Modeling the Lubrication of the Piston Ring Pack in Internal Combustion Engines Using the Deterministic Method

by

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Abstract

Piston ring packs are used in internal combustion engines to seal both the high pressure gas in the combustion chamber and the lubricant oil in the crank case. The interaction between the piston ring pack and the cylinder bore contributes substantially to the total friction power loss for IC engines. The aim of this thesis work is to advance the understanding of the ring liner lubrication through numerical modeling.

A twin-land oil control ring lubrication model and a top two-ring lubrication model are developed based on a deterministic approach. The models take into consideration the effect of both the liner finish micro geometry and the ring face macro profile. The liner finish effect is evaluated on a 3D deterministically measured liner finish patch, with fully-flooded oil supply condition to the oil control rings and starved oil supply condition to the top two rings. Correlations based on deterministic calculations and proper scaling are developed to connect the average hydrodynamic pressure and friction to the critical geometrical parameters and operating parameters so that cycle evaluation of the ring lubrication can be performed in an efficient manner. The models can be used for ring pack friction prediction, and ring pack/liner design optimization based on the trade-off of friction power loss and oil consumption.

To provide further insights to the effect of liner finish, a wear model is then developed to simulate the liner surface geometry evolution during the break-in/wear process. The model is based on the idea of simulated repetitive grinding on the plateau part of the liner finish using a random grinder. The model successfully captures the statistic topological features of the worn liner roughness. Combining the piston ring pack model and the liner finish wear model, one can potentially predict the long term ring pack friction loss.

Finally the thesis covers the experimental validation of the twin-land oil control ring model using floating liner engine friction measurements. The modeled ring friction is compared with the experimental measurement under different ring designs and liner finishes. The result shows that the model in general successfully predicts the friction force of the twin-land oil control ring/liner pair.

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1. Introduction

1.1 Project Motivations [1]

In the modern world, internal combustion (IC) engines are widely used in the area of transportation. The use of IC engines has been a major fossil fuel consumer, as well as an important air pollution contributor. As a result, two of the most important topics of the IC engine research are improving the efficiency of energy use and emission control.

1.1.1 Piston Ring Pack Friction in an Internal Combustion Engine

In a typical running cycle, mechanical friction loss accounts for around 10% of the total energy in the fuel for a diesel engine, as illustrated in Fig.1.1 [2]. Among the mechanical friction loss, piston ring pack is responsible for about 20%. Approximately 2–3% of the diesel fuel energy is lost through the frictional interaction between the piston ring pack and liner finish.

![Fig.1.1 Breakdown of Total Diesel Engine Energy, Mechanical Friction and Ring Pack Friction [2]](image)

Therefore there is a large potential in improving the engine efficiency by reducing the friction between the piston ring pack and the engine cylinder bore surface. Reducing ring pack friction also reduces the thermal load on the cooling system of the engine by reducing the amount of heat generated in the power cylinder. The challenge in finding the strategy for lowering ring pack friction is not to bring adverse effects in oil consumption, blow-by, excessive wear, and failure. And this requires a deep
understanding of the interaction between the solid surfaces of the ring face and cylinder bore surface with the existence of lubricant oil in between.

1.1.2 Control of Oil Consumption [4]

Oil consumption from the piston-ring-liner system contributes significantly to total engine oil consumption [3] [4]. Engine oil consumption is recognized to be a significant source of automotive engine emissions in modern engines. Unburned or partially burned oil in the exhaust gases contributes directly to hydrocarbon and particulate emissions [4] [5] [6]. Moreover, chemical compounds in oil additives can poison exhaust gas treatment devices and can severely reduce their conversion efficiency [4] [7] [8]. As a result, engine oil consumption is a very important index of modern engine performance and needs to be controlled properly.

Numerous studies have been carried out to analyze the impact of different parameters of the piston-ring-liner system on oil consumption. It was recognized that oil consumption is affected by the geometric details of the piston and rings [9] [10] [11] [12] [13], liner surface finish [14] [15] [16], cylinder bore distortion [17] [18], component temperatures [19], oil properties [20] [21], and engine operation conditions such as speed, load, and whether the engine operates in a steady state.

1.2 Piston Ring Pack [1]

In modern internal combustion engine designs, piston ring pack is usually consisted of three rings: (from the bottom to top) oil control ring, second ring (scraper ring) and top ring (compression ring) (Fig.1.2). There exist various designs for each ring. Fig.1.3-1.5 show some typical designs [22].
Fig. 1.2 Piston Ring Pack

Fig. 1.3 Oil Control Ring Designs [22]
Among the three types of oil control ring design, we will focus on the twin-land oil control ring. Twin-land oil control ring (TLOCR) is widely used in automotive diesel engines, and is gaining more and more applications in gasoline engines. The cross section of a typical TLOCR is shown in Fig.1.6. In order to seal the oil in the crank case
from the combustion chamber, the TLOCR tension is typically higher than the top two rings, and consequently its friction contribution is very important.

\[
\text{TLOCR Land}
\]

\[
\text{Land Width} = 0.2\text{mm}
\]

Fig. 1.6 Twin-land Oil Control Ring Cross Section

TLOCR is also critical in controlling the oil film thickness left on the liner, which is important for both the top two ring lubrication and engine oil consumption [23]. Thicker oil film thickness on the liner above the oil control ring enables stronger hydrodynamic support to the top two rings and generates less friction from the top two rings. However, it may increase the engine oil consumption. The trade-off between the top two ring lubrication condition and the oil consumption makes the oil control ring design optimization rather complicated. In order to optimize the TLOCR performance, a thorough understanding of the interaction between the TLOCR and cylinder bore liner finish is necessary.

The top two rings are important for blow-by control and regulation of the gas flows in the ring pack. High cylinder gas pressure makes them, particularly the top ring, a significant source for liner wear and oil transport. With the reduction of oil control ring tension in modern engine designs, the top two rings are becoming a more and more important source of friction power loss from the piston ring packs.

1.3 Surface Finish on Modern Cylinder Liners [1]

The liner surfaces of modern engines, manufactured with the typical three honing processes, usually consist of two different regions, the plateau part with a smaller root mean square (RMS) roughness and the valley part with a larger RMS roughness. A typical liner finish geometric profile is shown in Fig. 1.7. The plateau part of the surface
is formed by the final fine honing process, while the valley part comes from the early honing processes, and appears as the sparsely distributed grooves of typically several microns depth.

![Liner Surface Measurement](image)

**Fig.1.7 Liner Surface Measurement**

In general, when another surface, in this thesis the piston ring face, slides over the liner surface with a normal load, it is in the plateau part where all asperity contact occurs. With the existence of oil, high oil pressure also tends to be generated in the plateau area to drive the oil around the asperities. Due to this special topology, the composite RMS roughness generally used to represent the roughness of a nominally flat surface is not sufficient and should be replaced by $\sigma_p(r_{pq})$ [24], the RMS roughness of the plateau part and other statistical parameters to represent the valley area. Thus, the definitions of some other terminologies need to be clarified before the detailed technical discussion.

Nominal oil film thickness $h$: The nominal oil film thickness is defined as the minimum height of the nominal ring face profile minus the mean height of plateau part of the liner surface (See Fig.1.8). For a nominally flat ring face profile, it stands for the nominal gap
between the ring face and the liner plateau. For a curved ring face profile, it stands for the nominal minimum oil film thickness between the ring face and the liner plateau.

Film thickness ratio ($\lambda$ ratio): The widely accepted definition of $\lambda$ ratio is modified as the ratio of the nominal oil film thickness (defined above), to the RMS roughness of the plateau part of the liner surface, $\lambda = h/\sigma_p$.

![Curved Ring Profile](image1.png) ![Flat Ring Profile](image2.png)

**Fig.1.8 Nominal Oil Film Thickness $h$**

### 1.4 Modeling the Ring Liner Interaction [1]

A number of works were devoted to understand how liner surface features influence the ring pack friction as well as wear, and ways to improve its behavior through modifying the surface texture [25-33]. These works either intended to correlate the function / performance of the ring or ring pack with the statistical parameters derived from height distribution or neglected the unsteady nature of the oil redistribution between asperities. Furthermore, there have been no studies dedicated to the interaction between the oil control ring and the rough liner, which arguably is the most critical step toward understanding the effects of the liner finish on the outcome of the piston ring pack.

Unlike the top two rings, which both have a macro shape contributing to the hydrodynamic pressure generation between the ring face and the liner surface, the twin-land oil control ring usually exhibits flat running faces after running in. Due to the constraint between the two lands and a high normal load, the two flat faces are practically parallel to the liner surface. As a result, the hydrodynamic pressure generation between
the ring face and the liner, if any, is solely inter-asperity pressure, due to the interaction of the surface micro geometries, rather than the pressure developed with the macro shape of the ring running surfaces. The average hydrodynamic method, which is typically used in the numerical models for the top two rings, is based on the macro geometry of the running surfaces, thus would give zero hydrodynamic pressure for the oil control ring. Therefore, it is not able to correctly predict the behavior of the oil control ring liner interaction. Instead, the deterministic method based on the 3D measurement of the surface profile should be used.

The face profiles of the top two rings are curved in the ring axial direction. These curved profiles can be rather powerful in generating hydrodynamic pressure to balance the normal load, if there is sufficient oil supply. However, since the oil supply to the top two rings is controlled by the oil control ring and is typically limited to the inter-asperity level, it is rarely sufficient. Therefore the macro face profiles of the top two rings are in general not as effective as they were believed to be for ring liner lubrication. Both liner finish micro geometry and ring face macro profile may have significant effect on the top two ring lubrication.

1.5 Deterministic Hydrodynamic Modeling [1]

The deterministic method has been widely used in the numerical study of point contact lubrication. However, not much work has been performed using the deterministic method with proper boundary conditions for ring lubrication yet.

In order to apply the full deterministic method in evaluating the full stroke behavior of the ring liner interaction, two difficulties need to be addressed. First of all, 3D surface measurements usually have a limited measurement range, which can hardly be extended to the entire liner surface. To address this difficulty, Bolander et al. [34] presented a model based on the numerically created surface statistically equivalent to the real measured surface. However, the second difficulty, the trade off between the calculation efficiency and accuracy still remained as a major challenge to the researchers. In order to attain reasonable time efficiency, coarse meshes and large time steps have been used in some published works.
1.6 Scope of Thesis Work

The objective of this thesis work is to model the lubrication of the piston ring pack using the deterministic method.

The second chapter of this thesis introduces a deterministic hydrodynamic model proposed by Li et al. [35], and discusses the key assumption of the model.

The third chapter then applies the deterministic method to a twin-land oil control ring lubrication model [36]. The model accounts for the liner finish micro geometry effect, and it is based on a correlation approach.

The fourth chapter discusses the effect of macro face profile on an oil control ring, and compares the frictional performance of the twin-land oil control ring with the three-piece oil control ring under the same constraint of oil control.

The fifth chapter extends the application of the deterministic method and the correlation approach to a top two ring lubrication model. The model addresses the effect of both liner finish micro geometry and ring face macro profile. It also takes into account the effect of limited oil supply.

After the discussion of the ring pack friction models in chapter 3 and 5, the sixth chapter proposes a wear model to simulate the liner finish micro geometry evolution through wear. The model can be potentially used in prediction of long term behavior of ring pack friction.

The seventh chapter deals with model validation. A floating liner friction measurement [37] is used to compare with the model prediction and the results are analyzed.

The last chapter summarizes and concludes the thesis work and suggests potential future work on the topic.
2 A Deterministic Hydrodynamic Model [1]

This chapter introduces a deterministic method of hydrodynamic modeling for the interasperity flow field between the ring face and the liner finish. The method is originally proposed by Elrod [38], while Li et al. [35] first used it in the modeling of piston ring lubrication.

The chapter will start with a section to introduce the basic assumptions of the method, followed by the detailed explanation of the governing equations as well as the boundary conditions of the method for its usage in ring pack lubrication modeling. The last section of this chapter will discuss the full attachment assumption, a key assumption to the current deterministic approach.

2.1 Basic Assumptions

2.1.1 Assumptions for Lubrication Approximations

In this thesis, the numerical coordinate system is always attached to the ring unless otherwise stated.

In the area where there is enough oil to fully fill the gap between the ring face and the liner surface, the oil flow is governed by the Reynolds equation:

\[
\frac{d(\rho h)}{dt} = \nabla \cdot \left( \frac{\rho h^3}{12\mu} \nabla p \right) - \frac{V}{2} \frac{\partial (\rho h)}{\partial x}
\]

In the equation, \(\rho\) refers to oil density, \(h\) refers to the local clearance, \(\mu\) refers to the dynamic viscosity of the lubricant oil, \(p\) refers to the hydrodynamic pressure of the oil, and \(V\) is the sliding speed of ring. Here it is further assumed that the lubricant oil is incompressible so that the oil density \(\rho\) is a constant, and the following incompressible Reynolds equation applies. And the assumptions of lubrication approximations need to be satisfied.

\[
\frac{dh}{dt} = \nabla \cdot \left( \frac{h^3}{12\mu} \nabla p \right) - \frac{V}{2} \frac{\partial h}{\partial x}
\]
For the oil flow between the ring face and the liner surface, the assumptions of lubrication approximations are interpreted as the following (Fig.2.1):

\[
\frac{\Delta h}{\Delta x} \ll 1, \quad \text{Re}_h \frac{\Delta h}{\Delta x} \ll 1 \quad \text{and} \quad \frac{h^2}{\gamma T} \ll 1,
\]

so that all the inertia terms in the Navier-Stokes equation are negligible.

Here $\gamma$ refers to the kinematic viscosity of the lubricant oil, and $T$ is the characteristic time constant, which in this case can be chosen to be the time for each engine stroke. For a typical finished liner surface, the maximal $\frac{\Delta h}{\Delta x}$ is around 0.1. Since the Reynolds number of the situation is around 1, therefore $\text{Re}_h \frac{\Delta h}{\Delta x}$ is around 0.1. $\frac{h^2}{\gamma T}$ is in the scale of $10^{-5}$. As a result, all the conditions mentioned above are approximately satisfied, and the Reynolds equation applies.

### 2.1.2 A Cavitation Theorem

Lubricant oil cannot exist at pressures below its cavitation pressure. Once the pressure in the flow field drops to a critical value (the cavitation pressure), the oil will cavitate, separating into liquid and vapor. According to the Jakobson-Floberg-Olsson (JFO) theory, as described by Elrod [38], the oil domain can be divided into two distinct zones, a full film region and a partial film region.
In the full film region the oil flow is governed by the Reynolds equation as mentioned in the previous section. It is assumed that the cavitation region is composed of the mixture of liquid phase oil and oil vapor/air. The pressure is assumed to be constant and the oil film ratio (volume proportion of liquid phase) is the dependent variable. Because of the zero pressure gradients in the partial film region, the pressure driven flow term disappears in the cavitation zone. The liquid part of the oil is assumed to attach to both of the running surfaces and form a streaky shaped pattern. (See Fig. 2.2) The oil flow is governed by a pure hyperbolic oil transport equation:

\[
\frac{d(\rho h)}{dt} = -\frac{V}{2} \frac{\partial (\rho h)}{\partial x} \tag{35}
\]
2.2 The Numerical Approach [35]

By introducing an index variable to distinguish cavitation zone and full film zone, Elrod presented a universal numerical scheme to solve whole field [38]. This method avoids tracking the cavitation boundary and the result will automatically satisfy mass conservation. Other researchers reported this method shows numerical instability around the cavitation boundary [38] [39]. Payvar and Salant presented a way to avoid the numerical instability by controlling the index variable [40]. Instead of switching the index variable between zero and one, they used a small relaxation variable to control the stability. The method needs many more iterations to converge in the cases with cavitation than the ones without cavitation. In Li et al.’s model [35], the advantages of existing models are integrated together. Improvements were made to the iteration scheme to gain better robustness and efficiency.

Instead of using compressibility to relate density and pressure, Li et al. only introduce the index variable to switch between the Reynolds equation and oil transport equation. Without the huge lubricant compressibility coefficient, the density error of a point that switches from cavitation zone to full film zone will cause less numerical instability. [35] [44]

The index variable determines the state of a local grid point. To get a uniform governing equation, we need to write the pressure and density as functions of a universal dependent variable [40]. Define the universal dimensionless dependent variable $\phi$ and index variable $F$ as

$$p = F\phi p_{ref} + p_c$$

$$\rho = \rho_c + (1-F)\phi \rho_c$$

$$F = \begin{cases} 1 & \phi > 0 \\ 0 & \phi < 0 \end{cases}$$

in which $p_c$ stands for the cavitation pressure.

Then the Reynolds equation becomes
\[
\frac{d(1-F)h\phi}{dt} = \nabla \cdot (\eta \nabla F\phi) - \frac{V}{2} \frac{\partial(1-F)h\phi}{\partial x} - \frac{V}{2} \frac{\partial h}{\partial x} \frac{dh}{dt}
\]

in which

\[
\eta = \frac{P_{ref} h^3}{12 \mu}
\]

This is a convection-diffusion equation of \( \phi \) in the full film zone. It degenerates to a convection equation in the cavitation zone. Variable \( \phi \) has different physical meanings in different zones. In full film zone, it is a dimensionless pressure. In cavitation zone, its absolute value is the ratio of volume occupied by vapor/gas. The variable \( \eta \) serves as a diffusion coefficient. It is proportional to the cubic of film thickness and decreases dramatically around contact points. [35] [44]

Furthermore, instead of updating both \( F \) and \( \phi \) through a small relaxation number, Li proposed to only update \( \phi \). During iteration \( F \) serves as a switch function that takes zero or one as value.

When index variable \( F \) is fixed, the Reynolds universal equation loses nonlinearity, and iteration can converge quickly. [35] [44]

2.3 The Full Attachment Assumption

One of the key assumptions of the deterministic approach is that the liquid oil attaches to both of the running faces and forms cavitation streaks in the local partial film area (Fig.2.3). This is based on the belief that forming the streaks would help to minimize the surface interface area, thus minimizing the surface energy. In order for this to be true, the width of the streaks has to be much larger than the oil film thickness (Fig.2.3). This requires the characteristic surface slope in the circumferential direction to be small enough (Fig.2.4).

In the case of ring face with a macro profile and sufficient oil supply on the boundary (Fig.2.5), the macro face profile is the dominant geometry. The cavitation occurs in the downstream and the effective characteristic surface slope in the circumferential direction
is very small. The streaky cavitation behavior is observed and well documented with different experiment set-ups [41] [42].

Fig. 2.3 Full Attachment Assumption

Circumferential Direction

Fig. 2.4 Surface Profile in the Circumferential Direction
In order for a streak-like pattern to be formed in the flow field, a pressure gradient exists to push the oil away from the liquid / vapor or liquid / air interface (Fig.2.6). This pressure gradient is sustained by the surface tension and the pressure difference between the vapor / air domain and the low pressure in the liquid phase. The pressure in the liquid phase can not drop below zero, the air / vapor pressure depends on the environment which is also limited, and the surface tension is a physical property of the oil. By all means, the maximum pressure difference is limited.

On one hand, the streaks tend to become as wide as possible to reduce the number of streaks and thus reduce the total surface energy (Fig.2.7). On the other hand, larger streak width requires higher pressure difference. Therefore in a steady state condition, a stable equilibrium can be reached for the streak width when the ring sliding speed is low and oil temperature is high (low viscosity) as observed in experiments [41] [42]. When
the ring sliding speed is high and oil temperature is low (high viscosity), the equilibrium streak width decreases until the pattern breaks and chaotic transient cavitation pattern and oil separation may occur [43].

Fig. 2.6 Cavitation Streaks (Top View)

Fig. 2.7 Cavitation Streaks with Different Widths
Fig.2.8 shows the set-up of a numerical evaluation of the macro cavitation streaks in a steady state condition. The fully flooded leading edge has a boundary pressure of $P_l$, and the trialing edge has a boundary air pressure of $P_t$. The minimum pressure in the flow field is the cavitation pressure $P_c$. The ring face has a barrel shape with a radius of curvature $R$ and the liner surface is perfectly smooth. The geometry is uniform in the circumferential direction. The liner is moving with a speed of $V$ and the oil dynamic viscosity is $\mu$. Between the ring face and liner surface, the local space is either fully occupied by the liquid oil or by the air on the trailing edge side. In the liquid part, the flow field is governed by the steady state Reynold’s equation:

$$\nabla \cdot \left( \frac{h^3}{12 \mu} \nabla p \right) - \frac{V}{2} \frac{\partial h}{\partial x} = 0$$

Fig.2.8 the Set-up of a Numerical Evaluation of the Macro Cavitation Streaks
The boundary of liquid and air is determined by a no-flow-across-boundary rule and an adaptive numerical grid technique. Fig.2.9 shows the idea. A regular grid is used in the sliding direction (x direction). In the circumferential direction (y direction), the numerical grid is adaptive. Each y direction grid size is solved so that the mass conservation on a liquid-air boundary grid is satisfied and there is no flow across the boundary \( f_{i,j}^{in} = f_{i,j}^{out} \).

![Diagram showing the numerical evaluation of liquid-air boundary](image)

**Fig.2.9 Numerical Evaluation of Liquid-Air Boundary**

Table 2.1 lists the input specifications of an example. Fig.2.10 shows the corresponding pressure distribution for liquid or air. The liquid-air boundary is highlighted with the red curve. Fig.2.11 compares the circumferential maximum, minimum and average pressures with the pressure calculated with traditional method, which assumes uniformity of hydro pressure in the circumferential direction and oil separation at zero pressure gradient.
location. In this case, the circumferential variation of hydro pressure is large due to the relatively high trailing edge boundary pressure. The average hydro pressure can be significantly lower than tradition method’s prediction.

Three dimensionless groups can be formed to study the determination of half streak width $W$ (Fig.2.10):

$$\Pi_1 = \frac{\mu V}{(P_t - P_c) H}$$
$$\Pi_2 = \frac{W}{R}$$
$$\Pi_3 = \frac{R}{H}$$

The first dimensionless group captures the competition between the viscous shear and pressure difference. The rest of two groups simply capture geometry effect. The effect of ring width and leading edge boundary pressure is not considered.

The dependency of the three dimensionless groups for different test cases is plotted in Fig.2.12. As previously discussed, the streak width increases with the pressure difference $P_t - P_c$ and decreases with oil viscosity and ring sliding speed.

**Table 2.1 Input Specifications of an Example of Macro Cavitation Streak Evaluation**

<table>
<thead>
<tr>
<th>$P_t$ (bar)</th>
<th>1</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_c$ (bar)</td>
<td>10</td>
</tr>
<tr>
<td>$P_c$ (bar)</td>
<td>0</td>
</tr>
<tr>
<td>H (micron)</td>
<td>0.8</td>
</tr>
<tr>
<td>Ring Width (mm)</td>
<td>5</td>
</tr>
<tr>
<td>R (mm)</td>
<td>10</td>
</tr>
<tr>
<td>$\mu$ (pa.s)</td>
<td>5E-3</td>
</tr>
<tr>
<td>V (m/s)</td>
<td>3</td>
</tr>
</tbody>
</table>
Fig. 2.10 Liquid-Air Boundary Determination and Pressure Distribution

Fig. 2.11 the Circumferential Maximum, Minimum and Average Pressures
In the case of inter-asperity cavitation, the situation is not quite the same. For inter-asperity cavitation, liner roughness is the governing geometry. The local surface slope in the circumferential direction can reach as high as $\frac{1}{2}$ in radian (Fig. 2.13). While the cavitation streaks may not be able to survive, the real physics of cavitation pattern in the inter-asperity level is unknown.

However, what is important for deterministic modeling is not really the full attachment assumption, but rather the indication of this assumption in the oil flow rate. By assuming full attachment, one is actually assuming that the oil flux in the sliding direction is $\frac{1}{2} \rho \nu h$. This is equivalent to assuming that half of the oil is dragged by the moving surface and the other half stays with the static surface.
Imagine a steady state oil separation condition in Fig.2.14. The ring is sliding over the deterministic liner geometry and the flow field reaches a steady state. Oil separates into two parts in the deep valley and forms a cavity in between. The upper part sticks to the ring and the lower part forms a puddle staying with the liner. The control volume highlighted by the blue frame would have only oil inflow and no outflow. Therefore the upstream of the cavity would be filled and reaches the second state. However once the oil attaches to both running surface, half of the oil would again be dragged by the ring and the oil will inevitably separates again and returns to the first state. Therefore a steady state is reached.
Let's examine the flow rate in the cavitation area. Fig. 2.15 shows the comparison of the two different cavitation patterns, the separation form and the full attachment form. Take the same control volume, and one may realize that under the steady state, the flow rate in the cavitation zone (the outlet of the control volume) has to be the same since the oil inflows are the same. However, in order to reach the same oil flow rate, the oil film ratios (percentage of liquid oil occupation) in the two cavitation forms are not bounded to be the same. The full attachment form has the same oil film ratio as the separation form when the two separated puddles have the same oil film thickness. While in unsteady state,
the oil volume in the control volume is changing and therefore the oil flow rate also depends on the oil film ratio in the cavitation region, the full attachment assumption is merely an approximation of the separation form under the assumption that oil equally spits into two puddles when it separates as long as the deterministic modeling is concerned. Since it unifies the forms of convection flow rate formula in the full film and cavitation regions, it significantly simplifies the process of modeling and numerical solution. A more accurate model is quite difficult in this set-up. One has to assign a different model for the separation and full attachment partial film. Since the inter-asperity level separation condition in the cavitation area has yet been determined physically, any model that deals with different forms of cavitation would have to address this issue first.

![Diagram of two forms of cavitations](image)

**Fig.2.15 Two forms of cavitations**
2.4 Conclusion

This chapter introduces a deterministic approach to evaluate the flow field between two surfaces with relative motion, and discusses the key assumptions and the limitation of the approach. The next three chapters will discuss the twin-land oil control ring and the top two ring friction models developed based on the deterministic approach.
3 Twin-land Oil Control Ring Model [1]

This chapter introduces a twin-land oil control ring model for cycle friction prediction. It starts with a general introduction of the ring configuration and some important model assumptions. For a more thorough discussion of the model assumptions and the validation, one can check Chen's master thesis [1]. The introduction section is followed by the discussion of the hydrodynamic and asperity contact correlations used in the model, and how the cycle friction is computed from these correlations. Some examples of the model results are given in the third section. And the chapter ends with a section that discusses the surface filtering technique that removes the macro profile from the raw measurement of the liner surface geometry so that it satisfies the requirement of the deterministic calculation.

3.1 General Introduction

The twin-land oil control ring is generally considered to be the most important ring for friction power loss, both due to its own friction contribution, and due to the fact that it controls the oil film thickness on the liner that determines the oil supply for the top two ring lubrication. In a traditional design, the oil control ring could have about twice as much ring tension as the top two rings combined. In recent years, oil control ring tension has been reduced substantially. Yet in modern designs it still accounts for roughly half of the total ring tension. Therefore it is still an important source of engine friction power loss.

It is usually believed that the twin-land oil control ring exhibits a flat ring face on its two lands for its interaction with the liner finish. See Fig.3.1 and 3.2 for a ring face profile measurement. Especially after the ring is broken in, the ring face can essentially be treated as flat. And the roughness on the ring face is typically much smaller than that of the liner finish. Moreover, it is formed by the wear process in the running engines and therefore it is engine-dependent and not well defined. Therefore it is assumed in this chapter that the twin-land oil control ring face is flat and smooth.

Without a macro profile on the ring face, the twin-land oil control ring can only rely on the liner roughness and the inter-asperity hydro pressure to provide hydro support and lift.
the ring face from pure boundary contact. The inter-asperity hydro pressure generation can be evaluated using the deterministic method introduced in the previous chapter based on certain boundary conditions. Since the twin-land oil control ring typically works in a steady environment, i.e. sufficient oil supply and low ambient pressure, the boundary conditions are set accordingly in the model.

![Fig.3.1 Twin-land Oil Control Ring Face Profile](image)

![Fig.3.2 Twin-land Oil Control Ring Face (Worn)](image)
If we assume that the oil layer between the ring face and the liner finish is always in quasi-steady state, we can decouple the ring dynamic effect and the local force balance. The inter-asperity hydrodynamic pressure can be evaluated separately [1] and be fed into a cycle model in a correlation form together with an asperity contact model (also in a correlation form) to evaluate the cycle friction of the oil control ring. The next section will discuss the formation and computation of such hydrodynamic and asperity contact correlations.

### 3.2 Correlations

#### 3.2.1 Hydrodynamic Correlations

The inter-asperity hydrodynamic pressure generation can be evaluated using the deterministic method introduced in the previous chapter. Since the ring face motion in the radius direction is so slow and its influence is trivial except in a few degrees around top and bottom dead center area [1], its importance in friction power can be neglected. Therefore the hydrodynamic pressure between the ring face and the liner finish can be evaluated by sliding a flat ring face over the rough liner at a fixed oil film thickness $h$ (Fig.3.3).

Fig.3.4 shows the 3D deterministic measurement of a liner finish. The measurement is made with a mechanical stylus machine in a four micron resolution in both directions. The data is then filtered with the method introduced in section 3.4.

On the 3D liner measurement (Fig.3.4), the inter-asperity hydro pressure is deterministic for a given ring land width $w$, oil dynamic viscosity $\mu$, ring sliding speed $V$ and oil film thickness $h$ (Fig. 3.5, 3.6). The dependency of the average hydro pressure generation on the oil film thickness can be correlated in the following form:

$$
P_{\text{hydro}} = \frac{\mu V}{\mu_0 V_0} P_h \left( \frac{h}{\sigma_p} \right)^{-K_h} \quad (3.1)
$$

Here $P_h$ and $K_h$ are two constants based on the reference dynamic viscosity $\mu_0$ and reference ring speed $V_0$ used in the deterministic evaluation. The effect of oil dynamic viscosity and the ring sliding speed are assumed to be linear. Chen [1] has a more
detailed analysis of the validity of this assumption. Fig.3.7 shows a comparison between the correlation and the evaluated average hydro pressure.

The magnitude of the two constants $P_h$ and $K_h$ are critical for defining the performance of liner finish in hydrodynamic lubrication. $P_h$ defines a characteristic hydrodynamic pressure and $K_h$ defines how fast hydrodynamic pressure decays with the oil film thickness. The value of $P_h$ typically varies between $10^6$–$10^8$ Pa. The value of $K_h$ is mainly affected by the proportion of the plateau surface area (plateau ratio). For a liner finish with Gaussian surface height distribution, the value of $K_h$ is around 3. For a liner finish with approximately Gaussian distributed plateau and deep valleys, $K_h$ decreases with the decrease of the proportion of plateau area. This is because deep valleys have a larger roughness scale than the plateau area and thus the hydro pressure generation due to the deep valleys decays more slowly to the oil film thickness than the hydro pressure generation due to the plateau roughness.

Similarly the hydrodynamic shear stress can be correlated in the following form, in which $C_{f1}$, $C_{f2}$ and $C_{f3}$ are three constants.

$$f_{hydro} = \frac{\mu V}{h} \left( C_{f1} + C_{f2} \exp \left( -C_{f3} \frac{h}{\sigma_p} \right) \right) \quad (3.2)$$
Average film thickness \( h \)

Fig. 3.3 A Flat Ring Face over the Rough Liner

Fig. 3.4 A Liner Finish Measurement
Fig. 3.5 Hydrodynamic Pressure between the Ring Face and Liner Finish

Fig. 3.6 Hydro Pressure Generation of a Ring Face Sliding at a Constant Clearance
3.2.2 Asperity Contact Correlation [1]

Different asperity contact models can be implemented, since it’s decoupled with the hydrodynamic part. A typical one would be the asperity contact model developed by Greenwood and Tripp [45]. According to the Greenwood and Tripp model, nominal asperity contact pressure between two rough surfaces can be calculated by the following correlation,

\[ P_c'(\lambda) = a_p K' \int_{\lambda}^{\infty} (z - \lambda)^2 \phi(z) dz \]

in which

\[ K' = \frac{8\sqrt{2\pi}}{15} \left( \eta \beta r_p \right)^2 \frac{r_p}{\beta} \]

In the above two equations, \( P_c \) is the nominal asperity contact pressure between the two surfaces, \( a_p \) is the area ratio of plateau part to the entire surface, \( \eta \) is the asperity density...
per unit area in the plateau part, $\beta$ is the asperity peak radius of curvature in the plateau part, and $\phi(z)$ is the probability distribution of asperity heights.

The Greenwood and Tripp model assumes that contact is elastic, and the asperities are parabolic in shape and identical on the contacting surfaces. For deterministic modeling, all the surface geometry related parameters such as $a_p$, $\eta$, $\beta$ and $\phi(z)$ can be evaluated based on the 3D deterministic surface measurement.

The correlation of the asperity contact friction and the film thickness ratio can be expressed as

$$ f_c(\lambda) = C_{fc} A P_c(\lambda) $$

where $C_{fc}$ refers to the asperity contact friction coefficient, and $A$ is the oil control ring face area. The asperity contact friction coefficient depends on the material of the interacting surfaces and the lubricant oil. As an input value, it can be modified in the model and should be assigned with the measured value for specific applications. For the results presented in this thesis, it is assigned as 0.15 unless stated otherwise.

In reality the contact property of the surface is more complicated. Quite often the assumptions of Greenwood and Tripp model on surface asperity property don’t hold on real surfaces. Even the asperity property itself is hard to measure accurately and is subject to change in wear. Chapter 7 of model validation will discuss some effort of contact model calibration using experimental results. The model work for a more practical contact theory is on the list of the suggested future work.

### 3.2.3 Cycle Calculation

With the correlation in hydrodynamic pressure and asperity contact, the cycle friction calculation can be incorporated into a dynamic model that takes care of the load balance and ring dynamics. Tian’s thesis [46] has more details in this. The hydrodynamic sub-model in the original TLOCR cycle model is replaced by the hydrodynamic correlations described in the previous section.

Fig.3.8 to Fig.3.10 show some sample results of the cycle model. See the chapter of model validation for the detailed specification of the engine and the liner finish that is
evaluated in the model. Table 3.1 lists the hydrodynamic and boundary correlations for this particular liner finish. The oil control ring in the model has a land width of 0.15mm and a ring tension of 10.8N. The oil dynamic viscosity in the model is 0.00854 (pa.s) at 100 C. Fig.3.8 shows the ring friction of the two lands at 1000 rpm. Fig.3.9 shows the corresponding boundary friction, and Fig.3.10 shows the oil film thickness of the two lands. The increase of piston speed in the mid stroke causes a more effective hydrodynamic lubrication and lifts the ring face. The boundary friction reduces corresponding to the increase of oil film thickness. The increase of total fiction in the mid stroke indicates a hydrodynamic lubrication regime.

Table 3.1 Hydrodynamic and Boundary Correlations

<table>
<thead>
<tr>
<th>Correlation of Hydro Pressure</th>
<th>$P_{hydro} = \frac{\mu V}{\mu_0 V_0} \frac{h}{\sigma_p}^{\frac{1}{K_h}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_h$ (pa)</td>
<td>3.4586e+007</td>
</tr>
<tr>
<td>$K_h$</td>
<td>2.6475</td>
</tr>
<tr>
<td>$\sigma_p$ (micron)</td>
<td>0.071</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Correlation of Hydro Shear Stress</th>
<th>$f_{hydro} = \frac{\mu V}{h} \left( C_{f1} + C_{f2} \exp \left( -C_{f3} \frac{h}{\sigma_p} \right) \right)$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_{f1}$</td>
<td>0.9109</td>
</tr>
<tr>
<td>$C_{f2}$</td>
<td>1.4142</td>
</tr>
<tr>
<td>$C_{f3}$</td>
<td>0.9865</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Correlation of Contact Pressure</th>
<th>$P_{contact} = \begin{cases} P_c \left( \frac{z}{\sigma_p} \right)^{-K_c} &amp; \text{for } \frac{h}{\sigma_p} &lt; z \ 0 &amp; \text{for } \frac{h}{\sigma_p} \geq z \end{cases}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_c$ (pa)</td>
<td>1.267e+005</td>
</tr>
<tr>
<td>$z$</td>
<td>4.8</td>
</tr>
<tr>
<td>$K_c$</td>
<td>6.804</td>
</tr>
</tbody>
</table>
Boundary Friction Coefficient | 0.15

Fig. 3.8 TLOCR Friction at 1000 rpm

- Total friction force on the running surface of the lower land
- Total friction force on the running surface of the upper land
Friction force from boundary lubrication on the running surface of the lower land
Friction force from boundary lubrication on the running surface of the upper land

Fig. 3.9 TLOCR Boundary Friction at 1000 rpm
3.4 Surface Filtering Technique

The deterministic calculation requires the liner surface to be nominally flat. This is because we are only focusing on a very small area of the liner finish and the ring face is assumed to be not tilted. In reality the ring face will adjust its orientation locally to conform to the macro profile of the cylinder bore.

Traditionally the liner finish measurement is often filtered by a regular high pass Gaussian filter before any further use. However, due to the nature of multi-step honed surface geometry, the regular Gaussian filter is inadequate and often leaves some residual shape or creates some artificial shape near the deep valleys of the liner.

As an example, Fig.3.11 shows a raw 500 by 500 grids liner surface measurement of 2 micron resolution. The color scale shows the surface height in the unit of micron.
Fig. 3.12 shows the surface filtered with the traditional Gaussian filter. It is obvious that not only some residual shape is left on the surface, but also artificial bumps can be found on the plateau in densely grooved area. These artificial bumps have relatively large wavelengths comparing to the roughness micro structure and can wrongfully contribute to hydrodynamic pressure generation.

**Fig. 3.11 the Raw Liner Surface Measurement**
To address this issue, a new filter technique is developed. The idea is to filter the liner surface only based on the surface height in the plateau area. The plateau area is identified by a simple criterion such as the surface heights within a certain range (3 plateau roughness standard deviation) near the mean line of the plateau area. The surface can be repetitively filtered, each time starting from the raw data and with a better knowledge of the plateau classification. The process finishes when the identification of the plateau area does not change any more. The logic flow of the method is shown in Fig.3.13.

Fig.3.14 shows the surface in Fig.3.11 filtered with the new method. Fig.3.15 shows the waviness removed in the process. It is clear that the removed waviness has relatively large wavelengths and a high quality nominally flat liner surface is recovered by the new method.

In this thesis work, all the liner surfaces presented are filtered with the new method.
Fig. 3.13 the Logic Flow of the New Filter
Fig. 3.14 the Surface Filtered with the New Filter

Fig. 3.15 the Waviness Removed by the New Filter
3.5 Conclusion

This chapter introduces a twin-land oil control ring friction model based on the deterministic method discussed in chapter 2. It is assumed that the two ring lands have a flat face profile. This means all the hydrodynamic pressure generation would purely come from the liner roughness.

The flat face profile makes the TLOCR lubrication roughness dependent only, and any traditional methods fail in the friction prediction. The next chapter discusses the effect of a curved macro face profile on an oil control ring. Chapter 5 discusses the deterministic based modeling of the top two ring friction, in which both the liner roughness and ring face profile take an effect. In the top two ring model, the oil supply is defined by the oil control ring, whose performance is predicted using the TLOCR model discussed in this chapter.
4 Macro Face Profile Effect and Three-Piece Oil Control Ring

In this chapter, the effect of macro ring face profile with curvature on an oil control ring is discussed. The friction performances of ring faces with different curvatures are compared under the same constraint of oil supply.

4.1 Introduction

The TLOCR model discussed in the last chapter is based on a strict assumption that both ring lands have a flat ring face (no curvature). In reality, since the torsional stiffness of the ring is high, the break-in process of the ring tends to remove any curved shape of the ring face profile. Therefore even if the original ring face is curved, the curvature wouldn’t last for too long.

Unlike the TLOCR, the other major type of oil control ring, the three-piece oil control ring (TPOCR), typically has curved ring face profile. Typically the face profile of the TPOCR can be approximated as parabolic in the sliding direction and uniform in the circumferential direction. It can be defined using one face factor with the following equation.

\[ h_{\text{face}} = factor \times x_{\text{face}}^2 \]  

(Fig.4.1)

![Fig.4.1 TPOCR Ring Face Profile Definition](image_url)
4.2 Discussion

The face factor characterizes the sharpness of the ring face profile. The hydrodynamic pressure and oil film ratio of four ring faces of 0.2mm wide, same minimum oil film thickness and different face factors are shown in Fig.4.3 (See Fig.4.2 for the model setup). The results are evaluated with the deterministic method introduced in the previous chapter and are based on one particular liner finish. However the trends observed are more general. When the ring face gets shaper, the macro face effect gets more and more dominant. When ring face gets sharp enough, the inter-asperity cavitation effect is trivial and the effect of liner roughness in lubrication can be captured with the traditional average Reynold's equation. Fig.4.4 shows the average hydrodynamic pressure and oil film ratio profiles in the ring sliding direction for rings of different face factors. In general, a smaller face factor corresponds to a flatter and wider profile of hydrodynamic pressure generation and lower average oil film ratio.

**Full Oil Supply**

Fig.4.2 Face Profile Effect Model Set-up
Fig. 4.3 Hydro Pressure and Oil Film Ratio Distribution

Fig. 4.4 Hydrodynamic and Oil Film Ratio Profiles for Ring Faces with Different Face Factors
Fig. 4.5 shows the effect of ring face factor on the hydrodynamic pressure, hydrodynamic friction, hydrodynamic friction coefficient (hydro shear stress divided by hydro pressure) and the oil film thickness left on the plateau of liner finish (Fig. 4.6).

Starting from the flat profile, the hydrodynamic pressure first increases and then decreases with the ring face curvature at the same nominal oil film thickness. This indicates a stronger hydrodynamic pressure generation from a properly curved macro profile than from the roughness profile alone. The peak pressure location shifts to the right when the nominal oil film thickness increases, since the roughness effect decays faster with the oil film thickness than the macro profile effect. At the same nominal oil film thickness, the hydrodynamic friction decreases with the increase of ring face curvature due to the reduction of the effective high shear area, and the hydrodynamic friction coefficient exhibits a minimum with a proper ring face curvature.

Comparing to the perfectly flat ring face, a properly curved face profile tends to reduce the hydrodynamic friction. One may be curious why not use a curved ring face for TLOCIR. There are a few reasons:

First of all, the curvature on the ring face tends not to last long for TLOCIR as we have discussed previously.

Second, the exact curvature is hard to control in the manufacturing process, so the design optimization on the ring face curvature is barely meaningful.

Third, the curved ring face does a poor job in oil control. See Fig. 4.7, the oil film thickness left on the plateau of liner finish tends to increase with the curvature at the same nominal oil film thickness level. There are two reasons for this trend. See Fig. 4.8. The increase of ring face factor corresponds to a increase in the pressure gradient that pushes the oil flowing through the minimum oil film thickness location (increase of Couette flow). Meanwhile, the increase of ring face factor also increases the average oil film ratio at the minimum oil film thickness location that will increase the Poiseuille flow. Therefore if the same oil consumption constraint is imposed, the curved face ring would demand a much higher normal load, and the ring friction is not that low any more.
Fig. 4.5 Effect of Ring Face Factor at Different Minimum Oil Film Thicknesses
Fig. 4.6 Definition of Average Oil Film Thickness Left on the Plateau and in the Valley

\[
\text{Plateau Oil Film Thickness}
\]

- \( \frac{h}{\sigma_p} = 2 \)
- \( \frac{h}{\sigma_p} = 3 \)
- \( \frac{h}{\sigma_p} = 4 \)
- \( \frac{h}{\sigma_p} = 5 \)
- \( \frac{h}{\sigma_p} = 6 \)

Fig. 4.7 Face Factor Effect on Oil Film Thickness Left on the Plateau
The effect of oil control constraint can be illustrated with the following example.

The average overall oil film thickness left on the liner and the average oil film thickness left on the plateau are each used as the criterion for oil control. Comparing to the overall oil film thickness, the oil film thickness left on the plateau is more relevant since oil on the plateau is directly exposed to ring scraping, and is more likely to be consumed. However, the full attachment assumption of the deterministic method causes uncertainty in the calculation of oil film thickness left on the plateau. The space on the trailing edge of the ring face will bring the oil out of the valley and redistribute it onto the plateau. This causes a decreasing trend of the oil film thickness in the valley when ring face factor goes beyond certain level (Fig.4.9) and further increase of the oil film thickness left on the plateau. Oil film thickness left on the plateau can be overestimated for large ring face factors when the oil detachment actually occurs in the area of full attachment assumption. Therefore both oil film thickness definitions are used to illustrate the effect of oil control constraint.
Fig. 4.10 shows the constraint of oil control based on the two different definition of oil film thickness left on the liner. For both definitions, the oil film thickness left on the liner is controlled to the level of $h/\sigma_p = 3$. The corresponding minimum oil film thickness of different ring face factors is shown in Fig. 4.11. According to both criteria, a sharper ring face demands a smaller minimum oil film thickness in order to satisfy the constraint of oil control. The corresponding hydrodynamic pressure and hydrodynamic friction are shown in Fig. 4.12. According to either criterion, there is no advantage in friction for a curved ring face profile (TPOCR) than a flat ring face profile (TLOCR) under the same constraint of oil control.
Overall Oil Film Thickness

Plateau OFT vs Overall OFT

Fig. 4.10 Constraint of Oil Control for Two Different Oil Film Thickness Definitions

Minimum Clearance

Plateau OFT vs Overall OFT

Fig. 4.11 Minimum Oil Film Thickness under the Constraint of Oil Control
4.3 Conclusion

In this chapter, the effect of macro ring face profile with curvature on an oil control ring is discussed. The friction performances of ring faces with different curvatures are compared under the same constraint of oil supply. Although the curved ring face can typically reduce hydrodynamic friction coefficient at the same minimum oil film thickness level, it also allows more oil to pass. Therefore under the same constraint on oil control, no advantage is found in friction for the ring face with curvature.
5 Top Two Ring Model

This chapter introduces a top two ring model for cycle friction modeling. It starts with a general introduction of the ring configuration and some important model assumptions. Then as in the previous chapter, the hydrodynamic and asperity contact pressure correlations used in the cycle model are discussed. The chapter ends with a comparison of the deterministic based model results with the results of original Reynolds equation based model.

5.1 General Introduction

There are three major differences between the running conditions of the top two rings and the twin-land oil control ring. First of all, unlike the twin-land oil control ring, the top two rings typically have a curved running surface (Fig.5.1). The macro profile of the ring face contributes to the generation of hydrodynamic lift for the ring face. However, the oil supply to the top two rings is controlled by the oil control ring to the roughness level. This makes the lubrication oil supply dependent and diminishes the power of the macro profile in hydro pressure generation. The liner roughness micro structure therefore is as important as the macro ring profile for the ring lubrication. The last difference is that the top two rings, especially the top ring, often works in a high cylinder gas pressure environment. The gas penetration can affect the lubrication behavior of the top two rings.

The gas pressure effect in hydrodynamic pressure generation is neglected in this work, not just due to the uncertainty in physics [44], but also based on the argument that the lubrication effect is quite weak when the gas pressure is high. When the gas pressure is high, the ring endures high load from the back pressure, the sliding speed is low and it typically suffers limited oil supply since it’s usually in the upper part of the stroke. All these factors weaken the relative importance of the hydrodynamic support to the total load, and the load is mainly supported by the asperity contact. This is supported by the evidence that the liner finish is often polished near the top dead center region of the top ring after certain hours of running in an engine. Therefore the top ring friction in the high gas pressure region is mainly load dependent boundary friction which can be evaluated.
by multiplying the load by the boundary friction coefficient. As for how to bridge the high gas pressure area and the low gas area, this is a subject to future research.

**Top ring profile**  
**Second ring profile**

![Fig.5.1 Top Two Ring Profile (Worn)](image)

The ring profile and liner roughness effect as well as the oil supply effect in hydrodynamic pressure can be evaluated using the deterministic method. Here one major uncertainty is the size of the calculation domain. Given the oil left by the oil control ring, the wetting on the top two rings in reality is transient and irregular where surface tension of the oil plays an important role. However the deterministic method proposed in the previous chapter assumes a uniform calculation domain (See a comparison in Fig.5.2). It does not consider the surface tension effect and varying boundary of the full attachment region. The oil is assumed to attach to both the running surfaces in the entire calculation domain. This arbitrary setting deviates from reality and the increase of the calculation domain in the axial direction would arbitrarily increase the hydrodynamic friction due to
the full attachment assumption. This introduces an uncertainty in the model results. In this study, the calculation domain is selected to be a large enough constant to cover all the hydrodynamic pressure generation area while limited by the total available size of the ring contact area (The highlighted area in Fig. 5.1).

![Diagram of reality vs deterministic assumption](image)

**Fig. 5.2 Real Wetting and Calculation Domain**

### 5.2 Deterministic Calculation under Partial Oil Supply

The deterministic calculation on a curved ring profile with limited oil supply is very much the same with the calculation for the flat ring profile and flooded oil supply, except the inlet boundary is no longer full film. The oil supply is determined by the oil control ring. For one particular oil control ring oil film thickness, the oil control ring model predicts the distribution of oil film thickness left on the liner. This distribution is fed into the top two ring deterministic calculations based on the mass conservation. The boundary pressure is assumed to be a constant ambient pressure. In the calculation
domain the oil is assumed to always fully attach to both the solid surfaces. Therefore the boundary oil film ratio is given by

$$\rho_t = \min \left\{ \frac{2(h_{oil,t} + h_{res,t})}{h_{local,t}}, 1 \right\}$$

$$h_{res,t+1} = \max \left\{ h_{oil,t} + h_{res,t} - \frac{h_{local,t}}{2}, 0 \right\},$$

in which $h_{oil,t}$ denotes the oil film thickness on the liner in front of the calculation boundary, and $h_{local,t}$ refers to the local clearance on the calculation boundary. The factor 2 here comes from the full attachment assumption. If $\rho_t = 1$, the extra oil is put into the same circumferential grid on the boundary in the next time step, so that the total mass is conserved. The extra oil film thickness from the last step is captured by $h_{res,t}$.

The previous chapter discussed the effect of macro ring face profile with saturated oil supply. The oil supply condition is critical in determining the lubrication condition. Fig.5.3 shows the hydrodynamic pressure distribution and oil film ratio distribution of a curved ring face under both the partial oil supply and full oil supply. Unlike in the full oil supply case, the hydrodynamic pressure in the partial oil supply case is inter-asperity. Due to the oil supply condition, the strength of hydrodynamic pressure generation from the effect of ring face profile is largely compromised. The pressure generation is concentrated in the upstream area near the center of the calculation domain where nominal oil film thickness is small.

Fig.5.4 shows the average hydro pressure and oil film ratio profile across the ring width. The magnitude of the hydrodynamic pressure is significantly lower for the partial oil supply case and the peak of the hydro pressure is closer to the minimum oil film thickness location.

Fig.5.5 shows the average hydrodynamic pressure build-up for different minimum oil film thicknesses of both the two oil supply conditions. Not only the hydro pressure is order of magnitude lower for the limited oil supply case, the pressure is also decaying a lot faster with the increase of minimum oil film thickness. This effect is fairly important.
to determine the behavior of the top two rings. The high ‘stiffness’ of the hydro pressure’s dependence on minimum oil film thickness constrains the minimum oil film thickness in a small range determined by the oil supply. This makes the correlation approach introduced in the next section possible.

The biggest uncertainty in the deterministic calculation for the curved ring face profile under partial oil supply is the determination of calculation domain. Currently the size of the calculation domain in the ring sliding direction is arbitrarily chosen. In Fig.5.3 and 5.4 one can see that once the size of the calculation domain is largely enough, there is no hydro pressure generation near the boundary and further increasing the calculation size would not create more hydrodynamic lift. However, due to the full attachment assumption, the enlargement of the calculation domain arbitrarily increases the hydrodynamic friction and this causes uncertainty in friction calculation. On the other hand, a too small calculation domain may accumulate oil on the boundary, in which case the full attachment area is underestimated and the hydrodynamic lift is underestimated. In reality the optimal choice of the calculation domain depends on the oil supply, the minimum oil film thickness, the ring face profile, and etc.
Hydro pressure (log)  Oil film ratio (density)

Partial oil supply

Full oil supply

Fig. 5.3 Hydro Pressure and Oil Film Ratio Distribution (Partial Supply and Full Supply)
Fig. 5.4 the Average Hydro Pressure and Oil Film Ratio Across The Ring Width
(Partial and Full Oil Supply)

Fig. 5.5 the Average Hydro Pressure Generation (Partial and Full Oil Supply)
5.3 Correlations

5.3.1 Hydrodynamic Correlations

In the deterministic calculation for top two rings, the oil supply on the boundary of the calculation domain is determined by the oil control ring oil film thickness $h_{OCR}$ (see Fig.5.6), which is in general twice as the oil film thickness left on the liner used in the previous section. At a given level of oil control ring oil film thickness, the average hydrodynamic pressure generation on the ring face profile can be evaluated at different minimum oil film thicknesses $h_{prof}$. The ring face profile is assumed to be parabolic in the sliding direction and uniform in the circumferential direction. It can be defined using one face factor with the following equation once the calculation domain is fixed (Fig.5.7).

$$h_{face} = factor \times x_{face}^2$$ (5.1)

![Fig.5.6 Oil Supply Condition](image-url)
Fig. 5.7 Ring Face Profile Definition

Fig. 5.8 shows the variation of the average hydrodynamic pressure with minimum oil film thickness $h_{\text{prof}}$ for different ring face factors. The results are evaluated with the deterministic method and the boundary conditions discussed previously. The oil supply is defined by a given oil control ring oil film thickness (OCR OFT) $h_{OCR}$. The corresponding hydro pressure for an oil control ring with a flat profile of the same land width and sufficient oil supply is shown in the solid line in a log-log scale. When the face factor is equal to zero, the ring has the same ring face with the oil control ring. The hydrodynamic pressure exhibits two decay rates with the minimum oil film thickness in the two regions separated by the oil control ring oil film thickness $h_{OCR}$. When $h_{\text{prof}}$ is less than $h_{OCR}$, the oil supply is sufficient and the ring has identical hydrodynamic pressure generation with the oil control ring. When $h_{\text{prof}}$ is larger than $h_{OCR}$, the ring starts to starve more and more when the oil film thickness increases and the corresponding hydrodynamic pressure decays at a much higher rate. For the nonzero ring face factors, the hydrodynamic pressure decays at a similar high starvation rate around the supply level $h_{OCR}$. This is because the curved ring face allows more oil to pass than the flat ring face at the same minimum oil film thickness if the supply is sufficient. Since the hydrodynamic pressure acts stiffly and decays at a high rate with the increase of the minimum oil film thickness, the minimum oil film thickness should be roughly defined.
by the oil supply level, and can not deviate too far away from $h_{OCR}$ if the load is mainly supported by the hydrodynamic pressure. If otherwise the load is mainly supported by asperity contact, the hydrodynamic pressure wouldn’t be that important. Therefore we only need to correlate the hydrodynamic pressure of the top two rings at minimum oil film thickness near the oil supply level, and apply this correlation in the cycle model for load support analysis.

Fig.5.9 shows the behavior of hydrodynamic shear stress of the curved ring face. The general behavior is quite similar to the hydrodynamic pressure for the same reasons. We can also correlate the hydrodynamic stress of the top two rings at minimum oil film thickness near the oil supply level, and apply the correlation in the cycle model to calculate hydrodynamic friction.

Fig. 5.8 Hydro Pressure for Different Ring Curvatures
Fig. 5.9 Hydro Shear Stress for Different Ring Curvatures

Fig. 5.10 Hydro Pressure for Different Ring Curvatures at Three Oil Supply Levels
The starvation decay rate of the hydrodynamic pressure is different at various oil supply levels, which can be seen in Fig.5.10. The decay rate tends to increase with more oil supply, therefore it needs to be correlated with the oil supply level $h_{OCR}$. Taking that into consideration, the hydrodynamic pressure and shear stress of a particular face profile and liner finish can be correlated in the following form, which is the simplest form considering both the geometry and oil supply effects:

$$P_{prof} = \left(\frac{h_{OCR}}{h_{prof}}\right)^{K_p} \frac{\mu V}{(\mu V)_0} \left(a_p P_{0,OCR} \left(\frac{h_{prof}}{\sigma_p}\right)^{K_{OCR}}ight)$$

(5.2)

$$f = F_0 \frac{\mu V}{h_{prof}} \left(\frac{h_{OCR}}{h_{prof}}\right)^{K_f}$$

(5.3)

Equation 5.2 is the correlation for hydrodynamic pressure. The form is based on the hydrodynamic pressure correlation of the twin-land oil control ring with the same land width as the calculation domain of the profiled ring. For TLOCR, as introduced in the previous chapter, the hydrodynamic pressure correlation takes the following form

$$P_{OCR} = \frac{\mu V}{(\mu V)_0} P_{0,OCR} \left(\frac{h_{OCR}}{\sigma_p}\right)^{K_{OCR}}$$

(5.4)

in which $P_{0,OCR}$ and $K_{OCR}$ are two liner finish depend constants, and $\mu, V$ stand for oil dynamic viscosity and ring sliding speed. $(\mu V)_0$ is the reference value of $\mu V$ in the simulation. $\sigma_p$ is the standard deviation of the plateau part of the liner roughness. The correlation then defines a dependency of the hydrodynamic pressure $P_{OCR}$ on the oil control ring oil film thickness $h_{OCR}$.

For the profiled ring, two modifications are added onto the TLOCR correlation. Apart from the dependency of the hydrodynamic pressure $P_{prof}$ on the minimum oil film thickness $h_{prof}$ in the similar fashion, the oil supply here defined by $h_{OCR}$ also has an effect. The term $\left(\frac{h_{OCR}}{h_{prof}}\right)$ is defined as a filling factor and its influence is captured by the
power constant $K_p$. Since the pressure decay rate changes with the oil supply level, $K_p$ is not a constant and depends on liner finish, ring profile as well as oil supply $h_{OCR}$. For a particular liner finish and ring profile, $K_p$ only depends on $h_{OCR}$ and the dependency can be correlated in a linear fashion (a first order approximation). The second modification is the inclusion of a factor $a_p$ that captures the ring profile effect on hydrodynamic pressure generation. Given a liner finish and ring face profile, $a_p$ will be a constant.

Equation 5.3 is the correlation for hydrodynamic shear stress. Similar to the hydrodynamic pressure correlation, two terms are added into the basic shear stress formula to account for the ring face profile effect and oil supply effect. On top of the very basic hydrodynamic shear stress formula of $\frac{\mu V}{h_{prof}}$, $F_0$ is added as a constant to account for the ring face profile effect and $K_f$ is added to address the oil supply effect. For a given liner finish and ring face profile, $F_0$ would be a constant, while $K_f$ only depends on $h_{OCR}$ and the dependency can be correlated in a linear fashion (a first order approximation).

The correlations for the profiled ring are more complicated than those for the twin-land oil control ring. For a given ring profile and liner finish, the hydrodynamic pressure and shear stress have nonlinear dependency on both the minimum oil film thickness and oil supply, which is defined by the oil control ring and is varying along the piston stroke. Fortunately we don’t need a universal correlation valid for the entire space spanned by those two inputs, but rather a correlation valid only in the vicinity of each oil supply level (see Fig.5.11), since, as we argued before, this is the area where the ring can land and this is all we care about. In this small slot of interest, we can apply the first order approximation and the correlations developed so take the form of equation 5.2 and 5.3. The next section would discuss some typical values of the correlation coefficients.
5.2.2 Asperity Contact Correlation

The asperity contact correlation for the profiled ring is merely the contact model used in the twin-land oil control ring model integrated on the entire ring profile. One can check section 3.2.2 for details of the contact model.

5.2.3 Cycle Calculation

With the correlation in hydrodynamic pressure and asperity contact, the cycle friction calculation can be incorporated into a dynamic model that takes care of the load balance and ring dynamics. Tian's thesis [46] has more details in this.

5.3 Model Results

In traditional efforts of ring pack friction modeling, hydrodynamic pressure on a profiled ring face is usually evaluated with the original Reynolds equation or similarly average Reynolds equation, which is a modification of the original Reynolds equation by inclusion of roughness modifiers [44]. In this section, the model results using the deterministic based method introduced in this chapter and the results using the original
Reynolds are compared and the differences are analyzed and discussed. One needs to be aware that all the deterministic results shown in this section are liner specific.

For hydrodynamic pressure generation on the profiled ring face, the oil supply condition is critical (See Fig.5.12).

$$h_{prof} > h_{OCR}$$

$V$ $V$

$$h_{prof} < h_{OCR}$$

$V$ $V$

$$h_{OCR}$$

Oil $h_{prof}$

Oil $h_{prof}$

Fig.5.12 Two Situations

In the case when the minimum oil film thickness is higher than the oil supply level $h_{prof} > h_{OCR}$, Fig.5.14 shows the comparison between the hydrodynamic pressure evaluated with both original Reynolds equation and the deterministic method. The oil supply for the two methods here are the same. While the hydrodynamic pressure would be generated on the inter-asperity level when using the deterministic method, the original Reynolds equation, as would also the average Reynolds equation, predicts zero hydrodynamic pressure generation between the ring face and the liner finish. This is a common conclusion that original Reynolds equation or average Reynolds equation does not allow hydrodynamic pressure generation when the oil supply level is lower than the minimum oil film thickness, since any pressure generation would create a pressure gradient that pushes more oil to pass the minimum oil film thickness location (Fig.5.13).
Fig. 5.13 Oil Flux at the Minimum Oil Film Thickness Location

\[
\text{Flux} = \frac{1}{2} V h_{\text{min}} - \frac{h_{\text{min}}^3}{12 \mu} \frac{\partial P}{\partial X} > \frac{1}{2} V h_{\text{min}}
\]
In the case when the minimum oil film thickness is lower than the oil supply level $h_{prof} < h_{OCR}$, Fig. 5.15 shows the comparison between the hydrodynamic pressure evaluated with both the original Reynolds equation and the deterministic method. In this case, both methods would predict hydrodynamic pressure generation and most of the times the magnitudes are comparable. However, pressure would be built only in part of the ring face area depending on the value of minimum oil film thickness and oil supply according to the original Reynolds equation, while with the deterministic method, hydrodynamic pressure would usually be built across the entire calculation domain due to its assumption of the full attachment of oil and inter-asperity pressure generation. This creates more fuss when calculating the hydrodynamic friction. By assuming full attachment in the entire calculation domain, the deterministic method usually predicts
higher hydro friction than the original Reynolds equation, for which only the wetted area contributes to hydrodynamic friction.

Fig.5.15 Hydro Pressure Generation with Two Different Methods, $h_{prof} < h_{OCR}$

In a motoring condition, where the in-cylinder gas pressure is ambient, the friction performance of the top two rings depends the most on the oil supply, which is defined by the oil control ring. Table 5.1 lists the engine specification defined in the following two model cases. The liner finish used in the model and some roughness statistics are shown in Fig.5.16 and Table 5.2.

Table 5.1 Engine Specifications

<p>| Bore (mm) | 82.5 |</p>
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stroke (mm)</td>
<td>92.8</td>
</tr>
<tr>
<td>Engine Speed (rpm)</td>
<td>1000</td>
</tr>
<tr>
<td>Oil Dynamic Viscosity (pa.s)</td>
<td>9.945E-003</td>
</tr>
<tr>
<td>Boundary Friction Coefficient</td>
<td>0.1</td>
</tr>
<tr>
<td>Connection Rod Length (mm)</td>
<td>143.8</td>
</tr>
<tr>
<td>Top Two Ring Face Factor (m⁻¹)</td>
<td>10</td>
</tr>
<tr>
<td>Top Two Ring Calculation Domain (mm)</td>
<td>0.2</td>
</tr>
</tbody>
</table>

Top Two Ring Correlations

\[
P_{\text{prof}} = \left( \frac{h_{\text{OCR}}}{h_{\text{prof}}} \right)^{K_P} \frac{\mu V}{(\mu V)_0} \left( a_P P_{0,\text{OCR}} \right) \frac{h_{\text{prof}}}{\sigma_p}^{-K_{OCR}}
\]

\[
f = F_0 \frac{\mu V}{h_{\text{prof}}} \left( \frac{h_{\text{OCR}}}{h_{\text{prof}}} \right)^{K_f}
\]

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( P_{0,\text{OCR}} ) (pa)</td>
<td>3.1715e+007</td>
</tr>
<tr>
<td>( \sigma_p ) (micron)</td>
<td>0.071</td>
</tr>
<tr>
<td>( K_{OCR} )</td>
<td>2.5909</td>
</tr>
<tr>
<td>( K_P )</td>
<td>-1.2918 + 1.1080 (\frac{h_{\text{OCR}}}{\sigma_p})</td>
</tr>
<tr>
<td>( a_P )</td>
<td>0.3879</td>
</tr>
<tr>
<td>( F_0 )</td>
<td>0.6427</td>
</tr>
<tr>
<td>( K_f )</td>
<td>0.2655 + 0.0311 (\frac{h_{\text{OCR}}}{\sigma_p})</td>
</tr>
</tbody>
</table>
In the case when the oil control ring has a low unit pressure (0.2mm land width, 9N tension, 10.9 bar unit pressure), thus the oil supply is high, the ring can take advantage of the entire contact area for lubrication. Here it is assumed that the contact area has a limited size that is defined by the wear (Fig.5.17) and lubrication is limited to occur only
in the contact area. Outside of the contact area the face slope is too high and there never is enough oil to wet that area. In this case, when both of the two methods are taking full usage of the entire contact area, the two methods predict comparable minimum oil film thicknesses (Fig.5.18) and frictions (Fig.5.19). Here the same asperity contact model and boundary friction coefficient are used for total friction calculation with the two hydrodynamic sub-models. The oil supply level (OCR) is also plotted in Fig.5.17 and note that the minimum oil film thickness calculated with the original Reynolds equation is always lower than the supply level since that is the only situation where hydrodynamic pressure can be built. Above the travel area of the oil control ring, the oil supply for the second ring is assumed to be a constant level, the same as the oil film thickness at the TDC location of the oil control ring. In this case the original Reynolds equation predicts higher minimum oil film thickness and slightly lower friction. However this is not universal, but rather ring face profile and liner finish dependent.

Fig.5.17 Contact Area
Fig. 5.18 Oil Film Thickness (High Oil Supply)

Fig. 5.19 Friction (High Oil Supply)
Another extreme case is when the oil control ring has fairly high unit pressure (0.06mm land width, 20N tension, 80.7 bar unit pressure), thus the oil supply would be quite low. The original Reynolds equation stills predicts lower minimum oil film thickness except in the area of no wetting where the entire load is sustained by the asperity contact force (Fig.5.20). The hydrodynamic pressure is only generated in a small wetting area near the minimum oil film thickness location and boundary fiction is significant (Fig.5.21). The deterministic method however would consider the inter-asperity hydrodynamic pressure and still utilize the entire contact area. Therefore the average hydrodynamic pressure would be much higher and the minimum oil film thickness is also higher, which leads to a much lower boundary friction prediction. However since it assumes full attachment in the entire contact area, the hydrodynamic shear stress is generated in a much larger area which causes a much higher hydrodynamic friction (Fig.5.21). As a consequence, the overall friction levels are still comparable, only the compositions of boundary and hydrodynamic friction are different.

![Graph of Oil Film Thickness](image)

**Fig.5.20 Oil Film Thickness (Low Oil Supply)**
5.4 Conclusion

In this chapter, a deterministic based hydrodynamic model is introduced to evaluate the average hydrodynamic pressure and shear stress between the profiled top two ring face and the liner finish. Hydrodynamic pressure and shear stress are correlated with the minimum oil film thickness, oil supply, oil dynamic viscosity and ring sliding speed, given a liner finish and ring face profile. Those correlations are applied in the cycle model together with an appropriate asperity contact model to predict the cycle friction behavior of the top two rings. Some interesting model results and a comparison between the deterministic method and the traditionally used original Reynolds equation is discussed in the model result section. A major advantage of the model is that it can reflect the liner finish effect, and therefore can be used to compare frictions of different liner finish with the same ring design. This is not possible with the traditional methods.
We have to bear in mind that a few important assumptions and simplifications are made in the model.

First of all, the gas pressure effect on the hydrodynamics and the ring friction is neglected. This is saying the model is really just evaluating the top two ring friction in a motoring condition. The argument here is that the method is mainly developed to reflect the liner finish effect on friction, and when the gas pressure is high, the liner finish does not make a difference since the unit load is too high and the ring friction is mainly boundary. One might argue that even the boundary friction can be somehow affected by the liner finish. But that is certainly beyond the capability of this model. For the firing condition, what is really interesting is to find a way to bridge the boundary friction in the high gas pressure area with the friction in the low gas pressure area evaluated by the deterministic model. And that is one of the suggested future studies on this topic.

A second major assumption is on the calculation domain. Unlike the original Reynolds equation, with which the wetting area can be determined by some physical conditions, the deterministic method adopts arbitrary size of the full attachment area defined by the calculation domain. By applying a larger and larger calculation domain, one is effectively assuming a larger and larger wetting. A larger calculation domain usually means higher hydrodynamic lift since hydrodynamic pressure might still be generated sporadically between the roughness asperities even when the nominal clearance is high. However, it also means more hydrodynamic friction due to the increase of area in attachment. The true physics here is not quite clear and can be quite complicated. One may develop a way to determine the proper choice of calculation domain according to some physical rule. This may be another direction for the future work in this area.
6 A Simple Wear Model for Liner Topology Evolution

This chapter introduces a simple wear model for liner topology evolution. It starts with a
general discussion of the importance and background for such a model, followed by the
section to introduce the methodology, and ends with discussion of the model results and
some useful indications of the model.

6.1 General Introduction

The surface topology of liner finish changes with time due to solid-to-solid interaction
between piston rings/skirt and liner finish. In order to correctly account for the effect of
liner roughness in friction and oil consumption, it is critical to understand the liner
topology evolution in the break-in/wear process.

In this study, a simple wear model is proposed to simulate the roughness geometry
development in the break-in/wear process. The model applies an artificial asperity
interaction between the liner finish and an artificial grinder surface with certain
roughness to predict the worn liner roughness profile. By correctly choosing the grinder
roughness, one can match the height distribution of the predicted liner roughness to the
measured worn surface. The model is used on a measured roughness sample in their new
state to predict the broken-in/worn roughness profile. The deterministic model is then
used to compute the hydrodynamic pressure and shear stress between the twin-land oil
control ring and the predicted/measured worn liner finishes.

The result shows good correspondence in hydrodynamic performance between the
predicted and the measured worn liner finish, once the height distributions of the two
surfaces are matched.

6.2 Methodology

A grinder line is constructed as a white noise profile (a sequence of an independent
identically distributed Gaussian random variable) in the circumferential direction in the
same resolution of the liner surface measurement. The grinder line slides at a constant
nominal height above the liner finish (Fig.6.1). An uncertain proportion of each
interacting peak on the plateau is removed. The proportion to remove is chosen to be
follow a uniform distribution between 0 and 1, U[0 1]. The purpose of this randomness is to mimic the varying local sustainability to wear depending on the asperity size. In each grinding stroke a new random grinder surface is generated, which reflects the fact that liner finish interacts with different part of rings and particles in each stroke. The nominal height \( h \) of the grinder surface is determined as a constant value multiplied by the combined roughness of the grinder surface \( \sigma_g \) and the plateau part of the liner finish \( \sigma_p \).

\[
\sigma_{\text{combined}} = \sqrt{\sigma_p^2 + \sigma_g^2}
\]

This constant value is chosen to be small enough (e.g. 2) so that each stroke certain proportion of the liner asperities are removed. In this way, one can generate an artificial worn surface and the worn surface statistics can be controlled by manipulating the roughness of the grinder line and the number of strokes for grinding.

Fig.6.1 Grinder Line on the Surface Measurement
6.3 Discussion

Fig. 6.2 shows the surface topology of a sample new liner. During the break-in/wear process, the plateau part of the liner would evolve into some scratchy pattern while the valley part (deep grooves) will remain constant. The effect can be observed from the surface topology of the measured worn profile in Fig. 6.3. Fig. 6.4 shows the comparison of the surface height distribution density of the new and worn surfaces. From Fig. 6.4 we can see that the plateau area of the surface experiences a major change in height distribution while the valley part height distribution remains unchanged.

![Fig. 6.2 Surface Topology of a Sample New Liner](image-url)
Fig. 6.3 Surface Topology of the Measured Worn Liner

Fig. 6.4 Surface Height Distributions (New and Worn)
By applying the wear model introduced in the previous section, we can generate an artificial worn surface based on the new surface. With certain input parameters we can closely match the height distributions of the two worn surfaces (measured and artificial). Fig. 6.5 shows the surface topology of the artificial worn surface. A closer comparison between the worn and the artificial worn surface topology can be seen in Fig. 6.6. Visually one can tell that both surfaces exhibit a streaky pattern of wear marks in the ring-sliding direction. Fig. 6.7 shows the comparison of the height distributions of the three surfaces (new, worn and artificial worn). There are minor differences in the height distribution of the two worn surfaces, mainly due to the non-Gaussian distribution of the actual worn surface. By applying a heavier tail distribution to the grinder line, one can match the two height distributions better. However for simplicity this is not considered in this study.

![Fig. 6.5 Surface Topology of the Artificial Worn Surface](image)

- color scale length unit (μm)
  - 1400
  - 1200
  - 1000
  - 800
  - 600
  - 400
  - 200
  - 1000 1500 2000

- Axial direction (μm)
  - 0
  - 500
  - 1000
  - 1500
  - 2000

- Circumferential direction (μm)
  - 0
  - -13

Fig. 6.5 Surface Topology of the Artificial Worn Surface
Fig. 6.6 Worn and Artificial Worn Surface Topology

Fig. 6.7 Surface Height Distributions (New, Worn and Artificial Worn)

Fig. 6.8 shows the average hydrodynamic pressure and shear stress evaluated with the deterministic method. The method computes the flow field between a flat ring face and the liner finish at a fixed nominal clearance. As discussed in the previous chapters, this is the case when hydrodynamic pressure is generated among the roughness asperities and
the roughness alone is responsible for supplying hydrodynamic support. Therefore it is a good measure for roughness effect. The figure shows identical hydro pressure and shear stress generation at different nominal clearances for the two worn surfaces (measured and artificial). The figure also includes the curves of the correspondent measured new liner finish, and two other surfaces. The name ‘flat plateau’ features a surface with identical valleys and perfectly smooth plateau, and the name ‘new shrunk plateau’ features a surface with identical valleys and the plateau part shrunk from the original new surface to match the height distribution of the worn surface. These two surfaces are plotted in Fig.6.9 and Fig.6.10. All the results in Fig.6.8 are computed at a same constant oil viscosity and ring land width and ring sliding speed to exclude the relevant influence.

![Graph showing hydro pressure and shear stress vs. nominal clearance for different surfaces](image)

**Fig.6.8 Average Hydrodynamic Pressure and Shear Stress**
Fig. 6.9 Surface Topology of the 'Flat Plateau' Surface

Fig. 6.10 Surface Topology of the 'New Shrunken Plateau' Surface
The results presented in Fig. 6.8 are quite informative. The hydrodynamic pressure and shear stress between a flat ring profile and the liner finish is determined by the geometry of valley and plateau part of the liner finish. Among the five surfaces presented in Fig. 6.8, four of them, except the new one, share identical valley geometry and three of them share identical height distribution. The only significant difference in surface geometry among the worn, artificial worn and the shrunk new plateau surface is the topology of the plateau (see Fig. 6.11 for a comparison). The hydrodynamic pressure distribution for all the five surfaces can be seen in Fig. 6.12. The 'flat plateau' features no geometry on the plateau. The hydrodynamic pressure generated in the valley is maintained on the plateau and there is no further pressure built, therefore it defines a lower limit for the plateau geometry effect. The two worn surfaces (measured and artificial) feature the 'streaky' pattern in the sliding direction. The roughness on the plateau helps to further build the hydrodynamic pressure, although the orientation of the structure on the plateau diminishes the hydro pressure generation. Therefore these two surfaces exhibit slightly better hydro pressure generation capacity. The fact that the two worn surfaces share identical performance indicates the success of the wear model in capturing the topology effect of the worn liner finish as far as the hydrodynamic performance is concerned. Lastly, the cross-hatch pattern on the plateau of the 'new shrunk plateau' surface is quite effective in generating hydrodynamic pressure. As a consequence, the surface shows the strongest capacity in hydrodynamic pressure generation among the four. The new surface generates highest hydrodynamic pressure at the same nominal clearance since it has a rougher and cross-hatch patterned plateau.
Fig. 6.11 Surface Topologies (Worn, Artificial Worn, Shrunk New Plateau, and Flat Plateau)
The wear model successfully captures the 'streaky' structure on the plateau of the measured worn surface. Next we will take a closer look at the artificial wearing process that the model applies. In each grinding stroke, the plateau roughness is cut by a random and different grinder surface and the surface geometry evolution can be captured by two variables: the height of the peak of height distribution $h_m$ and the standard deviation of the plateau roughness $\sigma_p$. The first variable $h_m$ represents the evolution of the mean of plateau, which is determined by the grinder roughness $\sigma_g$, the nominal gap between the grinder and the liner surface $h_c$, and the number of strokes cut by the grinder. With a given $\sigma_g$ (0.3 micron) and $h_c$ (2 times of the combined roughness $\sigma_{combined}$), the variation of $h_m$ with strokes for the generation process of the artificial worn surface is shown in Fig. 6.13. During the cutting process, the peak of the plateau height distribution first shifts up then shifts down and in the end is approximately 0.8 micron below the new surface. This can also be observed in the height distribution (Fig.6.7). The second variable $\sigma_p$ represents the transition from the plateau roughness of the new liner finish to a worn
plateau defined by the roughness of the grinder surface $\sigma_g$ and the nominal gap between the grinder and the liner surface $h_e$. Again with a given value for those two parameters, the variation of $\sigma_p$ is shown in Fig.6.14. Unlike $h_m$, $\sigma_p$ experiences a transitional phase and stabilizes at a constant value (approximately 0.09 micron). This value defines the 'broken-in' roughness $\sigma_p$ of the plateau and would not change with further increase of strokes.

![Fig.6.13 Variation of $h_m$ with Strokes](image-url)
The final $h_m$ determines the relative height of the plateau and valley. It is in general not quite important as long as the break-in process has finished, since the magnitude of $h_m$ is typically negligible comparing to the depth of valleys. On the contrary, the 'broken-in' roughness $\sigma_{ps}$ is more critical since it affects the hydro pressure build-up in the plateau area. Fig.6.15 shows the height distribution of a new liner finish and three artificial surfaces created based on the same new surface with different 'broken-in' roughness. These surfaces share the same valley and final $h_m$, as well as the 'streaky' pattern on the plateau. The hydrodynamic pressure and shear stress are shown in Fig.6.16. It is clear that the rougher the plateau, the higher the hydrodynamic pressure/stress.

At this point, without the measurement of the worn surface, the 'broken-in' roughness $\sigma_{ps}$ is unknown. One may speculate that $\sigma_{ps}$ should have some correlation with the
original plateau roughness, since the wearing process in reality is a combined effect of both ring/piston liner interaction and third body particle contact. The ring roughness formed in the break-in process is a direct consequence of its interaction with the original liner finish, and the third body particle is mostly broken-off liner asperities. Therefore the size of both should be linked to the original plateau roughness. However, $\sigma_{ps}$ may also be related to load and material. More data needs to be examined in order to obtain any further knowledge in this.

![Fig.6.15 Height Distribution of the New Liner Finish and Three Artificial Worn Surfaces](image)

- new
- $\sigma_{ps} = 0.136 \mu m$
- $\sigma_{ps} = 0.092 \mu m$
- $\sigma_{ps} = 0.052 \mu m$
6.4 Conclusion

This chapter presents a simple wear model that simulates the plateau roughness evolution of the liner finish during the break-in/wear process. The study shows the hydrodynamic performance of the worn liner finish is quite predictable with the wear model, as long as the plateau roughness scale of the worn surface is known. The ‘streaky’ pattern on the plateau of the worn surface is stronger than the perfectly smooth plateau, but weaker than the cross-hatch pattern of the same height distribution. However the roughness scale on the plateau is unknown in reality. Meanwhile, empirical study shows that different spot on the worn liner may exhibit different roughness scale. Therefore a robust and effective approach can be to assume a reasonable spread on the worn plateau roughness scale, and use the wear model and deterministic method to provide a range of prediction for liner performance after break-in.

The simple wear model assumes the liner finish would always be cut by a grinder surface. In reality this may not be true. If the original liner is effective in generating
hydrodynamic pressure, the hydrodynamic lift may be high enough and the ring liner contact may be avoided where the ring travel speed is sufficiently high. This eliminates one source of wear for liner finish. Admittedly there is still debris coming from the top dead center area that can scratch the liner. Whether a fully developed ‘streaky’ pattern would be formed on the plateau for that particular liner is debatable.
7 Model Validation

In this chapter, a floating liner engine is used to validate the model results. The chapter starts with a brief introduction of the floating liner engine set-up. The test conditions for the model validation are then discussed. The measured engine frictions are compared with the model and the results are analyzed.

In this chapter, all the experimental work and data analysis are finished by Kai Liao and Dallwoo Kim from MIT Sloan Automotive Lab.

7.1 The Floating Liner Engine

The floating liner engine can be used for instantaneous friction force measurement between the piston assembly and the liner finish [47]. For this study, a modified single-cylinder floating liner engine is used. Fig.7.1 and Fig.7.2 shows a picture of the floating liner engine and the liner block. Fig.7.3 shows the plot of floating liner engine cross-section. The liner is supported by two pressure sensors, four top lateral stoppers and a circular bottom lateral stopper. The axial force on the liner is mainly supported by the pressure sensors, thus the reading from the sensors reflects the friction force between the piston assembly and the liner finish. The engine specifications of the floating liner are listed in table 7.1. The coolant inlet temperature, oil temperature and liner temperature are measured and recorded.
Fig. 7.1 Floating Liner Engine

Fig. 7.2 Floating Liner Engine Liner Block
Table 7.1 Floating Liner Engine Specification

<table>
<thead>
<tr>
<th>Engine Type</th>
<th>Single-Cylinder Four-Stroke Gasoline Engine</th>
</tr>
</thead>
<tbody>
<tr>
<td>Displacement [L]</td>
<td>0.496</td>
</tr>
<tr>
<td>Bore × Stroke [mm]</td>
<td>82.5 × 92.8</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>10</td>
</tr>
<tr>
<td>Maximum BMEP [MPa]</td>
<td>0.7</td>
</tr>
<tr>
<td>Maximum Engine Speed [rpm]</td>
<td>3000</td>
</tr>
<tr>
<td>Balance Shaft</td>
<td>Primary and Secondary</td>
</tr>
<tr>
<td>Piston Cooling</td>
<td>Oil Jet</td>
</tr>
<tr>
<td>Cyl.Temp [deg C]</td>
<td>30–120</td>
</tr>
<tr>
<td>Oil Temp [deg C]</td>
<td>30–100</td>
</tr>
</tbody>
</table>

Fig. 7.3 Floating Liner Engine Cross-Section Plot
7.2 Test Conditions

In this work, the effort has been focused on the validation of the TLOCR friction model introduced in chapter 3. There are a few reasons for the choice. First the TLOCR is typically the most important source of friction for the piston ring pack. Second the TLOCR lubrication is liner micro geometry determined, thus any traditional method available is powerless. The validation of the top two ring friction models would be left for future work in this field.

In order to more accurately measure the TLOCR friction, the floating liner engine runs under open cylinder head, motored condition. This minimizes the side force on the piston skirt. Furthermore, an unusually small piston is adopted (100 micron minimum clearance in the skirt area) to reduce the friction from the piston skirt area. On the ring side, the top two rings are removed to eliminate their friction contribution. Therefore the friction force measured would be dominantly TLOCR friction with a small piston skirt’s contribution. The friction force of piston alone (with all rings disassembled) is also measured so that the TLOCR friction can be obtained through subtracting the piston friction from the friction of piston-TLOCR combo.

Four different TLOCRs and one liner finish are available for the test. The four rings correspond to the combination of two different worn-in ring face profiles and three different ring tensions. The average land widths of the ring face profile and the ring tensions are listed in table 7.2.

<table>
<thead>
<tr>
<th>Type</th>
<th>Ring Land Width (mm)</th>
<th>Ring Tension (N)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A1</td>
<td>0.156</td>
<td>10.8</td>
</tr>
<tr>
<td>A2</td>
<td>0.156</td>
<td>18.5</td>
</tr>
<tr>
<td>A3</td>
<td>0.156</td>
<td>28.1</td>
</tr>
<tr>
<td>B1</td>
<td>0.23</td>
<td>10.8</td>
</tr>
</tbody>
</table>
The land widths are calculated as the average of the ring face profile measurement taken at nine different circumferential locations and both the upper and lower land. The worn-in profiles of the type A rings are shown in Fig.7.4-7.8.

Fig.7.4 is the worn-in land width measurement of the type A rings. The corresponding average land widths in table 7.2 are calculated as the average of the 18 values. Fig.7.5 shows the type A ring worn-in contact mark on the whole circumference. The even black wear mark shows that the ring has been properly worn-in. Fig.7.6-7.8 show the face profile of type A ring at nine circumferential locations. It is clear that the ring in general exhibits a flat profile.

![Profile measurement (OD face contact after test)](image)

<table>
<thead>
<tr>
<th>Angle</th>
<th>340°</th>
<th>315°</th>
<th>270°</th>
<th>225°</th>
<th>180°</th>
<th>135°</th>
<th>90°</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gap</td>
<td>0.42</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.18 mm</td>
<td>0.21 mm</td>
<td>0.19 mm</td>
<td>0.15 mm</td>
<td>0.15 mm</td>
<td>0.13 mm</td>
<td>0.17 mm</td>
<td>0.15 mm</td>
</tr>
<tr>
<td>0.16 mm</td>
<td>0.18 mm</td>
<td>0.16 mm</td>
<td>0.15 mm</td>
<td>0.15 mm</td>
<td>0.13 mm</td>
<td>0.17 mm</td>
<td>0.15 mm</td>
</tr>
<tr>
<td>0.10 mm</td>
<td>0.11 mm</td>
<td>0.10 mm</td>
<td>0.10 mm</td>
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<td>0.10 mm</td>
<td>0.11 mm</td>
<td>0.10 mm</td>
<td>0.10 mm</td>
</tr>
</tbody>
</table>

Fig.7.4 Type A Ring Worn-in Land Width Measurement
Fig. 7.5 Type A Ring Worn-in Contact on the Whole Circumference

Nb. Tension Load: 10.5N Gap: 0.42

Fig. 7.6 Type A Ring Worn-in Face Profile at Nine Different Circumferential Locations
Fig. 7.7 Type A Ring Worn-in Face Profile at Nine Different Circumferential Locations (Continue)
One liner finish is available for testing. Fig. 7.9 shows the liner finish measurement.

Some selected standard roughness statistics of each surface is listed in table 7.3. Here \( \sigma_p \) is the roughness standard deviation of the plateau part of the surface.
Liner temperatures are measured by the thermocouple shown in Fig.7.3. In the experiment, three different liner temperatures (60 C, 80 C and 100 C) are tested. SAE 5W30 fully formulated (with additives) oil is used for the entire test.
7.3 Result Analysis

Fig. 7 shows four friction traces, the measured friction of TLOCR-piston combo, the measured friction of piston alone, the measured friction of TLOCR as the difference between the previous two, and the modeled TLOCR friction. A1 type of ring (0.156 mm ring land width, 10.8 N ring tension) is used in the experiment and liner temperature is controlled at 100°C. The engine speed is 100 rpm. In the TLOCR model, a ring land width of 0.15 mm is used and a simple contact model is applied. The simple contact model is based on the work of Hu, Cheng, 1993 [48]. The contact pressure takes the following form:

\[
P_{\text{contact}} = \begin{cases} 
P_c \left( z - \frac{h}{\sigma_p} \right)^{-K_c} & \text{for } \frac{h}{\sigma_p} < z \\ 0 & \text{for } \frac{h}{\sigma_p} \geq z \end{cases}
\]

The \( P_c, z, \) and \( K_c \) are three constants. The original model is based on the assumption of Gaussian roughness height distribution. In reality the worn-in surfaces are non-Gaussian. Therefore for each liner surface measurement, the coefficient \( z \) is adjusted to match the model FMEP at the lowest engine speed (100 rpm) with the experiment result. Instead of using \( z = 4 \) as in the original model, \( z \) is adjusted to be 4.8 for the liner surface in the test. This makes the engagement of asperity contact at a higher oil film thickness. The contact friction takes the following form:

\[
f_{\text{contact}} = P_{\text{contact}} C_{fc}
\]

in which \( C_{fc} \) is the boundary friction coefficient and is 0.16 for this liner. For the rest of the study, the contact model is fixed for this liner finish.

Fig. 7.11 and 7.12 show the same friction comparison at 500 rpm and 1000 rpm engine speeds. The model captures the lubrication regime indicated by the experiment results. At 100 rpm, the ring is boundary/mixed lubrication. 500 rpm corresponds to a mixed lubrication regime, and at 1000 rpm, friction starts to take off in the mid-stroke, indicating a hydrodynamic lubrication regime. The model and experiment results match.
better with each other at lower engine speeds and when engine speed grows higher, there
starts to be some deviation in friction magnitude between the model and experiment
results.

![Friction Comparison Graph](image)

**Fig. 7.10 Friction Comparison for Al ring, 100 C Liner Temperature and 100 rpm Engine Speed**
Fig. 7.11 Friction Comparison for A1 ring, 100 C Liner Temperature and 500 rpm Engine Speed

Fig. 7.12 Friction Comparison for A1 ring, 100 C Liner Temperature and 1000 rpm Engine Speed
Fig. 7.13 shows the comparison between the experiment and modeled instantaneous friction coefficient vs. the none-dimensional parameter of $\mu V / (T / R)$. The none-dimensional parameter, defined as oil dynamic viscosity multiplied to instantaneous ring sliding speed and normalized by the normal ring load per unit length in circumferential direction, is an indicator of lubrication regime. Large values correspond to hydrodynamic regime and small values correspond to boundary lubrication regime. The data points in Fig. 7.13 correspond to instantaneous friction values in measurements of different engine speeds, ring tensions and liner temperatures. Fig. 7.13 covers the type A ring (0.156mm land width) and three different temperatures (60 C, 80 C, and 100 C).

Both the experiment and model results indicate a common stribeck behavior for the friction. The transition from boundary through mixed to hydrodynamic lubrication regime is clear with both experiment and model results. In general, it is a good match between the experiment and model results, except in the deep hydrodynamic regime.

The ring tension effect can be seen in Fig. 7.14 and 7.15. Both the model and experiment show a common hydrodynamic behavior for different ring tensions in hydrodynamic regime, while in mixed and boundary lubrication regime, lower ring tension leads to a slightly higher friction coefficient. This effect can be explained using the model:

In the hydrodynamic regime, the friction coefficient is roughly

$$C_f = C_{f,\text{hydro}} = \frac{\mu V}{h} \left( C_{f1} + C_{f2} \exp \left( -C_{f3} \frac{h}{\sigma_p} \right) \right) = \frac{\mu V_0}{\mu_0 V_0} \left( \frac{h}{\sigma_p} \right)^{-K_s} \left( C_{f1} + C_{f2} \exp \left( -C_{f3} \frac{h}{\sigma_p} \right) \right)$$

Here the friction coefficient is a unique function of the oil film thickness $h$, which can be determined by the none-dimensional parameter $\mu V / (T / R)$.

On the other hand, in the boundary and mixed lubrication regime, the friction coefficient is
\[ C_f = \frac{f_{\text{hydro}} + f_{\text{contact}}}{P_{\text{hydro}} + P_{\text{contact}}} = \frac{\mu V}{h} \left( C_{f1} + C_{f2} \exp \left( -C_{f3} \frac{h}{\sigma} \right) \right) + C_k P \left( z - \frac{h}{\sigma} \right)^{-k_c}. \]

It is a function of both oil film thickness \( h \) and \( \mu V \). Therefore the friction coefficient in the boundary and mixed lubrication regime needs to be determined by two non-dimensional parameters, and using \( \mu V / (T / R) \) alone is not enough.

Fig.7.16 shows the stribeck curve for the B1 type of ring (0.23mm land width and 10.8N ring tension) at two different liner temperatures. A similar discrepancy between the experiment and model results can be observed in the hydrodynamic regime.

**Fig.7.13 Instantaneous Stribeck Curve for Type A Ring, with Different Ring Tensions, at Different Liner Temperatures (Experiment and Model)**
Fig. 7.14 Instantaneous Stribeck Curve for Type A Ring, with Different Ring Tensions, at Different Liner Temperatures (Experiment)

Fig. 7.15 Instantaneous Stribeck Curve for Type A Ring, with Different Ring Tensions, at Different Liner Temperatures (Model)
The deviation between the model and experiment results in the hydrodynamic lubrication regime can be explained by many reasons both on the model side and the experiment side. On the model side, the following factors can contribute to the deviation:

1. Viscous heating
2. Uncertainty in shear thinning behavior at low temperatures
3. Liner finish measurement size limitation
4. Thermal layer on the liner

On the experiment side, the dynamic behavior of the lateral stoppers can be a major cause.

The oil temperature used in the model is the liner temperature measured by a thermocouple touching the exterior of the liner sleeve (Fig.7.3). In reality the friction work can heat up the oil in the ring-liner interface and the oil viscosity in the model is overestimated. The effect of the viscous heating can be roughly estimated in the following way.
Consider a simplified set-up in Fig. 7.17. Both the ring face and the liner is assumed to be perfectly flat with a gap \( h \) in between. Since the deviation is only in the hydrodynamic regime, we only need to consider the hydrodynamic friction here.

![Fig. 7.17 Set-up for Viscous Heating Effect Approximation](image)

The heat transfer in the flow field is governed by the following equation:

\[
\rho C_p \left( \frac{\partial T}{\partial t} + u(y) \frac{\partial T}{\partial x} \right) + k \frac{\partial^2 T}{\partial y^2} = \mu \left( \frac{\partial u}{\partial y} \right)^2
\]

Here \( \rho \) stands for the oil density; \( k \) stands for the heat conductivity of the oil; and \( \mu \) stands for the oil viscosity.

The boundary condition can be approximated as

\[
\begin{align*}
\left. \frac{\partial T}{\partial y} \right|_{y = h} &= 0 \\
T(y = 0) &= T(x = 0) = T_{\text{liner}}
\end{align*}
\]

by considering the fact that the ring face has a coating and keeps being heated, while the liner surface is constantly refreshed and the heat conductivity of the cast iron is 500 times of the oil.
Comparing the order of magnitude of the conduction and convection terms with the following typical values: $K = 0.14 \,[\text{W/m-K}]; \, L_w = 0.2\times10^{-3} \,[\text{m}]; \, \rho = 850 \,[\text{kg/m3}]; \, C_p = 2000 \,[\text{J/kg-K}]; \, U \sim 10 \,[\text{m/s}]; \, h \sim 0.5\times10^{-6} \,[\text{m}].$

$$\frac{k \frac{\partial^2 T}{\partial y^2}}{\rho C_p \left( \frac{\partial T}{\partial t} + u(y) \frac{\partial T}{\partial x} \right)} \sim \frac{k L_w}{\rho C_p U h^2} \sim 6.5$$

Therefore as a simplification, we can drop the convection term and only consider the conduction effect. We will have

$$\Delta T \sim \frac{\mu U^2}{k}$$

Here $\Delta T$ is the temperature rise of the oil due to viscous heating. The viscous heating effect is strong when liner temperature is low (viscosity is high) and piston speed is high. In the experiment the lowest liner temperature is 60 $^\circ$C and the highest engine speed is 1000 rpm. In such condition, the viscous heating effect only causes approximately 5% of drop in oil viscosity. Therefore the effect is trivial.

The shear thinning behavior of the oil used in the model is measured at high oil temperatures (140 $^\circ$C). The low temperature (as in the experiment) shear thinning behavior of the oil is assumed to be the same as under high temperatures. This may cause some overestimation of the oil viscosity and may partially explain the deviation.

The liner finish measurement used in the model is a very small patch. Whether this small patch represents the behavior of the entire cylinder bore especially after break-in is another uncertainty which may explain the deviation too.

The model relies on the deterministic surface measurement. However the anti-wear thermal layer formed due to the chemical additives in the lubricant oil can alter the surface micro-geometry and thus change the friction behavior of the ring liner interaction. This effect can be especially significant for smoother and finer liners, and thus might explain the deviation.
From the experiment side, the lateral stoppers (Fig. 7.3) which are supposed to only constrain the lateral motions of the floating liner and sustain the side forces may also take some axial force due to the dynamic behaviors of an over-determined system. This effect can be especially strong when the engine speed is high and the piston lateral force is large. In Fig. 7.18, the friction bump in the down stroke from the experiment data is quite strange. It is fairly difficult to understand from a model point of view why ring friction would behave so oddly.

Fig. 7.18 Friction Comparison for Al ring, 100 C Liner Temperature and 1000 rpm Engine Speed

7.4 Conclusion

In this chapter, the TLOCR model results are compared to the floating liner experiment results in a motoring test condition. Both the experiment and model results indicate a common Striebeck behavior for the friction. The transition from boundary through mixed to hydrodynamic lubrication regime is clear with both experiment and model results. In
general, it is a good match between the experiment and model results, except in the deep hydrodynamic regime. Several explanations and hypothesis are proposed to explain the deviation between the model and experiment in the hydrodynamic regime. Further study is needed to fully understand the phenomenon.
8 Conclusion

This chapter discusses the conclusion on this thesis work. The first part summarizes the entire thesis work, the second part draws some general conclusions and the third part proposes some potential future work following this thesis work.

8.1 Summary

The major content of this thesis is to develop a complete piston ring pack friction prediction model that incorporates the effect of both liner roughness micro geometry and ring face profile macro geometry. The liner finish micro geometry effect is evaluated with the deterministic model discussed in chapter 2, based on a 3D patch measurement of the liner surface.

The twin-land oil control ring model (TLOCR) is developed first. TLOCR is critical for both ring pack friction and oil consumption. Since the ring face usually exhibits a flat face profile after running-in, the traditional methods based on macro profiles predicts no lift from the hydrodynamic pressure. However, with the deterministic method based model discussed in chapter 3, the hydrodynamic pressure from inter-asperity flow can provide significant support for lubrication. This is verified by the experimental work by Kai Liao and Dallwoo Kim from MIT. The comparison between the model and experiment friction results is presented in chapter 7.

Chapter 4 discusses the effect of curved ring face profile under the flooded oil supply condition. The curvature of the face profile is a major feature of the three-piece oil control ring (TPOCR) that differentiates its behavior from TLOCR. In chapter 4, the friction performance of the TPOCR is compared to the TLOCR through deterministic modeling under the constraint on oil control. The study is based on one particular liner finish. There is no advantage in oil control ring friction to have a curved oil control ring face profile for that liner given the constraint on oil control.

The top two ring model discussed in chapter 5 considers both the ring profile effect and the liner finish micro geometry effect. This is important for top two ring friction modeling because the ring has a macro profile that would potentially enhance the
hydrodynamic pressure generation. However, since the oil supply to the top two rings is controlled by the oil control ring and is in the inter-asperity level, the macro profile effect is weakened and inter-asperity behavior with cavitation is important. The deterministic based model in chapter 5 covers both effects and uses the oil film left by the oil control ring as the oil supply for the top two rings.

The ring pack friction models developed in this thesis adopt a deterministic 3D liner finish measurement as input for liner finish micro geometry characteristics. However liner finish surface geometry changes with time due to wear, especially in the break-in process. This suggests that the friction prediction might be valid for only a short period of time. To address this issue, chapter 6 presents a wear model that can be used to simulate the surface geometry development in the wear process. Potentially the wear model, together with the ring pack models, can help to predict the long term friction behavior of the piston ring pack.

8.2 Conclusion

A few general conclusions can be drawn based on this thesis work:

1. The oil film thickness of the ring lubrication is in the order of surface roughness and therefore the deterministic method is needed to provide proper understanding.

2. The liner roughness micro geometry is critical for both TLOCR and top two ring lubrication. For TLOCR, due to the lack of curvature on the ring face profile, the liner roughness topology is the only geometry that generates hydrodynamic support. For top two rings, due to the roughness level oil supply, the liner roughness topology is as important as macro ring face profiles in hydrodynamic pressure generation.

3. Proper correlations can extend local deterministic calculation to whole cycle so that the integral effects and optimization of liner finish, ring pack, and lubricant designs can be performed in an efficient manner. While the correlations for the TLOCR is fairly straightforward, more complicated forms have to be deployed for the top two rings as more parameters on the rings as well as the oil supply condition are involved.
4. Comparison between the measurements and calculation showed promising sign of the method developed in this work.

5. The liner finish changes with engine running time, which affects the ring liner lubrication. It seems that a simple wear model is able to capture the change of hydrodynamic lubrication from the liner wear.

8.2 Potential Future Work

One major topic of potential works lies in the field of thermal layer. The models presented in this work rely on the deterministic surface measurement. However the anti-wear thermal layer formed due to the chemical additives in the lubricant oil can alter the surface micro-geometry and thus change the friction behavior of the ring liner interaction. This effect can be especially significant for smoother and finer liners, and thus might explain the deviation between the model and experiment results in the hydrodynamic regime presented in chapter 7.

Another potential topic is the asperity contact model. Currently as can be seen in the model validation work, the calibrated contact model usually deviates from the Greenwood and Tripp model, or any simplified model derived from it. The selection of a more reliable contact model is of great importance for precise modeling of ring pack friction.

There are a few uncertainties in the current top two ring model. The gas pressure effect is not well accounted in the deterministic method. The calculation domain is arbitrarily selected. Since the full attachment assumption is adopted in the model, this can arbitrarily over-estimate the friction force if the domain is too large. The solution can be to introduce a better domain selection criterion, or use a more sophisticated cavitation assumption that models the attachment and detachment. The model results should also be validated with the floating liner engine or other friction measurement equipment.

The wear model proposed in chapter 6 also needs further study. The major concern is the determination of the grinder surface characteristics. Currently the model is based on an arbitrary grinder surface. This is sufficient if only a range of the worn liner characteristics is needed since one can always use different grinder and get a certain
range of worn liner statistics. However if more precise prediction is needed, one would have to model the grinder characteristic with some explanatory variable.
References


