimple area to seal face area) range from 20 to 55%.
Within this range it has been shown that friction can be reduced by as much as 50%. These studies indicate that if the ratio of depth to dimple diameter is reduced below 0.02 a dramatic reduction in performance is seen. Also observed in one investigation [20] is the possibility that micro dimples can become filled with debris, affecting performance over time.

Despite the reduced performance predictions found in these studies for very small h/D ratios, a macro/micro feature (called a tapered channel), shows excellent performance. The term macro/micro feature is used to describe a feature that has a depth to size ratio, h/L (Fig.25) which is at least one order of magnitude smaller than current dimple configurations.

This geometry is well below what may be considered useful from a performance standpoint; however subsequent analysis and testing results (Ch.7 and 8) will show that the tapered channel geometry gives rise to significant cavitation affects as well as hydrodynamic load support resulting in low friction/wear characteristics while maintaining low leakage. Different configurations of the tapered channel geometry are shown in Figure 26.
The following will focus on the design, analysis, and optimization of this tapered channel geometry. Starting with the use of custom developed code, followed by analysis tools and testing. Results will show that it is possible to design unique macro/micro features capable of reducing friction by 65%. Further, such features will show significantly lower face temperature, debris resistance, low to zero measurable leakage, and low to zero wear when compared to an un-textured seal face.
In order to analyze geometry created through various mask configurations, a simulation program using Matlab® programming software was developed. The software was designed to simulate the actual laser manufacturing process by allowing the user to select any combination of imported mask shapes (designed and imported from AutoCad® using a script). Once the mask shape is selected, the user enters the seal face geometry followed by any combination of movements (Fig.27) and laser operations.

Figure 27: Map showing relative direction of each laser and mask axis

Once each set of movements is completed, the user is iteratively prompted for any additional laser operations, until the desired sequence of machining operations is completed. Once specified, the program performs the tasks, and creates the following outputs files:

1. File containing the resulting 3D coordinate geometry defining the seal interface with burned macro/micro feature geometry
2. File containing the machine language used by the laser to perform the exact laser operation that was just simulated
3. A .pro file (pattern profile in three dimensions, \( r, \theta, \) and depth) which is a specific data file used by a proprietary FEA/Fluid mechanics software to predict actual seal performance of the mechanical seal.

Originally, the simulation software used a finite difference approach in which mask shapes used were defined by a grid of equally spaced points (spaced 2.5\( \mu \)m apart to avoid pixilation due to the high aspect ratio of each macro/micro feature created). Points were captured that bounded the mask shape (See Figure 28), and then were compared to points on a master grid (which contained all points possible within the maximum size of the laser beam).

![Figure 28: Finite difference point method used for defining mask shape from dense population of grid points](image)

This initial approach was very cumbersome and slow. An iterative comparison of every point within the mask to every point within the master grid, and also accounting for each point on the grid already ablated during the current laser operation required significant computation. The reason for this original technique was due to the relative rotation or translation of the mask, which allowed the points (within the mask shape) to line up at some offset distance from the nodes defined in the master grid. This made it necessary not only to compare each point, but to provide a search boundary around
each point on the master grid to avoid missing ablation points. Without the search boundary ablation points would be missed (Figure 29) and with the search boundary each ablation point had to be accounted for to avoid multiple ablations due to overlapping search boundaries within each pulse. This approach also limited the 3D simulation size down to single macro/micro-feature geometry. Populating an entire seal face would be very time consuming, requiring a second operation where the macro/micro-feature was copied into a user defined array to populate the seal.

Figure 29: Simulated ablation points defining macro/micro feature missed prior to using search boundary

The code was revised, so that rather than encapsulating points within a grid, the mask shape itself would define the boundary (re-defined using a .lisp program, which takes in any AutoCAD shape, and subdivides it into multiple line segments along the shape boundary). By so doing, the processing time was reduced by orders of magnitude, allowing the user to simulate macro/micro-feature development across the entire sealing interface without requiring a second operation. This new approach came with a new set of challenges. The overlapped portion of each mask had to be defined dynamically. Once defined, the overlapped shape was then mapped onto the user
defined seal shape. This in turn was used to store the current ablation depth at every location on the seal face grid (called the master grid). The basic control logic algorithm behind this new line segment method occurs in the following steps:

1) User specifies the mask shape and beam shape that will be used (Figure 30), the laser wattage used, and then specifies the laser process (including the start point and end point of each axis moved during ablation, along with the step size of whichever axis is being tracked). The program automatically subdivides the movement of each axis specified into the same number of steps defined by the axis being tracked.

![Figure 30: User selects the predefined mask shape and beam shape](image)

2) Mask and table motion occurs as defined by the user to align the laser above the seal face in the desired home position. Once all axes have moved into their home position, the laser operation begins. The laser will fire based on the pulse rate given (defined by the step size mentioned in part 1). Thus the laser simulation occurs by moving all designated axis to the next step size location of each axis (each axis moves its designated step size in sequence). This is simulated by redefining the shape location based on the combined translation and rotation of each axis as designated by the user (Figure 31)
3) Next, the program returns the overlapped shape to be ablated into the master grid using bounding logic. This logic determines the resulting shape of the beam as it passes through some portion of the mask boundary characterized by boundary intersections. If there are no intersections, a second subroutine determines whether no ablation occurs due to a physical separation of the two boundaries, or whether full ablation occurs in the shape of the beam or mask bound within the other. If intersections do occur, they will occur in pairs due to physical limitations of actual mask geometry which are defined as polygons and cannot self-intersect. The following steps are taken to determine the overlapped shape:

a. The coordinates of each intersection are stored within an array and indexed within each boundary shape. This is accomplished by subdividing each intersecting line segment into two segments joined at the intersecting location (Figure 32). The intersecting coordinates within the array will later (step d) be used to link each section of the overlapped boundary together. The overlapped boundary sections are determined using "assessment points" (step b).
b. Along the perimeter of each shape (midway between each set of intersection points), two new points called “assessment points” are placed as shown in Figure 33. These assessment points are placed at a very small offset distance on either side of the shape boundary (defined to be perpendicular to the slope of the line segment closest to the assessment points). This ensures that there is one assessment point “inside” the boundary shape at each midway location and one point “outside”.

c. Each assessment point begins a horizontal line segment ending a finite distance past the furthest shape boundary in the positive X direction as
shown in Figure 34. The number of times each line segment intersects a particular boundary shape determines whether that assessment point falls “inside” or “outside” of said shape. If the number of intersections is odd, a marker returns true (designated by the number one) indicating that the point is “inside”. Figure 34 shows an example of how this mapping is displayed: Next to each point, there is a two digit marker. The first number indicates whether the assessment point is within the beam boundary (represented with a dashed line), and the second marker indicates whether the assessment point is within the mask boundary (represented by the solid line).

Figure 34: Odd/even mapping of assessment line segments intersection

d. Between each sequential pair of intersections, there can only exist one assessment point overlapped by both shapes (indicated as [1,1] in Figure 34). The boundary shape containing this assessment point is stored, while the other shape bound between said intersection points is rejected.
After considering all intersection pairs, the stored boundary shape segments are linked together at each intersection to form the overlapped shape (Figure 35). This shape constitutes the actual shape of the laser beam after passing through the mask at the specified orientation.

4) Next, two vertical columns of nodes are created. These columns start below the overlapped shape and extend up past the overlapped shape on both sides as shown in Figure 36. The nodes within each column are spaced at intervals corresponding to the vertical resolution of the master grid (which defines the seal interface).
5) The vertical bounding nodes on either end of the shape constitute new line segments. These segments are created purposefully at the Y-coordinate spacing of the master grid, and are used to redefine the boundary of the overlapped shape as horizontal line segments (rather than boundary line segments) where the nodal segments intersect the overlapped shape as shown in Figure 37.

Following this Y-coordinate transformation, the overlapped shape is then fully populated with points evenly spaced between each horizontal pair of Y-coordinates according to the X-coordinate grid spacing of the master grid as shown in Figure 38.
By populating the overlapped shape in this manner, these points map directly into the master grid without having to compare points, worry about point offset, or keep track of which points have already been ablated. Instead, each point within the overlapped shape is immediately updated in the master grid by incrementing the Z coordinates of those points to keep track of ablation.

6) This process continues iteratively (running through each user defined step until each pulse within the laser simulation code is completed). Upon completion, the master grid contains the number of times each point was ablated. These values are then multiplied by the material specific removal rate (factor of laser intensity, wavelength, pulse duration, and material properties) to define the actual depth at each location. Once completed, the information contained within the master grid provides actual 3-D point space data describing the seal face geometry. From this information, a 3-dimensional surface contour can be rendered by triangulating adjacent coordinate points to produce intersecting planes.
The simulation method just described was used in conjunction with a surface contour program called Surfer 9® to create the 3-dimensional tapered channel geometry as shown in Figure 39. Note that these features are depth magnified to allow the user to visualize the created geometry due to the high width to depth aspect ratio.

Figure 39: Tapered channel macro/micro-feature shown in thin segmented slice of the sealing interface (left) and isolated feature (right)
Ch.7: Macro/micro-feature Optimization using Computer Analysis

The next important step for implementing the macro/micro-feature is that of performance analysis. Prior to investing the time and money used to create these features, it's beneficial to analyze the features. In order to do this a finite element analysis should be conducted. For this analysis the .pro file (created within the simulation process of Ch.6) in conjunction with proprietary FEA/Fluid mechanics software was used. This software allows the user to enter in the appropriate loads, boundary conditions, fluid properties, and material properties (as discussed in Ch.2). These conditions are coupled with the periodic seal face geometry (.pro file) and run through an iterative analysis to find the steady state of the system.

Examples of analysis results are shown in Figures 40 and 41.

![Figure 40: Film thickness (left) and fluid state (right) example results using proprietary software](image-url)
In order to optimize the feature shown in Figure 39, each analysis was conducted under identical operating conditions varying only one geometric parameter at a time. Table 2 shows the operating conditions used:

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Water</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fluid Temp</td>
<td>38°C</td>
</tr>
<tr>
<td>Pressure</td>
<td>1.03 MPa</td>
</tr>
<tr>
<td>Shaft Speed</td>
<td>1500 rpm</td>
</tr>
</tbody>
</table>

Analysis was conducted first using an un-textured flat face (void of macro/micro features). This was followed by an optimization study for the optimum tapered channel geometry. Optimum depth was determined by comparing the results of tapered channels burned from 1.5 µm to 7.5 µm without changing width and length of the geometry. Results of this analysis are shown in Figures 42-44.
Figure 42: FEA results of temperature as a function of depth

From the results found in Figure 42, it is apparent that the temperature at the sealing interface is minimized when this particular macro/micro-feature has a depth of 4.5 µm.

Figure 43: FEA results of torque as a function of depth

Results from Figure 43 are consistent with the results of Figure 42, showing that torque is also minimized at a depth of 4.5 µm.
Again, the results shown in Figure 44 are as expected, as a reduction in torque and temperature are inversely related to the film thickness, which is directly related to leakage. The 40% increase in leakage shown, though significant relative to the flat face (0.0 µm depth), is extremely small. As a result, the analysis indicates that the 4.5µm depth macro/micro-feature analyzed is the optimum depth for this feature. In reviewing these results and comparing the fluid state at each depth, it is seen that the optimum feature is one which maintains a state of full cavitation. When cavitation is maintained across the area of this particular macro/micro-feature, an increase in hydrodynamic load support is realized. For the hydrodynamic feature analyzed full cavitation was achieved until reaching a depth of 6.0 µm. At this depth, the features closest to the outer diameter (operating at a higher velocity), lost full feature cavitation. A progression of this phenomenon is shown in Figure 45 where the darker region represents fluid cavitation.
Figure 45: Pictograph of fluid cavitation as function of depth in macro/micro-features closest to the seal interface OD

Once the optimum depth (4.5 µm) was achieved, the optimum percentage of area density covered by this macro/micro-feature was analyzed while holding the depth constant. The percentage of area density analyzed was 25%, 33%, and 45% of the sealing interface. Results of this analysis are shown in Figures 46-48.

Figure 46: FEA results of temperature as a function of percent area density coverage

From the results in Figure 46, it can be seen that a 33% density is optimum at the 4.5 µm depth already established.
Results from Figure 47, concur with the previous result, showing optimum density coverage of 33%.

Again, Figure 48 shows that the optimal density results in slightly more leakage than those feature configurations exhibiting higher temperature and torque. This is a direct result of the film thickness between the seal faces.
Ch.8: Macro/micro-feature Testing

A test program was set up to validate actual seal performance of the tapered channel macro/micro-features in two different mediums. The first series of tests were conducted in water using a friction tester designed to determine the accuracy of the seal model predictions established in Ch.7. In this test, tapered channels were burned into each seal interface at the range of depths previously analysed. The second test was conducted in liquid ethane, demonstrating the diversity of applications in which the tapered channel geometry might be used. All tests used a feature area density of 33%. A laser scanned image of one such seal interface is shown in Figure 49.

![Laser scanned image of tapered channel geometry laser ablated into the sealing interface](image)

Figure 49: Laser scanned image of tapered channel geometry laser ablated into the sealing interface

Water Testing

The first series of tests were conducted in water using a specialized friction tester shown in Figure 50. This tester was designed to measure dynamic friction of a single mechanical seal. This equipment is the property of Flowserv Corp, used with permission.
Figure 50: Specialized friction tester designed to measure running seal face torque (Flowserve)

A cad model highlighting the mechanical seal components within the friction tester housing is shown in Figure 51.

Figure 51: Test assembly used to validate macro/micro-feature performance

Every test was conducted using a mechanical seal with a 2.3 cm balance diameter. Face materials consisted of a carbon graphite rotating face running against a silicon carbide stationary face in the configuration shown in Figure 51 at the operating conditions found in Table 2. The duration of each test was 24 hrs. Initial testing was conducted on 5 different un-textured faces to establish a baseline from which all other tests were compared. All tests used identical grades and geometries of carbon rotating
faces running against identical grades of silicon carbide. A picture of one such set of faces is shown in Figure 52.

![Figure 52: Carbon rotating face (left) and silicon carbide stationary face (right)](image)

After running the baseline tests, tapered channel depth optimization was then conducted. In this study, each test was conducted using a different depth macro/micro-feature while maintaining a feature density of 33% as shown in Figure 53.

![Figure 53: Silicon carbide face with macro/micro-features laser burned into the interface](image)

Each face was traced for accuracy. An example of this is the 3 µm depth tapered channel shown in Figure 54. This trace was taken using a profilometer. The trace on the left was taken tangentially across a single feature, and the trace on the right was taken radially across a single feature as shown.

![Figure 54: Tangential (left) and radial (right) trace of a single macro/micro hydrodynamic feature 3µm in depth](image)
During each test thermocouples were attached to the inside of each stationary face at the sealing interface and at the fluid inlet and outlet of the seal chamber. Pressure transducers were connected to the seal chamber. A unique torque transducer was used to measure seal friction. Motor speed was also controlled using a variable frequency drive (VFD). All data was acquired real-time using data-acquisition through Labview®. A screen shot of our Labview® control and data-acquisition GUI is shown in Figure 55.

![Figure 55: Labview GUI showing data acquisition acquired during each test](image)

The raw temperature and torque test data is shown in Figures 56 and 57.
From the data shown in Figure 56 and 57, it is evident that there is a seal performance trend that changes as a function of taper channel depth. This information is further exemplified when graphing the average value of each parameter (shown in Figure 58, and 59)
Figure 58: Test results showing temperature as a function of depth

The average temperature test results of Figure 58 are very similar to the FEA analysis results of Figure 42.

Figure 59: Test results showing torque as a function of depth

The torque test results in Figure 59 are similar to FEA analysis results of Figure 43.

A comparison between a baseline test and the optimized tapered channel is shown in figure 60. This data demonstrates the correlation between torque and the temperature difference ($\Delta T$). The temperature difference is used for a more accurate comparison between tests, as it takes the difference between the sealing interface
temperature and the fluid inlet temperature. As expected, the $\Delta T$ is directly related to the torque due to the frictional heat generated at the sealing interface.

![Figure 60: Graph showing one baseline test compared with the optimized tapered channel.](image)

Results comparing all baseline tests to the optimized tapered groove show a significant improvement in performance. The average un-textured $\Delta T$ was $20.7 \pm 3.9 \, ^\circ C$ compared to the tapered channel average $\Delta T$ of $6.4 \pm 0.7 \, ^\circ C$ (a 69% reduction in face temperature). Dynamic torque for the un-textured seal averaged $2.8 \pm 0.45 \, N-m$ compared to the tapered channel torque of $0.98 \pm 0.14 \, N-m$ (a torsional reduction of 65%). The tapered channel topography also provided increased seal stability, evident by the reduction in temperature and torque fluctuations when compared to the baseline tests (See Figures 56, 57 and 60).

No leakage was observed in either the un-textured or tapered channel topography tests. Low leakage for the textured seal may be attributed primarily to the large regions of cavitation that affectively occupies flow space $^{[21]}$. Post-test inspection of the un-textured seal showed 1.8 microns of concave wear, indicating thermal distortion due to ID contact. Inspection of the tapered channel faces showed no measurable wear of either the micro-featured face (Fig.61) or the mating carbon. This observation was verified by taking post-test profile measurements.
Figure 61: Post-test image of tapered channel seal face

Figure 62 shows a comparison of the radial taper measurement profile for the un-textured and tapered channel carbon mating faces. Both faces were lapped flat to within one helium light band (≈ 0.3µm) prior to testing.

![Friction post-test radial trace of un-textured carbon face (left) and tapered channel carbon face (right)](image)

Ethane Testing

A second test was conducted using a mechanical seal with an 11.4 cm balance diameter of the same materials running on pure ethane at steady state operating conditions of 3600 rpm and 15 °C with pressure ranging from 3.7-8.2 MPa. A cad image created in Inventor® of the silicon carbide seal face with tapered channel ablated topography is shown in Figure 63.
The tapered channel design was run for a total of 94 hours through the entire range of pressures, while the un-textured seal failed in less than one hour at a single pressure of 4.4 MPa due to excessive face temperature. Figure 64 shows this comparison as a function of differential temperature ($\Delta T$).

**Figure 64: Comparison showing tapered groove performance (blue) under changing pressure profile vs. non-textured face running at 4.4 MPa**
Figure 64 shows how well the tapered channel surface topography ran on liquid ethane. The $\Delta T$ of the micro-featured face ran smoothly as it followed the pressure trend, whereas the $\Delta T$ of the un-textured seal face increased rapidly, failing over a time period of just one hour despite running at constant pressure. Figure 65 shows the direct comparison between the un-textured seal face and the tapered channel running under identical operating conditions (shown in Fig. 64 between the 45th and 51st hour of operation).

![Figure 65: Ethane test comparison](image)

When comparing the performance between the two seals running in ethane at 4.4 MPa (Fig.65), the difference is dramatic; the un-textured face design failed after just one hour of operation when the $\Delta T$ reached 55 °C (an increase of 46.2°C), whereas the tapered channel seal ran for 6 hours straight before going to the next pressure regime with an average $\Delta T$ of $-0.5 \pm 0.5$ °C. The negative $\Delta T$ can be attributed to the cooling effect of ethane as it changes phase across the seal face. Seal leakage was measured using both a mass flow sensor and rotometer, in both cases leakage was less than 0.6 L/hr throughout the test (the minimum resolution of the measuring devices). Further,
post-test inspection of the un-textured seal faces showed aggressive wear due to hard contact, whereas the tapered channel seal faces showed no measurable wear despite having run 94 hours at pressures ranging from 3.7-8.2 MPa.

Prior to conducting the tests described in this paper, four initial macro/micro-feature models were tested. These models all failed within a few hours of operation by excessive heat. After carefully tracing the features it was discovered that the laser machined silicon carbide (which is a photochemical process), exhibited some photothermal characteristics at the boundary edge surrounding each feature. As a result, a raised edge had formed around the perimeter of each macro/micro-feature boundary. An example of this raised edge is shown in Figure 66.

![Figure 66: Micro-dimple showing photo-thermal edge affects as depicted by the raised portion dimensioned as U][19]

The raised edge measured on these macro/micro-features, though only 0.3 µm in height (Δh), is an order of magnitude higher than the asperities across the seal face. As a result, the inclusion of these raised edges caused highly local distortion and wear, leading to 3rd body abrasion and increased contact wear. Post processing of laser ablated geometries is critical to good seal performance. No mention of such processing or need was found in current literature.
Ch.9: Discussion of Results and Conclusions

In comparing the FEA results of Ch.7, with the test results of Ch.8 (as shown in Figure 67 below), it is evident that there is good correlation between the FEA model used to predict seal performance and the actual seal performance.

![Figure 67: Comparison of FEA and actual test results of temperature and torque vs. depth](image)

Although the average temperature predicted by the FEA model was within 8% of actual seal performance, and the average torque was within 20%, trend was predicted well.

Based on these results, the following conclusions are made:

- Properly designed geometric forms called “hydrodynamic features” may be used to significantly improve mechanical seal performance
- The use of a laser is an excellent method for producing such features due to its accuracy, repeatability, adaptability and ease of use.
- Post processing to remove edge burrs is essential for proper sealing performance
• FEA models based on the Reynolds equation may be used to help predict the performance of hydrodynamic features (such as the macro/micro-feature tested in this Thesis), allowing the user to optimize seal performance prior to verification.

• Experiments in water correlate well with the design optimization analysis and testing in ethane shows the wide range of operating conditions for which this new feature can provide enhanced performance.

• Low contact and hence low wear are attributed to a more distributed hydrodynamic load support function. Likewise, low leakage is believed to be the result of optimized cavitation function. No debris was discovered in the micro-features at the conclusion of any testing.
REFERENCES:


BIBLIOGRAPHY:


