Numerical analysis of the effects of compression ring wear and cylinder liner deformation on the thermal mixed lubrication performance of ring-liner system

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Abstract. The lubrication performance of roughness compression ring-cylinder liner system (CRCL) with axial-asymmetric new/worn compression ring and out-of-round cylinder liner is studied in this paper. By considering the oil film thermal effect, cavitation phenomenon, and surface roughness, the energy equations, mass conservation Jacoboson-Floberg-Olsson (JFO) cavitation algorithm, generalized average Reynolds equation, and Greenwood-Tripp asperity contact model are employed to investigate the frictional behaviors of CRCL. The oil film thickness, friction forces, wear, and power loss of CRCL are investigated at various wear stages of compression ring. The effects of the magnitude of cylinder liner deformation on the frictional characteristics of CRCL are also analyzed with consideration of compression ring conformability. Numerical results show that the deformation of cylinder liner and the wear of compression ring have a significant influence on the lubrication performance. It also suggests that the oil film thermal effect should be considered in the numerical analysis to provide an accurate prediction on the frictional behaviors of CRCL.

Keywords: Frictional performance / compression ring-cylinder liner system / thermal analysis / out-of-round cylinder liner / JFO cavitation algorithm

1 Introduction

With the development of modern internal combustion engines (ICEs), more and more attention has been paid to the reductions of friction loss and fuel energy consumption because they affect emissions and fuel efficiency greatly [1–3]. In ICEs, about 20% of total friction loss and 5% of fuel energy consumption are caused by the friction of piston ring-cylinder liner system (PRCL) [4,5]. Therefore, reducing the friction of PRCL is crucial for ICEs to improve the emissions and fuel efficiency [6,7]. In order to minimize the friction of PRCL, the frictional characteristics of PRCL have been extensively studied in recent years.

For PRCL in working conditions, various regimes of lubrication (hydrodynamic, mixed, and boundary) will be encountered in an engine cycle because of the rapidly changing velocity, highly fluctuating thermal and structural loads [8]. Generally, the PRCL is under hydrodynamic regime in the majority of the engine cycle, and under mixed or boundary regimes at piston dead centers [9]. In consideration of the dominant lubrication regime of hydrodynamic in the engine cycle of PRCL, Jeng et al. [10] conducted a one-dimensional lubrication analysis for PRCL by solving Reynolds equation. The minimum oil film thickness between the piston ring and the cylinder liner was investigated for various engine speeds, piston ring tensions, and piston ring face profiles. Livanos et al. [11] investigated the effects of engine speed and load on the friction of compression ring using a piston assembly friction model. Higher minimum oil film thickness and lower friction loss were observed for the piston ring with smaller load. Taylor et al. [12] studied the oil film thickness and pressure of PRCL using a one-dimensional lubrication model with and without squeeze film effects. The squeeze film effects were observed to have a great impact on the oil film thickness at
reversal positions. Usman et al. [13] investigated the oil film thickness, friction force, and power loss of PRCL for the oil with different viscosity grades. The low viscous oils showed better tribological performance under engine start-up conditions.

By considering the mixed or boundary lubrication regimes of PRCL at piston dead centers, Hamatake et al. [14] studied the frictional characteristics of a piston ring pack for different monograde and multigrade oils. Ali et al. [15] investigated the effects of the piston ring dynamic behavior and oil viscosity on the frictional characteristics of PRCL. Their results showed that the low viscosity oil has small friction loss and oil consumption. More recently, Obert et al. [16] investigated the influence of oil supply and liner temperature on the friction coefficient of PRCL. Rahmani et al. [17] studied the power loss and emissions of compression ring at various liner temperatures. Their results demonstrated that the control of liner temperature can help to reduce the power loss and emissions, and the friction coefficient was not influenced by the oil supply rate and liner temperature.

In this study, in order to investigate the frictional characteristics of CRCL are analyzed at various wear stages of compression ring. The effects of liner out-of-roundness on the lubrication performance of CRCL are also analyzed with consideration of compression ring conformity.

2 Theoretical model

2.1 Geometrical model and expression of oil film thickness

Figure 1 shows the schematic diagram of CRCL. The compression ring moves up and down in the cylinder liner to provide a seal between the combustion chamber and the crankcase. The time-varying velocity of compression ring can be expressed as [17]:

\[ U \approx R\omega \sin(\omega t) + \frac{R^2\omega}{2l} \sin(2\omega t), \]

where \( R \) is the radius of crankshaft, \( \omega \) is the angular velocity of crankshaft, and \( l \) is the length of connecting rod.

In IC engines, the face profile of compression ring is usually asymmetrical in the axial direction [20,21], and the wear of compression ring is inevitable because of serious solid-solid contact caused by the high cylinder pressure and temperature [30,31]. In this study, in order to investigate the
influence of compression ring wear on the lubrication performance of CRCL, the asymmetrical compression rings in various wear degrees or after various service time (i.e., the un-wear compression ring (new compression ring), run-in compression ring, and embedded compression ring) are considered. In details, the run-in compression ring represents the ring after its initial run-in wear (after 10 000 km service), and the embedded compression ring represents the ring at the end of its useful life (after 150 000 km service). Figure 2a shows the axial face profiles of the new compression ring, run-in compression ring, and embedded compression ring. Because of different wear degrees or service time, the axial face profiles of run-in compression ring, embedded compression ring and new compression ring are quite different. The different axial face profiles of run-in compression ring, embedded compression ring and new compression ring will result in different values of $h_{\text{ring}}$, and subsequently result in different oil film thicknesses between the compression ring and cylinder liner, and lubrication performance. By measuring the face profiles with an optical Talysurf profilometer, the additional gaps between the compression ring and cylinder liner caused by the axial face profiles of run-in compression ring, embedded compression ring and new compression ring can be fitted by the polynomial equations [21].

Likewise, the additional gap between the compression ring and cylinder liner caused by the axial face profile of run-in compression ring $h_{\text{ring}}$ can be written as [21]:

$$h_{\text{ring}} = -197.16x^6 + 578.85x^5 - 583.44x^4 + 200.37x^3 + 56.366x^2 - 62.709x + 13.857.$$  

(3)

The additional gap between the compression ring and cylinder liner caused by the axial face profile of embedded compression ring $h_{\text{ring}}$ can be written as [21]:

$$h_{\text{ring}} = -6.493x^6 - 49.845x^5 + 143.22x^4 - 98.541x^3 + 27.351x^2 - 17.338x + 7.4986.$$  

(4)

It should be noted that the units of $x$ and $h_{\text{ring}}$ are millimetre and micrometre, respectively.

Moreover, the cylinder liner is usually deformed because of the thermal loads, high combustion pressures, manufacturing errors, and load difference between thrust and anti-thrust sides [22,23]. The cylinder liner deformation also changes the profile of cylinder liner, and consequently causes an additional gap between the compression ring and cylinder liner. Considering the conformability of compression ring, the additional gap between the compression ring and the cylinder liner caused by the cylinder liner deformation $h_{\text{liner}}$ can be predicted as [27]:

$$h_{\text{liner}} = \max\{0, \Delta R - U_n\},$$  

(5)

with

$$\Delta R = \left[ \sum_{n=0}^{n} \left( \frac{(\Delta x)_n \cos[n(\phi - \phi_n)]}{2} \right) \right] - \Delta R_{\min},$$  

(6)
\[ U_n = \frac{3(F_1 + F_{bp})r^2(2r - a)^2}{Eba^3(n^2 - 1)^2}, \]  

where \( R \) is the radius variation of cylinder liner caused by the liner deformation, \( R_{min} \) is the minimum variation in cylinder liner radius. \( U_n \) is the elastic deformation of compression ring caused by the conformability of compression ring, \( a \) is the radial thickness of compression ring, \( r \) is the nominal radius of cylinder liner, \( E \) is the Young’s modulus of compression ring, \( \Delta c \) is the maximum deformation of cylinder liner, \( \phi \) is the circumferential position, \( \phi_0 \) is the circumferential position of the maximum liner deformation, \( F_1 \) is the tension force of compression ring, \( F_{bp} \) is the backpressure force. \( n \) is the order of cylinder liner deformation. According to the works of Usman et al. [27], a typical value of \( n = 4 \) is used in this work. Figure 2b shows the schematic diagram of out-of-round cylinder liner with 4th-order deformation and the conformability of compression ring. For CRCL, both of the wear of compression ring and cylinder liner deformation change the profiles of compression ring and cylinder liner, and subsequently result in different gaps between the compression ring and cylinder liner. In the lubrication analysis of CRCL, different gaps between the compression ring and cylinder liner means different oil film thickness in CRCL, which consequently result in different hydrodynamic pressure distribution and frictional behaviors. The oil film thickness for CRCL with deformed cylinder liner and new/worn compression ring \( h \) can be expressed as:

\[ h = h_0 + h_{ring} + h_{liner}, \]

where \( h_0 \) is the minimum oil film thickness between the compression ring and cylinder liner.

It should be noted that the thermal effect will result in different temperature distribution in the compression ring [27], and subsequently causes a thermal expansion/distortion. The thermal expansion/distortion reduces the ring-end gap, alters the oil film thickness, and subsequently results in an additional tension pressure acting on the compression ring. Usman et al. [27] discussed the influence of thermal expansion/distortion of compression ring on the performance of CRCL. They pointed out that the expansion/distortion of compression ring usually varies within several microns, which will result in negligible changes in the ring-end gap, tension pressure, oil film thickness, and frictional performance of CRCL. Therefore, based on the discussion of Usman et al. [27], the compression ring expansion/distortion caused by thermal effect is not considered in this study. Similar neglect of the effect of thermal expansion/distortion of compression ring can be also found in the works [24,28].

2.2 Governing equation

In this study, the hydrodynamic pressure of oil film in CRCL considering the variable density and viscosity in the \( x, y, \) and \( z \) directions can be described by the generalized Reynolds equation. Moreover, the surface roughness affects greatly the lubrication performance of CRCL, especially under the mixed/boundary lubrication regime [9]. In order to simulate the effects of surface roughness on the lubrication performance, two different methods (i.e., the direct/deterministic method and the indirect/stochastic method) are usually adopted [32,33]. In this study, an indirect/stochastic method proposed by Patir and Cheng [34] is employed to simulate the effects of surface roughness on the lubrication performance of CRCL because of its low cost in computation time and easy realization [33]. The generalized Reynolds equation with consideration of Patir and Cheng’s approach and mass conservation JFO cavitation algorithm [35,36] is given as follows [37]:

\[ \begin{align*}
\frac{\partial}{\partial x} \left( \phi_0 G_{1x} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left( \phi_0 G_{1y} \frac{\partial p}{\partial y} \right) &= \frac{\partial}{\partial x} \left( \theta G_{2x} \frac{u_2 - u_1}{F_0} + \theta G_3 u_1 + \theta G_3 \phi_s \frac{u_2 - u_1}{2h} \right) \\
&+ \frac{\partial}{\partial t} (\theta G_3),
\end{align*} \]

with

\[ \begin{align*}
G_{1x} &= G_{1y} = \int_0^h \rho(z) \left[ \int_0^z \frac{1}{\eta(z')} d z' - \frac{F_1}{F_0} \int_0^z \frac{1}{\eta(z')} d z \right] d z, \\
G_{2x} &= \int_0^h \rho(z) \left[ \int_0^z \frac{1}{\eta(z')} d z' \right] d z, \\
G_3 &= \int_0^h \rho(z) d z = F_0 = \int_0^h \frac{1}{\eta(z)} d z,
\end{align*} \]

where \( x \) and \( y \) are the coordinates in the axial and circumferential directions, \( p \) is the hydrodynamic pressure of oil film, \( \rho \) is the oil density, \( \eta \) is the oil viscosity, \( t \) is the time. \( u_1 = 0 \) and \( u_2 = U \) are the velocities of cylinder liner and compression ring. \( \phi_s \) and \( \phi_p \) are the pressure flow factors, \( \phi_s \) is the shear flow factor, and they are defined by Patir and Cheng [34,38]. It should be noticed that the Patir and Cheng’s approach can be applied to arbitrary surface roughness structures with Gaussian distributed asperity heights [34]. For CRCL in actuality, the cylinder liner surface is usually cross-hatched and honed, and the asperity height of roughness liner surface in running-in condition usually obeys an approximate Gaussian distribution [39]. However, according to the work of Morris et al. [39], the Patir and Cheng’s approach is reasonable and acceptable in the simulation of roughness CRCL when the asperity height of roughness liner surface is assumed to be Gaussian distributed. This indicated that the Patir and Cheng’s approach can be applied to simulate the roughness CRCL, even though the asperity height of roughness liner surface is not strictly Gaussian distributed [39]. More accurate simulation on the roughness surface can be also conducted from the measured topography [40].

Based on the JFO theory, the saturation of the fluid \( \theta \) is given as follows [37]:

\[ \begin{cases}
 p > p_c \text{ and } \theta = 1 \text{ pressure region} \\
 p = p_c \text{ and } \theta < 1 \text{ cavitation region},
\end{cases} \]  

where \( p_c \) is the cavitation pressure.
The pressure boundary conditions are necessary for the solving of oil film pressure. In the current study, a fully flooded lubrication condition is assumed, and the pressure boundary conditions are given as follows [21]:

\[
\begin{align*}
\frac{\partial p}{\partial x} |_{x=s} &= p_l, \\
\frac{\partial p}{\partial x} |_{x=0} &= p_i,
\end{align*}
\]  

(11)

where \(p_i\) is the cylinder pressure, \(p_l\) is the gas pressure under the compression ring.

### 2.3 Asperity contact model

The lubrication regime of CRCL is usually mixed at piston dead centers. In this case, the asperity contact between the compression ring and the cylinder liner is inevitable. The tension force \(F_t\) and the backpressure force \(F_{bp}\) are balanced by the oil film force \(F_{oil}\) and the asperity contact force \(F_{asp}\), as shown in Figure 1. In order to investigate the asperity contact behaviors in CRCL, the well-known Greenwood-Tripp asperity contact model is employed to predict the asperity contact force \(F_{asp}\) and the asperity contact area \(A_c\) [21, 24].

\[
F_{asp} = \frac{8\sqrt{2}}{15} \pi (\kappa \beta \sigma)^2 E' \sqrt{\beta} A F_{2.5}(\lambda),
\]

(12)

\[
A_c = \pi^2 (\kappa \beta \sigma)^2 A F_2(\lambda),
\]

(13)

where \(E'\) is the equivalent elastic modulus of CRCL, \(\sigma\) is the comprehensive surface roughness of CRCL, \(\kappa\) is the density of asperity, \(\beta\) is the curvature radius of asperity, \(\lambda = h/\sigma\) is the oil film thickness ratio, \(A\) is the apparent asperity contact area. \(F_{2.5}(\lambda)\) and \(F_2(\lambda)\) are the statistic functions of roughness surface with Gaussian distributed asperities, they can be expressed as [41]:

\[
F_{2.5}(\lambda) = -0.0046\lambda^3 + 0.0574\lambda^4 - 0.2958\lambda^5 + 0.7844\lambda^2 - 1.0776\lambda + 0.6167,
\]

(14)

\[
F_2(\lambda) = -0.0018\lambda^5 + 0.0281\lambda^4 - 0.1728\lambda^3 + 0.5258\lambda^2 - 0.8043\lambda + 0.5003.
\]

(15)

### 2.4 Rheological relationship

The oil viscosity and density affect greatly the frictional performance. Generally, the oil viscosity and density are determined by the oil film temperature and pressure. The viscosity-pressure-temperature relationship and density-pressure-temperature relationship of oil film can be described by [42]:

\[
\eta = \eta_0 \exp \left\{ \ln \eta_0 + 9.67 \left[ -1 + (1 + 5.1 \times 10^{-9} p)^{-2} \left( \frac{T}{T_0} - 138 \right) \right] \right\},
\]

(16)

\[
\rho = \rho_0 \left[ 1 + \frac{0.6 \times 10^{-9} p}{1 + 1.7 \times 10^{-9} p} - 6.5 \times 10^{-4} (T - T_0) \right],
\]

(17)

with

\[
\begin{align*}
\eta_0 &= \kappa_{\eta}/[5.1 \times 10^{-9} (\ln \eta_0 + 9.67)], \\
\rho_0 &= \kappa_{\rho} (T_0 - 138)/(\ln \eta_0 + 9.67)
\end{align*}
\]

(18)

where \(\kappa_{\eta}\) is the viscosity-pressure coefficient of oil film, \(\kappa_{\rho}\) is the viscosity-temperature coefficient of oil film, \(T_0\) is the oil film temperature under atmospheric pressure, \(\eta_0\) and \(\rho_0\) are the oil film viscosity and density under atmospheric pressure when oil film temperature equals to \(T_0\).

### 2.5 Thermal analysis

During the running of CRCL, the friction heat causes an increase of oil film temperature, and then changes the viscosity and density. In this study, the three-dimensional energy equation is employed to calculate the oil film temperature distribution [42].

\[
c_p \left( \rho \frac{\partial T}{\partial x} + \rho \frac{v}{\mu} \frac{\partial T}{\partial y} \right) = k_p \frac{\partial^2 T}{\partial z^2} - T \frac{\partial \rho}{\partial T} \left( u \frac{\partial \rho}{\partial x} + v \frac{\partial \rho}{\partial y} \right) + \eta \left( \frac{\partial u}{\partial z} \right)^2 + \left( \frac{\partial v}{\partial z} \right)^2.
\]

(19)

Moreover, in order to calculate the temperature distribution of oil film accurately, the heat transfer equations for the solids (i.e., compression ring and cylinder liner) should be solved simultaneously because some heat is conducted to the solids. Because the solids densities and velocities in the \(x\) direction are constants, and the solids velocities in the \(y\) and \(z\) direction are zero, the heat transfer equations for the compression ring and cylinder liner can be rewritten as [42]:

\[
c_r \rho_r u_2 \frac{\partial T}{\partial x} = k_r \frac{\partial^2 T}{\partial z^2},
\]

(20)

\[
c_i \rho_i u_1 \frac{\partial T}{\partial x} = k_i \frac{\partial^2 T}{\partial z^2},
\]

(21)

where \(c_p\) and \(k_p\) are the specific heat capacity and thermal conductivity of oil film, \(c_i\) and \(k_i\) are the specific heat capacity and thermal conductivity of compression ring, \(c_r\) and \(k_r\) are the specific heat capacity and thermal conductivity of cylinder liner, \(\rho_r\) and \(\rho_i\) are the densities of compression ring and cylinder liner. \(u\) and \(v\) are the velocities of oil film in the \(x\) and \(y\) directions, and their expressions can be obtained according to the work of Liu et al. [42].

The heat flux continuity equations at the oil-ring interface and the oil-liner interface are given as follows [28]:

\[
k_x \frac{\partial T}{\partial z} |_{z=h} = k_p \frac{\partial T}{\partial z} |_{z=h}.
\]

(22)
For the thermal analysis of CRCL, the thermal boundary conditions are needed, and they are given as follows [28]:

\[
\begin{align*}
T &= T_{in} \text{ at inflow on inlet side} \\
\frac{dT}{dx} &= 0 \text{ at outflow on inlet side} \\
\frac{dT}{dx} &= 0 \text{ on outlet side} \\
T &= T_{liner} \text{ at cylinder liner surface} \\
T &= T_{ring} \text{ at compression ring surface}
\end{align*}
\]

where \( T_{ring} \) is the initial temperature of compression ring, \( T_{liner} \) is the initial temperature of cylinder liner, \( T_{in} \) is the inlet temperature of oil film. According to the work of Shahmohamadi et al. [29], the inlet temperature of oil film \( T_{in} \) is equal to the initial liner temperature \( T_{liner} \). Therefore, the initial temperature of cylinder liner \( T_{liner} \) should be known a priori for the thermal analysis of CRCL.

### 2.6 Performance parameters

The friction force in CRCL \( f_{total} \) consists of hydrodynamic friction force \( f_{oil} \) and asperity friction force \( f_{asp} \), and their expressions are given as follows [28]:

\[
f_{total} = f_{oil} + f_{asp},
\]

\[
f_{oil} = \iint_{\Omega} \left( -\frac{\mu U}{h} (\Phi_f + \Phi_{ls}) + \frac{h}{2} \frac{\partial p}{\partial x} \Phi_f \right) d\Omega,
\]

\[
f_{asp} = \tau_0 A_c + \alpha_0 F_{asp},
\]

where \( \Phi_f, \Phi_{ls} \) and \( \Phi_f \) are the friction-induced flow factors, \( \Omega \) is the oil film area, \( \tau_0 \) is the shear stress constant, \( \alpha_0 \) is the asperity friction coefficient.

In order to evaluate the wear of CRCL, the wear load on the compression ring defined by Gulwadi [43] is calculated. It can be expressed as:

\[
W_{load} = \frac{1}{\theta_f} \int_{\theta_0}^{\theta_f} p_{asp} |U| d\theta_c,
\]

where \( \theta_c \) is the crankshaft angle, \( \theta_0 = 360^\circ \) for a two-stroke ICE or \( 720^\circ \) for a four-stroke ICE.

The power loss of CRCL can be evaluated as [28]:

\[
P_{loss} = |f_{total} U|.
\]

### 3 Results and discussion

#### 3.1 Validation

Before studying the effects of compression ring wear and cylinder liner deformation on the thermal mixed lubrication performance of CRCL, it is necessary to verify the model used in our study. The validation has been made using Gu et al. [28] and our models. Figure 3 shows the oil film pressures on the middle cross-section of the compression ring calculated by Gu et al. [28] and our models under the same input parameters and working conditions. As shown in Figure 3, the oil film pressure predicted by the proposed model is in good agreement with the results of Gu et al. [28], which suggests the validation of the proposed model. In what follows, in order to provide insight into the effects of compression ring wear and cylinder liner deformation on the thermal mixed lubrication performance of CRCL, the minimum oil film thickness, friction force, wear load, and power loss are calculated by the presented model. Tables 1–3 provide all the necessary parameters for the performance analysis of CRCL. Figure 4 shows the velocity of compression ring and the cylinder pressure when the engine speed is 2000 r min\(^{-1}\).
Table 3. Mechanical, thermal and surface properties of CRCL.

| Parameter                      | Value      |
|--------------------------------|------------|
| Elasticity modulus of ring     | 250 GPa    |
| Elasticity modulus of liner    | 120 GPa    |
| Surface roughness of ring      | 0.4 μm     |
| Surface roughness of liner     | 0.4 μm     |
| Poisson’s ratio of ring        | 0.3        |
| Poisson’s ratio of liner       | 0.3        |
| Density of ring                | 7700 kg m\(^{-3}\) |
| Density of liner               | 7200 kg m\(^{-3}\) |
| Thermal conductivity of ring   | 25 W m\(^{-1}\) K\(^{-1}\) |
| Thermal conductivity of liner  | 55 W m\(^{-1}\) K\(^{-1}\) |
| Specific heat capacity of ring | 460 J kg\(^{-1}\) K\(^{-1}\) |
| Specific heat capacity of liner| 460 J kg\(^{-1}\) K\(^{-1}\) |

3.2 Influence of thermal effect

In order to illustrate the necessity of thermal effect in the lubrication performance analysis of roughness CRCL with axial-asymmetric new/worn compression ring and deformed cylinder liner, the minimum oil film thickness and friction under isothermal condition and thermal condition are calculated for various compression ring wear stages and cylinder liner deformations. For the convenience of comparative analysis, the differences of minimum oil film thickness, hydrodynamic friction, and asperity friction under isothermal analysis and thermal analysis are defined as:

\[
\begin{align*}
\Delta h_0 &= h_{0, \text{isothermal}} - h_0 \\
\Delta f_{\text{oil}} &= f_{\text{oil, isothermal}} - f_{\text{oil}} \\
\Delta f_{\text{asp}} &= f_{\text{asp, isothermal}} - f_{\text{asp}}
\end{align*}
\]

where \(\Delta h_0\), \(\Delta f_{\text{oil}}\), and \(\Delta f_{\text{asp}}\) are the differences of minimum oil film thickness, hydrodynamic friction, and asperity friction between isothermal analysis and thermal analysis. \(h_{0, \text{isothermal}}\), \(f_{\text{oil, isothermal}}\), and \(f_{\text{asp, isothermal}}\) are the minimum oil film thickness, hydrodynamic friction, and asperity friction under isothermal condition.

Figure 5 shows the differences of minimum oil film thickness at various wear stages of compression ring when the maximum deformation of cylinder liner \(\Delta c = 0\,\mu\text{m}, 15\,\mu\text{m}, 25\,\mu\text{m},\) and \(35\,\mu\text{m}\), respectively. As can be seen in Figure 5, the oil film thermal effect has a great influence on the minimum oil film thickness. With the oil film thermal effect taken into consideration, the minimum oil film thickness is reduced, and the reduction of minimum oil film thickness is different at each wear stage of compression ring. The difference of minimum oil film thickness decreases with the increase of compression ring wear. The new compression ring is found to have larger difference of minimum oil film thickness, especially at the middle of the strokes because of the high viscous shear of oil film. At power stroke, the maximum deformation of cylinder liner is small, the generated friction heat can be quickly conducted by the compression ring and the cylinder liner. Therefore, a small difference of minimum oil film thickness is observed for CRCL.

Figure 6 shows the differences of hydrodynamic friction and asperity friction for various maximum deformations of cylinder liner. As shown in Figures 6a, c and e, the oil film thermal effect has a great influence on the hydrodynamic friction of CRCL. With the oil film thermal effect taken into consideration, the hydrodynamic friction is reduced, and larger difference of hydrodynamic friction is concentrated at the middle of the strokes because of the high viscous shear of oil film. At power stroke, the maximum deformation of cylinder liner is found to have a great effect on the difference of hydrodynamic friction, and the difference of hydrodynamic friction decreases with the increase of maximum deformation of cylinder liner. However, the effect of cylinder liner maximum deformation on the difference of hydrodynamic friction is negligibly
small at intake, compression, and exhaust strokes. Moreover, higher difference of hydrodynamic friction is also found for the embedded compression ring, especially at the middle of the power stroke. As shown in Figures 6b, d, and f, the oil film thermal effect increased the asperity friction of CRCL, and it can be seen that the cylinder liner maximum deformation and compression ring wear have greatly influence on the difference of asperity friction. The difference of asperity friction increases with the increase of maximum deformation of cylinder liner. Compared with the new compression ring and embedded compression ring, smaller difference of asperity friction is observed for the run-in compression ring because of strong squeeze effect and small asperity contact area.

3.3 Influence of compression ring wear and cylinder liner deformation

Based on the above discussion, it is demonstrated that the thermal effect of oil film has a great influence on the minimum oil film thickness and friction in CRCL. Therefore, the thermal effect of oil film should be considered in the analysis of CRCL to provide more detailed information about the frictional behaviors. In order to study the influence of compression wear and cylinder liner deformation on the tribological performance of CRCL, the minimum oil film thickness, friction, wear load, oil film temperature, and power loss are evaluated under thermal condition for various wear stages of compression ring and maximum deformations of cylinder liner.

Figure 7 shows the minimum oil film thickness in CRCL for different maximum deformations of cylinder liner. What is observed is that the minimum oil film thickness decreases with the increase of maximum deformation of cylinder liner, and the decrease is significant at the middle of the strokes. Moreover, it can be also seen that the lubrication regime of CRCL is changing from hydrodynamic lubrication ($\lambda > 4$) to mixed lubrication ($\lambda < 4$) when the compression ring reaches near the dead centers at power stroke ($\theta_c = 360^\circ$ or $540^\circ$). Compared with the new compression ring, smaller range of mixed lubrication regime is found for the run-in compression ring and the embedded compression ring, especially near the bottom dead center of power stroke ($\theta_c = 540^\circ$).

Figure 8 shows the hydrodynamic friction and asperity friction in CRCL for different maximum deformations of cylinder liner. From Figures 8a, c, and e, what is observed is that the difference of hydrodynamic friction between various maximum deformations of cylinder liner is negligibly small at intake, compression, and exhaust.
strokes. However, at power stroke, the hydrodynamic friction decreases with the increase of maximum deformation of cylinder liner, and lower hydrodynamic friction is observed for CRCL with embedded compression ring. As shown in Figures 8b, d, and f, it can be seen that the cylinder liner maximum deformation and compression ring wear have greatly influence on the asperity friction in CRCL. The asperity friction increases with the increase of maximum deformation of cylinder liner. Compared with the new compression ring and embedded compression ring, smaller asperity friction is observed for the run-in compression ring.

The deformation of cylinder liner changes the tribological performance of CRCL. Two important changes are the wear of compression ring and the thermal effect of oil film. In order to investigate the influence of cylinder liner
deformation on the wear of compression ring and the thermal effect of oil film, the wear load of compression ring and the oil film temperature difference between deformed cylinder liner and circular cylinder liner are calculated. Figure 9a shows the peak values of wear load for different maximum deformations of cylinder liner. As can be seen in Figure 9a, the peak value of wear load on the compression ring is an increasing function of maximum deformation of cylinder liner, and the increasing rate decreases with the increase of cylinder liner maximum deformation. Compared with the new compression ring, lower peak values of wear load are observed for the run-in compression ring and embedded compression ring, especially for the run-in compression ring. The reason is that the asperity friction of run-in compression ring is lower. Figures 9b, c and d show the oil film temperature differences between deformed cylinder liner and circular cylinder liner for new compression ring, run-in compression ring, and embedded compression ring. What is observed is that the deformation of cylinder liner causes an increase of oil film temperature, but the increase is very small. Compared with new compression ring and run-in compression ring, lower increase of oil film temperature is observed for the embedded compression ring at power stroke.

Figure 10 shows the power loss of CRCL for different maximum deformations of cylinder liner. As can be seen in Figure 10, the cylinder liner maximum deformation has a less effect on the power loss of CRCL at intake, compression, and exhaust strokes. At power stroke, lower power loss is observed for the embedded compression ring, and the power loss of CRCL decreases with the increase of cylinder liner maximum deformation.

4 Conclusion

In this paper, a thermal-mixed lubrication model is proposed to evaluate the lubrication performance of roughness CRCL with axial-asymmetric new/worn compression ring and deformed cylinder liner. The effects of cylinder liner deformation on the minimum oil film thickness, friction force, power loss, and wear in CRCL are analyzed for various compression ring wear stages. The related conclusions can be made as follows:

The oil film thermal effect has a great influence on the minimum oil film thickness and friction of CRCL. The difference of hydrodynamic friction between isothermal analysis and thermal analysis is a decreasing function of cylinder liner maximum deformation at power stroke. The difference of asperity friction between isothermal analysis and thermal analysis is an increasing function of cylinder liner maximum deformation. Compared with the new compression ring, smaller difference of minimum oil film thickness and frictions between isothermal analysis and thermal analysis are found for the embedded compression ring and the run-in compression ring.

The minimum oil film thickness decreases with the increase of maximum deformation of cylinder liner, but the decrease is very small at power stroke. The asperity friction, the peak value of wear load, and the oil film temperature increase with the increase of maximum deformation of cylinder liner. The cylinder liner maximum deformation is also found to have a great effect on the hydrodynamic friction and power loss of CRCL, especially at power stroke. Moreover, compared with the new compression ring, lower hydrodynamic friction and power loss are observed for the embedded compression ring, and lower asperity friction is observed for the run-in compression ring.
Finally, it is noteworthy that the lubrication condition in this study is assumed to be fully flooded. In practice, the lubrication condition of compression ring-cylinder liner system is not only fully flooded, but also starved in some points of their strokes because of scraping effect of oil control ring. The effect of compression ring wear and cylinder liner deformation on the lubrication performance of CRCL under starved lubrication condition is the subject of ongoing work with consideration of oil supply.

Fig. 8. Friction forces in CRCL for different maximum deformations of cylinder liner: (a) hydrodynamic friction of new ring; (b) asperity friction of new ring; (c) hydrodynamic friction of run-in ring; (d) asperity friction of run-in ring; (e) hydrodynamic friction of embedded ring; (f) asperity friction of embedded ring.

Nomenclature

- $a$: Radial thickness of compression ring
- $A$: Apparent contact area
- $A_c$: Asperity contact area
- $b$: Axial width of compression ring
- $c_p$: Oil specific heat capacity
- $c_r$: Specific heat capacity of compression ring
- $c_t$: Specific heat capacity of cylinder liner
Fig. 9. (a) Peak value of wear load; (b) oil film temperature difference for new compression ring; (c) oil film temperature difference for run-in compression ring; (d) oil film temperature difference for embedded compression ring.

\[ \Delta c \]
Maximum deformation of cylinder liner

\[ E \]
Young's modulus of compression ring

\[ E' \]
Equivalent elastic modulus of ring and liner

\[ f_{\text{total}} \]
Total friction force

\[ F_{\text{asp}} \]
Asperity contact force

\[ f_{\text{asp}} \]
Asperity friction force

\[ f_{\text{asp}, \text{isothermal}} \]
Asperity friction under isothermal condition

\[ \Delta f_{\text{asp}} \]
Difference of asperity friction between isothermal analysis and thermal analysis

\[ F_{\text{hp}} \]
Backpressure of compression ring

\[ F_{\text{t}} \]
Tension force of compression ring

\[ f_{\text{oil}} \]
Hydrodynamic friction force

\[ f_{\text{oil}, \text{isothermal}} \]
Hydrodynamic friction under isothermal condition

\[ \Delta f_{\text{oil}} \]
Difference of hydrodynamic friction between isothermal analysis and thermal analysis

\[ F_{\text{oil}} \]
Oil film force

\[ F_{2.5}(\lambda), F_2(\lambda) \]
Statistic function

\[ h \]
Oil film thickness

\[ h_0 \]
Minimum oil film thickness

\[ h_0_{\text{isothermal}} \]
Minimum oil film thickness under isothermal condition

\[ \Delta h_0 \]
Difference of minimum oil film thickness between isothermal analysis and thermal analysis

\[ h_{\text{ring}} \]
The gap between ring and liner caused by the ring face profile

\[ h_{\text{liner}} \]
The gap between ring and liner caused by the liner deformation

\[ l \]
Length of connecting rod

\[ n \]
Order of cylinder liner deformation

\[ p \]
Oil film pressure

\[ p_a \]
Ambient pressure

\[ p_{\text{asp}} \]
Asperity contact pressure

\[ p_c \]
Cavitation pressure

\[ p_t \]
Gas pressure under the compression ring

\[ P_{\text{loss}} \]
Power loss

\[ p_t \]
Cylinder pressure

\[ r \]
Nominal radius of cylinder liner

\[ R \]
Radius of crankshaft

\[ \Delta R \]
Radius variation of cylinder liner caused by the liner deformation

\[ t \]
Time

\[ T \]
Temperature

\[ T_0 \]
Oil temperature under atmospheric pressure

\[ u \]
Oil velocity in x direction
Velocity of compression ring
Elastic deformation of ring caused by the ring conformability
Oil velocity in y direction
Wear load of compression ring
Axial coordinate in global coordinate system
Circumferential coordinate in global coordinate system
Radial coordinate in global coordinate system

Greek symbols
\(\alpha_0\) Asperity friction coefficient
\(\beta\) Curvature radius of asperity
\(\theta\) Saturation of fluid
\(\theta_c\) Crankshaft angle
\(\kappa\) Density of asperity
\(\kappa_p\) Oil thermal conductivity
\(\kappa_r\) Thermal conductivity of compression ring
\(\kappa_l\) Thermal conductivity of cylinder liner
\(\kappa_{vp}\) Oil viscosity-pressure coefficient
\(\kappa_{vt}\) Oil viscosity-temperature coefficient
\(\lambda\) Ratio of oil film thickness to comprehensive surface roughness
\(\eta\) Oil viscosity
\(\eta_0\) Oil viscosity under atmospheric pressure
\(\rho\) Oil density
\(\rho_0\) Oil density under atmospheric pressure
\(\rho_r\) Density of compression ring
\(\rho_l\) Density of cylinder liner
\(\sigma_{ring}\) Surface roughness of compression ring
\(\sigma_{liner}\) Surface roughness of cylinder liner
\(\sigma\) Comprehensive surface roughness of ring and liner
\(\tau_0\) Shear stress constant
\(\omega\) Angular velocity of crankshaft
\(\phi\) Circumferential position
\(\phi_n\) Circumferential position of maximum liner deformation
\(\Phi_{fi}, \Phi_{fs}, \Phi_{fp}\) Friction-induced flow factors
\(\Phi_s\) Shear flow factor
\(\Phi_{zr}, \Phi_{zg}\) Pressure flow factors
\(\Omega\) Area of fluid film

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