Developing an Approach Utilizing Local Deterministic Analysis to Predict the Cycle Friction of the Piston Ring-pack in Internal Combustion Engines

by

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Abstract

Nowadays, a rapid growth of internal combustion (IC) engines is considered to be a major contributor to energy crisis. About 20% of the mechanical loss in internal combustion engines directly goes to the friction loss between piston ring pack and liner finish. A twin-land oil control ring (TLOCR) deterministic model was developed by Chen et al. and it helps the automotive companies investigate the effects of liner finish, rings, and lubricants on friction and oil control of the TLOCR [2]. This work focuses on application of the TLOCR model and extension of the deterministic model to the top two rings.

First, there are some practical challenges in the application of Chen’s TLOCR deterministic model. Due to different wear condition on the same liner, surface roughness varies from spot to spot. A small patch of measurement cannot provide enough information and the change of plateau roughness makes the contact model unreliable. As a result, a multi-point correlation method was proposed to combine the information of different spots from the same liner and this method was shown to give better match to the experimental results.

A top-two-ring lubrication cycle model was developed based on the multiphase deterministic model by Li et al [30] and previous top-two-ring lubrication model by Chen et al [2][31]. The model is composed with two parts. First, the deterministic model is used to generate a correlation between the hydrodynamic pressure/friction and the minimum clearance with prescribed oil supply from the deterministic oil control ring model. It was found that within reasonable accuracy, the gas pressure effect on the hydrodynamic lubrication of the top two rings can be decoupled from the hydrodynamic lubrication. Thus, only single-phase deterministic model was needed to generate the correlation. This decoupling significantly reduces the computation time. Then, a cycle model was developed utilizing the correlation of hydrodynamic pressure/friction and the minimum clearance. The cycle model considers the effect of gas pressure variations in different ring pack regions as well as the dynamic twist of the top two rings. Finally, the models were used to examine the friction and lubrication of three different liner finishes in an actual engine running cycle.

Thesis Supervisor:

Dr. Tian Tian, Department of Mechanical Engineering
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1 Introduction

1.1 Project Motivation

Nowadays, a rapid growth of internal combustion (IC) engines is considered as a major contributor to energy crisis and environment problems. Increasing the engine’s efficiency and reducing its CO₂ emissions have a direct effect on the energy demands and environmental influence of the transportation sector. These goals are paramount in the entire automotive industry.

Reducing friction and oil transportation are the two major parts to help reduce energy loss and emission. In a typical diesel engine, approximately 10% of total fuel energy is dissipated due to mechanical loss, of which about 20% directly goes to the friction loss between piston ring pack and liner finish, as illustrated in Fig.1.1 [1]. As a result, there is still a large space for the automotive industry to reduce the energy loss by reducing the friction of the piston ring pack system. However, the difficulty is a series of trade-off between friction reduction, oil consumption and blow-by, which means that the reduction of friction at the same time should not introduce adverse effects in blow-by, oil consumption, excessive wear, and failure [2].

![Fig.1.1 Breakdown of Total Diesel Engine Energy, Mechanical Friction and Ring Pack Friction [1]](image)

Oil transport from the crank case to combustion chamber via piston-ring-liner system is a very important source of engine oil consumption that contributes significantly to automotive engine emissions [3] [4]. 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]. It has been recognized that the oil transport via piston-ring-liner system 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

A piston ring pack of modern engines is mostly composed of three rings (from top to bottom): Top ring (compression ring), second ring (scraper ring) and oil control ring (OCR), as shown in Fig.1.2.

In modern automotive industry, the twin-land oil control ring (TLOCR) is widely used in IC engines, especially in diesel engines. However, in North America, three-piece oil control ring is more popular in gasoline engines than TLOCR due to its low cost. In this work, we will focus on this type of oil control ring.
In order to seal the oil in the crank case, the TLOCR tension is typically higher than the top two rings, and consequently its contribution to the entire ring pack friction loss is significant. Additionally, TLOCR plays an important role in controlling the oil film thickness left on the liner, which is the source of oil supply to top two rings. Increasing oil film thickness of OCR is able to provide more oil to the top two rings and results in more oil supply through the liner to the top two rings, greater film thickness, and less friction. However, thicker oil film thickness may cause more oil enters the combustion chamber which on the other hand increases 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 [2]. Recently, with the help of TLCOR deterministic model by Chen, H. et al. [2] and the cycle model by Tian, T. et al. [22], automotive industry is attempting to reduced TLOCR ring tension and its ring-land-width so that they could reduce TLOCR friction and maintain its unit pressure and oil film thickness at the same time.

The top two rings are important for sealing the high pressure gas in the combustion chamber and 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. Under the effect of high cylinder pressure gas, the top two rings, especially the top ring, have a considerable effect on liner wear and oil transport. The high pressure boundary gas also affects the oil distribution inside the ring as well as the oil hydro-dynamic pressure which is reflected on the clearance between top-two-rings and the liner. The difficulty in modeling the top-two-ring is the high boundary gas pressure. The high pressure allows gas to penetrate the wetting region via the deep valley area and as a result, requires multiphase calculation in the model.

1.3 Surface Finish and Measurement on Modern Cylinder Liners

Cylinder Liner surface is finished with multi-stage honing process known as plateau honing which is composed of three stages: the first process is a base honing process using a coarse honing stick to generate deep valleys for lubrication retention. The second process is the finish honing process with a medium size abrasive grit on honing stick. Finally, plateau honing is
accomplished with very fine abrasive grits to generate plateau area [23]. Generally, it is the plateau part where two sliding surfaces interact with each other and with the existence of oil, high oil pressure also tends to be generated. As a result, it is important to define a parameter \( \sigma_p \) \((r_{pq})\) that describes the plateau RMS roughness [24].

Rough liners with deep valleys may give better hydro-dynamic behavior with thicker oil film thickness, but the boundary friction is large. Smooth liners with shallow valleys, on the other hand, may not provide good hydro-dynamic pressure generation ability, leading to lower oil film thickness and higher shear stress, but its boundary behavior is much better than rough liners. As a result, with the same ring tension, rough liner could help reduce hydrodynamic friction, but regarding the same oil film thickness or the same oil consumption, smooth liners, with reduced ring tension and thinner oil, appears to provide a better total friction behavior. [25]

These days, there are three major techniques to measure the liner surface: stylus, confocal microscope and white light scanning interferometry (WLI). The stylus profiler senses the surface height through mechanical contact where a stylus traverses the peaks and valleys of the surface with a small contacting force. The vertical motion of the stylus is converted to an electrical signal by a transducer [26] (see Fig.1.3). It is very sensitive to surface height and provides data to accurate position, but its lateral resolution is limited by stylus tip and the surface suffers plastic deformation during the measurement. The other two, confocal microscope and WLI utilize optical techniques. Confocal microscope uses an aperture (pinhole) to sense the height of a surface point with respect to the best focus position and tells the vertical scan height through the measured signal strength [26] (see Fig.1.4). WLI microscopy, Fig.1.5, uses a broadband light source. The light through a microscope objective is split into two parts, one directly to the surface and the other directed to a smooth reference mirror. The reflected beams are combined and produce interference fringes around the equal path condition which is detected by a camera detector. Scanning the surface vertically with respect to the microscope and detecting the optimum equal path condition at every pixel in the camera results in a topographic image [24]. Optical methods have the advantage of being non-contact
with the surface so that they do not bring any destruction and they are faster than stylus which is based on mechanical scanning. However, light scattering from the surface may affect the accuracy of the result and artificially generates individual spikes on the topographic image.

![Fig.1.3 Schematic Drawing of Stylus Method [27]](image1.png)

![Fig.1.4 Basic setup of a confocal microscope (Light from the laser is scanned across the specimen by the scanning mirrors. Optical sectioning occurs as the light passes through a pinhole on its way to the detector) [28]](image2.png)
1.4 Previous Work on Modeling the Ring liner Interaction Using Deterministic Method

Chen et al. developed the TLOCR deterministic model [2] which evaluates the average hydrodynamic pressure and shear stress between the oil control ring and the liner finish by sliding a flat ring face over the rough liner at a fixed speed and oil film thickness \( h \) (Fig.1.4). Then the dependency of the average hydro-dynamic pressure and shear stress generation on the oil film thickness can be correlated in the following form:

\[
P_{\text{hydro}} = \frac{\mu V}{\mu_0 V_0} P \left( h \right)^{\kappa_h} \quad \text{and} \quad f_{\text{hydro}} = \frac{\mu V}{h} \left( C_{f_1} + C_{f_2} \exp \left( -C_{f_3} \frac{h}{\sigma_p} \right) \right). 
\]

Here \( P_{\text{hydro}} \) and \( f_{\text{hydro}} \) stand for average hydrodynamic pressure and shear stress, respectively. \( P_h \) 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. \( \mu \) and \( V \) are instantaneous oil dynamic viscosity and ring sliding speed. \( h \) is the nominal oil film thickness and \( \sigma_p \) is the cylinder liner plateau roughness. Thus, \( h/\sigma_p \) represents oil film thickness normalized by cylinder liner surface roughness. In the expression of
hydrodynamic shear stress, $C_{f1}$, $C_{f2}$, and $C_{f3}$ are three non-dimensional parameters mainly
determined by the liner roughness microgeometry.

![Curved Ring Profile and Flat Ring Profile](image)

**Fig.1.4 Nominal Oil Film Thickness $h$ [2]**

Similarly, Chen developed another deterministic model for top two rings without considering boundary gas pressure [2]. There are several characteristics in the running condition of top two rings. First, unlike TLOCR, the oil supply to top two rings is not fully flooded and determined by the oil control ring. Second, top two rings typically have curved running surfaces which is important at hydro-dynamic pressure generation. However, due to partial oil supply, it does not dominate the pressure generation and the liner roughness micro structure is as important as the macro ring profile. The last but not the least, the boundary gas pressure difference across the top two rings, especially the top ring, is considerably large which allows gas flow to penetrate through or sometimes to be trapped inside the oil film. This phenomenon affects the lubrication behavior of the top two rings but was neglected in Chen's model. With the consideration of the first two characteristics, the model gives a correlation in following form:

$$P_{prof} = \left( \frac{h_{OCR}}{h_{prof}} \right)^{K_p} \left( \frac{\mu V}{\mu V_0} \right)^{\frac{1}{K_{OCR}}} \left( \frac{h_{prof}}{\sigma_p} \right)^{-K_{OCR}}$$

and

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

where $P_{prof}$ and $f$ are the average hydro-dynamic pressure and shear stress for top-two-ring, $h_{OCR}$ is the oil control ring nominal oil film thickness, $h_{prof}$ is the top-two-ring nominal oil film thickness, $K_p$ is power constant and depends on liner finish, ring profile as well as oil supply $h_{OCR}$, $a_p$ is another constant
that captures the ring profile effect on hydro-dynamic pressure generation, $P_{0, OCR}$ and $K_{OCR}$ are
two constants from TLOCR correlation which represents the effect of oil control ring. In the
shear stress correlation, $F_0$ is a constant addressing the oil supply and ring face profile effect
and $K_f$ is the power constant that depends on $h_{OCR}$.

To predict the lubrication behavior of top two rings under elevated gas pressure
environments with the effect of liner finish considered, Li et al. developed a multiphase
deterministic model [30]. This model considers the effect of gas penetration and trapped gas
behavior, and gives a robust result, but unfortunately it does not provide a correlation that can
be used for the cycle model.

1.5 Scope of Thesis Work

The objective of this thesis is to model the lubrication behavior of the piston ring pack
especially the top two rings using the deterministic method.

The second chapter based on Chen's previous work discusses some practical issues using
the TLOCR deterministic model [2].

The third chapter introduces the top-two-ring single phase deterministic model which
considers the effect of ring profile, liner surface roughness and partial oil supply. It also
discusses the numerical method and key assumptions of the model. In the end, this chapter
includes part of ring profile effect on the hydro-dynamic behavior between the ring and the
liner.

The forth chapter introduces the top-two-ring multiphase deterministic model that is an
improvement of the top-two-ring single phase deterministic model. Besides all the features
mentioned above, the multiphase model also considers the effect of elevated pressure gas. This
chapter also demonstrates a simplified model to correlate the multiphase model hydro-
dynamic pressure/stress as a function of the nominal clearance between the ring and the liner.
Finally, a cycle model that is based on the deterministic correlations is shown together with
some examples.
The last chapter summarizes and concludes the thesis work and suggests potential future work on the topic.
2 Practical Challenges for Applying TLOC R Model [31]

Chen et al. developed the deterministic TLOC R model [2][32] which uses a small patch measured on the liner as model input. However, due to different wear conditions, after a period of time, different points on a same liner will have different surface roughness. Therefore, a small patch from a single point may not contain enough information to represent the entire surface. It is important to gain knowledge on how the changing roughness on the liner influences the lubrication and friction of the TLOC R and how one can predict the changes of the friction due to liner wear. The objectives of this part of work are

1. To examine the hydrodynamic lubrication and asperity contact of a TLOC R and rough liners based on roughness measurements at different spots of the same liners as well as different liners with the same honing.

2. To provide the information obtained to another research effort to compare with the friction measurements using FLE.

2.1 Revisit of the Deterministic TLOC R Model

The deterministic TLOC R model was developed by Chen et al. [2][32]. This model evaluates the average hydrodynamic pressure and shear stress of a twin-land oil control ring under a reference sliding speed and oil viscosity at various ring-liner nominal clearances. Given the average hydrodynamic pressure and shear stress at different nominal ring-liner clearances, then it is able to derive a series of correlations that calculate the hydrodynamic pressure and shear stress under different sliding speed, oil viscosity and ring-liner nominal clearances.

\[ P_{\text{hydro}} = \frac{\mu \nu}{(\mu \nu)_0} P_h \left( \frac{h}{\sigma_p} \right)^{-K_h} \]  
\[ f_{\text{hydro}} = \frac{\mu \nu}{h} \left( C_{f1} + C_{f2} e^{-C_{f2} \frac{A}{h}} \right) \]

Here \( P_{\text{hydro}} \) and \( f_{\text{hydro}} \) stand for average hydrodynamic pressure and shear stress, respectively. \( P_h \) 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. \( \mu \) and \( V \) are instantaneous oil dynamic viscosity and ring sliding speed. \( h \) is the nominal oil film thickness and \( \sigma_p \) is the cylinder liner plateau roughness. Thus, \( h/\sigma_p \) represents oil film thickness normalized by cylinder liner surface roughness. In the expression of hydrodynamic shear stress, \( C_1, C_2 \), and \( C_3 \) are three non-dimensional parameters mainly determined by the liner roughness microgeometry.

To calculate the ring force balance equation, the boundary contact pressure between a TLOCR and a liner surface also needs to be considered. The boundary contact force is able to be calculated using the Greenwood-Tripp method [2][33][34][35] which has the following form:

\[
P_{\text{contact}} = \begin{cases} p_c \left( \Omega - \frac{h}{\sigma_p} \right)^Z & \text{for } \frac{h}{\sigma_p} < \Omega \\ 0 & \text{for } \frac{h}{\sigma_p} > \Omega \end{cases}
\]  \hspace{1cm} (eq. 2.3)

where \( P_c \) is a constant determined by the probability distribution of asperity height, the asperity shape, the plateau roughness and the material properties of the ring and the liner \( \Omega \) and \( Z \) are two constants. It is usually taken that \( \Omega = 4 \) and \( Z = 6.804 \) for the convenience of numerical calculation of the distribution function [33][34][35].

The ring tension is balanced by the sum of hydro-dynamic pressure and contact pressure generated between the TLOCR and the liner. The model input of the cylinder is usually obtained from the confocal microscope measurement.

2.2 Surface Roughness Effects on Contact Model

The boundary contact model inputs are strongly related to the surface roughness and wear conditions. Different measurement spots on the same liner surface can result in significant variations in boundary contact pressure calculation.

There are 3 sleeves with the same honing tested in the friction measurement and after the test 4 spots of each sleeve are measured (totally 12 spots). The four locations are: thrust side closed to top dead center (TDC Thrust), thrust side closed to bottom dead center (BDC Thrust), thrust side closed to mid stroke (Mid Thrust) and pin side closed to mid stroke (Mid Pin).
Surface measurement results are shown in Fig 2.1. The plateau roughness $\sigma_p$ and surface roughness $R_a$ from the measurement points on the three liner sleeves are shown in Table 2.1.

<table>
<thead>
<tr>
<th></th>
<th>#3-1</th>
<th>#3-2</th>
<th>#3-3</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>TDC Thrust</strong></td>
<td><img src="image1.png" alt="Image" /></td>
<td><img src="image2.png" alt="Image" /></td>
<td><img src="image3.png" alt="Image" /></td>
</tr>
<tr>
<td><strong>BDC Thrust</strong></td>
<td><img src="image4.png" alt="Image" /></td>
<td><img src="image5.png" alt="Image" /></td>
<td><img src="image6.png" alt="Image" /></td>
</tr>
<tr>
<td><strong>Mid Thrust</strong></td>
<td><img src="image7.png" alt="Image" /></td>
<td><img src="image8.png" alt="Image" /></td>
<td><img src="image9.png" alt="Image" /></td>
</tr>
<tr>
<td><strong>Mid Pin</strong></td>
<td><img src="image10.png" alt="Image" /></td>
<td><img src="image11.png" alt="Image" /></td>
<td><img src="image12.png" alt="Image" /></td>
</tr>
</tbody>
</table>

*Fig. 2.1 Surface Measurement of the Three Liner Sleeves at the TDC Thrust Side, BDC Thrust Side, Mid Stroke Thrust Side and Mid Stroke Pin Side (sliding direction from top to bottom and each patch is about 3mm by 3mm and the color bar unit is in meter) [31]*
Table 2.1 Plateau Roughness and Surface Roughness of the Three Liner Sleeves Measured at 4 Different Locations [31]

<table>
<thead>
<tr>
<th></th>
<th>TDC Trust</th>
<th>BDC Trust</th>
<th>Mid Thrust</th>
<th>Mid Pin</th>
</tr>
</thead>
<tbody>
<tr>
<td>#3-1</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\sigma_p$ ($\mu$m)</td>
<td>0.071</td>
<td>0.058</td>
<td>0.063</td>
<td>0.055</td>
</tr>
<tr>
<td>Ra ($\mu$m)</td>
<td>0.568</td>
<td>0.363</td>
<td>0.346</td>
<td>0.201</td>
</tr>
<tr>
<td>#3-2</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\sigma_p$ ($\mu$m)</td>
<td>0.092</td>
<td>0.08</td>
<td>0.116</td>
<td>0.041</td>
</tr>
<tr>
<td>Ra ($\mu$m)</td>
<td>0.301</td>
<td>0.247</td>
<td>0.314</td>
<td>0.267</td>
</tr>
<tr>
<td>#3-3</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\sigma_p$ ($\mu$m)</td>
<td>0.124</td>
<td>0.063</td>
<td>0.067</td>
<td>0.078</td>
</tr>
<tr>
<td>Ra ($\mu$m)</td>
<td>0.611</td>
<td>0.286</td>
<td>0.282</td>
<td>0.31</td>
</tr>
</tbody>
</table>

The FEMP calculation results using the mid stroke pin side measurement of the three liner sleeves are shown in Fig. 2.2.

![Fig. 2.2 FMEP Calculation Results of the Three Liner Sleeves Using the Mid Stroke Pin Side Surface Measurement as input [31]](image)
The calculation results in Fig. 2.2 demonstrate a significant variation among the three liner sleeves in FMEP at lower engine speed where boundary contact friction becomes more important than hydro-dynamic friction. It is illustrated in an analysis in the FMEP contribution from each module in Fig. 2.3.

![Graph showing boundary contact percentage vs. RPM for different sleeves.](image)

**Fig. 2.3 FMEP Contribution from Contact Model and Plateau Roughness of the Three Surfaces Measured at Mid Stroke Pin Side [31]**

Fig. 2.3 indicates that the FMEP is mainly from boundary contact at low engine speed so that the variation of the FMEP at the low engine speed should mainly result from the contact friction difference. In addition, the consistence of the plateau roughness \( \sigma_p \) and the FMEP at 100RPM further proves this statement: the rough surface with larger \( \sigma_p \) has more boundary contact friction than smooth surface with smaller \( \sigma_p \). Among all engine speed tested, boundary contact has the greatest contribution to FMEP at 100RPM. A study concentrating on 100RPM FMEP results can help define the deciding factor when boundary contact is dominating.
Fig 2.4 Correlation of FMEP at 100RPM and Plateau Roughness, FMEP at 100RPM Calculated Using Measurement from all 12 Spots on the 3 Liner Sleeves [31]

The FMEP results at 100RPM are taken to represent the boundary contact FMEP. It is plotted against plateau roughness at all 12 spots measured in Fig 2.4. The figure implies that boundary contact FMEP is strongly correlated with surface plateau roughness.

Therefore, when boundary contact dominates the total friction, liner surface plateau roughness determines the friction loss of the contacting pair.

### 2.3 Surface Roughness Effects on Hydrodynamic Model

Fig. 2.5 shows the instantaneous Strubeck curves obtained from the TLOCR model results using the measurement at different surface locations as input. The curves are compared with instantaneous Strubeck curve obtained from experimental result.

The x-axis is the Gumbel-Hersey-Number (GHN), $\mu$ is oil dynamic viscosity, $N$ is piston liner speed and $P$ is the unit load of the piston ring. The y-axis is friction coefficient. The solid lines
represent the TLOCR model results and marks represent experimental results. The experiment uses a 0.06mm land width TLOCR and the tension spring was changed to give the TLOCR different tensions. Lowering the liner temperature to 40°C and 60°C can extend the hydrodynamic region covered by the instantaneous Stribeck curve. It should be noticed that in the experimental results, piston friction has been subtracted and only friction results of 20-40 crank angle degree after BDC are used to obtain the experiment instantaneous Stribeck curve.

![Instantaneous Stribeck Curve](image)

**Fig. 2.5 Instantaneous Stribeck Curve from 0.06mm Land Width TLOCR Model Results Using Different Surface Inputs, Compared with FLE results [31]**

Fig. 2.5 indicates that hydro-dynamic pressure generation ability and shear stress calculated based on a small patch can have deviations from the average surface value. It is worth noticing the thrust side BDC surface input gives the calculation a good match with the experimental results from the same region. The observation firmly proves that the TLOCR model could give a robust result of hydro-dynamic pressure (shear stress) generation ability from a given surface input. However, results deviation is observed due to different surface inputs. If assuming that all of the ring tension load is balanced by hydrodynamic pressure, with eq. 2.1, it is easy to get the following equations:
\[ P_{\text{load}} = \frac{\mu U}{(\mu U)_0} P_h \left( \frac{h}{\sigma_{po}} \right)^{-K_h} \]  
(eq. 2.4)

\[ f_{h,\text{stress}} = \frac{h}{h} \left( C_{f1} + C_{f2} e^{-\frac{c_{f1}}{\sigma_{pl}}} \right) \]  
(eq. 2.5)

where \( P'_h = P_h \left( \frac{\sigma_u}{\sigma_{po}} \right)^K \), \( C'_{f3} = C_{f3} \frac{\sigma_{po}}{\sigma_p} \), and \( \sigma_{po} \) is the reference plateau roughness which is a constant. The friction coefficient is the ratio of friction force to the normal force. Then, the hydrodynamic friction coefficient is

\[ f_{c,h} = \left[ \frac{(\mu U)_0}{P'_h \sigma_{po}} \right]^{\frac{1}{a}} \left[ \frac{\mu U}{P_{\text{load}} \sigma_{po}} \right]^{\frac{a-1}{a}} \left[ C_{f1} + C_{f2} e^{-\frac{c_{f3}}{\sigma_{po}}} \right] \]  
(eq. 2.6)

With the definition of \( F_{h0} = \frac{(\mu U)_0}{P'_h \sigma_{po}} \) and \( G_0 = \frac{\mu U}{P_{\text{load}} \sigma_{po}} \), eq. 2.6 becomes

\[ f_{c,h} = F_{h0}^{\frac{1}{a}} G_0^{\frac{a-1}{a}} \left[ C_{f1} + C_{f2} e^{-\frac{c_{f3}}{\sigma_{po}}} \right] \]  
(eq. 2.7)

Fig. 2.6 shows the result of hydrodynamic friction coefficient as a function of \( G_0 \), which is a modified Gümbel-Hersey number.
Fig. 2.6 Hydrodynamic Friction Coefficient (In the legend, sleeve 1, sleeve 2 and sleeve 3 are 
#3-1, #3-2, #3-3 liner sleeve respectively; the second digit of 1, 2, 3 and 4 represent TDC thrust 
side, BDC thrust side, mid stroke thrust side and mid stroke pin side respectively)

From Fig. 2.6, it is noticeable that there is a considerable variance in hydrodynamic friction 
coefficient from different surface measurement with the same honing when $G_0$ is over 1.5 
which corresponds to heavy hydrodynamic region. The variance results from the term of $F_{ho}K_h^{\frac{1}{h}}$ 
and $\left[C_f + C_f^2e^{-C_f^2\left(\frac{h}{\sigma_{po}}\right)}\right]$. The term $F_{ho}K_h^{\frac{1}{h}}$ reflects the hydrodynamic pressure generation 
ability which is highly dependent on the local honing. The term $\left[C_f + C_f^2e^{-C_f^2\left(\frac{h}{\sigma_{po}}\right)}\right]$ is also 
present in eq. 2.2 as a correction factor to the hydrodynamic shear stress. The shear stress for 
smooth surface is $\frac{\mu u}{h}$. However, considering roughness effect, shear stress may not be 
generated within the deep valley area, and thus there comes such a term to modify the shear 
stress from a smooth surface to a rough surface. Therefore, this term should be highly
dependent on the percentage of plateau area, or the plateau ratio. In hydrodynamic region, due to exponential decay, $C_{f2} e^{-C_{f3} \frac{h}{\sigma_{p0}}}$ would quickly approach zero as the clearance increases, which means it should be $C_{f1}$ rather than $C_{f2} e^{-C_{f3} \frac{h}{\sigma_{p0}}}$ that is highly dependent on plateau ratio. This is shown in Fig 2.7.

![Cf1 vs. Plateau Ratio](image)

**Fig. 2.7** Correlation between $C_{f1}$ and Plateau Ratio (In the legend, sleeve 1, sleeve 2 and sleeve 3 are #3-1, #3-2, #3-3 liner sleeve respectively; the second digit of 1, 2, 3 and 4 represent TDC thrust side, BDC thrust side, mid stroke thrust side and mid stroke pin side respectively)

As shown above, the hydrodynamic behavior deviation is observed due to different surface inputs. The difference in plateau ratio and some other parameters related to local honing could both result in a difference in hydrodynamic pressure/shear-stress generation ability. Then, the challenge now becomes to find an input correlation that can represent the hydrodynamic pressure generation ability of the entire surface.
2.4 Multi-Point Correlation Method

The previous section shows the consistency between measurement and calculation in a local area, indicating the credibility of the model. A further question is how well this model can predict FMEP, which is dominated by the friction at mid-stroke and affected by surface roughness of all locations.

The experimental results of FMEP trend of the 0.06mm land width 28.5N tension TLOCR on three liners are shown in Fig. 2.8. In the figure, “10 hr”, “15 hr” and “extended” refer to the break-in time. #3-1 and #3-3 sleeves have been broken in using the 28.5N tension 0.06mm land width oil control ring under motored condition. During the break-in process, floating liner engine thrust side temperature and oil jet temperature has been kept at 60±1°C. The engine speed starts at 100RPM and increase by 100RPM every half an hour to 1000RPM. Within each half an hour, the result was recorded after the friction force became stabilized. The process continues for 15 hours for #3-1 liner sleeve, and 10 hours for #3-3 liner sleeve. FMEP results of #3-1 liner after 10 hours and 15 hours’ break-in are used. #3-3 liner results after 10 hours’ break-in are used.

#3-2 sleeve has been operated with various rings under both motored and fired conditions for an extended period. For the purpose of the current study, #3-2 sleeve is run under the same motored condition with that of the #3-1 and #3-3 sleeve for 5 hours. The last set of record of frictions from 100RPM to 1000RPM has been used for comparison. Piston friction has been subtracted from TLOCR friction results.
Fig. 2.8 Experiment FMEP trend of the 0.06mm land width 28.5N tension TLOCR on three liners of the same honing [31]

The experimental results in Fig 2.8 show that FMEP of different sleeves at different break-in time of the same honing are converging towards higher RPM. In the previous section, it has been explained that in high RPM region, hydro-dynamic friction dominates while in low RPM region, boundary contact friction becomes more important. This figure clearly shows that in an average sense, hydrodynamic behavior of three sleeves with the same honing is the same and the difference in FMEP results from asperities in contact.

Therefore, the following method is proposed to predict the FMEP of the TLOCR. First, obtain an overall hydrodynamic pressure (shear-stress) relation with the ring-liner clearance based on all the reasonable spots. Second, as the plateau roughness is changing during break-in, plateau roughness can be changed in a certain range to predict the evolution of FMEP during break-in.

The multi-point correlation method requires a definition of a representative plateau roughness ($\sigma_{p0}$) and plateau ratio ($r_{p0}$). These two values can be obtained by averaging the results from different measured spots.
\[ r_{pl0} = \frac{\sum_{i=1}^{N} r_{pl,i}}{N} \]  

(eq. 2.8)

\[ \sigma_{p0} = \sqrt{\frac{\sum_{i=1}^{N} r_{pl,i} \sigma_{pl,i}^2}{\sum_{i=1}^{N} r_{pl,i}}} \]  

(eq. 2.9)

where \( \sigma_{pl,i} \) and \( r_{pl,i} \) are plateau roughness and plateau ratio of each selected small surface patch.

The patch selection criteria are: 1. the patch should not have through scratch in the sliding direction; and 2. the patch should be representative of the intended honing pattern.

Then, the pressure (and shear-stress) and nominal clearance relationship from all the patches with normal wear is calculated using the deterministic model [2][32]. Single correlations in the form of eq. 2.1 and eq. 2.2 are obtained, with \( \sigma_p \) replaced by \( \sigma_{p0} \). The correlations of hydrodynamic pressure (and shear stress) with the clearance are shown in Fig. 2.9.
Fig. 2.9 Correlations of Hydro Pressure (and Shear Stress) and Clearance (In the legend, sleeve 1, sleeve 2 and sleeve 3 are #3-1, #3-2, #3-3 liner sleeve respectively; the second digit of 1, 2, 3 and 4 represent TDC thrust side, BDC thrust side, mid stroke thrust side and mid stroke pin side respectively) [31]

From Fig.2.9, one can conclude that at least under this honing condition, the hydrodynamic friction behavior of different surface patches with the same type of honing is quite similar. This is consistent with the experimental results. However, as shown in Fig. 2.1, due to different wear condition, the local roughness of these patches are very different. Therefore, the hydrodynamic behavior will have some variance, which has been shown in a previous section. Fig. 2.9 indicates that one particular type of honing determines the overall structure as well as the main trend of the hydrodynamic behavior. A change in the local structure will not significantly alter the overall trend, as shown in Fig 2.10.
In Fig. 2.10, the average hydrodynamic pressure of the original surface over the sliding distance is 1.33bar, of the surfaces with 3 scratches is 1.24bar and of the surface with many scratches is 1.15bar. This indicates that a local damage to the surface, as long as it does not ruin the whole structure, will not change the hydrodynamic pressure generation ability significantly.
In addition to the overall correlation obtained from all reasonable spots (good points), all spots, including those with severe scratches, are used to obtain another overall correlation. Friction results from both inputs are shown in Fig. 2.11.

Fig. 2.11 FMEP Calculated with ‘Good Points’ and ‘All Points’ Correlation Input, Using Multi-Point Correlation Method [31]

Fig. 2.11 shows the calculated FMEP of a 28.5N tension, 0.06mm land width TLOCR at 60°C. Calculation using the overall correlation from only reasonable points predicts the same result as that using the overall correlation from all the points at high RPM where hydrodynamic shear stress is dominant. At low RPM where asperities contact happens, deviations in the plateau roughness result in difference in FMEP.

Therefore, multi-point correlation provides robust results for different liners with the same type of honing. Experiments suggest limited hydrodynamic behavior variation over the break-in period with different liners of the same honing. Thus, the multi-point correlation method can be used to best represent the hydrodynamic behavior of a certain honing. The contact friction part is dependent on the exact stage of asperity wear.
2.5 Multi-Point Correlation Case Study

The following case compares experimental results with calculations that obtained from multi-point correlation inputs.

The friction difference between the experimental result and the calculation result with multi-point correlation inputs is shown in Fig. 2.12. The first graph is the result of a 28.5N tension 0.06mm land width TLOCR at 500RPM under 60°C liner temperature and the second graph is the result of a 10.5N tension 0.06mm Land Width TLOCR at 1000RPM under 60°C liner temperature. FMEP comparison between experiment and calculation results is shown in Fig. 2.13. In both Fig. 2.12 and Fig. 2.13, piston friction is subtracted from the experimental results.

![Fig. 2.12 TLOCR Model (Model) and Experiment (EXP) Friction Difference between 28.5N 0.06mm Land Width TLOCR and 10.5N 0.06mm Land Width TLOCR at 60°C Liner Temperature, 500RPM and 1000RPM respectively [31]](image)

In the comparison shown in Fig. 2.12, the variation in contact friction due to the difference of surface roughness at different points of the liner is balanced out by multi-point method. As a result, a good agreement between the experiment and model result is observed.
The effect of liner plateau roughness evolution is demonstrated in Fig. 2.13. The model calculates the FMEP of TLOCR with three plateau roughness (0.04μm, 0.066μm and 0.1μm) but with the same hydrodynamic pressure – clearance correlation (absolute clearance, not normalized by plateau roughness). It indicates the change of FMEP when contacting asperities are still evolving. The figure shows that hydrodynamic behavior is robust during the break-in process when the surface asperities are still changing. However, the contact behavior is evolving within the break-in process.

2.6 Conclusion

In this chapter, it has been evidently demonstrated that with the same type of honing, the contact model is highly dependent on the surface roughness. During the break-in process, the variation in friction behavior of the ring-liner system is mainly caused by the change of boundary contact friction. However, the hydrodynamic behavior with the same type of honing is roughly the same. It is demonstrated in Fig. 2.9 that the hydrodynamic pressure (shear stress) and nominal clearance correlations of different measurement patches almost fall on the same line. Nevertheless, the hydrodynamic behavior still has variance among these individual surface measurements. To get a more precise prediction of friction behavior, a multi-point correlation
is proposed to combine all the information of different measured patches with the same type of honing. As long as there is sufficient information about the roughness on overall structure that affects hydrodynamic pressure (shear stress), for example plateau roughness and plateau ratio, the calculation appears to match the measurement results reasonably.
3 Single Phase Top-two-ring Model

This chapter introduces the top-two-ring model based on deterministic method which is originally proposed by Elrod [36]. Li et al. [30] first introduced this method into the model of piston ring lubrication and Chen et al. [2] further developed the model and came up with a series of correlation between hydro-dynamic pressure and clearances as well as hydro-dynamic shear stress and clearances that can be used in Tian’s cycle model [22]. Single phase top-two-ring model incorporates the effects of macro ring profile geometry and partial oil supply but not the effect of boundary gas pressure.

3.1 Assumption for Single Phase Top-two-ring Model

3.1.1 Assumptions for Lubrication Approximations

Reynolds Equation is very useful in lubrication system which is:

\[
\frac{d(ph)}{dt} = \nabla \cdot \left( \frac{p h^3}{12 \mu} \nabla p \right) - \frac{V}{2} \frac{\partial (ph)}{\partial x} \tag{eq. 3.1}
\]

In this equation as well as the whole thesis, the numerical coordinate system is always attached to the ring. \( x \) is the sliding direction and \( y \) is the circumferential direction. Here, \( p \) 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 \( p \) is a constant, and the following incompressible Reynolds equation applies.

\[
\frac{dh}{dt} = \nabla \cdot \left( \frac{h^3}{12 \mu} \nabla p \right) - \frac{V}{2} \frac{\partial h}{\partial x} \tag{eq. 3.2}
\]
To apply Reynolds equation, the assumptions of the lubrication theory need to be satisfied:

\[
\left( \frac{\Delta h}{\Delta x} \right)^2 \ll 1, \quad Re_h \frac{\Delta h}{\Delta x} \ll 1 \quad \text{and} \quad \frac{(\Delta h)^2}{\nu t_c} \ll 1 \quad \text{Fig.3.1).}
\]

The first two assumptions state that the viscosity term in x direction and inertial terms are negligible. The third assumption is a well-defined limitation to neglect unsteady effects. \( \nu \) is the kinematic viscosity of the lubricant oil, \( \frac{(\Delta h)^2}{\nu} \) scales with the time for the velocity profile to diffuse to a steady-state shape across the gap and \( t_c \) is the characteristic time constant which could be interpreted as the time for each engine stroke. A modern liner surface that is finished with multi-stage honing process, is usually very smooth which means on the surface the maximal \( \frac{\Delta h}{\Delta x} \) is around 0.1. Oil viscosity and density under typical engine running condition is around 5e-3 Pa·s and 850 kg/m³. Therefore the Reynolds number is around 0.051 and \( Re_h \frac{\Delta h}{\Delta x} \) is around 0.005. The time scale to diffuse the velocity profile across the gap is around 5e-9s and the typical value for time constant \( t_c \) is around 0.01s and the term \( \frac{(\Delta h)^2}{\nu t_c} \) is of the order of 1e-7 to 1e-6. As shown above, these typical numbers firmly justify for the application of Reynolds equation to the lubrication between the ring-pack and the liner.

### 3.1.2 A Cavitation Theorem and Full Attachment Assumption

When lubricant oil pressure drops to a critical value (the cavitation pressure) which usually happens in deep valley region, liquid oil cannot exist alone and instead the oil cavitates and
separates into two phases: liquid and vapor. Based on the Jakobson-Floberg-Olsson (JFO) theory, Elrod divided the oil domain into two parts, a full film region and a partial film region [36]. In the full film region, both pressure flow and viscous flow exist as driving force and if assume constant liquid oil density, only the pressure is the dependent variables. In the partial film region, as described earlier, liquid oil phase and oil vapor/air coexist. It is assumed that the pressure is constant so that the pressure gradient is zero and only the viscous force drives the oil flow. Therefore, with the assumption of constant liquid oil density and negligible vapor oil density, the oil film ratio ($\Phi$, volume fraction of liquid phase) is the only dependent variable and the Reynolds equation collapses to the following form:

$$\frac{d(\phi h)}{dt} = -\frac{V}{2} \frac{\partial(\phi h)}{\partial x}$$

(eq. 3.3) [32]4

The oil flow is governed by a purely hyperbolic oil transport equation.

In the partial film region, there is another key assumption to describe its flow pattern: the full attachment assumption which assumes that the liquid oil attaches to both of the running faces and forms cavitation streaks in the local partial film area (Fig.3.2). Chen et al. believed that the formation of the streaks would help to minimize the surface interface area and thus minimize the surface energy [2].
3.1.3 Assumption for Deterministic Calculation Region

The width of top two rings is usually much larger than the oil control ring land width and the macro profile is significant (Fig.3.3). Furthermore, oil is supplied from the oil layer left by the OCR on the liner. Therefore, the wetting area hardly covers the whole ring and it becomes an uncertainty: the wetting area is transient and irregular where surface tension of the oil plays an important role. However, in the deterministic model, a fixed calculation region is needed (See a comparison in Fig.3.4). As a result, a relatively large calculation domain is applied to avoid artificial scrape and in this way the wetting area (the full film region) is evolved based on Reynolds equation and represented by the oil film ratio. Another assumption here is the neglect
of surface tension. This assumption deviates from reality and the increase of the calculation domain in the axial direction will artificially increase the hydro-dynamic friction due to the full attachment assumption.

![Fig.3.3 Top Two Ring Profile (Worn)](image)
3.2 The Numerical Approach [32]

As mentioned above, there are two parts in the lubricant oil flow between the ring and the liner: full film region and partial film region. Correspondingly there are two governing equations describing the two regions: Reynolds equation and the oil transport equation. Li et al. introduced a universal variable to switch between these two equations based on the schemes from Elrod and Payvar & Salant [32][36][37]. This method avoids tracking the cavitation boundary and the result automatically satisfies mass conservation. Using the universal variable instead of compressibility to relate density/oil film ratio and pressure helps improve the calculation efficiency. Furthermore without the huge lubricant compressibility coefficient, the
density error of a point that switches from cavitation zone to full film zone is able to cause less numerical instability [30] [32].

The universal 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 [30][32]. Define the universal dimensionless dependent variable $F$,

$$
P = \begin{cases} 
F & F \geq P_c \\
P_c & F < P_c 
\end{cases} \quad (eq. \ 3.4)
$$

$$
\phi = \begin{cases} 
1 & F \geq P_c \\
F_Pc & F < P_c 
\end{cases} \quad (eq. \ 3.5)
$$

Where $P$ is the pressure, $\Phi$ is the oil film ratio ($\Phi=1$ is the full film region and $0 \leq \Phi < 1$ is the partial film region) and $P_c$ represents the cavitation pressure.

In full film region ($F \geq P_c$), the Reynolds equation becomes:

$$
\frac{dh}{dt} = \nabla \cdot \left( h^3 \frac{\nabla F}{12 \mu} \right) - \frac{\nabla \cdot \mathbf{v} h}{2 \partial x} \quad (eq. \ 3.6)
$$

In partial film region ($F < P_c$), the oil transport equation becomes:

$$
\frac{d(Fh)}{P_c dt} = -\frac{\partial (Fh)}{P_c \partial x} \quad (eq. \ 3.7)
$$

The universal variable here also behaves as an index variable and it has different physical meanings in different zones. In full film zone, it is pressure. In cavitation zone, its absolute value is the fraction of volume occupied by liquid oil.

Instead of updating both $P$ and $\Phi$ through a small relaxation number, Li proposed to only update $F$. Thus when the full film and partial film regions are fixed, the Reynolds universal equation loses nonlinearity, and iteration can converge quickly. [30] [32]
3.3 Boundary Condition for Deterministic Calculation of Top-Two-Ring

Unlike the deterministic calculation of oil control ring [2], the inlet boundary condition of top-two-ring model is no longer full film. At a particular oil control ring clearance, the oil control ring model determines the distribution of oil film thickness left on the liner which needs to satisfy the mass conservation of trailing edge oil flow. This distribution is fed into the top-two-ring deterministic calculations as oil supply input and calculates the boundary oil film ratio based on mass conservation. The boundary pressure is assumed to be a constant ambient pressure. Therefore the boundary oil film ratio is given by

\[
\phi_t = \min\left\{\frac{2(h_{\text{oil},t} + h_{\text{res},t})}{h_{\text{local},t}}, 1\right\}
\]

\[
h_{\text{res},t+1} = \max(h_{\text{oil},t} + h_{\text{res},t} - \frac{h_{\text{local},t}}{2}, 0)
\]

where \(\phi_t\) represents the boundary oil film ratio, \(h_{\text{oil},t}\) denotes the oil film thickness on the liner in front of the boundary of computational domain, and \(h_{\text{local},t}\) refers to the local clearance on the boundary of computational domain. The factor 2 here results from the full attachment assumption. When the oil supply is sufficient, the inlet boundary condition could become fully flooded and the extra oil would be scraped 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 \(h_{\text{res},t}\).

3.4 Top-two Ring Model Correlations

Chen et al. developed correlations to relate the average hydro-dynamic pressure and shear stress to the nominal clearance between the ring and the liner [2]. These correlations could be used in the cycle model to calculate the friction behavior with a given ring shape and a surface liner finish.

In the deterministic calculation for top two rings, the oil supply is determined by the nominal clearance between the oil control ring and the liner, \(h_{\text{OCR}}\). With a given \(h_{\text{OCR}}\), the
average hydro-dynamic pressure generation of top-two-ring can be evaluated at different level of clearances between the top-two-ring and the liner, \( h_{\text{prof}} \). This is shown in Fig. 3.5.

![Fig. 3.5 Oil Supply Condition](image)

Fig. 3.5 Oil Supply Condition

Fig. 3.6 and Fig. 3.7 show the result of average hydro-dynamic pressure and shear stress with different nominal ring-liner clearances. When the ring profile is flat, the top-two-ring has the same ring face with that of oil control ring. At this time, if \( h_{\text{prof}} \) is less than \( h_{\text{OCR}} \), the oil supply is sufficient so that the top-two-ring single phase deterministic model has the same boundary condition as the TLOCR model. As a result, they should have the same hydro-dynamic behavior as well. However, if \( h_{\text{prof}} \) is larger than \( h_{\text{OCR}} \), the oil supply starts to starve and the hydro-dynamic pressure and shear stress decays at a much higher rate. For a ring with a curved ring profile, the hydrodynamic pressure (shear stress) decays at a similarly high starvation rate around the supply level \( h_{\text{OCR}} \) since the ring with a curved profile allows more oil to pass than the flat ring at the same nominal ring-liner clearance with sufficient oil supply.
Fig. 3.6 Hydro Pressure for Different Ring Curvatures [2]

Fig. 3.7 Hydro Shear Stress for Different Ring Curvatures [2]
With the consideration of the oil supply (determined by $h_{OCR}$), the ring shape effect and the nominal clearance between top-two-ring and the liner ($h_{prof}$), the hydrodynamic pressure and shear stress of a particular face profile and liner finish can be correlated in the following form:

$$P_{prof} = \left( \frac{h_{OCR}}{h_{prof}} \right)^{K_p} \frac{\mu V}{(\mu V)_o} \left( a_p P_{0,OCR} \right) \left( \frac{h_{prof}}{\sigma_p} \right)^{-K_{OCR}}$$  \hspace{1cm} (eq. 3.8)

$$f = F_0 \frac{\mu V}{h_{prof}} \left( \frac{h_{OCR}}{h_{prof}} \right)^{K_f}$$  \hspace{1cm} (eq. 3.9)

Here, $\mu$, $V$ are the instantaneous oil viscosity and ring sliding speed. $(\mu V)_o$ is the reference value of $\mu V$ in the simulation. $\sigma_p$ is the plateau roughness. $\left( \frac{h_{OCR}}{h_{prof}} \right)^{K_p}$ is defined as a filling factor influenced by oil supply level and $K_p$ is a linear function of $h_{OCR}$ for a specific ring profile and liner finish. $P_{0,OCR}$ is the same as $P_t$ in eq. 2.1

$$P_{hydro} = \frac{\mu V}{(\mu V)_o} P_h \left( \frac{h}{\sigma_p} \right)^{-K_h}$$  \hspace{1cm} (eq. 2.1)

$$f_{hydro} = \frac{\mu V}{h} \left( C_{f1} + C_{f2} e^{-C_{f2} \frac{h}{\sigma_p}} \right)$$  \hspace{1cm} (eq. 2.2)

and $a_p$ is the shape profile factor that is another linear function of $h_{OCR}$ for a specific ring profile and liner finish. $K_{OCR}$ is $K_h$ in eq. 2.1. $F_0$ is a constant to account for the ring profile effect on top-two-ring shear stress and $\left( \frac{h_{OCR}}{h_{prof}} \right)^{K_f}$ is added to address the oil supply effect, while $K_f$ works as $K_p$ in hydrodynamic pressure correlation and is a linear function of $h_{OCR}$ for a specific ring profile and liner finish.

### 3.5 Ring Profile Effect on Average Hydro-dynamic Pressure and Shear Stress of Top-Two-Ring Single Phase Model

The ring profile effect here mainly refers to the effect of shifting minimum clearance location. In a real engine cycle, the twist of the ring due to the high pressure gas and large initial force is not negligible and the twist will shift the location of lowest point especially for those rings with
flatter parabolic profile. Fig.3.8 shows the geometry of the ring. rb1 refers to the upwind width and rb2 refers to the downwind width.

Fig. 3.8 the Ring Geometry

Fig.3.9 and Fig.3.10 show the result of average hydro-dynamic pressure and shear stress as a function of $h/\sigma_p$ where $h$ is the minimum nominal clearance between the ring and the liner and $\sigma_p$ is the liner plateau roughness. The calculation uses different liner finish from the one shown in Fig.3.6 and Fig3.7. In these two figures, the legend is in format of ‘rb1+rb2’. For example, the first legend ‘400+200’ means that $rb_1=400\mu m$ and $rb_2=200\mu m$. 
Fig. 3.9 Average Hydro-dynamic Pressure with Shifting Lowest Point

Fig. 3.10 Average Hydro-dynamic Stress with Shifting Lowest Point
Fig. 3.9 shows that the shifting of the minimum clearance does not have a significant effect on the hydro-dynamic pressure generation ability except for the case in which $r_b_1=200\mu m$ and $r_b_2=400\mu m$ with smaller width of convergent area than others. The reason for this trend is that scraping starts to occur locally. The existence of local scraping close to minimum clearance reduces the wetting area. Fig. 3.11 shows the local volume ratio distribution of '400+200', '300+300' and '200+400' cases at the leading edge and Fig. 3.12 compares the whole oil volume ratio distribution between these three cases.

![Fig. 3.11 the Local Oil Volume Ratio at Leading Edge](image-url)
In Fig. 3.11, the black circles mark where local scraping happens. It is evident that compared to the first two cases, '200+400' case starts to have visible local scraping area. In Fig.3.12, The white lines roughly label the wetting region boundary where hydrodynamic pressure can be generated. These two figures together clearly show that shifting the lowest point very close to the leading edge will result in the existence of local scraping and furthermore reducing the wetting region especially the convergent wetting area. As a result, it may decrease the hydrodynamic pressure generation ability. However, from Fig.3.10, it is noticeable that shifting the lowest point does not really affect the shear stress.

As a result, the twist of ring does not really affect the top-two-ring single phase hydrodynamic behavior if it is assumed that the twist only shifts the lowest point of the ring and the lowest point is not close to the leading edge. This is a very important conclusion that will be utilized in later chapter for cycle model.

3.6 Conclusion

This chapter introduces the numerical method for the top-two-ring single phase model (the single phase model) and two correlations that relate the hydro-dynamic pressure generation
ability and shear stress to the nominal clearance between the ring and the liner. This single phase model evaluates the effect of ring profile, liner surface roughness and partial oil supply boundary condition but it does not include the elevated gas pressure effect. There is another important assumption about the calculation domain. A larger calculation domain usually means higher hydrodynamic pressure generation ability since hydrodynamic pressure might still be generated lightly between the ring and the roughness asperities even when the nominal clearance is high. However, it also means more hydrodynamic friction due to the increase of area of attachment. The true physics here can be complicated and has not been comprehensively understood yet. Further research is suggested.

This chapter also shows that the effect of non-negligible twist of the ring on hydro-dynamic behavior. Under the assumption that the twist of the ring only shifts the lowest point but does not change the parabolic profile shape, this non-negligible twist does not have remarkable effect on the hydro-dynamic behavior unless the minimum does not shift so close to the leading edge that shrinks the full film region. This is an important conclusion that will be used in the next chapter for cycle model calculation.
4 Multiphase Top-two-ring Model

The top-two-ring model without consideration of gas pressure effects was shown in previous chapter. However, in real engine running conditions, land pressures varies with time especially for top two rings. The high gas pressure tremendously changes the top two rings' friction behavior, oil transport and wear condition. In this chapter, a multiphase model with the consideration of elevated gas pressure environment for the top two rings will be presented as well as the key results and correlations.

4.1 Introduction

In real engine running conditions, land pressures are changing as shown in Fig. 4.1 at 2000 rpm and full load case. With large pressure differences between the leading edge and the trailing edge of the ring, gas could travel around and penetrate the oil through the valleys which determines the boundary pressure of full film region (shown in Fig. 4.2). As a result, the pressure in partial film region is no longer a constant as cavitation pressure because of the existence of high pressure air. Li et al. developed a multiphase model which captures all the features. [30] The assumptions of multiphase model are the same as those of the single phase model except for the boundary gas pressure. Li separated the oil calculation and gas calculation based on the fact that gas traveling speed is much higher than the sliding speed and then related gas pressure and density with ideal gas law so that it could tract the entrapped gas bubble pressure. [30] A remarkable point here is that when gas bubble expands, the partial pressure of the gas in the gas bubble will decrease while partial pressure of the oil vapor remains the same (determined by temperature, assuming a iso-thermal process). As a result, lower limit of the gas bubble pressure is the assumed oil vapor pressure due to the vapor comes out of oil. In summary, unlike the pure oil vapor bubbles, the bubble of gas and vapor mixture trapped in the liquid oil cannot disappear. [30]
Fig 4.1 IC Engine Running Condition at 2000 rpm, Full Load

(P1 - given cylinder pressure)
4.2 Governing Equations

Based on the Reynolds equation and full attachment assumption, in the gas-oil coexistence region, the governing equations for oil and gases are expressed respectively as follows,

\[
\text{Oil Phase:} \quad \frac{\partial (\phi_o h)}{\partial t} + \frac{U}{2} \frac{\partial (\phi_o h)}{\partial x} = \frac{\partial}{\partial x} \left( \phi_o \frac{h^2}{12 \mu_o} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left( \phi_o \frac{h^2}{12 \mu_o} \frac{\partial p}{\partial y} \right) \quad \text{(eq. 4.1)} [30]
\]

\[
\begin{align*}
1 & \quad 2 \quad 3 \quad 4
\end{align*}
\]
Gas Phase: \[ \frac{\partial (\phi_m \rho_g h)}{\partial t} + \frac{U}{2} \frac{\partial (\phi_m \rho_g h)}{\partial x} = \frac{\partial}{\partial x} \left( \phi_m \rho_g \frac{h^3}{12 \mu_g} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left( \phi_m \rho_g \frac{h^3}{12 \mu_g} \frac{\partial p}{\partial y} \right) \] (eq. 4.2) [30]

Here \( \Phi_o \) is the oil volume ratio and \( \Phi_m \) is the volume fraction of gas-vapor mixture, so
\[ \phi_o + \phi_m = 1 \] and \( \rho_g \) is gas density, \( \mu_o \) refers to oil dynamic viscosity and \( \mu_g \) refers to gas dynamic viscosity. The gas phase and oil phase share the same pressure in the partial film

region and term (2) is approximately of the same order of magnitude of term (3) and term (4). The difference between the comparison of (6), (7) and (8) and the comparison of (2), (3) and (4) is only the difference between the oil viscosity and gas viscosity. Considering \( \mu_g \) is much smaller than \( \mu_o \), we can conclude that term (6) is much smaller than term (7) and (8). Then gas phase collapse to the form:

\[ \frac{\partial (\phi_m \rho_g h)}{\partial t} = \frac{\partial}{\partial x} \left( \phi_m \rho_g \frac{h^3}{12 \mu_g} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left( \phi_m \rho_g \frac{h^3}{12 \mu_g} \frac{\partial p}{\partial y} \right) \] (eq. 4.3)

If we treat gas as ideal gas,

\[ P = \rho_g RT \] (eq. 4.4)

Substitute eq. 4.4 into eq. 4.3,

\[ 2 \frac{\partial (\phi_m \rho g)}{\partial t} = \frac{\partial}{\partial x} \left( \phi_m \frac{h^3}{12 \mu_g} \frac{\partial p^2}{\partial x} \right) + \frac{\partial}{\partial y} \left( \phi_m \frac{h^3}{12 \mu_g} \frac{\partial p^2}{\partial y} \right) \] (eq. 4.5)

According to Li’s theory, gas pressure determines oil boundary pressure and drives the oil flow which changes the oil volume ratio. Then the oil volume ratio determines the new gas traveling path. He also pointed out that since pressure wave is much faster than oil transportation (represented by \( \Phi_m \) or \( \Phi_o \)), the pressure variation in gas phase takes negligible time compared to oil movement. In addition the dynamic viscosity of gas is much smaller than that of oil, so during oil transportation gas is at quasi-steady state and the time derivative term is negligible [30]. Eq. 4.5 becomes:
4.3 The Separation of Hydro-dynamic Pressure in Multiphase Model

The multiphase model could give robust result of hydro-dynamic pressure under elevated gas pressure boundary condition. However, its drawback is distinct as well, the time consumption problem. The multiphase model calculation requires a large amount of time which is at least 4 times of the single phase model. There is a need to approximate the multiphase model result using a less time-consuming method.

4.3.1 The Separation of Hydrodynamic Pressure with Superposition Method

For a smooth surface and smooth curved ring, within the full film region the Reynolds equation is linear. In this case, the superposition method could be used to help separate the hydrodynamic pressure. However, when local cavitation or gas penetration occurs, the universal equations used to describe both partial and full film regions are non-linear. Thus, strictly speaking, it is impossible to apply superposition method to separate the pressure. However, considering the amount of time taken by the multiphase model calculation, it is interesting to examine what will be the error if decomposing the multiphase hydrodynamic pressure using superposition method. If the error is tolerable, tremendous savings in computation time can be achieved.

To apply superposition method, the whole system is divided into two sub-systems. The first one is the system with elevated boundary gas pressure at steady state (no time derivative term). The second one is the transient system without boundary gas pressure which is the same as single phase model. Therefore, the total hydro-dynamic pressure could be separated into gas steady pressure and single phase pressure:

\[ P_{tot} = P_{gas, steady} + P_{single} \]  \hspace{1cm} (eq. 4.7)
However the existence of local cavitation or gas penetration brings in the non-linearity to the universal equations so that we need to add a higher order term to correct this non-linear effect:

\[ P_{\text{tot}} = P_{\text{gas,steady}} + P_{\text{single}} + O(1) \quad \text{(eq. 4.8)} \]

The boundary gas pressure mainly drives the gas flow and it has more effect on the gas than on the oil. To simplify the calculation, a new variable is proposed called static gas pressure which is the pressure generated if only gas is used as lubricant between the ring and the liner. To calculate static gas pressure, gas phase Reynolds equation is used with \( \Phi_m=1 \). Substitute static gas pressure to replace gas steady pressure and combine the difference between these two with the higher order correction term. The equation becomes:

\[ P_{\text{tot}} = P_{\text{gas,static}} + P_{\text{single}} + \Delta \quad \text{(eq. 4.9)} \]

where \( \Delta \) refers to the correction term.

The ring geometry is shown in Fig.4.3. The ring width is 600\( \mu \text{m} \). The macro profile is parabolic and the minimum clearance is located at 400\( \mu \text{m} \) (\( r_{b1}=400\mu\text{m} \)). The ring profile is demonstrated in the following figures with dark blue line. This ring profile will be used for all the simulation in this chapter. Fig. 4.4 shows the result of average pressure over the steps where the total pressure is stable. The ring moves from right to left at the speed of 3m/s so the left side is the leading edge with the boundary pressure of 5bar and the right side is the trailing edge with the boundary pressure of 20bar.
Fig. 4.3 the Ring Geometry
Fig. 4.4 Calculation Result of Multiphase Model
In the first graph, the solid blue line is the average total pressure in circumferential direction and the line marked with red circles is the average gas static pressure in circumferential direction. In the second graph, the dynamic pressure is defined as

\[ P_{\text{dynamic}} = P_{\text{tot}} - P_{\text{gas,static}} \]  

(eq. 4.10)

and the difference between the dynamic pressure and the single phase pressure is the correction term shown in eq.4.9. The third graph shows the comparison of oil volume ratio between multiphase model and single phase model. The two curves of oil volume ratio distribution are rather close with each other which provides concrete evidence for the pressure separation despite the non-linear effect from the universal variable.

4.3.2 The Effect of Boundary Gas Pressure on Hydro-dynamic Pressure Separation

Fig.4.5 to Fig.4.8 shows the effect of boundary gas pressure. The leading edge pressure is fixed at 5bar and the change of the boundary gas pressure difference is accomplished by changing the trailing edge pressure from 5bar to 30bar. Thus the boundary gas pressure difference is from 0 to 25bar.
Fig. 4.5 Total Pressure with Different Boundary Gas Pressure Difference

Fig. 4.6 Gas Static Pressure with Different Boundary Gas Pressure Difference
Fig. 4.7 Dynamic Pressure with Different Boundary Gas Pressure Difference

Fig. 4.8 Oil Volume Ratio with Different Boundary Gas Pressure Difference
In all the figures, $p_w$ refers to the leading edge pressure and $p_e$ refers to the trailing edge pressure. With a constant leading edge pressure, an increase of the trailing edge pressure causes an increase in the total pressure difference. However, if focused on the comparison of the dynamic pressure and the oil volume ratio distribution, it is noticeable that with the increase of the gas pressure difference the deviation from the multiphase model to the single phase model (the correction term) becomes more and more significant, though they are still very close. This point reveals a fact that the increase of boundary gas pressure difference aggravates non-linear effect of the system.

4.3.3 The Effect of Ring Sliding Speed and Oil Viscosity on Hydro-dynamic Pressure Separation

Chen et al. demonstrated in his master thesis that it is really the multiplication of the ring sliding speed and the oil viscosity affects the hydro-dynamic pressure generation ability rather than either of the independent variables [34]. It suggests that instead of research on two variables independently it is more convenient to focus on only one variable. Here we will focus on sliding speed.

Fig.4.9 to Fig.4.11 show the effect of sliding speed which varies from 1m/s to 10m/s. The leading edge pressure is still fixed at 5bar and the trailing edge pressure changes from 5bar to 30bar by an incremental of 5bar.
Fig. 4.9 Calculation Result of Sliding Speed at 1m/s
Fig. 4.10 Calculation Result of Sliding Speed at 3m/s
Fig. 4.11 Calculation Result of Sliding Speed at 10m/s
The three figures demonstrate the effect of the multiplication of sliding speed and oil viscosity. When judging from the numbers, it is easy to notice that the hydro-dynamic pressure increases with the multiplication of these two parameters which is consistent with Chen's conclusion in his master thesis [34]. Another noticeable point is that when the sliding speed decreases, the variance of dynamic pressure and oil volume ratio between multiphase model and single phase model becomes more significant. This indicates that during the decrease of sliding speed, the non-linear effect becomes more important and the correction term becomes larger. The reason could be explained from the aspect of hydro-dynamic pressure. When the multiplication of sliding speed and oil viscosity decreases, the hydro-dynamic pressure between the ring and the liner decreases which reduces the resistance of the oil boundary to the high pressure gas flow so that the oil volume ratio is not able to keep its shape as single phase model and on the other hand aggravate the non-linear effect.

4.3.4 Conclusion

In general, the hydro-dynamic pressure of multiphase model could be separated into single phase hydro-dynamic pressure that describes the pressure generated by the oil transportation, gas static pressure that is from the elevated boundary gas pressure and a correction term that is from the non-linearity of the system. The above results show the robustness of this separation method under the condition that the ratio of boundary gas pressure difference to the single-phase hydrodynamic pressure is relatively small. However, when the ratio becomes large, the variance of the dynamic pressure and the oil volume ratio between multiphase model and single phase model is definitely not negligible and the separation method is not valid any more.

4.4 The Simplified Model and its Validation in Real Engine Condition

The above section introduces a method to separate the hydro-dynamic pressure from multiphase model. In this section, a simplified model will be proposed based on the pressure separation method.
The pressure separation method is given by eq. 4.9

\[ P_{tot} = P_{gas,static} + P_{single} + \Delta \]  \hspace{1cm} (eq. 4.9)

and in most cases when the ratio of single phase hydrodynamic pressure to boundary gas pressure difference is low, the correction term \( \Delta \) is negligible, so that under this conditions a simplified model is proposed:

\[ P_{tot} = P_{gas,static} + P_{single} \]  \hspace{1cm} (eq. 4.10)

The simplified model gives a possibility to approximate the multiphase model hydrodynamic pressure with that of single phase model and the gas static pressure. The reason for the approximation is that the time consumption for multiphase model is much larger than the summation of time required for single phase model and gas static pressure calculation.

As mentioned above, this simplified model works when the correction term is negligible. Therefore, two new variables are defined here as a criterion to judge its validation. The first one is the relative error, \( e \), which compares the correction term and the total pressure; the second one is called \( \beta \) that compares the boundary gas pressure difference and the single phase hydrodynamic pressure.

\[ e = \frac{P_{tot} - (P_{gas,static} + P_{single})}{P_{tot}} \]  \hspace{1cm} (eq. 4.11)

\[ \beta = \frac{\Delta P}{\mu U L} \]  \hspace{1cm} (eq. 4.12)

where \( \Delta P \) refers to the boundary gas pressure difference, \( \mu \) refers to the oil dynamic viscosity, \( U \) refers to the ring sliding speed, \( L \) refers to the wetting width of the ring and \( \bar{h} \) refers to the average clearance between the ring and the liner. It could be defined as a ratio of the oil volume within the ring to the wetting area. As explained before, the relative error is related to the ratio of the boundary gas pressure difference to the single phase hydrodynamic pressure. Thus \( e \) could be treated as a function of \( \beta \), which is plotted below:
The plot indicates that when $\beta$ is smaller than 1.5, the relative error is less than 5%, which is sufficiently small to be neglected in real engineering case. In other words, when $\beta$ is less than 1.5, the simplified model is a good approximation for engineering calculation. Using Tian’s 2D cycle model, it is not difficult to calculate $\beta$ in real engine cycles [22, 31]. Fig. 4.13 demonstrates the calculation result of $\beta$ under the engine condition of 2000rpm, full-load with a turbo charger.
In Fig. 4.13, the x-axis is the crank angle. From -360° to -180° is the intake stroke, -180° to 0° is the compression stroke, 0° to 180° is the combustion stroke and 180° to 360° is the exhaust stroke. β is smaller than 1.5 in most of the engine cycle except for a small period of time in expansion stroke when the pressure in combustion chamber is increased dramatically due to combustion process. However, even within this period, the maximum β value is still less than 2. Therefore, in general the simplified model is valid for real engine cycle condition which makes it possible to approximate the multiphase model hydrodynamic pressure using the single phase model and the gas static pressure. In this way, it is able to save considerable calculation time.

4.5 The Effect of Ring Clearance and Ring Profile on Static Gas Pressure

It can be figured out from eq. 4.6 that the static pressure is not affected by viscosity (neither of oil nor of gas) and ring sliding velocity. The static pressure is only a function of ring-liner clearance and ring profile shape. This section will discuss about the possible influence of these two factors.
4.5.1 The Static Pressure and Ring-liner Clearance

The ring-liner system works like a throttle for the gas flow. When the clearance decreases, the pressure drop across the gap is increasingly closer to a step function; on the other hand when the clearance increases, the pressure drop across the gap more resembles a linear function. Fig. 4.14 shows the trend.

The ring profile is the same as before, with the ring width of 600 μm and the lowest point is located at 400μm position. The leading edge pressure is 5 bar and the trailing edge pressure is 30 bar. The ring-liner nominal clearances are represented by the ratio of the clearance to the plateau roughness $\sigma_p$ which is 0.077μm.
The ring-liner nominal clearances change the static pressure distribution significantly but in terms of the average static pressure over the whole ring which is the actual factor in the ring dynamic calculation, the influence is not really considerable. It is because when the clearance is reduced, the static pressure is lower before the minimum clearance point but after this point, the static pressure with lower nominal clearance starts to exceed that with thicker nominal clearance. It compensates the difference before the minimum clearance. Fig. 4.15 shows the effect of clearance on static pressure.

![Graph showing average gas static pressure at different clearances](image)

**Fig. 4.15 Average Gas Static Pressure at different clearances**

In the figure, sigmap refers to the plateau roughness \( \alpha_p \). It is obvious that the change of the ring-liner clearance has negligible influence on the average static pressure when the gas boundary pressure difference is within 30 bar.
4.5.2 The Static Pressure and Ring Profile Effect

The ring profile effect here mainly means the effect of shifting minimum clearance location on gas static pressure. In real cycle engine, due to high gas pressure and large initial force, the twist of top two rings cannot be neglected. It changes the ring profile, mainly the minimum clearance location (especially for the rings with flatter parabolic profile shape). Fig. 4.3 shows the geometry of the ring. \( r_{b1} \) refers to the upwind width and \( r_{b2} \) refers to the downwind width. Fig. 4.16 shows the effect of shifting lowest point with the same parabolic shape.

![Static Pressure with Shifting Minimum Clearance](image)

**Fig. 4.16 the Effect of Shifting Minimum Clearance on Gas Static Pressure Distribution**

In Fig.4.16, the legend of ‘200+400’ means that \( r_{b1} = 200 \mu m \) and \( r_{b2} = 400 \mu m \) and the rest has the same format. This figure clearly illustrates that when the minimum clearance shifts toward the lower pressure boundary, the ring will have more area suffering higher boundary gas pressure and the average gas static pressure will increase. Fig. 4.17 shows the effect on average gas static pressure.
All the results point out a conclusion that minimum clearance shifting has a considerable effect on average gas static pressure. In real engine condition, the twist of the ring is mainly responsible for the shifting, which should be considered in the ring dynamics calculation.

### 4.6 The Separation of Shear Stress in Multiphase Model

Shear stress is basically the velocity derivative in the direction perpendicular to the average plateau height plane. The velocity profile comes from two parts: viscous flow and pressure flow, that is $f_s = f_v + f_p$. As explained in the last section, the pressure flow in multiphase model is separated into single phase pressure flow and gas static pressure flow which could be also applied in shear stress calculation. It also shows that when $\beta$ is small, the oil volume ratio distribution of single phase model is close to that of multiphase model and here it is assumed that these two are equal to each other.

$$f_s|_{multi} = f_v|_{multi} + f_p|_{multi} = \frac{\mu U \phi_{multi}}{h} + h \frac{\partial p_{tot}}{\partial x} \phi_{multi} \quad (eq. \ 4.13)$$
where $\frac{\mu U \phi_{\text{multi}}}{h}$ is the viscous flow shear stress and $h \frac{\partial P_{\text{tot}}}{\partial x} \phi_{\text{multi}}$ is the pressure flow shear stress. With the assumption of $\phi_{\text{multi}} = \phi_{\text{single}}$, eq. 4.13 becomes

$$f_{\text{sl multi}} = \frac{\mu U \phi_{\text{single}}}{h} + h \frac{\partial P_{\text{tot}}}{\partial x} \phi_{\text{single}}$$  \hspace{1cm} \text{(eq. 4.14)}

Substitute the simplified model into eq. 4.14, then

$$f_{\text{sl multi}} = \frac{\mu U \phi_{\text{single}}}{h} + h \left( \frac{\partial P_{\text{single}}}{\partial x} + \frac{\partial P_{\text{gas static}}}{\partial x} \right) \phi_{\text{single}}$$  \hspace{1cm} \text{(eq. 4.15)}

With the definitions of $f_{\text{v single}} = \frac{\mu U \phi_{\text{single}}}{h}$, $f_{\text{p single}} = h \frac{\partial P_{\text{single}}}{\partial x} \phi_{\text{single}}$ and $f_{\text{p gas static}} = h \frac{\partial P_{\text{gas static}}}{\partial x} \phi_{\text{single}}$, eq. 4.15 becomes,

$$f_{\text{sl multi}} = f_{\text{v single}} + f_{\text{p single}} + f_{\text{p gas static}} = f_{\text{sl single}} + f_{\text{p gas static}}$$  \hspace{1cm} \text{(eq. 4.16)}

$f_s$ represents the total shear stress, $f_v$ represents the shear stress due to viscous flow, $f_p$ represents the shear stress due to pressure flow, $\mu$ represents the oil dynamic viscosity, $U$ represents the ring sliding speed, $\Phi$ represents oil volume ratio, $P_{\text{tot}}$ represents the total hydrodynamic pressure in multiphase model and $P_{\text{gas static}}$ represents the gas static pressure. The subscripts ‘single’ means the parameter is from single phase model, the subscript ‘multi’ means the parameter is from multiphase model and the subscript ‘gas, static’ means the parameter is due to gas static pressure flow. To estimate multiphase model shear stress, the viscous flow shear stress is from single phase model and the pressure flow shear stress is from the summation of single phase model pressure flow and the gas static pressure flow.

With the above assumptions, eq. 4.16 gives an easier approach to estimate the shear stress in multiphase model. Fig. 4.18 and Fig. 4.19 demonstrate the comparison of single phase model shear stress/multiphase model shear stress and the comparison of multiphase model shear stress/the estimated shear stress using eq. 4.16 respectively. The ring profile is the same as described before. The leading edge boundary gas pressure is 5bar and the trailing edge boundary gas pressure is 30bar.
Fig. 4.18 the Comparison of Multiphase Model Shear Stress and Single Phase Model Shear Stress at Sliding Speed of 3m/s
In Fig. 4.18, a very important point is that the shear stress due to single phase model viscous flow (blue circle line) is quite close to the shear stress due to multiphase model viscous flow (blue solid line). It proves our assumption that the oil volume ratio distribution of single phase model is close to that of multiphase model. The difference between the single phase model shear stress and the multiphase model shear stress is mainly from pressure flow. However, the difference is compensated by adding the static pressure flow stress to the single phase model and the total shear stress from both sides are almost identical (in Fig.4.19) which is also due to a fact that the viscous flow shear stress of multiphase model is slightly larger than that of single phase model but the pressure flow shear stress of multiphase model is slightly smaller than that of the approximation method. These two terms balance out after summation.
The method is able to give a good approximation based on the assumption that $\phi_{multi}$ is equal to $\phi_{single}$. When $\beta$ becomes large, this assumption collapses and the error of the approximation is no longer negligible, as shown in Fig. 4.20 and Fig. 4.21.

**Fig. 4.20** the Comparison of Multiphase Model Shear Stress and Single Phase Model Shear Stress at Sliding Speed of 1m/s
The Comparison of Multiphase Model Shear Force and Simple Model Shear Stress ($v=1\text{m/s}$)

Fig. 4.21 the Comparison of Multiphase Model Shear Stress and Estimated Shear Stress Using Eq. 4.16 at Sliding Speed of $1\text{m/s}$

When the sliding speed is reduced to $1\text{m/s}$, the effect of gas static pressure becomes increasingly important; the oil volume ratio distributions of multiphase model and single phase model are no longer close to each other. Therefore, the approximation of the multiphase model shear stress using eq.4.16 is not valid.

4.7 Cycle Model Using Simplified Top-Two-Ring Multiphase Model Correlations

From previous sections, the total hydro-dynamic pressure could be written as:

$$P_{tot} = P_{gas, static} + P_{single} \quad (\text{eq. 4.10})$$
And $P_{\text{single}} = \left( \frac{h_{\text{OCR}}}{h_{\text{prof}}} \right)^{\kappa_p} \frac{\mu V}{(\mu V)_{\text{lo}}} \left( \alpha_P P_{\text{OCR}} \right) \left( \frac{h_{\text{prof}}}{\sigma_p} \right)^{-K_{\text{OCR}}}$ \hspace{1cm} (eq. 3.8)

At each crank angle, with the assumption that the system is axisymmetric and given the boundary pressure, ring tension, ring twist, piston speed and oil viscosity, it is easy to solve the force balance equation of the ring in radial direction. The ring geometry and the free body diagram of the ring are shown in Fig. 4.22.

Fig 4.22 Ring Geometry and FBD of the Ring in Radial Direction

$P_1$ and $P_2$ are the gas boundary pressure, $P_{12}$ is the gas pressure inside the ring groove and $\theta$ is the ring twist angle. Because of ring twisting, the lowest point shifts in sliding direction especially for those with flatter parabolic profile. As shown previously, it does not change the average single phase hydro-dynamic behavior significantly but it does have a significant effect on average gas static pressure and brings difficulty to correlate it. As a result, it does not require a modification to the single phase hydro-dynamic pressure/stress correlation but it is necessary to calculate the gas static pressure separately. With a given ring twist, ring profile and gas boundary pressure, the calculation is not difficult. As shown in the previous section, the ring clearance does not change the gas static pressure significantly, so neither does the roughness. Therefore, to save calculation time it is proper to assume the liner surface is smooth in the gas static pressure calculation. With the assumption that the ring is at quasi-steady state in the whole cycle, then the summation of the force should be zero. The force balance equation
together with the hydro-dynamic pressure and contact pressure correlations, help evaluate the ring-liner nominal clearance.

Then the hydro-dynamic friction could be obtained by substituting the clearance into shear stress correlation and the contact friction is calculated by multiplying the friction coefficient and the contact pressure. The total friction is the sum of hydro-dynamic friction and contact friction. In the last section, it proves that the multiphase model hydro-dynamic shear stress could be separated into single phase model shear stress and gas static pressure flow shear stress:

\[ f_{s|\text{multi}} = f_{s|\text{single}} + f_{p|\text{gas,static}} \]  
\[ f_{s|\text{single}} = P_0 \frac{\mu \nu}{n_{\text{prof}}} \left( \frac{h_{\text{OCR}}}{h_{\text{prof}}} \right)^{K_f} \]  

Due to the difficulty in correlating gas static pressure, the shear stress from gas static pressure flow has to be neglected. This assumption brings error to the friction calculation and needs to be addressed in future work.

4.7.1 Calculation Results of an Example Surface, Liner A

Fig. 4.23 shows the result of ring-liner clearance at each crank angle (degree). The liner is a relatively smooth liner (\( \sigma_p = 0.045 \mu m \)) with shadow valleys (\( R_v = 0.194 \mu m \)), labeled as Liner A. OCR tension is 28N, and second ring and top ring tension is 9N. OCR ring-land-width is 0.15mm and second ring and top ring effective lubrication width are both 0.6mm (this is a prediction from running Tian’s 2D-model first [22]). The engine runs at 2000rpm, full load, with a bore diameter of 82.5mm and a stroke of 92.8mm. The green lines are the results of oil control ring. If the top two rings function properly, the oil control ring should not suffer elevated gas pressure. Correspondingly in the figure the oil control ring clearance is quite symmetric. The blue line is the result of the top ring and the red line is the result of the second ring. These two rings are affected by the elevated gas pressure and during the compression stroke and expansion stroke when gas pressure increases dramatically, the top two rings is exposed to high
pressure gas in the ring groove which pushes the rings significantly close to the liner surface. As a result, the clearance of these two rings is rather small in these two strokes, which is shown in the figure.

Fig. 4.23 the Result of Ring Clearance at Each Crank Angle (Liner A)

Fig. 4.24 shows the result of friction force generated between the ring and the liner surface at each crank angle. The existence of high gas pressure at the end of the compression stroke and the start of the expansion stroke causes the small clearance of top two rings and it increases the contact pressure and contact friction significantly as well as the hydrodynamic friction. The friction of the second ring that is affected by elevated gas pressure less and the oil
control ring that is not affected by elevated gas pressure is roughly symmetric about the line of zero degree.

Fig 4.24 Friction Result at Each Crank Angle (Liner A)
Fig. 4.25 and Fig. 4.26 demonstrate a comparison between the cycle model based on multiphase model calculation and Tian's 2D-model [22]. Tian's 2D-model takes the liner roughness effect into consideration for oil control ring calculation but not for top-two-ring calculation. The cycle model based on multiphase model calculation uses Tian's 2D model oil control ring calculation result as an input and calculates the top-two-ring friction behavior with the consideration of liner roughness effect.

![Comparison of Nominal Clearance to 2D-Model (Liner A)](image)

**Fig. 4.25 a Comparison of Nominal Clearance to 2D-Model (Liner A)**
The difference between 2D-model and the current model is only the roughness effect. The liner surface used in the cycle calculation is Liner A with relatively smooth plateau roughness and shadow valleys ($\sigma_p=0.045\mu m$, RVK=0.194$\mu m$). This kind of surface is not favorable for retaining the oil on the liner (only small amount of oil is stored in the shadow valleys) and as a result the oil supply to the top two rings is limited. With the limited oil supply, the wetting area is small and concentrated around the minimum clearance. Thus, the roughness effect plays much more important role in hydrodynamic pressure generation than the ring profile effect.
which is shown in Fig. 4.27. Compared to ring profile effect with a large barrel drop, the roughness effect is not really an efficient way for hydrodynamic pressure generation.

The smooth liner actually has even worse ability to retain the oil on the liner but due to the purely smooth surface, all the hydrodynamic pressure is generated by the barrel shape. Therefore, the smooth liner has better hydrodynamic pressure generation ability. Fig. 4.28 shows the comparison of average hydrodynamic pressure between Liner A and a smooth liner.
under the same condition, namely, the same sliding speed, oil viscosity, leading edge pressure, trailing edge pressure, oil control ring clearance and profiled ring clearance.

![Graph showing Average Total Pressure Comparison between Rough Liner and Smooth Liner](image)

**Fig. 4.28 a Comparison of Average Hydrodynamic Pressure between Liner A and a Smooth Liner**

As shown above, for top two rings the smooth surface has better hydrodynamic pressure generation ability than Liner A. Thus, with the same ring tension, the clearance between the ring and the smooth liner should be larger than that between the ring and Liner A. It matches the results shown in Fig. 4.25. The nominal clearance between the ring and Liner A is smaller which leads to larger hydrodynamic friction and contact friction. In Fig. 4.26, the smooth liner generates larger contact friction in expansion stroke. It is because in 2D-model, the contact area is the entire surface but in the current cycle model, there is an assumption that the
contact pressure could be only generated on plateau area. Therefore it has less contact area and as a result it has less contact pressure and contact friction.

4.7.2 Calculation Results of an Example Surface, Liner B

To compare the roughness effect of the current model, the friction prediction of another liner surface measurement is done here. This liner is labeled as Liner B. It has a relatively small plateau roughness ($ap=0.04\mu m$) and deep valleys ($Rvk=0.879\mu m$). OCR land-width is 0.2mm and the top ring and the second ring effective lubrication width are 0.6mm, the same as those used of Liner A. The engine running condition and ring tension are the same as those used in Liner A calculation. Fig. 4.29 and Fig. 4.30 show the results of nominal clearance and friction.

![Graph showing ring clearance at each crank angle for Liner B](image)

*Fig. 4.29 the Result of Ring Clearance at Each Crank Angle (Liner B)*
Fig 4.30 Friction Result at Each Crank Angle (Liner B)
Fig. 4.31 to Fig. 4.32 compare the results of Liner B between the current model and 2D-model.

**Fig. 4.31 a Comparison of Nominal Clearance to 2D-Model (Liner B)**
Fig. 4.32 a Comparison of Friction Force to 2D-Model (Liner B)

Liner B with deep valleys starts to become favorable for keeping the oil left by the oil control ring (oil could be stored in the deep valleys). If we assume that top two rings could take advantage of the oil stored in the valleys, the oil supply to the top two rings becomes relatively sufficient, leading to larger wetting area. Thus, the ring profile effect plays a more important role than roughness effect in hydrodynamic pressure generation and the hydrodynamic pressure generation ability of Liner B is better than that of a smooth liner. As a result, the ring clearance calculated by the current model is larger than that calculated by 2D-model which is reflected in Fig. 4.31. However, this does not include the end of compression stroke and the entire expansion stroke. The reason is that during this period of time, the pressure inside the ring groove is significant and it pushes the ring quite close to the liner. Under this condition, the
roughness effect on hydro-dynamic pressure generation starts to play an important role which is similar to what happens to Liner A.

Fig. 4.32 shows the friction comparison between the friction of the current model and 2D-model. Fig. 4.31 shows that the clearance calculated by the current model is higher than that calculated by 2D-model. As a result the hydrodynamic friction from the current model should be smaller than that from 2D-model. However, Fig. 4.32 shows the opposite trend. The reason is believed to be the full attachment assumption. The full attachment assumption artificially increases the area that can generate hydrodynamic shear stress to the entire ring region but the 2D-model states that only the fully flooded area that can generate hydrodynamic shear stress. This bias needs more research in the future.

4.7.3 Calculation Results of an Example Surface, Liner C

The third liner is labeled as Liner C with relatively rough plateau ($\sigma_p=0.079\mu m$) and deep valleys ($R_{\text{pk}}=1.250\mu m$). The OCR land-width is 0.2mm and the top ring and the second ring effective wetting width are 0.6mm, the same as the ones used for Liner A and B as well as other conditions. Fig. 4.33 to Fig. 4.34 show the results of nominal clearance and friction and Fig. 4.35 to Fig. 4.36 compare the results of Liner C between the current model and 2D-model.
Fig. 4.33 the Result of Ring Clearance at Each Crank Angle (Liner C)
Fig 4.34 Friction Result at Each Crank Angle (Liner C)
Fig. 4.35 a Comparison of Nominal Clearance to 2D-Model (Liner C)
Liner C has much deeper valleys ($R_{vk}=1.250 \mu m$) and it favors the most for keeping the oil on the liner left by the oil control ring. As a result, its oil supply to top two rings is the most sufficient and the hydrodynamic pressure generation ability of top two rings are the best among all the three liners. Fig. 4.35 shows that compared to smooth liner, the nominal clearance of top two rings are much higher which matches what was mentioned above.

4.7.4 The Comparison among the Three Liners

To gain a more direct sense of how the roughness affects the top two ring friction behavior, this chapter demonstrates a comparison of the calculation results by the current model from all the
three liners shown in previous sections. Fig 4.37 shows a comparison of the surface structure. As mentioned above, Liner A has a relatively smooth plateau and shadow valleys, Liner B has a relatively smooth plateau and deep valleys and Liner C has a relatively rough plateau and deep valleys.

![Image of surface structure comparison](image)

**Table 4.1** Ring-Land-Width or Effective Wetting Width Used in the Current Cycle Model

<table>
<thead>
<tr>
<th></th>
<th>Liner A</th>
<th>Liner B</th>
<th>Liner C</th>
</tr>
</thead>
<tbody>
<tr>
<td>OCR land-width (mm)</td>
<td>0.15</td>
<td>0.2</td>
<td>0.2</td>
</tr>
<tr>
<td>Second Ring Effective Wetting Width (mm)</td>
<td>0.6</td>
<td>0.6</td>
<td>0.6</td>
</tr>
<tr>
<td>Top Ring Effective Wetting Width (mm)</td>
<td>0.6</td>
<td>0.6</td>
<td>0.6</td>
</tr>
</tbody>
</table>

*Fig. 4.37 Surface Measurement of Liner A, Liner B and Liner C (the color bar unit is in meter and the sliding direction is from top to bottom)*

Table 4.1 shows a comparison of ring width or effective wetting width used in the current cycle calculation.
Fig. 4.38 shows a comparison of average hydrodynamic pressure as a function of ring-liner clearance among three liners. Solid pink lines represent the results of Liner A, solid blue lines represent the results of Liner B and solid red lines represent the results of Liner C. The lines marked with circles represent the results for OCR and the lines marked with squares represent the results for top-two-ring when OCR clearance is around 0.24\(\mu\)m. The figure clearly shows that the order of the hydrodynamic pressure generation ability of top two rings from high to low is Liner C, Liner B and Liner A especially in the case when the top-two-ring clearance is larger than the oil control ring clearance. It matches the calculation result shown in previous sections.

![Fig. 4.38 Average Hydrodynamic Pressure as a Function of Ring-liner Clearance](image-url)
Fig. 4.39 shows a comparison of friction results among three liners. The blue lines represent top ring results, the red lines represent second ring results, the green lines represent OCR results and the orange lines represent total friction results. The solid lines represent the results from Liner A, the dashed lines represent the results from Liner B and the dotted lines represent the results from Liner C.
From Fig. 4.39, it is noticeable that for top two rings, the hydrodynamic friction from Liner C is less than that from Liner B and the hydrodynamic friction from Liner B is less than that from Liner A. The reason is that if we assume the top two rings could take advantage of the oil stored in the valleys, then Liner C and Liner B which has deep valleys so that they are more favorable for keeping the oil on the liner left by the oil control ring. Thus the oil supply to the top two rings is more sufficient and it leads to higher hydrodynamic pressure generation ability which could give higher ring-liner clearance and at the same time reduce the hydrodynamic friction. However, for the top ring, when the high gas pressure pushes the ring very close to the liner, the contact friction from Liner C is larger than that from Liner B because of larger plateau roughness. This is the reason why in compression stroke and expansion stroke, Liner C has higher friction than Liner B.

For the oil control ring, even though Liner C could provide more hydrodynamic pressure as shown in Fig. 4.38, the plateau roughness of Liner C is larger than that of Liner A and Liner B which results in larger contact friction. At this engine speed and oil control ring tension, the oil control ring does not always operate in fully hydrodynamic region and contact friction is important near bottom dead center and top dead center, especially for Liner C. As a result, the friction from Liner C is larger than that from Liner A and Liner B. The difference of the oil control ring friction between Liner A and Liner B is probably due to the difference of ring-land-width. The oil control ring-land-width used in Liner A calculation is 0.15mm but that used in Liner B calculation is 0.2mm.

Fig. 4.40 shows a comparison of total friction mean effective pressure (FMEP) among all the three liners and Fig. 4.41 shows a comparison of boundary contact FMEP among all the three liners.
Fig. 4.40 a Comparison of Total FMEP (1, 2 and 3 correspondingly refer to Liner A, Liner B and Liner C)
Fig. 4.41 a Comparison of Boundary Contact FMEP (1, 2 and 3 correspondingly refer to Liner A, Liner B and Liner C)
These two figures illustrate that the results of current model is not always consistent with that of 2D-model. Looking at the total FMEP of the entire ring pack (Fig. 4.40), the 2D model says that Liner A could provide better friction behavior than Liner B and Liner B could provide better friction behavior than Liner C. However, the current model shows an opposite trend that Liner C has better friction behavior than Liner B and Liner B has better friction behavior than Liner A. As a result, considering roughness effect on top two rings, it would change the friction behavior from that without considering roughness effect and it has a possibility to change the trend for the entire ring pack.

4.8 Conclusion

This chapter introduces the top-two-ring multiphase model (the multiphase model) to evaluate the average hydro-dynamic pressure and shear stress generated between the ring and the liner. This model is an improvement from the single phase top-two-ring model (the single phase model). It considers not only the effect of ring profile, liner surface roughness and partial oil supply but also the elevated gas pressure [29]. Due to the fact that the calculation time of multiphase model is quite long, there is a need to reduce the modeling time. Hence, a simplified model based on a separation of multiphase model hydro-dynamic pressure and shear stress method is introduced. In general, this simplified model separates the multiphase model hydro-dynamic pressure/shear stress into the single phase model hydro-dynamic pressure/shear stress and gas static pressure flow/shear stress. Then the chapter demonstrates that the simplified model is valid when the ratio of the single phase hydro-dynamic pressure over the boundary gas pressure difference is relatively small.

A cycle friction behavior calculation is then introduced based on the simplified model correlation and some results are shown to prove the robustness of the cycle calculation and the simplified model. With the three examples of friction calculation of different liner roughness, the current model could reflect their different roughness effect and the results are robust. If we assume that top two rings could take advantage of the oil stored in the valleys, the surfaces with deeper valleys which are a favorable factor to keep oil left by the oil control ring can give
better oil supply to top two rings. Hence, it can provide better hydrodynamic pressure
generation ability and as a result reduce their friction.
5 Conclusion

5.1 Summary and Conclusion

The objective of this thesis is to develop a complete model using deterministic method to simulate the friction behavior between a piston ring pack and a liner, especially for the top two rings. It has also discussed the effect of roughness and ring profile, elevated gas pressure and partial oil supply.

Chen developed a deterministic TLOCR model which uses a small patch of surface measurement as an input to model the friction behavior for the whole cycle but this method runs into some difficulties after the liner has been run for a while. The liner surface micro structure would change from spot to spot due to different wear condition and a small patch of measurement cannot provide enough information. Thus during the evolution process of the liner roughness, the contact model is not accurate. Then, a multiple-point-correlation method is proposed and the evolution process of the liner roughness is simulated by changing the plateau roughness so that the contact model could reflect the reality. Some experimental results are shown here to verify the method.

Unlike oil control rings, the oil inlet boundary condition of top two rings is not fully flooded and its profile plays an equal role as the liner roughness in hydro-dynamic behavior between the ring and the liner. The elevated gas pressure is also important but it has not been considered in Chapter 3. Therefore, the deterministic top-two-ring model in this chapter is called the single phase top-two-ring model (single phase model). A numerical method for single phase top-two-ring deterministic calculation introduced by Li. has been shown as well. Furthermore, correlations that relate single phase hydro-dynamic pressure and shear stress to the minimum nominal clearance between the ring and the liner are demonstrated as well as the effect of ring profile on hydro-dynamic behavior. The effect of shifting the lowest point on hydrodynamic behavior is discussed as well in this chapter. It seems that as long as no local scraping occurs, shifting the lowest point does not have a significant effect on hydrodynamic pressure generation.
The forth chapter introduces an improved top-two-ring model that considers the elevated pressure effect. This model is called the multiphase top-two-ring model (multiphase model). The governing equations for the multiphase model were developed by Li et al. Based on the superposition method, the multiphase model hydro-dynamic pressure and shear stress could be separated into two parts: one from single phase model, the other from static pressure flow. If the ratio of the single phase hydro-dynamic pressure over the boundary gas pressure difference is relatively small, this method is valid. In the chapter, the current cycle model based on the simple model correlation is also shown at the end with three example surfaces with different roughness characteristics. The three examples prove that the current cycle model could reflect the roughness effect very well and it can give a robust result. Finally, a conclusion has been made that the surfaces with deeper valleys which is a favorable factor to keep the oil left by the oil control ring, can give better oil supply to top two rings. Hence, if assuming that top two rings could take advantage of the oil stored in the valleys it can provide better hydrodynamic pressure generation ability and as a result reduce their friction.

5.2 Potential Future Work

The implementation of experiments to verify the cycle model is the next stage of work. The friction measurement under fired condition is difficult because the combustion process varies from cycle to cycle and it is difficult to obtain the average land pressure. The friction test measures the total friction including the friction between the piston and the liner but this friction source has not been considered in the ring pack friction model. A potential solution is to run the engine without any ring and measure the piston only friction, but due to the variance in combustion process it is uncertain to conclude that the ring pack friction is the total friction minus the piston only friction. Another problem is to determine the calculation region. The ring width of top-two-ring is much larger than that of oil control ring. As a result, even under fully flooded boundary condition, it is difficult for the wetting area to cover the whole ring width. Then it is important to determine the wetting area as a calculation region instead of the whole ring width because as shown in Chapter 3, enlarging the calculation region would increase the friction from calculation.
Another potential future work is to include the hydro-dynamic shear stress due to static pressure flow in the cycle model. The static pressure variation in real engine cycles makes it difficult to correlate its shear stress. In the current model, the shear stress due to static pressure flow is not considered which deviates from the reality and requires future work.
Reference


