The Influence of Barrel Offset on Cylinder Liner-Piston Ring Lubrication State

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Abstract. The shape of the barrel surface is an important parameter of the piston ring, and it is also one of the key factors determining the lubrication state of the cylinder liner-piston ring friction pair. To research the influence of barrel surface shape on the lubrication performance of cylinder liner piston ring, a transient three-dimensional lubrication model of cylinder liner piston ring is established by combining the average Reynolds equation and green wood formula. The lubrication performance of this friction pair under different working conditions and different cylinder surface shape is analysed, which provides the basic numerical research for the shape optimization of piston ring.

1. Introduction
The friction loss of cylinder liner piston ring accounts for a large part of the energy loss of internal combustion engine, which has a great influence on the performance of internal combustion engine [1]. As the volume of internal combustion engine is smaller and smaller, the power is higher and higher, the speed is faster and faster, and the working conditions of cylinder liner piston ring are more and more severe, the influence of the friction pair on the efficiency and reliability of internal combustion engine is more and more important [2].

Piston ring lubrication properties are affected by the work conditions and oil performance, but also directly linked with the structural parameters of the piston ring [3]. The cylinder surface of piston ring is one of the conditions for forming hydrodynamic lubrication. The position and size of cylinder surface bulge will lead to different lubrication states under the same working condition, and the lubrication performance will directly affect the efficiency and durability of internal combustion engine, so it is very necessary to study the shape of cylinder surface. The first ring is taken as research object, and the comprehensive surface roughness, cylinder deformation in the circumferential direction, lubricant viscosity and other factors are taken into account.
2. Constructing Lubrication Numerical Model

2.1. Three-dimensional Average Reynolds Equation [4-5]

\[
\frac{\partial}{\partial x} (\rho \phi \frac{h^3}{\mu} \frac{\partial p}{\partial x}) + \frac{\partial}{\partial y} (\rho \phi \frac{h^3}{\mu} \frac{\partial p}{\partial y}) = 6U \frac{\partial (p\bar{h}_T)}{\partial x} + 6U \sigma \frac{\partial (p \phi \sigma)}{\partial x} + 12 \frac{\partial (p \bar{h}_T)}{\partial t} \tag{1}
\]

where \( \phi_x \) is the pressure flow factor in x direction; \( \phi_y \) is the pressure flow factor in y direction; \( \phi_s \) is shear flow factor; \( \sigma \) is the Comprehensive roughness; \( p \) is average film pressure; and \( \bar{h}_T \) is the expected value of actual film thickness.

2.2. Pressure and Shear Flow Factor [6-7]

The pressure flow factor \( \phi_x \) and \( \phi_y \) are the ratio of the average pressure flow of rough surface and the pressure flow of smooth surface. Rough surface direction factor in x and y directions are reciprocal each other, therefore:

\[
\phi_x = \begin{cases} 
1 - Ce^{-\gamma} & \gamma \leq 1.0 \\
1 + CH^{-\gamma} & \gamma > 1.0 
\end{cases} \tag{2}
\]

\[
\phi_y (H, \gamma) = \phi_x (H, \frac{1}{\gamma}) \tag{3}
\]

Shear flow factor \( \phi_s \) could show the affection of additional flow when the two rough surface have relative slip. The formula of \( \phi_s \) is:

\[
\phi_s = (\frac{\sigma_1}{\sigma})^2 \Phi_s (H, \gamma_1) - (\frac{\sigma_2}{\sigma})^2 \Phi_s (H, \gamma_2) \tag{4}
\]

\[
\Phi_s = \begin{cases} 
A_h \exp(-a_2 H + a_2 H^2) & H \leq 5.0 \\
A_h \exp(-0.25 H) & H > 5.0 
\end{cases} \tag{5}
\]

2.3. The Radial Equilibrium Equation of Piston Ring

\[
F_g + F_z = F_p + W_a \tag{6}
\]

where \( F_g \) is gas pressure back of piston ring; \( F_z \) is piston ring tension; \( F_p \) is general radial lubrication film pressure.

2.4. Asperity Contact Model

When the height of the rough peak is Gaussian distribution, the contact force \( W_a \) and the actual contact area \( A_c \) of the rough surface can be obtained by using the rough surface contact theory proposed by Greenwood [8]:

\[
W_a = \frac{16 \sqrt{\pi} \eta \sigma}{15} \int \beta \phi_z (H) dA \tag{7}
\]
\[ A_i = \pi^2 (\eta \beta \sigma)^2 \int_A F(H) dA \]  

(8)

where supposed \( \eta \beta \sigma = 0.04, \sigma/\beta = 10^{-3} \), and \( E \) is composite modulus of elasticity, \( F_{2S}(H) \) and \( F_2(H) \) could be found in reference [6].

2.5. Temperature Model

This paper uses Woschni relation to calculate temperature:

\[ T(x) = T_{\text{tbc}} - (T_{\text{tbc}} - T_{\text{tde}}) \times (x/S)^{0.5} \]  

(9)

where \( T_{\text{tbc}}, T_{\text{tde}} \) are the temperatures of cylinder liner’s TDC and BDC, and \( S \) is the stroke of piston ring.

3. Calculation Result and Analysis

In the example, the parameters are set as follows: the cylinder diameter is 50 mm, the rod length is 192 mm, the crank radius is 57 mm, the speed of rotation is 4000 rpm, the piston ring height is 3 mm, the piston ring radial thickness is 2 mm, the cylinder liner roughness is 1.6 um, the piston ring roughness is 0.8 um, the difference between long axis and short axis of oval cylinder liner is 10 um, and the lubricating oil viscosity is 0.13 pa\( \cdot \)s. The pressure of air chamber and the pressure between the first, second ring are shown in figure 1.

![Figure 1. The pressure of air chamber and the pressure between the first, second ring.](image)

3.1. Barrel Shapes of Different Cases

Figure 2 shows the different barrel surfaces of three examples.

![Figure 2. The barrel surface shapes of different cases.](image)
In Example 1, the piston ring barrel surface height $\delta_0$ counted as 5.0 $\mu$m, no offset, in Example 2 barrel surface height calculation unchanged barrel surface offset $\delta_s = 0.45$ mm, in examples 3 barrels height unchanged barrel surface offset $\delta_s = -0.45$ mm, X-axis positive direction deviating from the combustion chamber.

### 3.2. Minimum Oil Thickness Analysis

The influence of Piston barrel surface offset ratio on minimum film thickness is shown in figure 3. The minimum value of minimum film thickness appears on the compression stroke near TDC during entire operating cycle in all three examples, it is because the dramatic surge cylinder pressure, piston rings speed close to zero and decrease the viscosity of lubricants and other factors, at this moment, the oil thickness has been less than two surface integrated roughness, indicating that the contact of asperity body on two solid surface of piston rings and cylinder liners has occurred, piston rings have been in mixed lubrication state. The calculated results are very consistent with the actual operation. The main wear occurs at TDC and BDC, and BDC is little better than TDC.

![Figure 3. Film thickness distribution in different crank angle.](image)

During downward movement of the piston, i.e., the intake and power strokes, with the positive offset, the length of Convergence wedge reduces, the length of the diverging wedge increases, it is not conducive to the formation of the film, the minimum film thickness is reduced compared symmetric barrel surface. During upward movement of the piston, i.e., the compression and exhaust strokes, the length of Convergence wedge increases, the length of the diverging wedge decreases, it is conducive to the formation of the film, and the minimum film thickness is increased compared symmetric barrel surface. The influence mechanism of piston ring surface barrel negative offset on minimum oil film is similar, but the phenomenon is just the opposite. During the intake and power strokes, the minimum film thickness is increased compared symmetric barrel surface, during the compression and exhaust strokes, the minimum film thickness is reduced compared symmetric barrel surface.

### 3.3. Friction Analysis

The effect of piston barrel surface offset on fluid friction is shown in figure 4. During downward movement of the piston, i.e., the intake and power strokes, the positive offset decreases the film thickness, but also increases the fluid friction acting on the piston ring. During upward movement of the piston, i.e., the compression and exhaust stroke, the positive offset increases the film thickness, but also reduces the fluid acting on the piston ring friction. The influence of piston ring barrel offset on fluid friction is corresponded to the minimum oil film thickness. The effect of piston ring surface barrel offset on total friction is shown in figure 5. Positive offset increases the maximum friction near TDC.
negative offset decreases the maximum friction near TDC. With the parameters given in this paper, the minimum friction power consumption will be obtained when the symmetrical barrel shape is selected.

4. Conclusion
According to the above results, the following conclusions could be obtained:

The influence mechanism of piston ring surface barrel offset on piston-ring lubrication and friction performance is similar in compression, exhaust, intake and power stroke, but the result is just the opposite. Positive offset increases the film thickness and decreases the fluid friction during the upward movement, during the downward movement the oil film thickness decreases, the fluid friction increases, the maximum frictional force increases, the action of the negative offset is opposite. Therefore, during the design of the surface shape of the piston ring barrel, the working conditions and piston ring friction losses must be taken into account to select the best barrel surface shape.

Figure 4. Fluid friction in different crank angle.  
Figure 5. Total friction in different crank angle.

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References
[1] Tung S C and McMillan M L 2004 Automotive tribology overview of current advances and challenges for the future Tribology International 37 (7) 517-536
[2] Wen S Z and Huang P 2002 Principles of Tribology Beijing, Tsinghua University Press
[3] Ye X M 2004 Numerical Investigation and Application of Three-Dimensional Lubrication Performance in Piston Ring Pack Huazhong University of Science and Technology
[4] Patir N and Cheng H S 1978 An average flow model for determining effects of three-dimensional roughness on partial hydrodynamic lubrication Transaction of ASME, Journal of Lubrication Technology 100 (1) 12-17
[5] Patir N and Cheng H S 1979 Application of average flow model to lubrication between rough sliding surfaces Transaction of ASME, Journal of Lubrication Technology 101 (2) 220-230
[6] Liu K, Gui C L and Xie Y B 1997 The study of circumferential non-uniformity of piston ring Lubrication Transaction of CSICE 15 (3) 281-289
[7] Liu K and X B 1995 A study of surface flow factors and mixed lubrication property of piston ring-cylinder of engine Chinese Internal Combustion Engine Engineering 16 (3) 66-72
[8] Greenwood J A and Tripp J H 1971 The contact of two nominally flat rough surfaces Proc. Inst. Mech. Eng. 185 (1) 625-633