Multiphase Oil Transport at Complex Micro Geometry

by

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Abstract

Dynamic sealing systems, such as mechanical face seals and piston rings in internal combustion engines are critical components of modern machines. These sealing systems present a unique challenge in controlling the lubricant supply and flow in the contact areas in order to minimize leakage, friction, and wear. Due to the sealing requirement, the minimum oil film thickness is in the order of surface roughness, which is one critical design parameter for the sealing systems. On the other hand, the wavelength of the surface geometrical features ranges from the size of the asperities of the surface roughness, which is in the order of a few microns to the size of the sealing components, which is in the order of millimeters to hundreds of millimeters. It is helpful for engineers to have a good understanding of lubricant transport across a large range of length scales. The aim of this thesis is to establish efficient and robust hydrodynamic lubrication models that are able to handle arbitrary complex geometries, flexible boundary conditions, and penetration of foreign gases.

In this thesis, first a general oil transport model was developed. The model considers the variation of oil volume occupation fraction and establishes the dynamic mass flow balance in all locations. Instead of using inefficient small relaxation coefficients to assure convergence, we adjust the local linearization scheme according to local full film or partial film status. The model also applies quick contour detection algorithm to avoid the problems caused by equation’s singularity around contact points and slow convergence caused by complex contact patterns. Furthermore, the models can be easily adapted to different scales. With the strong link between the numerical scheme and critical physical processes, the model eases the analysis of complicated results. This model has served as fundamental block of applications that predict and optimize the performance of metal face seals and piston ring pack liner system.

Based on the success of single specie two phase oil transport model, a new multi phase oil transport model have been developed with reasonable assumptions about oil/gas mixture coexistence pattern and oil contact pattern. This multi phase model expands our analytical capability to some important but formerly not reachable areas like the starving oil supply boundary condition, pressurized gas boundary condition and the influence of gas penetration to oil film between mechanical components. Some
preliminary investigations about the influence of gas penetration to the lubrication of the piston rings in internal combustion engines has been carried out. The results demonstrate that this model is able to preserve the oil mass conservation while capturing the gas penetration, gas pressure variation, and its interaction with the liquid oil. Furthermore, the results show that the liner finish effects becomes more and more prominent when the ring face profile becomes flatter and flatter with either fully-flooded or starved oil supply boundary condition.

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Contents

1 Introduction ................................................................. 19
  1.1 Background and Motivations ......................................... 19
     1.1.1 Metal Face Seals ............................................ 20
     1.1.2 Piston Rings in Internal Combustion Engines ............... 22
  1.2 Friction and Leakage .................................................. 23
  1.3 Previous Modeling Work ................................................ 24
     1.3.1 Modeling Work on Metal Face Seals ......................... 24
     1.3.2 Modeling Work on Piston Ring/Liner Lubrication .......... 25
  1.4 Objectives of the Thesis Work ...................................... 27
     1.4.1 Flexible Metal to Metal Face Seals ....................... 27
     1.4.2 Ring/Liner Lubrication .................................... 28

2 Major Factors of Modeling Consideration ............................. 29
  2.1 Geometry on Different Scales ....................................... 29
     2.1.1 Surface Profile ............................................ 30
     2.1.2 Surface Roughness ........................................ 31
     2.1.3 Surface Texture .......................................... 31
  2.2 Lubrication regimes .................................................. 33
     2.2.1 Hydrodynamic Lubrication .................................. 34
     2.2.2 Mixed Lubrication ......................................... 34
     2.2.3 Boundary Lubrication ..................................... 35
  2.3 Physical Phenomena in Lubrication Areas ......................... 35
     2.3.1 Oil Transport Mechanism ................................... 35
3 A Robust and Efficient Single Specie Two Phase Unsteady Lubrication Model with Adaptive Scaling

3.1 Introduction

3.1.1 Cavitation and Boundary Condition

3.1.2 Past Works on Universal Cavitation Algorithm

3.2 New Universal Cavitation Algorithm

3.2.1 Universal Variable Definition

3.2.2 Spatial Discretization Scheme

3.2.3 Assemble the Jacobian Matrix

3.2.4 Physical Constrains and Iteration Scheme

3.2.5 Boundary Conditions

4 Model Application One: Metal Face Seals

4.1 Deformation Model

4.1.1 The Geometry of Metal Face seal

4.1.2 2D Deformation Model of Ring Surface

4.1.3 Surface Measurement and Stitching Patches

4.2 Fluid Model

4.2.1 Reynolds Equation in Polar Coordinate

4.2.2 Assemble the Jacobian Matrix

4.2.3 Patch Measurement Based Flow Factor Improvement

4.3 Model Results

4.3.1 Model Configuration

4.4 Conclusions

5 Model Application Two: Twin Land Oil Control Ring

5.1 Twin Land Oil Control Ring Model
5.1.1 Statement of the Problem ...................................... 73
5.1.2 Contact Preprocessing ......................................... 74
5.1.3 Fully Flooded Leading Land and Starving Oil Supply at Trail-
ing Land .......................................................... 78

5.2 Results and Analysis ............................................... 78
5.2.1 Mass Conservation ............................................. 82
5.2.2 Oil Redistribution .............................................. 83
5.2.3 Pressure Noise Processing ..................................... 86
5.2.4 Oil Accumulation ............................................... 89

5.3 Conclusion ............................................................ 92

6 Multi-phase Unsteady Oil Transport Model for Piston Rings Under
Starved Conditions ....................................................... 95

6.1 Boundary Conditions for the Lubrication of the Piston Top Two Rings
in IC Engines .......................................................... 95
6.1.1 Oil Supply Conditions for the Top Two Rings ............... 96
6.1.2 Gas Pressure Conditions for the Top Two Rings ........... 96

6.2 Criteria and Assumptions .......................................... 98
6.2.1 Objectives of Model ........................................... 98
6.2.2 Oil Gas Coexistence Pattern .................................. 99

6.3 Governing Equations .............................................. 100
6.3.1 Spatial Discretization Scheme ................................. 102
6.3.2 Assemble the Jacobian Matrix ................................. 103

6.4 Iteration Scheme of Two Species Model .......................... 106

6.5 Applications to Oil Transport of Top Two Rings ............... 107
6.5.1 Ring Geometry .................................................. 107
6.5.2 Model Inputs .................................................. 111
6.5.3 Sample Results ................................................. 112

6.6 Conclusion ............................................................ 131
7 Conclusions

7.1 The Lubrication Models Developed in this Thesis Work .......................... 133
7.2 Future Work ......................................................................................... 135
List of Figures

1-1 Metal face seal structure. ............................................. 20
1-2 Power cylinder and ring pack system .............................. 23

2-1 Surface profile of metal face seal and piston ring. ............. 30
2-2 Surface profile and roughness. ...................................... 32
2-3 Surface tension and leakage. ....................................... 39
2-4 Oil gas coexistence patterns ....................................... 40

3-1 Surface slide on surface ............................................. 43
3-2 Control volume and value locations .............................. 48

4-1 Flexible metal to metal face seal ................................... 56
4-2 Seal profile and geometry parameters ............................ 58
4-3 Flexible metal to metal face seal ................................... 60
4-4 Pressure distribution after two cycles ............................ 66
4-5 Oil supply at ID after two cycles ................................. 67
4-6 Oil leakage at OD after two cycles ............................... 67
4-7 Oil exchange rate at radial direction after two cycles .......... 68
4-8 The non leakage pattern in flat band after two cycles .......... 69
4-9 The viscous friction variation from startup in three cycles .... 70

5-1 Ring pack face profile ............................................... 72
5-2 Oil control ring sliding above liner ............................... 75
5-3 Contact patterns for different $\lambda$ .............................. 76
5-4 Liner surface measurements and the position of ring in three different phase in calculation

5-5 Local clearance, hydrodynamic pressure generation and density distribution

5-6 Hydrodynamic pressure generation in three different phase

5-7 Relative residual at sliding phase

5-8 A cross section cavitation zone

5-9 Pressure driven flow over clearance contour

5-10 Pressure driven flow over pressure contour

5-11 Pressure driven flow and surface grooves

5-12 Statistical local momentum of pressure

5-13 The geometry of TLOCR sliding on liner

5-14 Oil volume fraction, pressure driven flux and clearance

5-15 Oil scraping rate in TLOCR pocket

6-1 Power cylinder and schematic of oil flow

6-2 Oil transport of top two rings

6-3 Oil contact patterns in power cylinder

6-4 Flow chart of two species model

6-5 Working environments of top two rings

6-6 The geometry of ring and liner

6-7 Pressure conditions for top two rings

6-8 Sketch of pressure averaged in circumferential direction

6-9 Oil development at crank angle $45^\circ$

6-10 Oil development at crank angle $-45^\circ$

6-11 Velocity and minimum clearance for rings with $a = 10, 35, 70$

6-12 Compare multi phase model and Reynolds equation model at crank angle $45^\circ$

6-13 Clearance and oil volume ratio between piston ring and rough liner at crank angle $45^\circ$
6-14 Pressure and oil volume fraction averaged at circumferential direction  125
6-15 Compare multi phase model and Reynolds equation model at crank angle 45 ................................. 127
6-16 Pressure distribution of tailing pressurized gas boundary .......... 128
6-17 Pressure distribution of OCR .............................. 129
6-18 Oil volume fraction in OCR ............................... 130
List of Tables

2.1 Most common roughness statistical parameters ............... 32
Nomenclature

\( \alpha \) Universal variable to be solved

\( \Delta y \) Grid space at y direction

\( \kappa \) Artificial compressibility

\( \mu \) Dynamic viscosity

\( \phi \) Oil volume fraction

\( \phi_g \) Volume fraction of gas

\( \phi_o \) Volume fraction of oil

\( \sigma \) Surface tension coefficient (N/m)

\( \theta_x \) Twist angle around x axis

\( \theta_y \) Twist angle around y axis

\( g \) Cavitation index variable

\( h \) Oil film thickness

\( H_0 \) Reference clearance

\( K \) Flow conductivity

\( p \) Oil pressure

\( R_a \) Arithmetic average of absolute roughness height
\( R_p \)  Maximum roughness peak height

\( R_q \)  Root mean square (RMS) roughness height

\( R_v \)  Minimum roughness valley height

\( R_{sk} \)  Skewness of absolute roughness height

\( U_z \)  Surface squeezing velocity

\( Z(x, y) \)  Seal surface height measured at lot resolution profilometer

\( Z_p(x, y) \)  Patch surface height measured at high resolution profilometer
Chapter 1

Introduction

1.1 Background and Motivations

Tribology plays an important role in practical industrial applications. A thin film of lubricant is deliberately introduced to separate two surfaces in relative motions. The hydrodynamic pressure buildup in surfaces converging zone supports part of total load, reduces the asperity contact pressure, hence reduce the friction and wear. Lubricant also serves as additive carrier that helps protecting surface, maintaining viscosity under large temperature variation etc.[16]. But the importance of lubricant does not mean the more oil the better. Because mechanical components are often subjected to intrinsically contradictory constrains, we need not only delivery oil to the location to protect surface but also control the oil flux to minimize leakage or oil consumption. Nevertheless, it is not so easy to predict oil transport when surface clearance reaches small scale ($0.1\mu m \sim 1\mu m$). There are always complex surface textures on mechanical surfaces. Some of these surface textures are caused by manufacture processes; some are intentionally designed to improve performance. After the load applied, the surface will deform. Additionally, the space between surfaces keeps varying when surfaces slide to each other. All of these make it a critical yet difficult problem to understand and to model the oil transport between the complex geometry formed by surface texture. Here, we list two common components that heavily utilize oil transport modeling to optimize parameters and improve designs.
1.1.1 Metal Face Seals

Mechanical face seals are widely used in industry. The specific seal this work is focused on is a metal face seal. It is used extensively in rotating housings, lubricated joints, axles of mining trucks, etc. As showed in Figure 1-1, it includes two identical metal rings and two rubber toric rings installed in a metal housing. The metal faces, which form a dynamic sealing interface, are highly polished for good lubrication. The rubber rings can roll between the ramps of metal rings and housing. Hence the rubber rings apply uniform and constant load even when seal moving inside housing due to impact or other reasons. That is also why it is called floating seal sometimes. The main function of this metal face seal is to seal the lubricant lubricating bearing system in wheel axels of heavy duty truck, rollers of track type tractor. Several seal systems of different sizes work together to do the job. While the inner seals maybe fully immersed in lubricant, the seal at outer flank may face a much tougher running condition. It seals the lubricant inside and prevents mud and gas mixture penetrating.

Figure 1-1: Metal face seal structure.
The metal face seal we are interested in was designed for small tractors. The seals are required to last thousands of hours without leakage. With the prerequisite of zero leakage, a smaller load on seals is desired to minimize the friction and wear. Though the relative simple design of the seal achieved great success, there is not too much of fundamental understanding of its mechanism. Especially on the area of inter asperity oil transport. Step by step, the design of this metal face seal was scaled up for larger machines. This change brings challenges to old seal design. The diameter of larger seals could be up to \( \sim 3m \). The larger seals behave not so consistently. Their performances have large variation even when the seals are manufactured in the same procedures. There are no reliable quality control parameters to identify the seals that would fail before we put them in filed. In a batch of seals that satisfy same requirements, some can last thousands of hours; some fail immediately. It is necessary to understand the fundamental working mechanism of the seals. Only based on such kind of understanding, we can find the correct performance control parameters.

There are two major kinds of failing mechanisms, leaking and scoring. When the seals are in relative motions, the surface geometry features on seals drive the oil in the narrow space between seals. If there is enough local oil pumping mechanism, even a scratch on seal surface that is as narrow as human hair could cause excessive leakage. The leakage increases oil consumption and finally cause machine failure due to lack of lubrication. Scoring failure usually happened in very short time. Seal before scoring runs like normal seals. At some spot, the friction and temperature rise rapidly over the limit. Following it, excessive leakage and debris in oil can be observed during this scoring process. At last the seal will stuck or run at very high friction. After dissembled the scored seals, we can find surface damages and burning spots at the sealing surface. Scoring usually happens at some local area. Then the generated debris ploughs and damages the surface. When new seals go through the break-in period, there are always minor scorings. Minor scoring helps seal surface match each other in better way and builds up the oxidized layer which is important to protect the surface in contacts.

In order to minimize leakage, the sealing surfaces of the pair of seals need to
be pressed against each other by normal load. On the other hand, the lubricant need to be transported into the sealing interface to reduce friction, heat and wear generated by the surfaces’ relative rotation. This intrinsic contradiction between leakage and lubrication presents general challenge in designing durable mechanical face seals. Although there have been tremendous improvements over the past several decades, the durability and performances of seals remain unpredictable. Though the cost of seal itself is not very much, the maintenance and delay of project cost a lot. The mining trucks which seals serve in are designed to work 24 hours a day.

The seal design appears simple, but the mechanisms of its operation and failure modes are largely unknown. As a result, seal design and tests have to go through tremendous amount of trial and error procedures for new applications. It is believed that gaining fundamental understanding of the operation and failure modes through theoretical modeling is essential to efficiently developing this type metal to metal face seals for future applications.

1.1.2 Piston Rings in Internal Combustion Engines

Power cylinder, which is composed with piston, piston rings, and liner, is the kernel part of internal combustion engine. The piston ring pack can be viewed as a dynamic sealing system. During the engine cycle, the good sealing of combustion gas is the precondition and guarantee to extract energy. The piston ring serves as the seal in the power cylinder. Like other dynamic sealing systems, it faces the same intrinsic challenge of controlling the lubricant supply and flow in the contact areas to minimize leakage, friction, and wear. The existence of highly pressurized gases further complicates the problem. The interaction between the pressurized gases and the residual oil among surface asperities and honing grooves is complex yet lack of thorough understanding. The pressurized gases could either push the residual oil out from asperities to lubricate the surfaces or push the residual oil totally out of the narrow space between piston ring and liner, resulting the loss of lubrication. Furthermore, the running conditions of high temperature and high speed in piston ring/liner system as well as the existence of the small scales of surface features make it extremely hard to
experimentally observe the interaction between the pressurized gases and the residual oil through experiments. Models based on proper assumptions become powerful tools for the research in this field.

1.2 Friction and Leakage

Friction is an important factor to influence performance of mechanical components. The friction compromises the energy for machines to do useful work, and hence lowers the efficiency and performance. Friction also generates heat, which can degrade the lubricant and deform the component, which in turn affects the lubrication and friction behavior of the components. Leakage could happen when components are either in relative motion or not. But the mechanisms of leakage could be quite different in these two situations. For components that are in static seals, there are two factors to cause leakage. One is pressure gradient, the other is an open channel to let lubricant pass through. For components that are in dynamic seals, the leakage mechanism could be much more complex. The clearance between surfaces and pressure gradient keeps changing and so are the direction and amount of lubricant flux are in constant variation either. As a result, it is not necessary to have an open channel at any
moment to have a leakage, as the lubricant can travel from inlet to outlet step by step via dynamically changing clearances and pressure gradient. Only after we understand the oil transport between surfaces in relative motions that we can predict leakage correctly.

All components mentioned above sections have a common character that they are designed to seal the lubricant in dynamical environments. Friction and leakage are two major objects we want to minimize to improve components' performances. Unfortunately, the methods we can apply to minimize one are often in contradiction to the methods that minimize the other one. For example, to minimize the leakage, applying large load on component to achieve a total sealing or smaller clearance between surfaces is good method. But this method will directly increase friction. Therefore, the key object of component design is to maintain the balance between leakage and friction in a dynamic environments. Without the model that provides correct quantitative results, it is impossible to reach the balance.

1.3 Previous Modeling Work

1.3.1 Modeling Work on Metal Face Seals

Similar to most of the lubrication problems, geometry of the contact surfaces plays an important role. From the current design specifications and manufacturing processes, the geometry of the sealing surface exhibits characteristics at both macro and micro levels. At the macro scale, the surface height varies within the so-called flat band and the transition region. Along the circumferential direction, the surface height variation is dominated by 2nd order sinusoidal wave and its magnitude is in the order of a few microns for a seal with a diameter in the order of 100mm. Additionally, the profile in the radial direction varies along the circumference. Hereafter, the height variation of the sealing surface is defined as waviness. The waviness of the sealing surface is controlled to be within limit from the current processes. However, rarely two seal rings are found to have the same characteristics at this level of details and it has been found
from bench tests that waviness difference could contribute to the performance variations of the seals in leakage and scoring. G. Costa proved the existence of waviness in our seal and investigated the effects of waviness on hydrodynamic lubrication.[3] Pascovic and Etsion developed an analytical study for aligned and misaligned symmetrical double seals.[17][18] Knoll developed a three-dimensional numerical simulation using finite element method, which accounted for waviness effects.[14] Person et al. used finite difference method considering waviness and misalignment effects.[22] Tournerie et al. extended their model to include the heat transfer through the stator and the rotor.[32] These previous studies have dealt with full film lubrication without considering cavitation and unsteady effects. Furthermore, at the micro level, there exists roughness that is determined by the polishing process, which is the final manufacturing step. The height of the roughness is in the order of 0.1 microns and the correlation lengths are in the order of 1 micron in the radial direction and 10 microns in the circumferential direction. These micro features were also found to affect the performance of the seals.[7] Hong's work considered the thermal effects on the lubrication in sealing band, and gave scoring failure criteria.[11] But his work was based on the seal lubrication of steady state. There is still a lot unexplored area about impact of the lubricant transport on seal performance. Wang improved the thermal model based on fully unsteady deterministic lubrication model.[35] [34] The model was applied to analyze the performance of flexible metal to metal face seal. The results showed that the contact wetness, oil exchange rate and surface temperature distribution within sealing band are critical to seal scoring and leakage.[36]

In summary, there are many uncertainties to predict the performance of the flexible metal-to-metal face seal (FMMFS). This thesis work is aimed at understanding the performance of FMMFS, through advanced oil transportation modeling.

1.3.2 Modeling Work on Piston Ring/Liner Lubrication

Cross-hatch liner finish has been used in automotive engines for quite a long time. It is known that proper liner finish is essential for survival and durability of the engines. Further performance parameters such as friction and oil consumption are
also sensitive to liner finish.[27][10][9] Despite its critical role in determining the power cylinder system performance, little is known about the mechanisms that control the interaction between a liner and its mating parts, particularly piston rings. This lack of understanding about the interaction of piston rings and rough liner prevents one from having an efficient way to find design solutions to optimize power cylinder system in friction, wear, and oil consumption.

In a modern engine, plateau honing is commonly used. On the plateau honed liner, there are two distinctive features, namely, deep valleys (here mentioned as deep grooves) created during rough honing and the plateau part that has much less asperity size. It is often discussed in the community that one of the functions of the valley parts is to act like an oil reservoir for the plateau part that has direct interaction with the liner. In Hill’s work,[10] it was found that valley part has negative effects on oil consumption. Existence of valleys is also considered to be critical to avoid scratches and scuffing on the liner.[27] However, the physical mechanisms of the effects of valleys have never been discussed to the authors’ knowledge. Furthermore, effects of liner roughness height distribution and cross-hatch angles were also the subject of a great deal of theoretical and experimental works.[13][31][8]

Tian did extensive researches about the dynamics of the rings, ring/groove interaction, gas flows and design parameters about ring pack.[29] Thirouard implemented a two-dimensional Laser-Induced Fluorescence(LIF) visualization system to observe the oil transport in piston ring pack. Models were developed based on experimental observation to analyze oil flow in and between the piston region such as oil transport in lands, grooves and ring gaps.[28] With more and more understanding of oil transport mechanism in large features of ring pack/line system, the urgency of understanding oil transport mechanism in roughness level micro features getting higher and higher to understand the source of the oil accumulation on the piston as well as the lubrication and friction between the piston rings and the liner.
1.4 Objectives of the Thesis Work

From the examples presented in the previous sections, we can notice the performance of system is always determined by entangled factors. The design optimization is constrained by different running conditions and requirements. Besides other significant factors, the oil transport mechanism in a narrow space between surfaces that has multiple scales and complex features plays a central role. A robust and versatile model can help one gain fundamental understanding of the oil transport mechanism. And its adaptation to different applications can enhance the design and optimization capability of engineers.

The ultimate object of this thesis work is to develop a fully unsteady deterministic model that can fundamentally handle oil transport in complex surface geometries. The model should maintain the mass conservation of oil and give correct oil volume faction and pressure distribution in all sorts of running conditions. The model should also correctly handle the cavitation phenomenon, starving oil supply boundary conditions and pressurized gas boundary conditions. Possessing such abilities allows the model to help engineers understand the lubricant transport under the influence of vapor and gas in large geometry scale range, complex roughness characters, designed surface patterns, wide speed range, and different oil properties. The results of the model can be used to calculate the important performance indexes of mechanical system such as total friction, oil exchange rate and leakage. While the main objective of this work is to develop general fundamental lubrication models, the models will be applied to two concrete applications, namely, the modeling of flexible metal to metal face seals and ring/liner lubrication. The development of this model followed a path from easy to difficult, conciseness to complete. Based on the specific requirements of these two listed applications, the staged path to develop the model is listed below.

1.4.1 Flexible Metal to Metal Face Seals

The model development path for FMMFS project:

1. Develop a steady state hydrodynamic lubrication model to gain basic under-
standing on the effects of waviness on oil transport and pressure generation.

2. Develop a hydrodynamic sub model to study the effects of surface roughness on oil flows.

3. Develop a contact model to explain seal mechanical distortion, change of asperity statistic parameter due to asperity contact, and the influence of these surface changes to the oil flows.

4. Develop an unsteady hydrodynamic lubrication model to simulate pumping and squeezing of the oil in the sealing area formed by two wavy rough surfaces rotating to each other.

5. Develop simple one way out flow boundary condition to handle the seal in contact with air at outside boundary.

6. Integrate sub models to form a single component multi phase deterministic lubrication model that can handle surface roughness and deformations.

### 1.4.2 Ring/Liner Lubrication

The model development path of piston ring/liner lubrication project.

1. Develop a deterministic mass conserve twin land oil control ring model to explain the oil accumulation in the pocket of twin land oil control rings.

2. Develop a fast solver for simplified compressible Reynolds equation to solve gas pressure in given open channel.

3. Develop a model that correctly solve the movement of pressurized gas-oil interface.

4. Develop the protection algorithm to prevent model crash at large gas flow rate.

5. Integrate sub models to form the two species multi phase deterministic lubrication model that can handle interaction of pressurized gases and oil.
Chapter 2

Major Factors of Modeling Consideration

Before we get into the details of modeling, a very important work is to analyze the major factors influencing the performance of mechanical components we are going to model. The focus of the thesis work is to develop a versatile and robust model to provide fundamental understanding of multi phase oil transport mechanism in micro structure. The model finds its applications in metal face seal system and ring/liner system. But its role is not limited in these two areas. In this chapter we list the major factors of oil transport in micro structure, categorize them and point out their impacts to modeling. The relations between the factors are sorted out and organized. For those factors with significant impacts on but will not cause difficulty to modeling, there will be open interfaces for adoption of those factors. But they will stay out of the focus in the discussion of thesis.

2.1 Geometry on Different Scales

Since the oil flows is transported between the surfaces in relative motion, the first step for modeling is defining the shape of surface. Due to either design objectives or manufacturing tolerance, the real surface deviates from mathematical ideal flat. The scales of the deviations are range from the size of component down to interatomic
2.1.1 Surface Profile

The first kind of length scale is in the order of mm in x-y directions and microns in the z direction. It is named surface profile in this thesis. Surface profile is the deviation extends at the largest length scale which is comparable to the scale of whole surface. Figure 2-1 shows the surface profile in metal face seal and piston ring. The deviation itself could be at microscopic range such as the waviness of the metal face seal, or at significantly larger scale like the taper band profile or coning angle of flat band. The surface profile could be created through unintentionally manufacture process like seal waviness created by lapping process, or created intentionally to fulfill certain functions such as the ring radial profile designed to enhance hydrodynamic load support ability, oil scraping ability.

The mathematical description of surface profile is a smooth non flat surface. From
measurement side of view, it is an averaged virtual plane of measured data. (Fig. 2-2)

2.1.2 Surface Roughness

The second kind of deviation is named surface roughness in this thesis. Surface roughness is the surface height deviation from the smooth virtual reference plane. Usually this virtual reference plane is surface profile. The larger the deviation is, the rougher the surface is. Unlike the surface profile, the roughness is typically short wavelength height variation. If we look into a small patch on surface, the measured surface height deviation from local virtual profile is considered to be a random variable. The features of roughness are described by its statistical parameters. Table 2.1 lists the definitions of most common roughness statistical parameters.

For surface only contains profile and roughness, due to the large difference of scales, it is a common approach to consider the their influence to lubrication separately. Since the geometry of roughness is complicated, the effects of roughness is also complicated. But not all details of roughness are necessary to modeling the whole system. Then building up a connection between the effects of roughness and characters of roughness is a natural and feasible solution. This connection always comes from a deterministic model that considers all details of roughness in small patch. After carefully analysis the results, most critical statistical characters are picked out to construction the relation between effects of roughness and characters of roughness. This kind of relation generally depends on the applications.

2.1.3 Surface Texture

The third kind of deviation is intermediate scale between the two described earlier. It is named surface texture in this work. Surface texture is the deviation formed by group of units. The units of surface texture could be varied in size and distributed either uniformly or irregularly. Each unit shares certain common geometry feature. The scale of single unit is between the surface profile and roughness, while the whole
Table 2.1: Most common roughness statistical parameters

<table>
<thead>
<tr>
<th>Roughness parameter</th>
<th>Description</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>$R_a$</td>
<td>Arithmetic average of absolute height</td>
<td>$R_a = \frac{1}{A} \int</td>
</tr>
<tr>
<td>$R_q$</td>
<td>Root mean square(RMS) height</td>
<td>$R_q = \sqrt{\frac{1}{A} \int z^2 ds}$</td>
</tr>
<tr>
<td>$R_{sk}$</td>
<td>Skewness</td>
<td>$R_{sk} = \frac{1}{AR_q^3} \int z^3 ds$</td>
</tr>
<tr>
<td>$R_p$</td>
<td>Maximum peak height</td>
<td>$R_p = \max(z)$</td>
</tr>
<tr>
<td>$R_v$</td>
<td>minimum valley height</td>
<td>$R_v = \min(z)$</td>
</tr>
</tbody>
</table>

Figure 2-2: Surface profile and roughness.
group’s scale extends at the scale comparable to the whole surface.

Before mechanical components are integrated into systems, certain surface processing procedures are critical to performance of components. From the early age of inner combustion engine, people noticed creating certain honing grooves on the liner surface can improve the performance and durability of IC engine. The created groove shape texture on surface can serve as debris trap and prevent the ploughing effect from increasing the friction and damaging the surface. Further more, the residual oil inside the grooves can be dragged out to supply the plateau area when oil supply is insufficient. With realization of the connection between surface texture and performance, new surface texture design were created. Except traditional mechanical way to generate surface texture, new technics like laser texture processing were introduced to surface process.

When the surface texture is formed by uniformly distributed units of same geometry, we can apply the similar approach to build up relations between its geometry parameters and effects to lubrication. But when the surface texture is formed by irregularly distributed units of different size, the existence of surface texture will greatly increase the difficulty of modeling oil transport in micro geometry. Unlike the surface profile whose effects to lubrication can be separated from effects of surface roughness, the effects of surface texture usually interferes with the effects of surface roughness because the size difference between them is not large enough to separate their effect. On the other hand, the larger size and wide coverage of surface texture make it very hard to get statistical significant relations from the deterministic model’s results on small patch. The model of oil transport in micro geometry must consider the characters of different surface features.

### 2.2 Lubrication regimes

The lubrication could be divided to different types according to the size of film thickness $h$. A mechanical components in working situation, its surface could work at different lubrication regimes. Each lubrication regimes has its own characters and
difference influence to oil flow, friction and wear. Through well design, we can put
different part of component surface, and guide oil movement. When the oil moves
in a proper way, we can achieve a continuous dynamic contact pattern to prevent
leakage while protecting the surface from excessive wear. There are three major flow
regimes, that we will describe in following sections.

2.2.1 Hydrodynamic Lubrication

When the film thickness is much larger than the surface roughness $h \gg R_a$, the
surfaces are totally separated by lubricant film. This lubrication regime is Hydro-
dynamic lubrication regime. In hydrodynamic lubrication regime, we can ignore
the influence of surface roughness, only consider the influence of surface profile. In
this regime, load is mainly supported by hydrodynamic force. Friction is caused by
the viscosity force of lubricant. Surface is well protected while the large clearance
between surface leaves plenty of space for lubricant to move around and help forming
leakage.

2.2.2 Mixed Lubrication

When the oil film thickness is comparable to the surface roughness $h \sim R_a$, the
surfaces get in contact at a few high peaks of roughness. This lubrication regime is Mixed
lubrication regime. In mixed lubrication regime, the load is supported by
surface contacts and hydrodynamic force generated by lubricant between roughness.
In this regime, the effect of roughness is very important. If the surface texture exists,
it is hard to separate the effects of roughness and texture. The friction in this region could be modeled a combination
of dry friction caused by surface contact and viscous friction generated by oil film. In
mixed lubrication regime, the pressure and oil distribution are very sensitive to the
variation of oil thickness.
2.2.3 Boundary Lubrication

When the oil film thickness is smaller than the surface roughness $h < R_a$, the surface contacts constantly occur at peaks of surface roughness. This lubrication regime is **boundary lubrication regime**. In boundary lubrication regime, most of load is supported by surface contacts, most of friction is generated by the friction of surface contacts. There is a oxide layer covers the surface in contact, decrease the friction and protect surface from damaged by several local wear and heat. Though the surface contacts constantly occur at boundary lubrication regime, for a single surface roughness peak, the surface is not always in contact. Due to the relative motion between surfaces, the contact pattern and oil distribution are in constant changing. When surface is not in contact, the inter asperities lubricant flow carries additives and cool down the surfaces heated in last contact.

2.3 Physical Phenomena in Lubrication Areas

2.3.1 Oil Transport Mechanism

In the lubrication theory, the film thickness is significantly small relative to length scale at other direction. The Reynolds number of tribological flow we are interested is around 0.1 to 10. The flow is laminal with this Reynolds number range. The oil transport mechanism follows the Reynolds equation 2.1 which is a simplification of Navier-Stokes Equations based on the criteria $Re h/L \ll 1$.

$$\frac{\partial}{\partial x} \left( \frac{h^3}{12\mu} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left( \frac{h^3}{12\mu} \frac{\partial p}{\partial y} \right) = \frac{U_1 - U_2}{2} \frac{\partial h}{\partial x} + \frac{\partial h}{\partial t} \quad (2.1)$$

The two terms at right side of Reynolds equation are active terms caused by the relative movement between surfaces. The first one $\frac{U_1 - U_2}{2} \frac{\partial h}{\partial x}$ is the Couette flow caused by sliding at the tangential direction of surface. The second term $\frac{\partial h}{\partial t}$ is the squeezing term caused by surface movements at normal direction of surface. The
terms at left side of Reynolds equation are Poiseuille flow caused by the pressure difference in lubricant. Put aside other phenomena, the Reynolds equations could be understood as following way: The relative movements of surfaces drive oil to move between surfaces, a specific pressure distribution will be generated to satisfy the mass conservation requirement.

We can categorize the oil flows from another perspective. The oil flow has two major directions. One is the surface sliding direction, the other is perpendicular to surface sliding direction but follow the tangent of surface. The Couette flow can only generate oil flow in the surface sliding direction, while the Poiseuille flow can drive oil in both directions. The flow at sliding direction plays an active role. It is the original driven force of oil transport. But that does not mean the flow at perpendicular direction is not important. Without flow at perpendicular direction, the oil will stay at the same sliding path until the surface sliding direction deviated from original direction.

In special situation, the flow at perpendicular direction decides the mechanical components success or fail. Each mechanical components has its own oil supply mechanism. The oil could comes from either along the sliding direction or perpendicular to it. For piston ring pack, the oil supply comes from the sliding direction while the oil supply of metal face seal comes from the perpendicular direction. In those cases, perpendicular flow directly relates to oil supply and leakage. Hence it is extremely important for those mechanical components like metal face seal.

The Poiseuille flow contains the flow components both at sliding direction and perpendicular direction. The ratio between those two components could be very large in specific applications. Like in the metal phase seal application, the width of sealing band at radial direction is only around three millimeters while the length sealing interface at circumferential direction could as long as hundreds of millimeters and several meters. These kinds of extreme length ratio decides the pressure driven flow at sliding direction is much smaller and insignificant compared to the pressure driven flow at perpendicular direction. As a model designed to explore general oil transport problem, simplification and optimization based on such kind of case related criteria.
will not be considered. All terms are kept and discretized.

2.3.2 Cavitation - A Partial Film Phenomenon

The relative motion of surfaces changes the clearance between surfaces. These changes tend to squeeze the oil out at the location where clearance is decreasing, and to fill the oil when the clearance is increasing. To maintain the mass conservation, a pressure gradient is generated to produce proper oil flow. There is no upper bound of oil pressure if we consider the oil as incompressible fluid. Hence at the locations where surface squeezes oil out we can always get enough pressure rise. But the oil pressure can not decrease without limit as the oil will vaporize at a lower enough pressure. This oil physical property applies a lower bound on oil pressure. At those locations where oil pressure reached the lower bound, there is not sufficient pressure gradient to suck oil in. Hence with the increase of clearance between surfaces, a void filled by oil vapor and dissolved gas will appear. This phenomenon is called cavitation.

In the cavitation region, oil only occupies part of the space between surface. Therefore, in this thesis, we also call cavitation region as partial film region. The rest of space is occupied by the oil vapor or dissolved gas. The pressure at cavitation pressure is decided by contents of mixture. When dissolved gas dominates, the pressure is close to ambient atmosphere pressure. When the flashed oil vapor dominates, the pressure is the vaporization pressure. For most of applications, the pressure in cavitation region is a constant pressure between zero and atmosphere pressure. Compare to the hydrodynamic pressure generated by surface movements, this pressure difference does not cause significant influence on components’ performance. In the cavitation region, the only oil transport mechanism is Couette flow.

Cavitation changes the whole picture of oil transport in complex geometry. The Reynolds equation that defines oil transport in full film region is a linear partial differential equation about single unknown variable oil pressure. The flow in cavitation region follows a hyperbolic oil transport equation that describes the balance between Couette flow and squeezing term. The difficulty lies on how to decide the full film region and cavitation region. For specific location, it is full film or partial film de-
pends on the balance between flow in different mechanism. The Couette flow and the squeezing flow are only related to local status. But the Poiseuille flow is related to the pressure distribution. Consider the compressibility of normal lubricants, the pressure influence spreads in a speed fast enough to treat pressure as a instant and global influence. Hence the local oil occupation status is coupled with the situations at far field and boundaries. The uncertainty of location and size of cavitation region make the oil transport problem a nonlinear problem.

2.3.3 Surface Tension and Oil Contact Pattern

The surface tension plays multiple roles in oil transport problem. The first role of surface tension is prevent leakage. Take metal face seal as an example (Figure 2-3). When the oil reach the outside edge of metal face seal, it need a pressure difference to overcome the surface tension force. The necessary magnitude of pressure difference is influenced by lots of factors such as local surface roughness and seal movements. In this work, we do not dig into the details about the exact magnitude. A estimation of the range of the pressure difference is a more practical object for the purpose of modeling leakage. In the best situation, the offset of metal face rings’ centers could be small enough $d \ll R_2$. With the consideration of $R_1 \gg R_2$ we have the pressure difference needed to break surface tension force is:

$$\Delta P = \sigma \left( \frac{1}{R_1} + \frac{1}{R_2} \right) \approx \frac{2\sigma}{h}$$

This give us a very important method to judge whether the leakage will happen from the result of oil transport model. Given the pressure boundary condition $p = p + 2\sigma/h$ at outside edge of metal face seal, we can get the local oil flux toward outside. Positive flux mean oil is being squeezed out, negative flux means oil is being sucked in. Integrate the oil flux over time, we can get the local amount of oil attached on edge of metal face seal.

$$Q(\theta, t) = \int_0^t q(\theta, t) \, dt$$

(2.3)
For seal at steady running condition, if total accumulated flux in one rotation is positive, the oil attached on edge of seal will increase progressively and cause leakage at last. The negative accumulated flux means the gas or oil from boundary will penetrate in, but it does not mean the seal will not leak. The maximum value of accumulated flux in one rotation decides whether seal has burst local leakage at rotation. If the maximum accumulated flux in one rotation is larger than what surface tension can hold $\max(Q(\theta, t)) > \pi h^2/2$, the seal will reach local burst leakage. In each rotation, small amount of oil will leak from the spot on edge. Though as a total effect, the seal is sucking gas in at the same spot.

### 2.3.4 Gas Lubricant Coexistence Patterns

In lots of working environment, the mechanical components can not get fully oil supply from boundary. In addition to lubricant, the gas around the boundary could flow into the clearance between surface. Once gas get into the space between surfaces, it could interact with lubricant and change the oil transport and performance of components.
The coexist pattern of oil and gas can leads to various influence. The oil could have fully attachment with both surfaces. In this case, gas could be fully surround by lubricant and entrapped in the space between surfaces. The entrapped gas can build up certain amount of pressure to prevent oil supply get into the region it occupied and push out excess oil when surrounding pressure decrease. The oil and gas could have separate between the surface, each get in contact with one surface 2-4. In this case, gas has an connected channel to flow freely. The dynamic viscosity of gas at room temperature is only $1.983 \times 10^{-5} Kg/m \cdot s$. This is around three orders smaller than the dynamic viscosity of oil. A small pressure gradient could generate relatively much larger gas flow. Hence if the oil and gas attach to one surface separately, the pressure in region occupied by mixture is not sensitive to the clearance variation caused by surface movement until the gas volume fraction has been compressed to sufficient small number. When the clearance is large enough, the gas could form bubbles fully separate from surfaces and enclosed in oil. The gas bubbles’s influences to mixture’s properties are great. Without the gas bubble, oil’s compressibility is very small. That is why a thin film of lubricant can support huge load. With the gas bubbles inside, the mixture’s compressibility is related to the gas volume fraction and can not be treated as incompressible fluid. The gas bubbles also change other properties of mixture such as viscosity and heat transfer coefficient. Furthermore, this kind of gas bubble oil mixture has unstable and nonuniform properties because its unsteady nature. That make it very hard to model and measure.
Chapter 3

A Robust and Efficient Single Specie Two Phase Unsteady Lubrication Model with Adaptive Scaling

In this chapter, we introduce our new numerical model for the unsteady hydrodynamic lubrication with consideration of partial film existence. The new model preserves liquid oil mass conservation at all time steps. The center piece of the this model is the development of an iteration scheme to effectively handle the different properties of the governing equations in different oil flow regimes and the corresponding problems caused by switching between them. By doing so, we avoided the oscillations at the interface of full and partial films without using very small relaxation factors. Consequently, this new model converges fast and robustly. Furthermore, a fast contour detection algorithm was utilized to minimize the negative influence of the contact patterns.
3.1 Introduction

3.1.1 Cavitation and Boundary Condition

Reynolds equation for liquid film lubrication is linear and straightforward to solve. What complicates the problem is the existence of partial film or cavitation regions in the lubrication problems involving complex patterns in local clearance. Because lubricant can not sustain pressure below its saturation or vapor pressure, when local clearance between surfaces increase so suddenly that the local pressure reaches this lower bound. Then, if the local clearance cannot be completely filled by the liquid oil, vapor and gases are released. If the gas is fully from the dissolved gas in oil, the amount of gas is small enough to fully dissolve into oil again when pressure is higher than saturation pressure. Therefore, unless the pressure difference between saturation and vapor pressure is important, we can treat the vapor and gas in the same way. They can come in and out of oil in negligible amount of time.

Cavitation phenomenon divides the modeling domain to full film region and partial film region (cavitation region). In the full film region, Reynolds equation governs the oil transport. Without losing generality, we consider two surfaces with surface profiles $h_1(x, y, t), h_2(x, y, t)$ (Figure 3-1). The surfaces slide in $x$ direction at speeds $U_1, U_2$, respectively.

If we fix the coordinate at the top surface and align the $x$ direction with the relative sliding direction of surfaces, the Reynolds equation in Cartesian coordinate is in the form of 3.1

$$
\frac{\partial (\rho h)}{\partial t} = \frac{\partial}{\partial x} \left( \rho \frac{h^3}{12\mu} \frac{\partial P}{\partial x} \right) + \frac{\partial}{\partial y} \left( \rho \frac{h^3}{12\mu} \frac{\partial P}{\partial y} \right) - \frac{U}{2} \frac{\partial (\rho h)}{\partial x} \tag{3.1}
$$

$$
U = U_2 - U_1 \tag{3.2}
$$

$$
h(x, y, t) = h_1(x, y, t) - h_2(x, y, t) \tag{3.3}
$$

Neglecting the compressibility of the liquid oil, we consider that in the full film
zone oil density is a constant $\rho_0$ in equation 3.1. In the cavitation zone, the gap is filled by a mixture of vapor/gas and liquid lubricant\[4\][25]. Hence, the pressure is assumed to be a constant $P_c$. Because of the zero pressure gradient, the pressure driven flow term disappears in the cavitation zone. The governing equation is a pure hyperbolic oil transport equation.

$$\frac{d(\rho h)}{dt} = -\frac{V}{2} \frac{\partial(\rho h)}{\partial x}$$  \hspace{1cm} (3.4)

At the rupture locus of cavitation zone, the flow continuity requires a zero pressure gradient and the pressure equals cavitation pressure. This boundary condition is called Swift-Stieber boundary condition or Reynolds boundary condition\[26\].

$$\frac{\partial P}{\partial n} = 0$$  \hspace{1cm} (3.5)

Here, $n$ is the normal direction of cavitation boundary. Positive direction is point out of cavitation zone.

At the reformation locus of cavitation zone, the Couette flow across the locus
has discontinuity. The boundary condition is provided by Jakobsson and Floberg at first \cite{12}. Elrod and Adams extend it to more general problems \cite{6}.

\[
\frac{h^3}{12\mu} \frac{\partial P}{\partial n} = \frac{U h}{2} \left( 1 - \frac{\rho}{\rho_0} \right)
\]

(3.6)

### 3.1.2 Past Works on Universal Cavitation Algorithm

The full set of cavitation zone boundary condition are inner boundary condition. To apply it, a cavitation boundary must be located based on current pressure distribution. Then the problem is reformulated to get the new pressure distribution. To simplify the iteration process, Elrod developed a universal scheme to solve the whole field \cite{5}. In his scheme, Elrod introduces a new variable that is valid in both full and partial film regions such that one obtains universal variable $\alpha$ to be solved in whole domain. In full film zone, $\alpha$ represents the oil density ratio. In cavitation zone, $\alpha$ represents the oil volume fraction $\phi$.

\[
\phi \equiv \frac{\rho}{\rho_c} = \alpha
\]

(3.7)

\[
P = P_c + \kappa(\alpha - 1)
\]

(3.8)

Here, the $\rho_c$ is the oil density at cavitation pressure. The $\kappa$ is an artificial compressibility used to relate pressure with density.

By introducing an index variable $g$ to distinguish cavitation zone and full film zone, Elrod presented a universal numerical scheme to solve whole field.

\[
g = \begin{cases} 
1 : & \text{full film zone} \\
0 : & \text{cavitation zone} 
\end{cases}
\]

(3.9)

Using the cavitation index $g$ to turn off Poiseuille flow in cavitation zone and replacing density and pressure by functions of $\alpha$, one can transform the Reynolds equation to the following form.
This method avoids tracking the cavitation boundary and the result will automatically satisfy mass conservation. However other researchers reported this method had numerical instability around the cavitation boundary[5][33]. Additionally, the artificial compressibility \( \kappa \) needed to be tuned for different cases to avoid the discrete system from becoming too stiff.

Payvar and Salant presented an alternative method to avoid the numerical instability by controlling the index variable iteration scheme[21]. Instead of switching the index variable \( g \) between zero and one, they introduced a small relaxation factor(0.01) to update the index variable for better control of numerical stability. Furthermore, the universal variable is defined in a new way,

\[
\frac{\rho}{\rho_c} = 1 + (1 - g)\alpha \\
\frac{P - P_c}{P_{ref}} = g\alpha \tag{3.11}
\]

Here, the \( P_{ref} \) is a preset reference pressure. With this definition, the universal variable \( \alpha \) represents the dimensionless pressure in full film zone. In cavitation zone, \( \alpha \) does not have direct physical meaning, but the \( (1 + \alpha) \) represents the oil volume fraction. This deliberately designed universal variable has smooth transition when the cavitation index switch between zero and one. However, because the relaxation factor has to be set to a small value to ensure the numerical stability, this method needs much more iterations to converge in cases with cavitation phenomenon than cases without cavitation phenomenon.
3.2 New Universal Cavitation Algorithm

3.2.1 Universal Variable Definition

Recognizing the shortcomings of the existing works, we developed a new universal numerical scheme to solve the Reynolds equation with existence of partial film in a more robust and efficient manner. The starting point of the new numerical model originates from the physical meaning of the Reynolds equation, namely, mass conservation. The Reynolds equation simply says the change of oil mass in control volume equals the difference between the oil in and out. Following this idea, the governing equation in cavitation zone is no different from a Reynolds equation with zero Poiseuille flow. While all universal cavitation algorithms utilize the cavitation index variable to turn the Poiseuille flow on and off, they differ in two aspects. The first one is how to map pressure and oil volume ratio into single universal variable to be solved. The second one is how to update the universal variable and reformulate the discrete system of algebraic equations when the cavitation index is switched.

In the present work, the advantages of existing models are integrated together. Improvements were made on both universal variable and iteration scheme to gain better robustness and efficiency.

First, the central problem of oil transport modeling is the oil mass flow balance. This nonlinear problem should be linearized to a discrete algebra system about the universal variable to be solved. The process of solving universal variable is actually a process of tuning the universal variable to change oil mass flux until it reaches balance. Since pressure gradient decides the oil mass flux balance in full film region and the oil volume fraction decides the oil mass flux balance in cavitation region, it is a natural decision to integrate dimensionless pressure in full film region and oil volume fraction in cavitation region to form the universal variable.

\[
\alpha = \begin{cases} 
\frac{P - P_c}{P_{ref}} + 1 & : \text{full film zone} \\
\frac{\rho}{\rho_c} = \phi & : \text{cavitation zone}
\end{cases}
\]  

(3.12)

46
Defining the universal variable in this way reserves the clear physical meaning. We can directly relate the universal variable with oil flux. When oil flux variation in whole domain is not large, we can use constant pressure like cavitation pressure as $P_{ef}$ to normalize pressure. When oil flux variation is large, we can define the reference pressure according to the scale of Couette flow or squeezing flow. When Poiseuille flow has same scale at both directions, we can define $P_{ef}$ like equation.

\[
P_{ef} = \begin{cases} 
\frac{6\mu U_0 \Delta x}{H_0^2} & : \text{Couette flow dominate} \\
\frac{6\mu U_x (\Delta x)^2}{H_0^3} & : \text{Squeezing flow dominate}
\end{cases}
\] (3.13)

Here, $H_0$ is the reference clearance, $\Delta x$ is the grid space at x direction, $U_x$ is the surface squeezing velocity. This universal variable definition can assure the universal variable varies at similar scale in full film region and cavitation region. Hence the small numerical error will not cause oscillation when flow switch status. From numerical point of view, it is a preconditioning to discrete algebra system based on physical consideration.

The magnitude of universal variable is designed to be in the order of unity when the oil fills the whole local clearance but has cavitation pressure. That ensures a continuous universal variable transition between full film status and cavitation status. The universal variable itself could serve as cavitation index variable. We can easily judge the status of local flow from whether the universal variable larger than one. The flow is at full film region when $\alpha > 1$, cavitation region when $\alpha < 1$.

Since the universal variable has different physical meanings in full film region and cavitation region, the relation between the oil flow and universal variable should change when the flow switches status. The iteration scheme should take a special care. The details will be explained in section 3.2.4.
3.2.2 Spatial Discretization Scheme

To solve the Reynolds equation, we need to discretize it to a system of algebraic equations. Here we use the Reynolds equation in Cartesian coordinate as an example to show the procedure. At first, the computational domain is divided to a number of small control volumes. Figure 3-2 shows the relation among control volume, nodes, and interfaces of the control volumes. The variables we are interested in such as pressure and oil volume fraction are located at the center of control volume. The pressure gradient and oil flux are defined at the interface between control volumes.

Dividing equation 3.1 by $\rho_c$, we can rewrite Reynolds equation to an equation about the oil volume conservation

$$\frac{\partial (\phi h)}{\partial t} = \frac{\partial}{\partial x} \left( \phi \frac{h^3}{12\mu} \frac{\partial P}{\partial x} \right) + \frac{\partial}{\partial y} \left( \phi \frac{h^3}{12\mu} \frac{\partial P}{\partial y} \right) - \frac{U}{2} \frac{\partial (\phi h)}{\partial x} \quad (3.14)$$

The $\frac{h^3}{12\mu}$ in Poiseuille flow terms has property similar to heat conduct coefficient or diffusion coefficient. It represents how much oil can flow through the interface.
between control volumes. Following the method of processing heat conduct coefficient designed by Patankar[19], we introduce the flow conductivity $K$ as

$$K = \frac{h^3}{12\mu} \quad (3.15)$$

We combine the oil viscosity into the flow conductivity. Therefore when the flow model couples with thermal model, the influence of temperature to viscosity can be easily plugged in.

The flow conductivity is a property attached with interface, which is a harmonic mean of the flow conductivities at the adjacent nodes.[19]. The capital subscripts mark the values on nodes. Lowercase subscripts mark the values on interfaces.

$$K_P = \left. \frac{h^3}{12\mu} \right|_P$$

$$K_E = \left. \frac{h^3}{12\mu} \right|_E \quad , \quad K_W = \left. \frac{h^3}{12\mu} \right|_W$$

$$K_S = \left. \frac{h^3}{12\mu} \right|_S \quad , \quad K_N = \left. \frac{h^3}{12\mu} \right|_N$$

$$K_e = \frac{2K_PK_E}{K_P + K_E} \quad , \quad K_w = \frac{2K_PK_W}{K_P + K_W}$$

$$K_s = \frac{2K_PK_S}{K_P + K_S} \quad , \quad K_n = \frac{2K_PK_N}{K_P + K_N} \quad (3.16)$$

Integrating Equation 3.14 in a control volume yields

$$\frac{\partial (\phi_P h_P)}{\partial t} \Delta x \Delta y = Q_e + Q_w + Q_n + Q_s \quad (3.17)$$

The volume flow at y direction $Q_n, Q_s$ contains only Poiseuille flow. They are discretized with the central difference scheme
\[ Q_n = K_n \frac{P_N - P_P}{\Delta y} \Delta x \]  
\[ Q_s = K_s \frac{P_S - P_P}{\Delta y} \Delta x \]  

(3.18)  
(3.19)

\( \phi \) in Poiseuille terms of equation 3.14 disappeared in equations 3.18 and 3.19 because the oil volume fraction of Poiseuille flow is always 1 when oil pressure is larger than cavitation pressure.

The volume flows at x direction \( Q_e, Q_w \) have both Poiseuille flow and Couette flow. The Couette flow should be discretized in upwind difference. Usually the computational grid inherits the uniform data grid from surface measurement instrument. If we select the time step \( \Delta t = \Delta x/U \), the upwind difference will not influence the precession of result too much.

\[ Q_e = K_e \frac{P_E - P_P}{\Delta x} \Delta y - \phi_p U \frac{h_e}{2} \Delta y \]  
\[ Q_w = K_w \frac{P_W - P_P}{\Delta x} \Delta y - \phi_w U \frac{h_w}{2} \Delta y \]  
\[ h_e = \frac{h_E + h_P}{2}, \quad h_w = \frac{h_W + h_P}{2} \]  

(3.20)

3.2.3 Assemble the Jacobian Matrix

Integrating the equation 6.3.2 from \( t^i \) to \( t^{i+1} = t^i + \Delta t \), we have

\[ \omega \phi_p^{i+1} h_p^{i+1} - (1 - \omega) \phi_p^i h_p^i = \frac{\omega(Q_e + Q_w + Q_n + Q_s)^{i+1} \Delta t}{\Delta x \Delta y} \]  
\[ - (1 - \omega)(Q_e + Q_w + Q_n + Q_s)^i \Delta t \]  

(3.21)

When \( \omega = 1 \), it is a implicit time scheme. When \( \omega = 0.5 \), it is a Crank-Nicolson scheme.

Assembling all the discretized terms together, we have
\[ f(\alpha^{i+1}) = g(\phi^{i+1}, P^{i+1}) - S_P = 0 \] (3.22)

\[ g(\phi^{i+1}, P^{i+1}) = \omega \left[ \frac{\phi_p^{i+1}h_p^i - (Q_c + Q_w + Q_n + Q_s)^{i+1}\Delta t}{\Delta x\Delta y} \right] \] (3.23)

\[ S_P = (1 - \omega) \left[ \phi_p^i h_p^i - \frac{(Q_c + Q_w + Q_n + Q_s)^i\Delta t}{\Delta x\Delta y} \right] \] (3.24)

Applying the chain rule, we obtain the Jacobian matrix of 3.22.

\[
\begin{align*}
\frac{\partial f}{\partial \alpha^{i+1}} &= \frac{\partial f}{\partial P^{i+1}} \frac{\partial P^{i+1}}{\partial \alpha^{i+1}} + \frac{\partial f}{\partial \phi^{i+1}} \frac{\partial \phi^{i+1}}{\partial \alpha^{i+1}} \\
\frac{\partial f}{\partial P_N^{i+1}} &= -\omega \frac{K_n}{\Delta y^2} \Delta t \\
\frac{\partial f}{\partial P_S^{i+1}} &= -\omega \frac{K_s}{\Delta y^2} \Delta t \\
\frac{\partial f}{\partial P_E^{i+1}} &= -\omega \frac{K_e}{\Delta x^2} \Delta t \\
\frac{\partial f}{\partial P_W^{i+1}} &= -\omega \frac{K_w}{\Delta x^2} \Delta t \\
\frac{\partial f}{\partial P_p^{i+1}} &= -\left( \frac{\partial f}{\partial P_N^{i+1}} + \frac{\partial f}{\partial P_S^{i+1}} + \frac{\partial f}{\partial P_E^{i+1}} + \frac{\partial f}{\partial P_W^{i+1}} \right) \\
\frac{\partial f}{\partial \phi_p^{i+1}} &= \omega \left( h_p^{i+1} - U \Delta t \frac{h_p^{i+1}}{2\Delta x} \right) \\
\frac{\partial f}{\partial \phi_w^{i+1}} &= -\omega U \Delta t \frac{h_w^{i+1}}{2\Delta x} \\
\end{align*}
\] (3.25)

From the definition of universal variable \( \alpha \) equation 3.11, we have

\[
\begin{align*}
\frac{\partial P_S^{i+1}}{\partial \alpha_S^{i+1}} &= g_S P_{ref} \,, \quad \frac{\partial P_N^{i+1}}{\partial \alpha_N^{i+1}} = g_N P_{ref} \\
\frac{\partial P_E^{i+1}}{\partial \alpha_E^{i+1}} &= g_E P_{ref} \,, \quad \frac{\partial P_W^{i+1}}{\partial \alpha_W^{i+1}} = g_W P_{ref} \\
\frac{\partial \phi_p^{i+1}}{\partial \alpha_p^{i+1}} &= (1 - g_p) \,, \quad \frac{\partial \phi_w^{i+1}}{\partial \alpha_w^{i+1}} = (1 - g_w) \\
\end{align*}
\] (3.26)
Plugging 3.27 and 3.26 into 3.28, we have the all elements of Jacobian matrix.

\[
\frac{\partial f}{\partial \alpha^{i+1}_N} = -\omega g_N P_{ref} \frac{K_n}{\Delta y^2} \Delta t
\]
\[
\frac{\partial f}{\partial \alpha^{i+1}_S} = -\omega g_S P_{ref} \frac{K_n}{\Delta y^2} \Delta t
\]
\[
\frac{\partial f}{\partial \alpha^{i+1}_E} = -\omega g_E P_{ref} \frac{K_n}{\Delta x^2} \Delta t
\]
\[
\frac{\partial f}{\partial \alpha^{i+1}_W} = -\omega g_W P_{ref} \frac{K_n}{\Delta x^2} \Delta t - \omega(1 - g_w) U \Delta t \frac{h_w}{2 \Delta x}
\]
\[
\frac{\partial f}{\partial \alpha^{i+1}_P} = -\omega g_P P_{ref} \left( \frac{K_n}{\Delta y^2} + \frac{K_s}{\Delta y^2} + \frac{K_e}{\Delta x^2} + \frac{K_w}{\Delta x^2} \right) \Delta t
\]
\[
+ \omega(1 - g_P) \left( h_p - U \Delta t \frac{h_e}{2 \Delta x} \right)
\]

(3.28)

All variables in 3.28 are at time step \( t^{i+1} \) with or without superscript \( i + 1 \).

### 3.2.4 Physical Constrains and Iteration Scheme

After obtaining the full Jacobian matrix of the discrete algebraic equations system, we can use Newton’s iteration to solve the universal variable. Then, physical variables such as \( P \) and \( \rho \) can easily be extracted. Recognizing that the universal variable has different physical meanings when the flow switches status, we developed a new scheme to avoid oscillations and losing physical meaning.

The nonlinear property of the oil transport problem comes from the constraints on the dependent variables. The oil volume fraction should be larger than zero and smaller or equal to one. The pressure should be always higher or equal to the cavitation pressure. The property of oil flux term should be checked to ensure they represent the physical flux correctly. Mathematically, this problem can be elegantly described as an optimization problem with constraints. In this way, the modeling can utilize existing tools and ensure the robustness. But the shortcomings are that the dual variables will take extra memory and computational power and we will loss the direct connection between physical meaning of terms and modeling variables. Hence
we sought for current solution of checking the constrains between iterations. This method can give us direct physical picture of oil transport and provide convenience to handle endless local flow pattern which is hard to handle. But it needs deliberate design of iteration scheme.

Instead of updating both $g$ and $\alpha$ through small relaxation number, we only update $\alpha$ in iteration. $g$ serves as a switch function and takes only zero or one as value. When a node switches status between full film and cavitation, we can not update $\alpha$ according the calculated value. Because when $\alpha$ changes from 0.5 to 1.5, the equations mean the mass balance could be maintained if the oil in control volume tripled. If we directly update $\alpha$ to 1.5, then the real physical meaning of this action is the mass balance is maintained after local pressure increased to $0.5P_{ref} + P_c$. That is totally different from what equations mean. Furthermore, because $\alpha$ has different meanings in cavitation zone and full film zone, the tolerable error in one zone could be unacceptable in the other zone. Therefore, when a node switches status between full film and cavitation, our current iteration scheme sets $\alpha$ to one, and then decides whether $g$ should switch between 0 and 1, based on the local net oil flux calculated from updated $\alpha$. In this kind of way, we can avoid to use small relaxation factors and the oscillation caused by flow switching status. When the index variable $g$ is not changing anymore, the Reynolds equation 3.1 becomes linear, and the result will converge quickly.

3.2.5 Boundary Conditions

Because the definition of universal variable is directly related to the oil volume fraction and pressure, it is fairly straightforward to process the normal boundary conditions. The value of boundary nodes will be used to calculate $S_p$ and $f(p^i, P^i)$ during Newton's iteration, without any involvement with any elements of Jacobian matrix.

There is a special case that needs to be mentioned. For starving oil supply boundary condition, the boundary node could have pressure higher than the cavitation pressure and partial film at the same time. Since we need not to update the universal variable at boundary, it will not cause any trouble if we directly use pressure and oil
volume fraction instead of extracting it from universal variable during iteration. The only additional step for us to take is to modify the Poiseuille flow. For example, if node $W$ is a boundary node. Term $Q_w$ in 3.20 should be calculated as

$$Q_w = \begin{cases} 
K_w \frac{P_w - P_P}{\Delta x} \Delta y - \phi_w U \frac{h_w}{2} \Delta y & : P_W \leq P_P \\
\phi_w K_w \frac{P_w - P_P}{\Delta x} \Delta y - \phi_w U \frac{h_w}{2} \Delta y & : P_W > P_P 
\end{cases} 
$$  \hspace{1cm} (3.29)

By doing so, we can partly handle the starving oil supply boundary condition when pressure at boundary is not so far away from the cavitation pressure. However, when the gas pressure is significantly higher than the cavitation pressure under starved conditions, we need to use a more sophisticated gas-oil multi phase model to be discussed later.
Chapter 4

Model Application One: Metal Face Seals

The flexible metal-to-metal face seal (FMMFS) is a machines component that seals the lubricant in load supporting axis (Figure 4-1). The major function of FMMFS is sealing the lubricant while keeping the friction low. This function requires FMMFS maintaining the balance between leakage and friction. But this balance is hard to achieve because these two requirements contradict to each other. The higher load helps decreasing leakage but it increases the possibility of scoring since higher load leads to larger friction and heat. To find the optimized parameters, a model that provides quantitative results of hydrodynamic pressure and oil distribution for any given sealing surface geometry is highly desirable.

4.1 Deformation Model

4.1.1 The Geometry of Metal Face seal

The FMMFS is formed by two identical metal rings pressed together by two identical rubber o-ring. The whole set of structure is installed in housing ramp of the seal. Unlike most of the seal designs that a seal pair is composed with one with softer material than the other, the two rings of the seal pair have identical materials and
they are designed to wear down simultaneously to maintain the sealing. When the clearance between seal housings is changed by vehicle movements, the rubber toric rings roll on the seal ramp to keep the load on seal constant. The identical geometry and material of rings are designed to wear down evenly and keep the seamless sealing interface.

From the view of the section that crosses the center of FMMFS(Figure 4-2), the seal profile has three bands. From the center of the seal to outside the seal, the three bands are taper band, transition band and flat band.

The taper band is where the oil supply comes from. It is used to minimize the axial loading while maintaining proper unit pressure on the flat band. Additionally, as the flat band experiences continuous wear, it gradually extends to the original taper band. Therefore, the taper angle controls the rates of increase of the radial width of the flat band and decrease of unit pressure, and is a critical design parameter.

The transition band provides a bridge between the taper and flat bands. The clearance at taper band is around millimeters; therefore it does not directly influence the load supporting and sealing ability of seal. The clearance at transition band
gradually decreases from millimeter to micron level. This smooth transition area ensures the sealing locus varying continuously when the thermal expansion effect twists the seal surfaces open. The most important geometry parameter of transition band is its radius of curvature.

The flat band is the place where surface contact locates. Finally, the flat band supports the axial loading and provides the sealing. Although nominally flat, this flat band exhibits complex geometry. For a seal ring with approximately 100mm diameter, the height variations of the flat band at free state can be from microns to 10s of microns depending on the manufacture processes. When thermal expansion effect twists seal surfaces open, the angle between flat bands could be up to 0.1 degrees[11]. The twist angle of flat band is concavity when this twist moves surface contact locus away from seal center, convexity when towards the seal center. Together, the transition band and flat band are sealing band. The sealing band is where the hydrodynamic pressure, friction and wear are generated. All seal geometry variation for design purpose lead to geometry variation in sealing band. The FMMFS model takes geometry of sealing band as input. The output of FMMFS model helps designer to check the effects of design.

In addition to the profile of sealing band, the roughness at sealing band plays an important role to seal performance too. The seals with different roughness features have different performances. Past experiences show that once the surface roughness of the flat band is over certain limit, the seal leaks right at the beginning.

4.1.2 2D Deformation Model of Ring Surface

The load on seal is usually in the order of $N/mm$. This unit means there is $1N$ load applied on every millimeter length along circumferential direction. Though the width of sealing band is roughly $3mm$, the width of actual contact locus is around $1mm$. Hence when load on seal is $1N/mm$ the average pressure in contact area is around 10 bars. This pressure is far less than the yield strength of steel which is in the order of $GPa$. From this observation, it is nature to assume the surface texture and
roughness do not change too much with the applied load. Because the circumference of the seal is much larger than the width at radial direction, the seal is much flexible at circumferential direction than at radial direction. Though the load on seal is not large enough to change the surface micro geometries, it is enough to deform the ring profiles at circumferential direction. The finite element analysis of seal deflection shows the deflection of seal is in the order of $10\mu m$ when load is $1N/mm$. This load is still lower than the typical load in applications. The measured amplitude of seal waviness is around 2 microns. That means when all loads are applied, the waviness of seal has been flatten out.

Based on this assumption, the ring deformation model treats the seal as a group of individual sections along the circumferential direction. Each section has its own radial profile and longitudinal roughness extending at circumferential direction. The ring sections are subject to two kind of deformations. The first one is the twist around the circumferential direction. The twist angle is the result of balance between thermal expansion effect and torque of rubber ring applied on seal. This twist angle comes from experiment results at different temperatures. In the deformation model, the twist angle is an input parameter. The second ingredient of ring deformation
is the shift of clearance between ring surfaces. For surface with 1D roughness, Lee gave a model of the relation between load at unit length and dimensionless clearance normalized by the roughness $R_q$ number. For any given clearance between surfaces, this model can calculate the asperity contact force along radial locus. Integrate the asperity contact force; we get the total contact force on the section.

With the given twist angle, the contact force, which is the applied load minus the hydrodynamic force, decides the clearance between surface. The hydrodynamic force is sensitive to the clearance between ring surfaces. Therefore an iteration to detect the correct clearance is necessary. The iteration starts from current twist angle and hydrodynamic force. If the sum of contact force and hydrodynamic force is larger than applied load, the clearance will increase. If it is smaller than applied load, the clearance decreases. When this sub iteration converged, the clearance will generate the needed contact force to balance applied load with hydrodynamic force. Once the clearance for each section on circumferential direction is decided, the geometry of seals will change according to clearance on each section. The hydrodynamic pressure calculation in the next iteration will be based on the modified clearance.

4.1.3 Surface Measurement and Stitching Patches

One important input to apply the model is the surface geometry in the flat band and the outer transition band. With the available optical profilometer, we needed to stitch different measured patches to obtain the geometry (waviness) of the region we are interested. Realizing the existing stitching program coming with the equipment was not accurate enough for our purpose (Figure 4-3a), we developed our own stitching procedure and algorithm. We used a low spatial resolution ($0.25\,mm$) profilometer to measure the geometry at profile level that includes the height variation at circumferential direction and the geometry in transition band. Another high spatial resolution ($20\,\mu m$) profilometer measures the geometry at texture level that includes the height variation at radial direction. The patches of high spatial resolution profilometer are mapped to low spatial resolution result. The stitching algorithm shifts and twists the patches to match them with the low spatial resolution result (Figure 4-3b).
Given the low resolution measurement $Z(x, y)$ and high resolution measurement on a patch $Z_p(x, y)$, the stitching algorithm is an optimization problem of getting the minimum value of object function $Err$ by tuning three parameters. The three parameters are the height shift of patch $h_0$, the twist angle $\theta_x$ around $x$ axis and the twist angle $\theta_y$ around $y$ axis.

\[
Err \equiv \iint (Z(x, y) - Z_p(x, y) - h_0 - y \tan(\theta_x) - x \tan(\theta_y))^2 \, dx \, dy \tag{4.1}
\]

### 4.2 Fluid Model

#### 4.2.1 Reynolds Equation in Polar Coordinate

The oil transport in FMMFS follows the Reynolds equation in Polar coordinate. For a seal rotates at angular velocity $\Omega$, it is in the form of equation 4.2
\[
\frac{\partial}{\partial r} \left( \rho r \frac{h^3}{12\mu} \frac{\partial p}{\partial r} \right) + \frac{\partial}{\partial \theta} \left( \frac{\rho h^3}{12\mu} \frac{\partial p}{\partial \theta} \right) = \frac{\partial}{\partial \theta} \left( \rho \phi \Omega r \right) + \frac{\partial (\rho \phi h)}{\partial t} \tag{4.2}
\]

Dividing equation 4.2 by \( \rho_c \), we rewrite equation 4.2 as

\[
\frac{\partial (\rho \phi h)}{\partial t} = \frac{\partial}{\partial r} \left( \rho r \frac{h^3}{12\mu} \frac{\partial p}{\partial r} \right) + \frac{\partial}{\partial \theta} \left( \frac{\rho h^3}{12\mu} \frac{\partial p}{\partial \theta} \right) - \frac{\partial (\rho \phi \Omega r)}{\partial \theta} \tag{4.3}
\]

Comparing equation 4.3 with equation 3.14, we notice that two equations are in similar form. Therefore, the discretization of equation 4.3 will follow the same analysis described in the last chapter.

The flow conductivity at \( r \) direction and \( \theta \) direction are introduced as

\[
K_r = \frac{rh^3}{12\mu}, \quad K_\theta = \frac{h^3}{12\mu r} \tag{4.4}
\]

Unlike the Reynolds equation in Cartesian coordinate, the flow conductivity is related with flow direction here. The flow conductivities on nodes are:

\[
K_{r,P} = \left. \frac{rh^3}{12\mu} \right|_p, \quad K_{\theta,P} = \left. \frac{h^3}{12\mu r} \right|_p
\]

\[
K_E = \left. \frac{h^3}{12\mu r} \right|_E, \quad K_W = \left. \frac{h^3}{12\mu r} \right|_W
\]

\[
K_S = \left. \frac{rh^3}{12\mu} \right|_S, \quad K_N = \left. \frac{rh^3}{12\mu} \right|_N \tag{4.5}
\]

The flow conductivities at interfaces are calculated as

\[
K_e = \frac{2K_{\theta,P}K_E}{K_{\theta,P} + K_E}, \quad K_w = \frac{2K_{\theta,P}K_W}{K_{\theta,P} + K_W}
\]

\[
K_s = \frac{2K_{r,P}K_S}{K_{r,P} + K_S}, \quad K_n = \frac{2K_{r,P}K_N}{K_{r,P} + K_N} \tag{4.6}
\]
Integrating the equation 4.2 on the control volume gives

\[
\frac{\partial (\phi hr)}{\partial t} \Delta \theta \Delta r = Q_e + Q_w + Q_n + Q_s \quad (4.7)
\]

The volume flow at \( r \) direction \( Q_n, Q_s \) are discretized in central difference as

\[
Q_n = K_n \frac{P_N - P_P}{\Delta r} \Delta \theta, \quad Q_s = K_s \frac{P_s - P_P}{\Delta r} \Delta \theta \quad (4.8)
\]

Variable \( \phi \) disappeared since it always equals one in Poiseuille flow terms.

The volume flows in \( \theta \) direction \( Q_e, Q_w \) are composed of both Poiseuille flow and Couette flow. Discretizing Poiseuille flow in central difference and the Couette flow in upwind difference, we have

\[
Q_e = K_e \frac{P_E - P_P}{\Delta r} \Delta \theta - \phi_p \Omega r_P \frac{h_e}{2} \Delta r
\]

\[
Q_w = K_w \frac{P_W - P_P}{\Delta r} \Delta \theta - \phi_w \Omega r_P \frac{h_w}{2} \Delta r
\]

\[
h_e = \frac{h_E + h_P}{2}, \quad h_w = \frac{h_W + h_P}{2} \quad (4.9)
\]

### 4.2.2 Assemble the Jacobian Matrix

Integrating the equation 4.6 from \( t^i \) to \( t^{i+1} = t^i + \Delta t \), we have

\[
\omega r_P^{i+1} \phi_P^{i+1} h_P^{i+1} - (1 - \omega) r_P^i \phi_P^i h_P^i = \frac{\omega (Q_e + Q_w + Q_n + Q_s)^{i+1} \Delta t}{\Delta \theta \Delta r} - \frac{(1 - \omega) (Q_e + Q_w + Q_n + Q_s)^{i} \Delta t}{\Delta \theta \Delta r} \quad (4.10)
\]

Putting all the discretized terms together, we obtain

\[
f(\alpha^{i+1}) = g(\phi^{i+1}, P^{i+1}) - S_P = 0
\]

\[
(4.11)
\]
where,
\[
g(\phi^{i+1}, P^{i+1}) = \omega \left[ \phi_P r_P h_P^{i+1} - \frac{(Q_e + Q_w + Q_n + Q_s)^i \Delta t}{\Delta \theta} \right]
\]
\[
S_P = (1 - \omega) \left[ \phi_P r_P h_P^{i+1} - \frac{(Q_e + Q_w + Q_n + Q_s)^i \Delta t}{\Delta \theta} \right]
\]

(4.12)

According to chain rule, we have
\[
\frac{\partial f}{\partial \alpha^{i+1}} = \frac{\partial f}{\partial P^{i+1}} \frac{\partial P^{i+1}}{\partial \alpha^{i+1}} + \frac{\partial f}{\partial \phi^{i+1}} \frac{\partial \phi^{i+1}}{\partial \alpha^{i+1}}
\]

(4.13)

From equation 4.12, we can get terms in equation 4.13
\[
\frac{\partial f}{\partial P_N^{i+1}} = -\omega \frac{K_n}{\Delta \theta^2} \Delta t
\]
\[
\frac{\partial f}{\partial P_S^{i+1}} = -\omega \frac{K_s}{\Delta \theta^2} \Delta t
\]
\[
\frac{\partial f}{\partial P_E^{i+1}} = -\omega \frac{K_e}{\Delta \theta^2} \Delta t
\]
\[
\frac{\partial f}{\partial P_W^{i+1}} = -\omega \frac{K_w}{\Delta \theta^2} \Delta t
\]
\[
\frac{\partial f}{\partial P_P^{i+1}} = -\left( \frac{\partial f}{\partial P_N^{i+1}} + \frac{\partial f}{\partial P_S^{i+1}} + \frac{\partial f}{\partial P_E^{i+1}} + \frac{\partial f}{\partial P_W^{i+1}} \right)
\]
\[
\frac{\partial f}{\partial \phi_P^{i+1}} = \omega \left( r_P^{i+1} h_P^{i+1} - \Omega r_P^{i+1} \Delta t \frac{h_P^{i+1}}{2 \Delta \theta} \right)
\]
\[
\frac{\partial f}{\partial \phi_W^{i+1}} = -\omega \Omega r_P^{i+1} \Delta t \frac{h_P^{i+1}}{2 \Delta \theta}
\]

(4.14)

From the definition of universal variable \( \alpha \) equation 3.11, we have
\[
\frac{\partial P_S^{i+1}}{\partial \alpha_S^{i+1}} = g_S P_{ref}, \quad \frac{\partial P_N^{i+1}}{\partial \alpha_N^{i+1}} = g_N P_{ref}
\]
\[
\frac{\partial P_E^{i+1}}{\partial \alpha_E^{i+1}} = g_E P_{ref}, \quad \frac{\partial P_W^{i+1}}{\partial \alpha_W^{i+1}} = g_W P_{ref}
\]
\[
\frac{\partial P_P^{i+1}}{\partial \alpha_P^{i+1}} = g_P P_{ref}
\]
\[
\frac{\partial \phi_P^{i+1}}{\partial \alpha_P^{i+1}} = (1 - g_P), \quad \frac{\partial \phi_W^{i+1}}{\partial \alpha_W^{i+1}} = (1 - g_W)
\]

(4.15)
Plugging 4.15 and 4.14 into 4.16, we have the all elements of Jacobian matrix.

\[
\begin{align*}
\frac{\partial f}{\partial \alpha^N_{i+1}} & = -\omega g_N P_{ref} \frac{K_n}{\Delta r^2} \Delta t \\
\frac{\partial f}{\partial \alpha^S_{i+1}} & = -\omega g_S P_{ref} \frac{K_n}{\Delta r^2} \Delta t \\
\frac{\partial f}{\partial \alpha^E_{i+1}} & = -\omega g_E P_{ref} \frac{K_n}{\Delta \theta^2} \Delta t \\
\frac{\partial f}{\partial \alpha^W_{i+1}} & = -\omega g_W P_{ref} \frac{K_n}{\Delta \theta^2} \Delta t - \omega (1 - g_w) \Omega r_w \Delta t \frac{h_w}{2 \Delta \theta} \\
\frac{\partial f}{\partial \alpha^P_{i+1}} & = -\omega g_P P_{ref} \left( \frac{K_n}{\Delta r^2} + \frac{K_s}{\Delta r^2} + \frac{K_e}{\Delta \theta^2} + \frac{K_w}{\Delta \theta^2} \right) \Delta t + \omega (1 - g_p) \left( h_p r_p - \Omega r_p \Delta t \frac{h_e}{2 \Delta \theta} \right)
\end{align*}
\]

Without extra superscripts, all numbers in 4.16 are at \(t^{i+1}\).

The iteration scheme is the same as described in section 3.2.4

### 4.2.3 Patch Measurement Based Flow Factor Improvement

The surface roughness of FMMS has a unique character from the machining and wear processes. First, the last process of the machining, namely, polishing gives the longitudinal roughness along the circumference. The relative movements between rotor and stator include rotation at circumferential direction, off center shifting at radial direction, separating and squeezing at axial direction. In all these movements, the rotation is the most significant one. The grinding effect generates surface roughness extending at the circumferential direction. This non-isotropic surface roughness enhances Poiseuille flow at circumferential direction while decreases Poiseuille flow at radial direction.

The model considers this effect by modify the flow conductivity at circumferential direction and radial direction according to surface roughness measurement. This method is in essence similar to Patir and Chen’s flow factor model[20]. While the flow-factor type of method is questionable when the clearance is a few times of the
roughness standard deviation, for this unidirectional roughness is at least reflects the flow resistance difference in two directions. Further work needs to examine if more advanced approach needs to be developed to consider the roughness effects and it is beyond the scope of the current work.

4.3 Model Results

More detailed results combining experimental observation and theoretical studies were conducted by Wang who incorporated the lubrication model developed in this work into his thermal-lubrication model. [35] [36][34] Here we only demonstrate the effectiveness of the hydrodynamic lubrication model in describing the oil transport in the sealing band. The seal has to seal the lubricant and minimize the fiction. There are two major working conditions of the seal. For the inner seal, both inner side(ID) and outer side(OD) is filled by lubricant. For the outer seal, only ID is filled by lubricant, the OD is in contact of air. The work in this chapter is focused on the applications that both sides of seal are fully filled by lubricant. The special one way flow boundary condition was applied at OD. Though this boundary condition can not model the gas penetration and the outer locus of oil fully filled region. It helps to avoid the unrealistic oil supply from OD when OD of outer seals are in contact with air.

4.3.1 Model Configuration

To demonstrate the lubrication model, the problem is simplified as for a given geometry of rotor and stator surface, running conditions, initial condition and boundary condition, we calculate the oil distribution and pressure distribution at specific time. The inner radius of the seal we are modeling is 43mm, the width of the flat band is 3mm. The surface profile data of rotor and stator come from optical profilometer measurement. The dynamic viscosity of oil is 0.005 Pa·s. The rotation speed of seal is 667RPM. The average clearance between rotor and stator is 0.55μm. The cavitation pressure is 0.5 bar. It needs to be mentioned that the setting of parameters is to demonstrating the ability of the model and the parameters are not necessary the
actual running parameters.

The model calculates the oil volume fraction and pressure distribution. Based on these two primary output, multiple seal performance indicators are defined to help seal design. We showed the pressure distribution after two cycles in Figure 4-4. The clearance at ID side is much larger than the clearance in flat band. Therefore, the pressure mainly buildup in the flat band. At the moderate running speed, the hydro pressure could be at the order of 10\textit{bar}.

Figure 4-5 shows the oil supply at the ID side. Though the ID boundary condition is full film boundary condition, the pressure build up in the flat band keeps pushing part of oil from flat band through ID boundary. This effect has two impacts. At one side, it decrease the oil supply from the ID side. At another side, it helps the oil circulation in the flat band and keep the oil in flat band fresh. The performance of seals depends on the balance of these two impacts. The model provides quantitative results help to find the boundary of operational parameters.
Figure 4-5: Oil supply at ID after two cycles

Figure 4-6: Oil leakage at OD after two cycles
Figure 4-6 shows the oil leakage at the OD side. For a seal both ID and OD side are filled by oil, this leakage is not necessary a bad effect. But for seal in contact with air at OD, the leakage of oil is a number we want to minimize to zero. There are two oil leakage spike at OD side for these pair of seals.

Even when seal can get constant oil supply from ID, the oil may not go through at the radial direction far enough to lubricate all possible contact spots. To investigate the quantity of oil circulation at radial direction, the oil exchange rate is defined as a factor as the sum of absolute value of oil flow rate at radial direction (Figure 4-7). The oil exchange is large at the radial location close to oil supply. When the location getting deeper and deeper into the flat band, it gets harder to have sufficient oil supply. Most of oil will be circulated back to ID before it reaches the working spot.

The applications of seals show the leakage can happen at very short time scale. In the full cycle of seal rotation, the major leakage may happen at several specific phases. In another word, if the local oil exchange rate is huge, this could be a
good indication of possible leakage. The non leakage pattern is such a performance indication defined as logical value that marks whether the local oil exchange rate is larger than the leakage threshold (Figure 4-8). All spots that oil exchange rate is small enough are considered as spots oil can not pass. At any given moment, if there is no continuous passage connects ID and OD, the seal is considered as in good performance at the moment.

With the exact value of local Couette flow and Poiseuille flow, we can calculate the viscous friction in seal. Figure 4-9 shows the viscous friction in three full cycles start from a sudden start of seal. The friction reach it maximum at the starting point, then decrease to a steady average value with the excessive oil has been expelled from the flat band.
Figure 4-9: The viscous friction variation from startup in three cycles

### 4.4 Conclusions

More detailed work on FMMF seals were presented by Hong and Wang [11][34] who integrated different versions of the hydrodynamic lubrication model into their system models. The content presented shows that the numerical scheme developed in this work enables one to effectively handle the complex oil transport phenomena in FMMF seals. While exact leakage and scoring tendency is still difficult to be fully predicted, the model prediction with simplified boundary condition and analysis on oil exchange rate can still provide insight to their understanding.
Chapter 5

Model Application Two: Twin Land Oil Control Ring

Examining the interaction between piston rings and the liner, one can notice that there is a large extent of length scales involved. First, the axial width of the running surface of the top two rings range from 1-4mm for the top two rings. On the other hand, the axial width of the running surface of the oil control ring is in the order of 0.1mm to achieve high unit pressure (Figure 5-1). In terms of face profiles, the top ring has in general a barrel face with a height drop in the order of 10 microns over the extent of the running surface. The second ring usually has a taper face design. The taper part does not have much lubrication function and the lower worn part exhibits a combination of barrel shape and linear line after certain amount of running time[30], and plays an essential role interacting with the liner. The barrel drop of the worn part is usually in the order of 1 micron. Certainly, as the worn width increases, the feature becomes more prominent. A three-piece oil control ring, which is widely used in SI engines, has fairly sharp running surface profile due to the rails’ freedom twisting within the clearance[31]. Finally, the Twin-Land Oil Control Ring (TLOCR) has virtually flat running surface, largely due to the constraint between the two lands under sufficient tension.

Although a TLOCR has flat running surface, examination of a worn liner generally
Figure 5-1: Ring pack face profile
shows that at BDC of the oil control ring location, there is more wear on the liner than
the mid-stroke. This general evidence suggests existence of hydrodynamic pressure
generation when the TLOCR travels at mid-stroke. On the other hand, lubrication
theory based on either original or averaged Reynolds equation indicates that flat
surface cannot generate hydrodynamic pressure rise in the lubrication area.

The essence of the averaged Reynolds model[20] is to treat the roughness effects
as a minor disturbance to the lubrication process dominated by macro profile. As
being discussed earlier, the average Reynolds model has certain validity when being
applied to the top two ring lubrication as the macro profile has more than one micron
height difference across the axial extent of the running face, which is much greater
than the roughness height on the liner. Cautions still need to be made as in reality
the majority portions of the top two rings are only lubricated by the oil left by the oil
control ring. As a result, the real lubrication area may be much less than the nominal
axial width of the ring. Within the small lubrication area, the height difference may
be in the same order of liner roughness height.

This work is intended to bring a new approach to fundamentally examine the
effects of liner finish at the micro level. To do so, a deterministic method was applied
to study the interaction of the flat land running surface of a TLOCR and a rough
liner. First, the model basics are discussed and consistency of the model is examined.
Then, results and discussions are focused on lubricant oil transport and hydrodynamic
pressure generation within asperities and their network.

5.1 Twin Land Oil Control Ring Model

5.1.1 Statement of the Problem

In this application, we applied a deterministic model to calculate the lubricant trans-
port between the running surface of the TLOCR and liner. There are two lands with
flat top surfaces on the TLOCR. In a properly functioned TLOCR, the difference of
force on the two lands of TLOCR should be able to generate a torque to twist the
TLOCR lands parallel to liner surface. As a result, in general the running surface profile of the TLOCR is found to be flat. Consequently, if there is any hydrodynamic force generation between the flat surface and the rough liner, it should come from the micro geometrical features on liner roughness instead of macro profile.

A more complete approach would be that two flat surfaces sliding on a rough liner (Figure 5-2). However, at the scale of surface roughness, the free surface oil on the liner between the two lands can redistribute, driven by the surface tension. Maintaining flow continuity between two lands requires resolution of the film free-surface redistribution and it will be discussed in the future publications. In this application, we mainly focus on the bottom land during engine down stroke (Figure 5-2). The measured surface used has 4 or 1 micron spatial resolution in both axial and circumferential directions. The land width of TLOCR is 0.15 or 0.2 mm. 2 to 4 mm in the circumferential direction were used in different cases. Doing so gives the calculation domain an aspect ratio greater than 10. Therefore, the simulation can be considered to be sufficient in terms of covering the features along the circumferential direction.

In the calculation, both sliding speed and nominal clearance between the ring and liner are constant. In real applications, sliding speed is changing and the ring tension is fixed rather than clearance. In the following paper by Chen et al., a correlation method was developed to extend the current local calculation to entire engine cycle.[1][2]

5.1.2 Contact Preprocessing

Additional to the distinction of the full-film and cavitation zones, possibility of direct contact between the ring and liner asperities have to be considered. Needless to say that the number of contact points increases with applied load. Figure 5-3 shows the contact patterns of different film thickness ratio (Dark points are contact points). One can see that these contact points are not evenly distributed and quite complex.
Figure 5-2: Oil control ring sliding above liner
Figure 5-3: Contact patterns for different $\lambda$
in computational domain.

At the contact points, local film thickness is zero and thus Reynolds equation becomes singular and loses its validity. There are many possible ways to avoid singularity caused by individual contact points. For example, one can set a very large diagonal coefficient at the row of the matrix corresponding to the contact points. Doing so could cure the matrix singularity caused by individual contacts. However, when the nominal clearance is small, there are always possibilities that some points with non-zero clearance surrounded by contact points that forms a continuous closed circle. These points circled by the contact points are then isolated from outside of the contact points. Consequently a zero pivot matrix occurs, which cannot be detected by this method. Another option is to set local film thickness to a small value at contact points. By doing so, all the points in the calculation regime are assembled into the matrix and thus the method is not economical. Furthermore, setting the $h$ at contact points to a small number, one would obtain a very small flow conductivity $K$ in the corresponding row, which leads to an ill matrix. As a result, the number of iterations required for convergence increases dramatically. In summary, these two options were considered and then eventually abandoned in favor of the following approach.

In this model, an efficient and robust method was developed to handle the complexity caused by contact points. In this method, the surfaces are preprocessed before assembling the matrix. If variable flow conductivity $K$ between two neighboring points is greater than a threshold number, they are considered to be connected. Only points reach boundary through a connected path are in calculation. The isolated points simply keep their initial pressure and mass. The argument is there still are pressure generation and mass transport inside those isolated zones. However, considering there is not any mass exchange between connected zone and isolated zones. The results in connected zone, which is our main concern, still maintain correctness, though further improvements at contact points are possible and important.
5.1.3 Fully Flooded Leading Land and Starving Oil Supply at Trailing Land

Boundary conditions at the leading and trailing edges need to consider both pressure and liquid oil percentage. Constant pressure boundary condition is applied at the leading edge and the trailing edge (1 bar in the calculation shown later). At the leading edge, full film boundary condition is applied.

The boundary condition for liquid oil percentage at the trailing edge needs a bit extra work although directly setting it as full film would not affect the results in any significant way. Nonetheless, here at the trailing edge, following the discuss in section 3.2.5, the boundary condition for liquid oil percentage is set in such a manner that allows both partial and full films.

5.2 Results and Analysis

To give an illustration of the model results, the model was applied to a TLOCR with 0.2mm land width sliding on a measured liner surface patch shown in figure 5-4. The surface is measured in a 4μm mesh. Starting from the right end of the surface, the flat ring land of 0.2mm width is sliding over the liner surface to the left at a sliding speed of 3m/s. The oil flow mass balance and hydrodynamic pressure generation are calculated inside the ring land domain phase by phase, while in each phase, the ring moves to the left by the distance of one grid size (4μm). Oil viscosity is fixed at 0.005Pa.s, and the average film thickness, defined as the shift between the ring land surface and the mean plateau surface of the liner, is maintained in each calculation.

Figure 5-5 plots the local clearance, hydrodynamic pressure generation and density distributions at a specific ring location, phase 500 (see figure 5-4 for the ring location). Here the density is defined as the local oil volume ratio, and density equal to one refers to full film, and density equal to zero means no oil locally. As shown in figure 5-5, local cavitations are mainly distributed in the valley area, and continuous pressure build up could be observed in the plateau area. The complex liner surface geometries make
Figure 5-4: Liner surface measurements and the position of ring in three different phase in calculation
the cavitation pattern rather complex and make it a necessity to solve the system by deterministic method.

Figure 5-6 shows the local clearance profiles and pressure generations in the ring domain at three different ring locations, namely phase 300, 500 and 700, whose ring positions on the liner is illustrated in figure 4. The average film thickness is maintained at 0.628µm, which for this specific liner would generate a very small asperity contact pressure. The average hydrodynamic pressure in this case could be around 5 bars at a low speed of 3m/s. Considering the linearity of hydrodynamic pressure to the sliding speed[1], in the mid stroke, when the sliding speed would be over 15m/s, substantial hydrodynamic pressure (over 20 bars) could be generated by the liner micro geometries to sustain the normal ring load.
Figure 5-6: Hydrodynamic pressure generation in three different phase
5.2.1 Mass Conservation

To demonstrate that the code solves the equations correctly, mass conservation as well as rupture and reformation conditions need to be examined. The mass conservation was examined using different sizes of control volumes. Doing so can ensure that the model not only converges at local spot, but also damps out the error accumulation from individual points. The results show that for all the sizes of control volumes, relative residual is in the order of $10^{-6}$ as controlled. Figure 5-7 shows the relative residuals at all sliding phase for the case discussed in the previous section. Here the relative residual is defined as the ratio of residual flux in unit time to the local volume.

Another important check is whether the pressure and density satisfy JFO cavitation boundary condition. At the film rupture location, the pressure gradient is zero.

$$\frac{\partial P}{\partial n} = 0$$  \hspace{1cm} (5.1)

At the reforming location, the boundary condition is
Here, $n$ is the direction normal to cavitation boundary, from cavitation zone pointing to full film zone.

Figure 5-8 shows the clearance, pressure and oil density plot at one section extended on TLOCR sliding direction. At the rupture loci, pressure gradient is zero and density is continuous. At the reformation locus, both pressure gradient and density are discontinuous. There are large numbers of cavitation zones in results. They all satisfy the JFO boundary condition well.

5.2.2 Oil Redistribution

There are two components in oil transport between OCR and liner, namely, the sliding driven flow and pressure driven flow. The sliding driven flow is always present as long as two surfaces have relative sliding speed difference. The pressure driven flow is generated to balance the difference of sliding driven flow caused by clearance difference in order to preserve mass conservation. At the locations with pure sliding driven flow, pressure gradient is zero, the lubricant or lubricant and gas mixture only moves in the axial direction. On the other hand, at the locations with the presence of pressure gradients, the oil flow in the axial direction is either enhanced or reduced by the pressure gradient in the axial direction. Additionally, the pressure gradient in the circumferential direction can push or suck lubricant side way and redistribute the lubricant along the circumference. Pure gas bubble could be driven by small pressure gradient and flow at circumferential direction. But the mobility of gas in mixture is limited by surrounding lubricant and only follows the flow of surrounding lubricant. Large pressure driven flow always appears when there are contact points on the sliding direction. Low clearance points act quite similar to contact points impeding flow passing by even though they are not real contact points. Figure 5-9 shows pressure driven flow direction over local clearance contour (the numbers on the plot are the film thickness in microns). One can see that there are several low
Figure 5-8: A cross section cavitation zone
clearance points at the left side, which are connected with each other and form a continuous oil-blocking belt. When the oil is dragged to the connected asperity belt from high clearance upstream, it is expelled in front of these low clearance points. Consequently, a large continuous pressure rise is generated along the asperity belt (Figure 5-10). Contrarily, oil can easily pass isolated individual asperities without introducing significant pressure rise.

Further observation shows the correlation between the pressure driven flow and surface pattern. Figure 5-11 shows the pressure driven flow at left and local clearance at right. The darker points have larger clearance. The black lines are horning grooves. Compared to the larger clearance in the groove/valley areas, the points in plateau area act like low clearance points. When sliding driven flow drags the lubricant in the groove valleys to the plateau area, large pressure driven flow is built up to balance the sliding flow. Across those large pressure gradient belts, the pressure is increased.
to a higher level in the plateau area and maintains significant hydrodynamic pressure in the downstream plateau areas.

Quite different from intuition, the large pressure driven flow in groove/valleys is not along the valleys and it is mostly in the opposite of sliding direction. Only at the transition from the valleys to plateau, the pressure flow shows components along the circumferential direction. Nonetheless, the existence of the pressure driven flow at the transition from valleys to plateau shows the importance of the valleys in oil supply to and pressure rise in the plateau areas. This point will be further illustrated in a later section.

5.2.3 Pressure Noise Processing

In the pressure results, there are always some points with pressure spikes at Giga-Pascal level. The smaller the nominal clearance is, the more these isolated pressure spikes are. These isolated pressure spikes occur at the locations with the film thickness
Figure 5-11: Pressure driven flow and surface grooves
less than 0.1 micron. They are in general not real and the results of insufficient local spatial resolution. Although at these locations, the mass flow balance are not disturbed due to their small clearance, the magnitude of hydrodynamic pressure is important to the determination of clearance. We applied two methods to remove those unrealistic pressure spikes. In the first method, we set up a threshold clearance limit such that the hydrodynamic pressure at the points with less clearance than the threshold is the average of hydrodynamic pressure at the surrounding points.

In the second method, we detect the pressure continuity. When a point pressure is higher than the sum of a threshold pressure and the average pressure of its four direct neighbor points, we start a local pressure solver. This solver takes the five points’ clearances and the pressures of the four neighbor points as input, interpolates the surface profile on a finer mesh, and calculates the average pressure in that local position with a finer resolution. The pressure result from this local solver is used to replace the original pressure. The question turns to how to setup correct threshold clearance or pressure. In figure 5-12, we plot out the local momentum of the pressure distribution. The local momentum is defined as pressure times its possibility density function of the pressure, which indicates the contribution of each specific pressure value to the total hydrodynamic pressure generation. Higher local momentum means greater contribution to average pressure. The two spikes around 1 bar refer to boundary pressure and cavitation pressure. The original pressure calculation shows that local pressures with large values contribute a lot to the average pressure. The other two curves show the pressure contribution after pressure spikes removed. Those three curves show almost the same profile at pressure lower than 100 bar. Hence, we picked 100 bar as the pressure threshold, which shows quite reasonable result. For all the cases we studied, the pressure momentum show stable continuous variations with parameter perturbation and thus it seems to be a quite useful statistical tool to separate the real pressure from the pressure noises caused by near-contact points.
5.2.4 Oil Accumulation

To explore the phenomenon of oil accumulating between the two lands of TLOCR, a special TLOCR model was built up based on the single specie two phase unsteady lubrication model. The geometry specifications are as the Figure 5-13 shows. The ring lands are perfect smooth surfaces with 0.16mm width. The distance between the upper land and bottom land is 1mm. The ring slide at speed of 5m/s on the liner. The model contains two piece of computational domain. Each one represent one land. At the leading edge of the bottom land, the oil is fully supplies since where the oil supply directly comes from the oil reservoir. The model calculates how does the oil pass through the bottom land and leave it at the trailing edge of bottom land. The roughness on liner builds up pressure. This pressure drives oil to redistribute through the clearance in plateau area an horning grooves. When the oil leaving the trailing edge, some location is fully flood while some location is only partially filled.

Figure 5-14 shows a small section on the trailing edge of bottom land. The oil volume ration show the how much of local clearance is filled by oil. The pressure flux has been converted to the oil film height increments after the oil leave the trailing
Figure 5-13: The geometry of TLOCR sliding on liner
Figure 5-14: Oil volume fraction, pressure driven flux and clearance
edge of bottom land. Both this virtual oil height and the local clearance has been normalized according to the maximum clearance in this section.

After leaving the trailing edge of bottom edge, the Couette flow will only leave the oil at the half of the local clearance. When the upper land approaching, this amount of oil can easily get into the clearance between ring and liner. But with the extra contribution of pressure driven flow, the oil thickness after leaving trailing edge of bottom land may exceed the local clearance. In this TLOCR model, the combined residual oil thickness at trailing edge of bottom land serves as the oil supply boundary condition at leading edge of upper land. Compare the oil flux at boundary and oil supply, the model can get how much oil was scraped by upper land and accumulated. In figure 5-15 the results show for this specific case, oil scraping rate is that oil accumulates $0.06\mu m$ when the TLOCR travels every millimeter.

With the accurate, robust and efficient single specie two phase unsteady lubrication model, we can easily build special models for specific applications. This shows the model has the ability to help understanding the practical applications and improving the design.

5.3 Conclusion

A general deterministic hydrodynamic lubrication model was modified to study the interaction between a Twin Land Oil Control Ring (TLOCR) and a liner with cross-hatch liner finish. Efforts were made to customize the general model to simulate the particular sliding condition of TLOCR/liner interaction with proper boundary conditions. The results show that model is consistent, robust, and efficient. The lubricant mass conservation was justified and discussed. Then analysis was conducted on the lubricant transport between the deep grooves/valleys and plateau part of the surface to illustrate the importance of deep grooves in oil supply to the plateau part and hydrodynamic pressure generation. Furthermore, since the TLOCR land running surface is completely flat and parallel to the nominal liner axis, the liner finish micro
Figure 5-15: Oil scraping rate in TLOCR pocket
geometry is fully responsible for the hydrodynamic pressure rise, which was found to be sufficient to support significant portion of the total ring radial load. Moreover, continuity and size of asperity network were found to be critical to hydrodynamic pressure rise. The model is deemed to be adequate to study the liner finish effects in IC engines, which is one of the most critical areas in engine friction, wear, and oil consumption.
Chapter 6

Multi-phase Unsteady Oil Transport Model for Piston Rings Under Starved Conditions

In the power cylinder system, one can find all kinds of oil gas coexistence patterns listed in section 2.3.4 and more. The interaction of the oil control ring and a rough liner without consideration of the gas pressure effects has been studied by Chen and Li[1]. In this chapter, we focus on the modeling of oil transport with gas existence in top two rings and oil control ring with the effects of the gas pressures at trailing edge.

6.1 Boundary Conditions for the Lubrication of the Piston Top Two Rings in IC Engines

The lubrication condition of the top two rings, namely, the oil supply and the pressure boundary conditions, depends on the oil control ring behavior as well as the dynamics and gas flows of the ring pack. These boundary conditions play a predominant role in determining the lubrication of the top two rings.
6.1.1 Oil Supply Conditions for the Top Two Rings

In order to describe the oil supply in ring pack/liner system, the system was divided and named as the schema in Figure 6-1. The oil supply in power cylinder comes from the oil reservoir below the oil control ring. Then, the oil control ring, if properly designed, immediately scrape off the excessive oil and leaves an oil film whose thickness that is in the order of liner roughness level during a down stroke. While there is possibly oil flow from the piston lands to the liner locally and occasionally, our best knowledge is that the top two rings are primarily lubricated by the thin oil left by the oil control ring during down-strokes. In another word, the thin oil left by the OCR determines the friction behavior of the top two rings. Therefore, fresh oil needs to be supplied continuously in optimized amount to ensure the lubrication and minimize the oil consumption. The oil supply from the reservoir could be treated as fully supplied. The oil control ring controls the amount of oil supply and prevent excessive oil supply. In expansion stroke, the oil supply to top two rings comes from the oil pass oil control ring through linear/ring clearance, ring gap and ring groove. In compression stroke, the oil supply to the top two rings comes from the residual oil attached on liner that is left by top two rings left in the intake stroke. This small amount of residual oil is the oil actually generating lubrication. To control and improve the oil amount, the model of oil transport between liner and ring clearance is necessary.

6.1.2 Gas Pressure Conditions for the Top Two Rings

The top two rings are designed to seal the combustion gas. Unlike the oil control ring, the oil supply of top two rings comes from the residual oil attach on the liner and oil pass through the the grooves hold the rings and the gaps on the rings. Starving oil supply boundary is frequently observed in top two rings dynamics (Figure 6-2 ). The space not occupied by oil is full of gas. Unlike other mechanic components, the gas pressure at boundary could be much higher than cavitation pressure. Therefore,
Figure 6-1: Power cylinder and schematic of oil flow

Figure 6-2: Oil transport of top two rings
the gas at boundary does not only follow the movements of oil. It could play an active role to push oil around and change the oil transport between rings and liner. In the compression stroke, top ring seals the the gas in combustion chamber, the compressed gas can build up pressure of several bars. In the power stroke, the high pressure combustion gas pushing at the trailing edge of top ring. The pressurized gas penetrates in the space and changes the hydrodynamic lubrication and oil transport. For this kind of boundary condition with pressurized gas and starving oil supply, the simple processing introduced in section 3.2.5 in incapable of handle it. A two species multi phase unsteady lubrication model is necessary to study oil transport with the existence of pressurized gases.

6.2 Criteria and Assumptions

6.2.1 Objectives of Model

The multi phase flow is a very wide territory. It is impossible to fully discuss such a big topic in this thesis. In this work, we will focus on the topic related with oil transport in lubrication. In all the influence caused by gas oil interaction, there are two effects we are mostly interested in and want the model capable of handling.

First, when entrapped in the oil, the gas behaves very differently from the oil vapor. The oil vapor particle travels at sound speed through the clearance between surfaces which is in the order of micron. Without gas, the time needed to condense or vaporize between the liquid and vapor can be considered as negligible compared to the time scale for the lubrication. Therefore, a constant pressure is maintained in a vapor pocket. However, due to the limit of saturate pressure and diffusion rate of gas into oil, the extra gas introduced from environment can not be dissolved into the oil in the lubrication time scale. Instead, the entrapped gas bubble will be pressurized when compressed. The entrapped gas bubble will increase the local pressure above cavitation pressure while compressed. On the other hand, when gas bubble expands, the partial pressure of the gas in the gas bubble will decrease while partial pressure of
the oil vapor remains the same (determined by temperature, assuming a iso-thermal process). As a result, the gas bubble will maintain cavitation pressure due to the vapor comes out of oil. In summary, unlike the pure oil vapor bubbles, the bubble of gas and vapor mixture trapped in the liquid oil cannot disappear and the model should be able to keep track of the amount of the entrapped gas.

Second, a channel may exist to allow the gas to flow through the lubrication area from one boundary to the other. (may need a figure to explain these two patterns) When gas passing through an open channel inside full film region that connected two boundaries at different boundary pressure, the gas pressure will follow its own conservation law to build up a continuous pressure varying path. What is the pressure? Will this open channel expand, maintain or collapse? The cavitation model can not answer these questions. But the two species multi phase model should give reasonable answers.

6.2.2 Oil Gas Coexistence Pattern

As discussed in section 2.3.4, the oil gas coexistence pattern is a complex issue. There are mainly three types of oil-gases coexistence patterns and they all appear in the power cylinder systems. In this modeling work, based on the topic we are interested in and the experimental observations (Figure 6-3), we put the focus of modeling on oil fully attachment pattern. The gas bubble fully separated from surfaces mainly happens in large clearance between surface. The pattern of oil attached on one of the surfaces are mostly observed in lands between rings. Since the gas flow will change the free surface of oil, this pattern can be distinguished from the fully attachment pattern on LIF captured movie. The computational grid size is inherited from surface measurement. It is of at size of 1μm to 4μm. In a control volume at such a size, if there is oil and gas coexist, the model assume there is only single continuous interface between gas and oil. The shape and orientation of this interface is decided by the status of central control volume and neighboring control volumes.
6.3 Governing Equations

The basic idea of the model is to maintain the mass conservation of both oil and gas in each control volumes. Based on the full attachment criteria, in the gas-oil coexistence region, the governing equations for oil and gases are expressed respectively as follows, the oil phase follows the Reynolds equation 6.1.

$$\frac{\partial (\phi_o h)}{\partial t} + \frac{U}{2} \frac{\partial (\phi_o h)}{\partial x} = \frac{\partial}{\partial x} \left( \phi_o \frac{h^3}{12 \mu_o} \frac{\partial P}{\partial x} \right) + \frac{\partial}{\partial y} \left( \phi_o \frac{h^3}{12 \mu_o} \frac{\partial P}{\partial y} \right)$$  \hspace{1cm} (6.1)

$$\frac{\partial (\phi_m \rho_g h)}{\partial t} + \frac{U}{2} \frac{\partial (\phi_m \rho_g h)}{\partial x} = \frac{\partial}{\partial x} \left( \phi_m \rho_g \frac{h^3}{12 \mu_g} \frac{\partial P}{\partial x} \right) + \frac{\partial}{\partial y} \left( \phi_m \rho_g \frac{h^3}{12 \mu_g} \frac{\partial P}{\partial y} \right)$$  \hspace{1cm} (6.2)

Here the $\phi_m$ is volume fraction of gas vapor mixture, and $\phi_m + \phi_o = 1$. In full film or full gas zone, one of the two equation will disappear, the other will automatically degenerate to the proper governing equation. Because the gas phase and oil phase share the same pressure in the region of gas oil coexist, the fluid with higher dynamic
viscosity has smaller Poiseuille flow. Compare the right side of equation 6.2, 6.1. Considering that the dynamic viscosity of gas is three orders of magnitude less than that of the oil \((\mu_o \gg \mu_g)\), we know if there is any pressure variation in gas oil coexisting region, the pressure is mainly decided by gas phase.

Consider about the left sides of equation 6.2, 6.1 are in the same order, we know the gas phase can not generate significant pressure variation that can push oil phase. Therefore, we can neglect the Couette flow in equation 6.1. The governing equation of gas phase becomes

\[
\frac{\partial}{\partial x} \left( \phi_m \rho_g \frac{h^3}{12\mu_g} \frac{\partial P}{\partial x} \right) + \frac{\partial}{\partial y} \left( \phi_m \rho_g \frac{h^3}{12\mu_g} \frac{\partial P}{\partial y} \right) = \frac{\partial (\phi_m \rho_g h)}{\partial t} \tag{6.3}
\]

If we treat gas as ideal gas,

\[
P = \rho_g RT \tag{6.4}
\]

Plug equation 6.4 into equation 6.3, we have

\[
\frac{\partial}{\partial x} \left( \phi_m \frac{h^3}{12\mu_g} \frac{\partial P^2}{\partial x} \right) + \frac{\partial}{\partial y} \left( \phi_m \frac{h^3}{12\mu_g} \frac{\partial P^2}{\partial y} \right) = \frac{\partial (2\phi_m Ph)}{\partial t} \tag{6.5}
\]

When equation 6.5 reach the steady state of gas phase, it becomes a simple Laplace equation.

\[
\frac{\partial}{\partial x} \left( \phi_m \frac{h^3}{12\mu_g} \frac{\partial P^2}{\partial x} \right) + \frac{\partial}{\partial y} \left( \phi_m \frac{h^3}{12\mu_g} \frac{\partial P^2}{\partial y} \right) = 0 \tag{6.6}
\]

Put things altogether, we have the full picture of gas oil interaction which will decides our numerical iteration scheme.

1. Gas decides the inner and external boundary pressure of oil phase.

2. Oil transport follows the Reynolds equations and pressure boundary decide by
gas.

3. The void space not filled by oil is the transport path of gas phase.

4. When gas exists in control volume, the lower bound of oil pressure is gas pressure.

5. The lower bound of gas pressure is the cavitation pressure

### 6.3.1 Spatial Discretization Scheme

To discretize equation 6.1. We use the same control volume, nodes and interfaces geometry configuration showed in figure 3-2. Without further mention, we will neglect the subscript of oil in all terms.

The flow conductivity $K$ follows the same definition in section 3.2.2. Since the local oil volume fraction will influence the oil connected area on interface, at first glance, it seems more convenient and reasonable to couple the $\phi_o$ into flow conductivity at first glance. But with deeper observation, we will notice the oil volume fraction should be treated in upwind scheme instead of central scheme. This is the most important difference between single specie and two species model. In single specie model, wherever the Poiseuille occurs, the oil volume fraction is one. In two species model, the gas pressure could be higher than oil pressure in neighborhood. Therefore, the Poiseuille should be treated as a convection term instead of a diffusion term.

Integrate the Reynolds equation on the control volume, the equation becomes

$$\frac{\partial (\phi_p h_p)}{\partial t} \Delta x \Delta y = Q_e + Q_w + Q_n + Q_s$$

The oil flow rates at the interfaces of control volume are

$$Q_n = K_n \frac{P_N - P_p}{\Delta y} \Delta x \phi_n$$

$$Q_s = K_s \frac{P_S - P_p}{\Delta y} \Delta x \phi_s$$
\[
Q_e = K_e \frac{P_e - P_P}{\Delta x} \Delta y \phi_e - \phi_P U \frac{h_e}{2} \Delta y \\
Q_w = K_w \frac{P_W - P_P}{\Delta x} \Delta y \phi_w - \phi_W U \frac{h_w}{2} \Delta y
\] (6.8)

Define the logical variable to mark pressure gradient direction on interface.

\[
F_n = \begin{cases} 1 : P_N > P_P \\ 0 : P_N \leq P_P \end{cases}
\]

\[
F_s = \begin{cases} 1 : P_S > P_P \\ 0 : P_S \leq P_P \end{cases}
\]

\[
F_e = \begin{cases} 1 : P_E > P_P \\ 0 : P_E \leq P_P \end{cases}
\]

\[
F_w = \begin{cases} 1 : P_W > P_P \\ 0 : P_W \leq P_P \end{cases}
\] (6.9)

The oil volume fraction of Poiseuille flow terms in equation 6.8 are decided through upwind oil volume fraction.

\[
\phi_n = \phi_N F_n + \phi_P (1 - F_n)
\]

\[
\phi_s = \phi_S F_s + \phi_P (1 - F_s)
\]

\[
\phi_e = \phi_E F_e + \phi_P (1 - F_e)
\]

\[
\phi_w = \phi_W F_w + \phi_P (1 - F_w)
\] (6.10)

6.3.2 Assemble the Jacobian Matrix

Integrate the equation from \( t^i \) to \( t^{i+1} = t^i + \Delta t \), we have

\[
\omega \phi_P^{i+1} h_P^{i+1} - (1 - \omega) \phi_P^i h_P^i = \frac{\omega(Q_e + Q_w + Q_n + Q_s)^{i+1} \Delta t}{\Delta x \Delta y} \\
- \frac{(1 - \omega)(Q_e + Q_w + Q_n + Q_s)^i \Delta t}{\Delta x \Delta y}
\] (6.11)
When $\omega = 1$, it is an implicit time scheme. When $\omega = 0.5$, it is a Crank-Nicolson scheme.

Assemble all the discretized terms together, we get

$$f(\alpha^{i+1}) = g(\phi^{i+1}, P^{i+1}) - S_P = 0$$
$$g(\phi^{i+1}, P^{i+1}) = \omega \left[ \phi_p^{i+1} h_p^{i+1} - \frac{(Q_e + Q_w + Q_n + Q_s)^{i+1} \Delta t}{\Delta x \Delta y} \right]$$
$$S_P = (1 - \omega) \left[ \phi_h^{i+1} - \frac{(Q_e + Q_w + Q_n + Q_s)^i \Delta t}{\Delta x \Delta y} \right] \quad (6.12)$$

According to chain rule, we have

$$\frac{\partial f}{\partial \alpha^{i+1}} = \frac{\partial f}{\partial P^{i+1}} \frac{\partial P^{i+1}}{\partial \alpha^{i+1}} + \frac{\partial f}{\partial \phi^{i+1}} \frac{\partial \phi^{i+1}}{\partial \alpha^{i+1}} \quad (6.13)$$

From equation 6.12, we can get terms in equation 6.13

$$\frac{\partial f}{\partial P_N^{i+1}} = -\omega \phi_n \frac{K_n}{\Delta y^2} \Delta t$$
$$\frac{\partial f}{\partial P_S^{i+1}} = -\omega \phi_s \frac{K_s}{\Delta y^2} \Delta t$$
$$\frac{\partial f}{\partial P_E^{i+1}} = -\omega \phi_e \frac{K_e}{\Delta x^2} \Delta t$$
$$\frac{\partial f}{\partial P_W^{i+1}} = -\omega \phi_w \frac{K_w}{\Delta x^2} \Delta t$$
$$\frac{\partial f}{\partial P_P^{i+1}} = - \left( \frac{\partial f}{\partial P_N^{i+1}} + \frac{\partial f}{\partial P_S^{i+1}} + \frac{\partial f}{\partial P_E^{i+1}} + \frac{\partial f}{\partial P_W^{i+1}} \right)$$
$$\frac{\partial f}{\partial \phi_N^{i+1}} = -\omega F_n \frac{K_n}{\Delta y^2} (P_N^{i+1} - P_P^{i+1}) \Delta t$$
$$\frac{\partial f}{\partial \phi_S^{i+1}} = -\omega F_s \frac{K_s}{\Delta y^2} (P_S^{i+1} - P_P^{i+1}) \Delta t$$
$$\frac{\partial f}{\partial \phi_E^{i+1}} = -\omega F_e \frac{K_e}{\Delta x^2} (P_E^{i+1} - P_P^{i+1}) \Delta t$$
$$\frac{\partial f}{\partial \phi_W^{i+1}} = -\omega \Delta t \left[ U \frac{h_w}{2 \Delta x} + F_w \frac{K_w}{\Delta x^2} (P_W^{i+1} - P_P^{i+1}) \right]$$
\[
\frac{\partial f}{\partial q^{i+1}_P} = -\omega(1 - F_P)\Delta t \left[ \frac{K_n}{\Delta y^2} (P^{i+1}_N - P^{i+1}_P) + \frac{K_n}{\Delta y^2} (P^{i+1}_S - P^{i+1}_P) + \frac{K_n}{\Delta x^2} (P^{i+1}_W - P^{i+1}_P) \right] \\
+ \omega \left( h^{i+1}_P - U \Delta t \frac{h_e}{2\Delta x} \right)
\] (6.14)

From the definition of universal variable \( \alpha \), we have

\[
\frac{\partial P^{i+1}_S}{\partial \alpha^{i+1}_S} = g_S P_{ref}, \quad \frac{\partial P^{i+1}_N}{\partial \alpha^{i+1}_N} = g_N P_{ref} \\
\frac{\partial P^{i+1}_E}{\partial \alpha^{i+1}_E} = g_E P_{ref}, \quad \frac{\partial P^{i+1}_W}{\partial \alpha^{i+1}_W} = g_W P_{ref} \\
\frac{\partial P^{i+1}_P}{\partial \alpha^{i+1}_P} = g_P P_{ref} \\
\frac{\partial \phi^{i+1}_N}{\partial \alpha^{i+1}_N} = (1 - g_N)\Delta t, \quad \frac{\partial \phi^{i+1}_S}{\partial \alpha^{i+1}_S} = (1 - g_S)\Delta t \\
\frac{\partial \phi^{i+1}_E}{\partial \alpha^{i+1}_E} = (1 - g_E)\Delta t, \quad \frac{\partial \phi^{i+1}_W}{\partial \alpha^{i+1}_W} = (1 - g_W)\Delta t \\
\frac{\partial \phi^{i+1}_P}{\partial \alpha^{i+1}_P} = (1 - g_P)\Delta t
\] (6.15)

Plug 6.15 and 6.14 into 6.13, we have the all elements of Jacobian matrix.

\[
\frac{\partial f}{\partial \alpha^{i+1}_N} = -\omega g_N P_{ref} \frac{K_n}{\Delta y^2} \Delta t - \omega(1 - g_N)F_n \frac{K_n}{\Delta y^2} (P^{i+1}_N - P^{i+1}_P)\Delta t \\
\frac{\partial f}{\partial \alpha^{i+1}_S} = -\omega g_S P_{ref} \frac{K_n}{\Delta y^2} \Delta t - \omega(1 - g_S)F_s \frac{K_n}{\Delta y^2} (P^{i+1}_S - P^{i+1}_P)\Delta t \\
\frac{\partial f}{\partial \alpha^{i+1}_E} = -\omega g_E P_{ref} \frac{K_n}{\Delta x^2} \Delta t - \omega(1 - g_E)F_e \frac{K_s}{\Delta x^2} (P^{i+1}_E - P^{i+1}_P)\Delta t \\
\frac{\partial f}{\partial \alpha^{i+1}_W} = -\omega g_W P_{ref} \frac{K_n}{\Delta x^2} \Delta t - \omega(1 - g_W)\Delta t \left[ U \frac{h_w}{2\Delta x} - F_w \frac{K_w}{\Delta x^2} (P^{i+1}_W - P^{i+1}_P) \right] \\
\frac{\partial f}{\partial \alpha^{i+1}_P} = -\omega g_P P_{ref} \left( \frac{K_n}{\Delta y^2} + \frac{K_s}{\Delta y^2} + \frac{K_e}{\Delta x^2} + \frac{K_w}{\Delta x^2} \right) \Delta t \\
+ \omega(1 - g_P) \left[ h_p - U \Delta t \frac{h_e}{2\Delta x} - (1 - F_P) \frac{K_n}{\Delta y^2} (P^{i+1}_N - P^{i+1}_P) \Delta t \right]
\]
\[-(1 - F_p) \frac{K_s}{\Delta y^2} (P_S^{i+1} - P_P^{i+1}) \Delta t - (1 - F_p) \frac{K_e}{\Delta x^2} (P_E^{i+1} - P_P^{i+1}) \Delta t \]
\[-(1 - F_p) \frac{K_w}{\Delta x^2} (P_W^{i+1} - P_P^{i+1}) \Delta t \]

Without extra superscripts, all numbers in equation 6.16 are at \( t^{i+1} \).

### 6.4 Iteration Scheme of Two Species Model

The flow chart of the model is listed in Figure 6-4. The cycle starts from the oil phase calculation. Most steps of oil phase iteration is the same as single specie model except the local low bound of pressure is not a constant anymore. Rather, the lower bound of the pressure is either the oil vapor pressure or the local gas pressure, whichever one is greater.

Once the oil phase calculation converged, a gas phase calculation is carried out to get gas distribution. The gas phase calculation has a distinguish difference with oil phase calculation. The oil phase calculation has two steps, pressure calculation and oil volume ratio calculation. The variation of pressure happens in much smaller time scale than the time scale of oil volume ratio variation.

\[ V_{sound} \gg U \quad , \quad \frac{\Delta x}{V_{sound}} \ll \frac{\Delta x}{U} \]  

(6.17)

Therefore, the calculation of oil pressure does not involve time derivatives. It is looking for a steady solution of pressure distribution in full film zone at that moment. The calculation of oil volume ratio involves time derivatives.

Since sound speed of gas is much larger than oil transport speed. The pressure variation in gas phase takes negligible time. The dynamic viscosity of gas is much smaller than oil. Therefore the gas transport also take negligible time compare to the oil transport. Considering about these two factors, we need not to calculate gas pressure and gas mass separately. The gas pressure and mass of gas in control volume are calculated simultaneously.
Before start the gas phase calculation, the pressure of gas-vapor mixture should be updated according to the new volume fraction of mixture from oil phase calculation. The volume fraction of mixture $\phi_m$ does not vary in gas phase calculation. When we discretizing equation 6.5, $\phi_m$ is combined into flow conductivity coefficient.

$$K = \frac{\phi_m h^3}{12 \mu}$$  \hfill (6.18)

### 6.5 Applications to Oil Transport of Top Two Rings

The multi phase oil transport model can handle the applications that pressurized gas coexists with oil and plays an important role. The lubrication of the top two rings in the power cylinder system presents one application.

The oil supplies of top two rings come from the residual oil attached on liner after oil control ring scraped by. When oil supply is insufficient at boundary, the pressurized gas is possible to reach the minimum clearances area between ring and liner. The existence of gas could decrease the hydrodynamic force generated by ring sliding. It could also pump more oil in and generated more hydro dynamic force. The force balance on the ring decides whether ring will collapse or be pressed onto liner and maintain sealing. The coupling between gas transport and oil transport is unclear and has significant yet unknown influence. The multi phase oil transport model is such a tool designed to do quantitative analysis.

#### 6.5.1 Ring Geometry

The working environments of top two rings are shown in Figure 6-5. The surface patch we consider in model is relative small compare to the whole liner. Therefore, it is convenient to use a local coordinate in modeling. The axial direction is negative $x$ direction in the local coordinate. Circumferential direction is marked as $y$ direction.
Figure 6-4: Flow chart of two species model
in the local coordinate. The direction normal to liner point to ring is $z$ direction in the local coordinate. The patch size at $y$ direction is small in applications. Therefore the liner curvature does not play a significant role.

Each type of engine has its own ring pack design. For the work in this thesis, we focus on the region around the location with the minimum nominal clearance between the ring and the liner and the clearance profile of this region can be approximated as a parabolic without loss of generality.

$$h_r = ax^2$$ (6.19)

Each ring profile is defined by its $a$. In following applications, if without specific note, $a$ is 70 for top ring, 50 for second ring. To isolate the liner finish effect, we assume that ring is perfectly smooth. However, the liner surface is full of roughness and horning grooves. Different liner surface measurements are used in following ap-
The actual clearance between ring and liner is the combination of ring profile $h_r$, linear roughness $h_l$ and the average gap $h_0$ as in Figure 6.20.

$$ h = h_r - h_l + h_0 $$

The gap is a result of force balance on ring. It changes according to the total support generated between ring and liner and many other factors. In this work, the force balance of ring is not in our consideration. The gas is usually a known input from other models.

The barrel shape of ring profile forms a minimum clearance locus between two ring edges. This minimum clearance locus divides ring to two zones. The zone at the side of ring sliding direction is converge zone. From the view of coordinate fixed on liner, the local clearance is decreasing with ring sliding movement. The ring profile generates hydrodynamic force in the converge zone. The zone at other side
of minimum clearance locus is the diverge zone. The clearance increases with ring sliding movements in this zone. The ring edge locates at the side of converge zone is leading edge. The ring edge locates at the side of diverge zone is trailing edge.

The model was first applied to applications that ring slides on smooth liner surface. These applications have the advantage that the influence of ring profile is separate from the influence of liner roughness. Further more, we can compare the results of the multi phase model with Tian’s model which is widely used in industry applications[29]. The differences between models can show us the influence of pressurized gas, the directions where we should improve the multi phase model at.

6.5.2 Model Inputs

The power cylinder is a complex system includes rings, piston and liner. The oil and gas transport power cylinder follow multiple different paths and mechanisms. The transport of oil and gas influences the dynamics of power cylinder system. The oil and gas transport between ring and liner is an important part of this system. However it is extremely difficult and costly to get the necessary data from power cylinder system. The daunting number of designs and running parameters drives people to develop efficient models to analysis the performance of ring packs.

The working cycle of piston has four different strokes according to crank angle. The strokes are intake, compression, expansion and exhaust. The piston and ring pack move upward in compression and exhaust strokes, downward in intake and expansion strokes. There are several critical parameters decides the oil and gas transport between ring and liner. These parameters are pressures at leading edge \( P_l \), pressure at trailing edge \( P_t \), ring sliding speed \( v \) and ring gap \( h_0 \). These critical parameters vary during engine strokes. It is very hard to get these parameters through experiment. In the work of this thesis, Tian’s ring pack model provides these critical parameters.

In Figure 6-7, we list the pressure inputs for top two rings. The location of pressures are defined as in Figure 6-2. The actual pressure at leading edge \( P_l \) and pressure at trailing edge \( P_t \) come from \( P1, P2, P3 \) according to the crank angle and the piston ring. Other parameters such as oil viscosity \( \mu \) and oil supply amount \( \rho_{oil} \)
are subject to similar variation. Before we apply the multi phase oil transport model to specific ring and crank angle, we drive the required parameters from the results of full ring pack model.

6.5.3 Sample Results

The model was first applied to observe the influence of pressurized gas boundary to oil film build up between piston ring and liner. The clearance between top two rings and liner varies from the ring profile scale (0.1mm) to liner roughness scale (0.1/\text{mum}) along the ring width. If the residual oil on liner is sufficient, at certain spot, the oil will fill all the clearance between ring and liner. Once the full film zone formed, hydro dynamic pressure will rise in the full film zone and change the force balance on rings. The details of of how the full film zone is very interesting and important topic. We want to know the width of full film zone with different

- thickness of the residual oil
Figure 6-8: Sketch of pressure averaged in circumferential direction

- minimum clearance between the ring and the liner

- pressure difference between leading edge and trailing edge

- ring sliding speed

We are not going to discuss the details of the influences introduced by parameters listed above in this thesis. But we will show the multi phase model is capable of answering such kind of questions with simple examples. The results varies in both circumferential and axial direction. Because the length scale at circumferential direction is much larger than the axial direction, we are more interested at the variation of the total pressure, gas pressure and oil volume fraction along the ring width. The axial direction is $x$. If without specific notes, the positive direction of $x$ is always pointing from leading edge to trailing edge in following applications. The ring is sliding above the liner as shown in Figure 6-8. All the quantities are averaged at circumferential direction.
Oil Development at Crank Angle 45

The first condition is the top ring sliding on a smooth liner at crank angle 45. It is the expansion stroke of engine. The piston and ring move from top to the bottom at the speed of 7.12 m/s. The effective dynamic viscosity after considering the shear-thinning effect of oil is 0.004 Pa·s. The minimum clearance is 0.29 μm, locates at $x = 0$. The pressure at leading edge is 5.54 bar. The pressure at trailing edge is 11.13 bar. The residual oil at liner is 0.167 μm. Since we only calculate a short time period around specific crank angle, we set the oil thickness the same at the leading edge and trailing edge. The oil thickness has been converted to the local oil volume fraction before used as input. For a ring with profile that $a = 70$, the clearance at boundary is several microns, which is much larger than the local residual oil thickness. The real contact pattern at boundary is residual oil attached on liner but separated from the ring. Only when the clearance reach a threshold number, the oil reattachment will happen. The oil reattachment itself is a very important and difficult topic. It is not the work of this thesis. In the multi phase model, we assumed the oil is fully attached with both surfaces all the time. The pressure gradient at boundary is zero since gas can move freely between ring and liner. The oil supply flow is composed only by Couette flow. Therefore, to ensure the oil supply is the same. The local oil volume fraction should be twice as the local residual thickness.

In Figure 6-9, we show the oil development at crank angle 45. At the initial stage, there is no oil between the ring and liner. The gas can travel from trailing edge to leading edge through the space between ring and liner. There is also no hydrodynamic pressure. The total pressure is equivalent to the gas pressure. The gas pressure is solved from the transport equation of gas phase. Therefore, the gas pressure is influenced by the profile of the clearance between ring and the liner (Figure 6-9a).

With the ring slide on liner, more and more oil comes into the clearance between ring and liner. The oil occupies part of the clearance between ring and liner and
(a) Initial stage

(b) before oil filling the clearance between ring and liner
(c) oil fully flooded the clearance between ring and liner

(d) end stage

Figure 6-9: Oil development at crank angle 45

116
narrow the channel that gas pass through. Therefore the transition of pressure from trailing edge pressure to leading edge pressure happens in a shorter length in axial direction (Figure 6-9b).

Finally, the oil fully filled the clearance between the ring and the liner at certain axial location. This full film region blocks the gas transport path that connects trailing edge with leading edge. Hydro dynamic pressure start to building up in the full film region. The gas pressure in full film region is zero since there is no gas there. The gas pressure in calculation domain is almost the same as the boundary gas pressure where it connects to (Figure 6-9c).

The pressure difference between trailing edge and leading edge pumps part of oil from trailing side to leading side. This effect enhances the oil sealing at the minimum clearance between the ring and the liner. The full film band around middle becomes wider and reach steady status at last (Figure 6-9d).

**Oil Development at Crank Angle -45**

The second condition is the top ring sliding on a smooth liner at crank angle -45. It is the compression stroke of engine. The piston and ring move from bottom to the top at the speed of 7.08m/s. The dynamic viscosity of oil is 0.004Pa·s. The minimum clearance is 0.30μm, locates at $x = 0$. The pressure at leading edge is 2.86 bar. The pressure at trailing edge is 1.18 bar. The residual oil at liner is 0.167μm. The parameters at crank angle -45 are close to the parameters at crank angle 45 except the boundary gas pressure. At crank angle 45, the high trailing edge pressure is caused by combustion gas. The high leading edge pressure at crank angle -45 is caused by the compressed air. The different mechanisms make the pressure difference between boundaries at crank angle 45 is much larger than the pressure difference at crank angle -45.

In Figure 6-10, we listed the oil development at crank angle -45. It follows similar development as at crank angle 45. However, the gas pressure gradient points from trailing edge to leading edge at crank 45. The pressurized gas will pump oil away
Gas pressure, \( t = 0.56497 \) micro second

Total pressure, \( t = 0.56497 \) micro second

Oil volume fraction, \( t = 0.56497 \) micro second

(a) Initial stage

Gas pressure, \( t = 96.0452 \) micro second

Total pressure, \( t = 96.0452 \) micro second

Oil volume fraction, \( t = 96.0452 \) micro second

(b) before oil filling the clearance between ring and liner
Gas pressure, $t = 101.6949$ micro second

Total pressure, $t = 101.6949$ micro second

Oil volume fraction, $t = 101.6949$ micro second

(c) oil fully flooded the clearance between ring and liner

Gas pressure, $t = 169.4915$ micro second

Total pressure, $t = 169.4915$ micro second

Oil volume fraction, $t = 169.4915$ micro second

(d) end stage

Figure 6-10: Oil development at crank angle -45
from the minimum clearance towards the leading edge. Mean while, the ring movement keeps accumulating oil at the leading side. This two oil transport mechanisms competed with each other and enhanced the oil development at leading side. At crank angle $-45$, the pressurized gas keeps pumping oil out of leading side toward trailing side. Therefore, at the same time from the initial condition ($t = 101.12\mu s$), the oil has fully flooded the minimum clearance at crank angle 45 (Figure 6-9c) while there is still a partial path for gas to transport at crank angle $-45$ (Figure 6-10c). The multi phase model can not only handle the pressurized boundary condition but also capture the effects of gas pressure gradient correctly.

**Compare with the Original Reynolds Equation Model**

Before we developed the multi phase model, we can use original Reynolds equation to solve the problem of ring sliding on smooth liner. The gas pressure is used as the pressure boundary condition for the full film region. The location and width of full film region comes from mass conservation consideration[29]. The minimum clearance between the ring and the liner is decided by the force balance on ring. Therefore, the minimum clearances of rings with different parabolic coefficient are different while the sliding speeds and oil supplies are the same. In figure 6-11, we showed the velocity and minimum clearance between the ring and the liner at different crank angles.

We will pick crank angle 45 to compare the Reynolds equation model and multi phase model. It is the expansion stroke of engine. The piston and ring move from top to the bottom at the speed of $7.12 m/s$. The dynamic viscosity of oil is $0.004 Pa\cdot s$. The minimum clearances are $0.3144 \mu m$ for $a = 10$, $0.3023 \mu m$ for $a = 35$ and $0.2936 \mu m$ for $a = 70$. The minimum clearances locate at $x = 0$. The pressure at leading edge is 5.54 bar. The pressure at trailing edge is 11.13 bar. The residual oil at liner is $0.167 \mu m$.

In Figure 6-12, we showed pressure of rings with different profiles ($a = 10, 35, 70$) slide on smooth liner at crank angle 45. Both models take the same inputs, and the results are very close to each other.
It is not surprise to see both models give the same result. Because for smooth liner, once the oil accumulated around the minimum clearance, the full film zone blocks the gas transport path. At this situation, as long as we can track the boundary between gas and full film correctly. The Reynolds equation model can give the right pressure in full film zone.

The multi phase model solves both oil transport equation and gas transport equation. There is no special requirements for the location of the boundary between full film zone and partial film zone. The boundaries between full film and partial film are natural output of the model. Therefore, the multi phase model can handle much more complex geometry and automatically give the location and size of full film region instead of take them as prescient information. But it is important to see the Reynolds equation model and multi phase model give the same result for simple oil distribution case.
Figure 6-12: Compare multi phase model and Reynolds equation model at crank angle 45
**Influence of Surface Roughness and Surface Pattern**

In real case, there are plenty of surface roughness put on liner intentionally. In Figure 6-13a, we showed the one of clearances between the ring and rough liner. The darker region means smaller clearance. The brighter liner is the large clearance caused by horning grooves. The surface roughness and the surface pattern leave plenty of spaces for gas to penetrate in. They also induce cavitations and separate the full film zones.

We showed the oil distribution of a ring with $a = 30$ sliding on rough liner (Figure 6-13b. The bright zones around $x = 0$ are filled by the oil. The dark grooves across the bright zones are the partially filled horning grooves. The horning grooves provide connected paths for gas to penetrate and influence the pressure and oil distribution around the horning grooves. These phenomenon are important and are out of the ability of Reynolds equation model.

When the ring is sliding on the rough liner, the surface features under the ring...
change all the time. In Figure 6-14, we showed the pressure averaged at circumferential direction at 158.54 micro seconds after the ring sliding on the rough liner from no oil initial condition. This calculation time is long enough for the oil flow in whole filed reaches steady status. Nevertheless, the local pressure and flow pattern are still changing with the change of surface roughness and pattern. Therefore, even we averaged the pressure at circumferential direction, there are still pressure variations along the axial direction.

To study the influence of surface roughness and pattern, we conducted calculations for the top ring at 45 degrees after the TDC of the expansion stroke. Except the liner profile, all the input parameters are the same as we used in section 6.5.3. To remove the time variation caused by roughness, we averaged all the results in 112 micro seconds after flow reached steady status. These values are compared with the results of the same ring sliding on smooth liner.

Comparing the results of rough liner and smooth liner (Figure 6-15), the first thing we noticed is the gas penetration. Unlike the smooth liner, the rough liner has plenty of surface features to let gas penetrate in (Figure 6-13b). Therefore the partial pressure of gas is not zero around the minimum clearance. Though most of the area are filled by oil, gas can still transport though the grooves and roughness. The partial pressure around minimum clearance is neither a linear profile nor a hydrostatic distribution. It is because the gas pressure gradient depends on the effective cross section area allow gas pass.

The pressurized gas has two major influence on oil transport. At first, it could pump oil into small clearance location and enhance the sealing. But the existence of gas could also decrease the ability of mixture generating hydro dynamic force. Compare the results of the smooth liner and the rough liner, we observed both effects. In all three rings with different parabolic coefficients \((a = 10, 35, 70)\), we noticed that because of gas penetration, lots of area get oil supply pumped by pressurized gas. The area that generated hydro pressure are expanded in all three cases (Figure 6-15a, 6-15b, 6-15c). At the same time, the existence of gas decreasing the ability of
Figure 6-14: Pressure and oil volume fraction averaged at circumferential direction
mixture generating hydro dynamic force. For a flat ring profile \((a = 10)\), less gas can penetrate in. Therefore the peak pressure is higher than the peak pressure of smooth liner (Figure 6-15a). For rings with larger \(a\), more gas can penetrate in. Therefore the peak pressure is lower than the peak pressure of smooth liner (Figure 6-15b, 6-15c). The final effect depends on the balance of gas penetration’s two major influences.

The gas penetration changed the mechanism of hydro dynamic pressure generation. For smooth liner, the hydro pressure is mainly generated by the ring profile. Therefore the location of peak pressure is offset from minimum clearance noticeably. For rough liner, the hydro pressure is generated by all three scales of surface features. The local surface features that generated inter asperities hydro pressure are evenly distributed at both sides of minimum clearance. Therefore the location of peak pressure is closer to minimum clearance in rough liner cases than in smooth liner cases. This will change the center of hydro force, hence influences the force balance of piston rings.

The influence of liner roughness depends on the clearance between piston ring and liner. The liner roughness generates significant inter asperities hydro pressure when the clearance between piston ring and liner is comparable to standard deviation of roughness. The effect of liner roughness is negligible when the clearance between piston ring and liner is much larger than standard deviation of roughness. Therefore, a flatter ring profile builds up hydro pressure at a wider range. This effect is important to the two rings because the worn effect will make the ring profile getting flatter than original profile.

**Gas Pressure Influences on the Oil Control Ring Lubrication**

As shown in Figure 6-7, the third land pressure could be elevated to above 1 bar especially during the expansion stroke and this will influence the lubrication of the upper land. The single-specie TLOCR model presented earlier is not applicable for this elevated gas pressure effect. In this section, we will use this multi-phase model to examine the effects of the gas. The two-piece oil control ring has flat profile. When
Figure 6-15: Compare multi phase model and Reynolds equation model at crank angle 45
the flat and smooth oil control ring slide above a rough liner, the major mechanism of generating hydrodynamic pressure is the micro cavitation phenomenon happens in among the roughness on liner surface. We applied the multi phase model on a two pieces oil control ring.

Figure 6-16 shows the total pressure (averaged along circumferential direction). The ring width is 0.2mm. In both cases, the the ring has higher pressure at the trailing edge. Oil control ring has plenty oil supply at leading edge. Therefor the oil volume fraction is 1 at leading edge. To investigate the penetration of pressurized gas, we set the oil volume fraction at trailing edge to 0.4. This setting creates an entrance for pressurized gas. There are two pressure differences in all the cases. The lower pressure difference is 1bar. The higher pressure difference is 4bar. For each pressure difference, we tested the minimum clearance between the ring and the liner equals to 2, 3, 4 and 4 times of the \( \sigma \) of liner roughness. We want to see the details of pressure variation with different minimum clearance. The huge pressure of \( h_{min} = 2\sigma \) dwarfs the pressure in other cases. Therefore, the pressure of \( h_{min} = 2\sigma \) is not shown in Figure 6-16.
The results show the pressure generation is increasing with the minimum clearance decreasing. The pressurized gas lifts the hydrodynamic pressure within ring width. Because the gas will decrease the mixture’s ability to generate hydrodynamic pressure. The more gas penetrated into the ring width, the pressure within ring width is more close to the boundary gas pressure. The results show with larger pressure difference and higher minimum clearance, the gas can penetrate into ring width deeper. For the cases with 4bar trailing edge pressure difference, the gas almost penetrated the whole ring width for \( h_{\text{min}} = 5\sigma \) case, while there is still plenty of hydrodynamic pressure in the \( h_{\text{min}} = 3\sigma \) case. When the gas penetrated deeper and deeper into the ring width, the width of full film region blocks gas transport become narrower and narrower. Therefore the pressure gradient inside it caused by gas pressure difference becomes huge. At last the pressurized gas will break the thin film in negligible time and connect the gas at both sides. All these happen in the time scale gets smaller and smaller. Current model cannot handle this situation. Therefore, in current multi phase model, gas can penetrate into ring width and make the full film with as thin as several grids. But it cannot catch up the phenomenon that the thin full film breaks because that requires a much smaller time step. This could be a very important and
interesting direction to improve current model.

In Figure 6-17, we showed the total pressure in ring width include the $h_{min} = 2\sigma$ case. Unlike the average Reynolds equation model that will give zero hydrodynamic pressure for flat ring profile, the flat ring can generate significant hydrodynamic pressure through inter roughness cavitation. The pressurized gas enhances the hydrodynamic pressure generation when the gas penetration is not so deep.

In Figure 6-18, we showed the oil volume fraction distribution in ring width. The minimum clearance is $4\sigma$. The ring setting blocks gas penetration for pressure difference 1bar while the gas can penetrate through for pressure difference 4bar. The results show the gas penetration does not mean the gas excludes all oil. The oil volume fraction is higher than 0.5 even for the gas penetrated case. As long as there is enough space that connected with the boundary, the penetrated gas can get supply from boundary and keep penetrating.
6.6 Conclusion

The single phase lubrication model based on Reynolds equation with consideration of cavitation is not able to describe the type of sliding lubrication under starved oil supply condition with the surrounding gas pressurized, such as the lubrication of the upper piston rings and the liner. Recognizing this limitation, we developed a multi phase deterministic lubrication model and applied it to the lubrication of the top two rings with consideration of the micro-geometry of the liner finish.

As the dynamic viscosity of the gases is three order of magnitude less than the oil, the gases redistribute the mass much faster than the oil. The model takes full advantage of this difference in the transport rate between the gases and the oil. More specifically, we model the liquid oil transport with the fully unsteady Reynolds equation and solve the gas transport and its pressure distribution based on steady state hydrostatic flow solution.

The multi phase deterministic hydrodynamic lubrication model has been developed to study the influence of pressurized gas to ring pack and liner system. Based on the criteria that the time scale of density variation in gas phase is much smaller than in oil phase, the model treats oil phase as a fully unsteady mass balance equation and treats gas phase as a steady equation of pressure distribution. As a result, the multi phase can keep mass balance in oil phase and calculate pressure variation in gas phase simultaneously. The model considers the oil-gas coexist pattern as that both phases are fully attached with surfaces in relative motions. The mass fluxes at the gas-oil interface are treated differently at oil phase calculation and gas phase calculation. At oil phase calculation, the interface allows both Couette flow and Poiseuille flow of oil pass, the gas pressure decides the boundary pressure at the interface. At gas phase calculation, the location and the movements of interface between oil and gas serves as a moving boundary of the channel that gas flows by or the moving boundary of the bubble that gas is confined in.

The multi phase model was applied to study the oil transport in ring pack and liner system. The results show that the pressurized gas will move the center of hydro force
closer to the geometry center of ring profile. The pressurized gas also pushes residual oil in the roughness in plateau area of liner and helps generating inter asperities hydro pressure. The same effects of enhancing inter asperities hydro pressure were observed when we apply the multi phase model to study flat twin land oil control ring. Because of the narrow land width of twin land oil control ring, we also observed the gas penetrates through the most of the land width of oil control ring at large boundary pressure difference. Though the model can not simulate gas penetration breaks the last thin oil film and connect the leading edge and trailing edge, it still provides valuable information about when the pressurized gas can fail the sealing of piston ring.

This multi phase model can give quantitative results about how pressurized gas changes the oil transport and pressure distribution. Upon further investigation, the model can help establish more in-depth understanding of the lubrication and friction between the rings, particularly, the top two rings, and the liner with the consideration of the micro geometry of the liner finish. This information is critical to design of many mechanical components and is unprocurable to traditional single phase oil transport models. Though the multi phase model’s criteria and results are lack of substantial experiments supports, it still opens a field that is important and challenging.
Chapter 7

Conclusions

7.1 The Lubrication Models Developed in this Thesis Work

In the study of mechanical components that are lubricated by oil film, the pressure distribution is naturally the first and most important quantity that researchers want to achieve. Because the pressure variation transports quickly in the oil, for the full film region, the pressure distribution is solved essentially as a steady state result for given geometry and its variation rate when compressibility of oil is negligible. The cavitation phenomenon changes this simple scenario. When local pressure is lower than cavitation pressure, partial film develops and the oil fraction of the partial film area needs to be resolved while the pressure is a known constant. The dynamics of the interface between oil zone and cavitations becomes the center of modeling to maintain the mass conservation and cavitation criteria simultaneously.

The unsteady single specie mass-conserved model developed in this work focuses on the robustness and efficiency of oil transport modeling. The model considers the unsteady effect of oil volume ratio variation. It uses the time variation trend of both the volume ratio and pressure to judge whether the control volume belongs to cavitation region or full film region. With this physics-based algorithm, the model avoids the instability caused by switching status between cavitation and full film in a
control volume. Furthermore, we achieve robustness of the model without sacrificing efficiency. Therefore its efficiency is greatly increased. In all different kinds of surface configurations, it converges stably and maintains mass conservation correctly. The unsteady single specie mass-conserved model was applied in different applications on metal face seal and piston ring pack liner system. The results proved its viability at different scales.

The existence of pressurized gases further complicate the problem. For normal oil-cavitation interface, pressure gradient always drives the oil from the full film region to the cavitation region. On the other hand, the pressurized gas could push the oil from the partial film region with the pressurized gas towards the full film region. Therefore, the dynamics of interface between full film zone and partial film zone is much more complex and needs special treatments.

The multi-phase unsteady lubrication model developed in this work assumes that the oil is fully attached to both solid surfaces and there exists a continuous oil-gas interface in control volumes. The model considers the transport mechanism of both gas and oil. It keeps mass conservation of the oil at all situations to determine the pressure distribution of the oil film. It uses gas pressure as the boundary pressure at the gas-oil interface for the calculation of the oil pressure in the full film region. In the region with partial oil film, the model calculates two quantities. One is the oil volume fraction; the other is the gas pressure distribution. The calculation of oil volume faction follows the similar method of unsteady single phase mass conservation model. The difference is the Poiseuille flow caused by gas pressure variation is considered. Though in most of oil partial film region it is negligible, it is critical to decide the dynamics of interface between oil full film and oil partial film. The pressure distribution in oil partial film is decided by gas pressure. When the multi phase unsteady model is calculating the pressure distribution in gas phase, it treat the interface between oil full film region and oil partial film region as a moving boundary with zero gas flow rate. The multi phase model keeps mass conservation of enclosed gas bubble until the volume fraction of gas is smaller than preset critical small number. When gas volume fraction is smaller than preset number, the gas bubble will dissolve into oil.
The model does not track the transport of dissolved gas. Therefore, the dissolved gas will never come back.

Overall, this work shows that with proper scaling, problems of the hydrodynamic lubrication with or without consideration of the pressurized gases can be solved in a robust and efficient manner no matter how complex is the clearance geometry.

### 7.2 Future Work

The multi phase model developed in this work is based on the criteria that oil is fully attached with both surfaces in the partial film zone and there is a clear interface between full film zone and partial film zone. These criteria come from the observation of laser induced fluorescence (LIF) imaging system and many other previous observations. The film thickness observed by LIF imaging system is from microns to hundred of microns. Consider about the surface tension force tends to form the smallest interface area, it is natural to infer that the same oil contact pattern is more stable at smaller inter asperities scale. Further observations about the oil contact pattern in inter asperities scale could clarify this problem and lay down a solid foundation for modeling work in this area.

No need to mention, such kind of experiment is challenging. A possible way is using models based on physics such as lattice Boltzmann method to derive the oil contact pattern at sub micron scale. The lattice Boltzmann method is based on statistical physics. It can model the surface tension purely based on inter molecular force between surface and oil. The cavitation phenomenon could also be modeled as phase transitions by inter molecular force between oil and vapor. And the surface tension effects and phase transition solely depends on the surface tension coefficient and oil’s equation of status ([23],[24]). There is no preset model criteria required. The difficulty of this approach lies mainly at two directions. The first, the lattice Boltzmann method use cubic lattices to cover the computational domain. In lubrication problems, the scale at film thickness direction is much smaller than other two directions. Therefore, to achieve enough resolution at film thickness direction requires a large
amount of lattices. But this difficulty is not formidable once we consider about the parallelized algorithm and low cost in-lattice operations of lattice Boltzmann method. The other difficulty is the pressure difference in lubrication problem could easily be as large as tens to hundreds of cavitation pressure. The compressible effects in lattice Boltzmann method will be exaggerated in such kind of situations. To overcome this difficulty, new oil’s equation of status is necessary. This equation of status should cover large pressure difference yet only cause small density variation.

Another possible oil gas coexistence pattern is the oil and gases are evenly mixed. The oil filled with tiny gas bubbles can be treated as a special kind of fluid with varying compressibility and viscosity. The compressibility and viscosity of such a mixture are functions of the volume fractions of oil and gas. The multi phase model based on these criteria could be solved by single phase lubrication model coupled with a transport equation of gas in oil. A.O.Lebeck developed a foamy oil model to predict performance of zero leakage metal face seal[15]. He also reported the instability caused by nonlinear prosperities of foamy oil. The foamy oil model is not applicable with open system such as liner piston ring system, where the interaction time is small and mixing mechanism is not clearly present. However, further researches at this direction are still important and necessary for other mechanical applications.
Bibliography


