Three-Dimensional Piston Ring–Cylinder Bore Contact Modeling

Increasing durability, preventing knocking combustion, improving fuel efficiency, and reducing pollutant emission characterize the needs for modern internal combustion engine design. These factors are highly influenced by the power cylinder system design. In particular, the piston ring to cylinder bore contact force distribution around the circumference of the piston rings must be optimized under all running conditions. To accomplish this, the ring manufacturers make the ring curvature nonconstant along the circumference. Most existing analytical tools are not able to simulate the variation along the ring circumference. In order to improve the understanding of this contact distribution and provide a high-fidelity ring design tool, a three-dimensional finite element piston ring model was developed to accomplish this variation. The modeling procedure and results are presented in this work. Experiments using a commercially available ring with negative ovality were conducted to validate the model. The ring free-shape profile and the ring cross section geometries were used as inputs to the model. Typical piston ring groove and cylinder wall temperatures were also model inputs to characterize thermal influences on the ring/bore interface forces. The ring/bore conformability was analyzed as a function of the ring radial displacements, cylinder bore constraint forces and thermal load changes to the ring. The model output showed radially separation gaps between the ring front face and the bore. This analysis provides an insight to evaluate the piston ring design. Together with an optimizer, the model can be used as a ring design tool to predict the ring free shape with a specified constraint force distribution pattern. Examples are given to demonstrate the capabilities of this numerical analytical tool. In addition, the 3D ring model can be used to improve the accuracy of existing lubrication, friction, and wear analysis tools and therefore improve the entire internal combustion engine power cylinder system design. [DOI: 10.1115/1.4030349]
beam elements with consideration of the ring lapping process [9]. Tomanik and Bruno developed a ring radial force distribution measurement method with pin gauge that shows the contact force variations at different circumferential locations [10]. Tomanik also proposed a new criterion for ring conformability [11]. Tejada and Padial showed different ring pack configurations’ impact on oil consumption and blowby [12]. However, the ring–cylinder bore interaction, including the separation gaps for the noncontact nodes and constraint forces for the in-contact nodes, remains a less understood topic in internal combustion engine study.

This work describes a 3D analytical ring model to addresses the pressure/force distribution and the ring conformability between the ring–cylinder bore interface. The main contribution of this work is that it solves the problem from the minimum strain energy point of view without any unrealistic boundary condition assumptions. The penalty method is used to find the contact pairs between the ring and the cylinder liner. The finite element model is described in the Modeling Approach section below, along with experimental measurement validation. Dynamic analysis including the influence of the piston secondary motions [13] is not the focus of the work.

Modeling Approach

In this section, the mathematical description of the finite element ring model is presented. Ring mesh is generated based on the ring outer diameter (OD) from the coordinate measuring machine (CMM) measurement and the cross-sectional geometries. The penalty method used to solve the ring–cylinder bore contact is also discussed with a force release approach.

Finite Element Description

The ring at its free shape is meshed with eight-node hexahedral elements. For each node, there are three degrees of freedoms (DOF) as the displacements in the X, Y, and Z directions in the global coordinate system. Thus an element has a total of 24DOFs as can be found in Fig. 1.

Figure 1 shows a cubic element, which is the parent element of the hexahedral elements. All the hexahedral elements are mapped to the parent element, whose shape functions are predefined. The details of the formulation can be found from Refs. [14,15]. For each element, there are also 24 equivalent nodal loads corresponding to the DOFs. The finite element formulation for each element can be expressed as

\[ K_i \cdot q_i = f_i \]  

(1)

where \( K_i \) is the stiffness matrix for the \( i \)th element, \( q_i \) and \( f_i \) are the DOFs and load vector for the \( i \)th element. Detailed formulation of the components for the \( K_i \) matrix and \( f_i \) vector can also be found in Ref. [15]. For each element, \( K_i \) is a 24 \( \times \) 24 matrix while \( q_i \) and \( f_i \) are both 24 \( \times \) 1 vectors.

After assembling all the elements based on their connectivity relations, the finite element expression for the system that defines the ring can be obtained

\[ K \cdot q = f \]  

(2)

The force vector \( f \) should take into consideration all loads acting on the ring, which include static ring tension load due to its free shape, gas pressure load around the ring, temperature gradient induced thermal load, friction load, etc. As the goal of this work is to calculate the static contact load due to the ring free shape, other loads’ influences are not discussed here except for ring tension and thermal load. However, the ring tension is still an unknown here since it is not clear if a certain node will be in contact against the cylinder wall or not. The method of searching for the in-contact nodes will be discussed in the following section, Force Release. The free-shape ring mesh is shown in Fig. 2.

Due to the symmetric property of the ring, the boundary condition for the ring is that the nodes at the ring back cross section (opposite to the ring gap) are fixed in the X-direction only in the ring plane. These nodes are allowed to move in the radial direction due to constraint forces or the reaction forces from its adjacent node. Additionally, the nodes can also displace in the axial direction (Z-direction) if the ring cross section is nonsymmetric.

Penalty Method

The penalty method is widely used for solving contact problems [16]. Compared to the other popular method, the Lagrange multiplier method, the order of the system is not increased using the penalty approach. Thus, the penalty method was chosen to solve the present ring–cylinder bore contact problem.

The base constrained optimization problem can be formulated using the principle of minimum total potential energy. This principle is a fundamental concept used in structure analysis analyzing structure deformation. It states that a structure should deform to a stationary state that minimizes its total potential energy, including the elastic strain energy and potential energy from the applied force. This principle is used to formulate the ring–cylinder bore contact problem. The formulation can then be expressed as

To find nodal displacement, \( q \) that minimizes the ring potential energy defined as

\[ K \cdot q = f \]  

(2)
\[ \Pi = \frac{1}{2} q^T K q - q^T f \quad (3) \]

and subject to the constraint
\[ r_i \leq R_B \quad (4) \]

where \( r_i \) is the radius of the \( i \)th cross section at the ring face and \( R_B \) is the radius of the cylinder wall.

The minimization of the potential energy requires the derivative of \( \Pi \) with respect to \( q \) vanishing, which is
\[ K q = f \quad (5) \]

This constraint states that the piston ring outer radius should stay in contact (for the “=” case) or within (for the “<” case) the cylinder wall surface. It is not realistic for the piston ring locating beyond the cylinder wall, which is the case when \( r_i > R_B \). At the same time, the constraint implies that if the node on the ring front face is in contact with the cylinder bore, the constraint force should be nonzero and along the radial direction pointing inward to the center of the cylinder bore; on the other hand, if the node on the ring front face is not in contact with the cylinder bore, the constraint force should be zero. It is obvious that high nonlinearity is involved in the present contact problem. The penalty method is used to solve \( q \) by instead solving a sequence of specially constructed unconstrained optimization problems. That is, with the penalty method, an additional term that accounts for constraints is introduced into the system potential energy found in Eq. (3) as
\[ \Pi_p = \frac{1}{2} q^T K q - q^T f + \frac{1}{2} \lambda^T g \quad (6) \]

where \( \lambda \) is the penalty number and \( g \) represents the node geometric constraint and can be expressed in the matrix form as
\[ g = A q - c \quad (7) \]

Here, \( c \) is the gap between the node’s initial free state position, and the final deformed state and the matrix \( A \) projects the nodal DOF to the gap. Now, the minimization of the modified potential energy accounting for the penalties requires the derivatives of \( \Pi_p \) with respect to \( q \) to vanish, which yields the following relation:
\[ [K + A^T \lambda A] q = f + A^T \lambda c \quad (8) \]

From the above equation, the penalty method converts the geometric nonlinearity into material nonlinearity by modifying the potential energy term. The penalty method introduces a penalty term into the system potential energy formula as expressed in Eq. (6). If the penalty number \( \lambda \) is equal to zero, then Eq. (6) becomes identical to Eq. (3). The penalty number \( \lambda \) is typically a very large number, such that a large cost is added to the objective function when the solution points lie outside the original feasible region. Thus by minimizing the modified potential energy accounting for the penalty term, the ring displacement is found and the constraint is satisfied. The recommended range for the penalty number \( \lambda \) found in literature [17] is
\[ \lambda = \bar{\lambda} \max [\text{diag}(K)]; \bar{\lambda} \in [10^5, 10^6] \]

A flowchart for solving the ring–cylinder bore contact problem using the penalty method can be found in Fig. 3.

In this approach, geometric relations and constraint force directions are used to check the constraint violation during the penalty method iteration. The steps are outlined as follows:

1. Build/update the global stiffness matrix and force vector.
2. Calculate the nodal displacement under the current force condition.
3. Calculate the constraint force for the nodes that are in contact with the cylinder bore.
4. Check the constraint violation: for the nodes not under constraint force, they should not move beyond the cylinder bore surface; for the nodes under constraint force, the constraint force direction should be inward, pointing to the center of the cylinder bore.
5. If no constraint mentioned in step 4 is violated, the stationary state of the ring is found; otherwise, return to step 1 until all the constraints are satisfied.

The nodes used for the constraint violation check are the upper and lower ones at the specified cross sections on the ring front.
face as shown in Fig. 4. If the upper and lower nodes satisfy the constraint criterion, it is assumed the corresponding cross section is also satisfied.

For a ring with a symmetric cross section under uniformly distributed temperature, the upper and lower constraint forces should be equal in magnitude and in the same direction. Different constraint forces develop when a twisting moment on the ring occurs, due to cross section, asymmetry, temperature gradient, nonuniform gas pressure, etc.

**Force Release Approach**

One challenge with the penalty method is to identify the correct contact pair on the two surfaces: the constrained nodes on the ring front face and the corresponding infinitesimal constraint surface on the cylinder wall. For the constrained nodes on the ring, the locations at the ring deformed state are not known. For the constraint surface, either for a circular bore or a distorted bore with a known distortion pattern, the geometry and the normal direction of any infinitesimal surface can be found. The unknown is which infinitesimal constraint surface will be in contact with the constrained nodes after the ring deformation.

In order to overcome the difficulties in identifying the contact pairs, a force release approach is adopted by initially applying a pulling force at the ring tips as shown in Fig. 5. Initial forces are applied at the ring tips. These forces are reduced gradually after each iteration and vanish eventually. The direction of the initial force is along the line that connects the ring tips, and its magnitude should be chosen with care. The reason is that the ring under this force should be as close to the constrained cylinder bore surface as possible. One simple criterion in choosing the initial force is that under the initial pulling force, all nodes at the ring front face should not move beyond the constraint bore surface and at the same time, the ring end gap should be greater or equal to zero, so that no overlap occurs at the ring tips. The finite element model can be used to find the appropriate initial force. The flowchart using the force release method is shown in Fig. 6. The “contact analysis” part here refers to the procedures described in Fig. 4.

The purpose of the initial force is only to assist in finding the infinitesimal constraint surface on the cylinder liner that restrains the corresponding ring node. This force must vanish eventually due to its fictional nature, and it needs to be released gradually to guarantee the convergence of the constraint infinitesimal surface. In this work, the force is halved at each step until the force difference between the two consecutive steps is less than 0.1 N. And in each step, the penalty method approach is employed to obtain the constraint pair in the current step. The flowchart for this force release approach is shown in Fig. 6.

**Ring Free-Shape Optimization**

A piston ring free-shape profile can be optimized based on its contact pressure/force distribution when installed into the cylinder bore, since different applications require different pressure/force distribution patterns. For example, a uniform distributed profile is favored for 4-stroke engine application; while for the 2-stroke engine, the main concern is the ring tip protrusion so the contact pressure/force is desired to vanish at the ring tips. The ring free shape optimization is completed through an iterative approach. It is achieved with the 3D ring–cylinder bore contact analytical tool and an optimizer. The design space is the ring radius at different circumferential locations. A package such as HEEDS can be used as the optimizer.

**Ring Thermal Load**

During engine operation, the piston rings, especially the top compression ring, are subject to a high-temperature thermal load. The ring front face is flooded with the lubrication oil film while other boundaries are exposed to gas combustion products. There is a thermal convection path from the combustion chamber, through the piston crown to the ring groove, and through the ring to the liner and to the water jacket. For this study, the convective boundary condition used is shown in Fig. 7. This boundary condition is for the EcoMotors International’s opposite piston opposite cylinder engine inner piston top compression ring. This boundary condition represents the average liner temperature, despite of cylinder liner temperature variation along the axial direction.

This convective boundary is axisymmetric uniform for the ring about the cylinder center line. The temperature variation across the ring is governed by the heat conduction equation

\[
\frac{\partial T}{\partial t} - k \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) = 0
\]

However, for a piston ring application as described in this study, the ring is moving rapidly with the piston’s motion at over 1000 rpm. Thus the rate of temperature variance at any point can be considered to be zero since heat spread process takes time and it would not be able to change much in such a short duration between engine cycles. In this case, the temperature variation does not depend on time. Based on this assumption, the time
variant problem can be simplified as a steady-state problem and the governing equation becomes

$$k \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) = 0$$  \hspace{1cm} (10)

with a convective boundary condition

$$-k \left( \frac{\partial T}{\partial x} n_x + \frac{\partial T}{\partial y} n_y + \frac{\partial T}{\partial z} n_z \right) - h(T - T_\infty) = 0$$  \hspace{1cm} (11)

Finite element analysis (FEA) is used again to calculate the temperature distribution for the piston ring. With the additional thermal load, the constitutive relation for an isotropic linear elastic material [14] can be written as

$$\sigma = E \varepsilon + \sigma_0$$  \hspace{1cm} (12)

where \( \sigma_0 = (E\varepsilon\Delta T / 1 - 2\nu) \).

From the ring thermal analysis, the ring deformed shape under the above thermal load can be obtained. This shape is considered as the “temperature-influenced” free shape. Both the ring free shape at room temperature and the temperature-influenced free shape will be studied for the ring–cylinder bore contact analysis.

**Experiment Verification**

A test rig capable of measuring the ring contact load at variable circumferential locations is shown in Fig. 8. The instrument can also be adapted for different bore diameters by adjusting the radius of the load cells.

By setting the load cells at the desired circumferential locations and adjusting them to the cylinder wall radius, the ring test rig is configured to measure the ring–cylinder wall contact at the load cell locations. This test rig with five load cells is used to verify the numerical model. It should be noted that five load cells may be not adequate to represent the continuous contact force distribution between the ring and the cylinder wall, but they can be used for model validation.

The piston ring is installed to the ring test rig for contact forces measurement as shown in Fig. 9. The ring is aligned with the five load cells as the ring back (opposite to the ring end gap) is in touch with one load cell. Two load cells are located close to the ring tips; and the other two are located ±90 deg from the ring end gap.

The contact loads between the ring and the load cells are simultaneously recorded using a data acquisition system. The results from the measurement will be presented in the Results and Discussion section below.

**Results and Discussion**

One example given in this section is a keystone compression ring. First, the model predicted ring–cylinder wall contact forces

$$\alpha_1 = \alpha_2 = 5.6^\circ$$

$$\text{Ring Face}$$

$$2.59 \text{ mm}$$

$$4.02 \text{ mm}$$
are compared with the measured results using the ring test rig for validation. Then the ring–bore contact is analyzed at room temperature. Finally, the ring temperature distribution is calculated under the convective boundary condition, and the contact forces with temperature influence are obtained.

The ring is a top compression ring for a 2-stroke engine application. The cross section of the ring is symmetric and is shown in Fig. 10.

The main parameters describing the ring are listed in Table 1.

As discussed previously in the Modeling Approach section, the ring OD profile at its free state is a necessary input for the FEA program. This OD profile is measured with a CMM machine and the normalized radius as shown in Fig. 11 for half of the ring. The other half of the ring is symmetric to this half shown in Fig. 11.

The measured ring OD coordinates are shifted with respect to the center of the cylinder bore. In this way the nodes on the ring front face at the ring back have radius equal to the cylinder bore radius. The OD radii at all circumferential locations are normalized by this OD radius at the ring back. Thus, the normalized radius plot for ring free shape shows that the radius increases from the ring back and peaks at around 140 deg away from the ring back. After this location, the normalized radius decreases again till the ring tip. The ring OD coordinate is measured with an increment less than 1 deg.

Another input for the FEA program is the locations at which the constraint forces are applied. Five circumferential locations were specified as potential contact locations; they are located at the load cell positions to compare the numerical results with the experimental results. Defining the ring back at 0 deg and the middle of the end gap as ±180 deg, the five locations are at 0 deg, ±90 deg, and ±167 deg, respectively. The model-predicted and measured contact forces at the five cross sections are normalized by the force at 0 deg and the results are shown in Fig. 12. It was found that the model-predicted contact forces show good agreement with the measured forces at these five locations, especially at the ring back and ring tips. Contact forces at ±90 deg show less agreement. However, similar contact force distribution is found between the predicted forces and measurement. That is at ±90 deg the highest constraint forces are observed, while at ±167 deg the constraint forces are the lowest.

The difference between the model-predicted and measured contact forces may be attributed to the following factors: first, ring modulus of elasticity used in the model, as the material properties from other chemical compositions and surface treatment are not considered in the model; second, the measured ring free shape also introduces some error due to the accuracy of the CMM machine; third, as it was observed from the measurement that the ring may not contact with the load cells at their centers, thus the contact locations may be slightly changed.

The above comparison between the numerical result and the measurement is only for validation of the FEA model. It has been verified that the FEA model is able to predict the contact forces between the ring front face and the cylinder wall. However, the above analysis only considers five circumferential locations. The result may not represent the ring/bore contact sufficiently. More constraints are necessary in order to obtain the contact force distribution.

The same ring was then analyzed with thirteen constraint locations along the circumference at room temperature. The free-

<table>
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<tr>
<th>Table 1 Main parameters for the ring</th>
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<td>Ring material</td>
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<tr>
<td>Modulus of elasticity</td>
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<tr>
<td>Poisson’s ratio</td>
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<tr>
<td>Cylinder bore diameter</td>
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<td>Coefficient of thermal expansion</td>
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<td>Thermal Conductivity</td>
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<td>Ring/gas convective coefficient</td>
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<td>Ring/oil film convective coefficient</td>
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Fig. 11 Normalized ring radius at its free shape (half-ring)

Fig. 12 Normalized calculated and measured contact forces

Fig. 13 Free and deformed shapes
shape ring and the deformed ring without temperature influence are shown in Fig. 13.

The green mesh in Fig. 13 represents the free-shape ring while the red mesh represents the deformed ring shape under the cylinder bore constraints without temperature influence. It is obvious that the ring is pushed inward from its free state. The constraint forces that push the ring to its deformed position are shown in Fig. 14. The blue bar (left) and red bar (right) represent the constraint forces at a certain circumference location at the upper and lower corners at ring face as shown in Fig. 4. Green (square) and purple (circle) points show the separation gaps between the ring face and the cylinder bore at the top and bottom edges, respectively.

From Fig. 14, it is found that the two contact forces at the same cross section are identical since the ring has a symmetric cross section and there is no twisting moment on the ring. The plot also shows that the constraint force at the ring back is the highest. At the cross sections approximately 30 deg away from the ring back, the lowest constraint forces are found for the sections that contact against the cylinder wall directly. The constraint forces at the ring tips vanish, such that the ring separates from the cylinder wall in its front face at its two tips. The separation gap is defined as the radial distance between the cylinder wall ID and the ring tip OD. A 34 μm separation gap is found for this specific ring from the FEA model.

This ring–bore separation has been verified by experimental observation using a light-tightness method (ISO 6621-2). A schematic diagram of the experiment and the measuring setup can be found in Fig. 15.

After the ring was installed into the cylinder liner, a light source was provided on one side of the ring and detected on the other side. Thus light can be observed if a separation gap exists between the ring front face and the cylinder liner. For other parts of the ring that conform to the cylinder liner, no light would be observed. This measurement observation showed that light was detected at the ring end gap up to about 15 deg away from the end gap in both directions (Fig. 16). This means the ring separated from the cylinder wall near the ring tips. The light-tight measurement confirms the FEA result.

Another way to check the finite element analysis result is using the total force acting on the ring. As it is a static problem, the total force on the ring should be zero in X, Y, and Z directions. The XY plane defines the ring plane and the Z direction is the axial direction. For the ring, the total force is verified to be zero along both X and Y directions.

The magnitude of the constraint forces and the separation gaps are shown in Table 2. The approximate sign in the right column means the location close to 180 deg. As the ring end gap is not fully closed, the location is not exactly 180 deg. The up and low locations refer to the top and bottom edges of the ring cross section shown in Fig. 4.

The result shows good agreement with the work done by Tománek and Bruno [10] using a pin gauge device to measure the constraint forces and Liu et al. [9] using beam elements. In Tománek’s work, eleven load cells were used; in the present work, a total of 13 cross sections were investigated. Another difference

<table>
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<th>Table 2 Constraint forces and the separation gap size along ring circumference</th>
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<tr>
<td>Location</td>
</tr>
<tr>
<td>Force, up (N)</td>
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<tr>
<td>Force, low (N)</td>
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<tr>
<td>Gap, up (µm)</td>
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between Tomanik’s work and the present study is the ring free shape. Although both rings have negative ring ovality, the ring used for the present study is for a 2-stroke engine. Thus in this case, the main concern is to avoid the ring tip protrusion into the ports.

Thermal Load Influence

In order to obtain a more accurate analysis of the piston ring deformation and subsequent force distribution at engine operating conditions, the thermal load needs to be considered due to the fact that the ring is subject to high temperatures. About 31% of piston heat flow is through the ring pack for a high-duty diesel engine as claimed by Mierbach et al. [18]. Under the boundary condition defined in Fig. 7, the temperature distribution for the entire ring was obtained and shown in Fig. 17.

As the temperature distribution does not vary along the ring circumferential direction as assumed, the temperature only depends on the radial and axial location. This temperature distribution for one arbitrary cross section can be found in Fig. 18.

From Figs. 17 and 18, the ring average temperature at the prescribed condition is 214 °C, which is much higher than room temperature (25 °C). The inner upper corner has the highest temperature, while the lowest temperature is found at the outer lower corner. However, the temperature gradient across the ring is very small, due to the small dimensions in both the radial and axial directions of the ring. These small lengths do not provide high thermal resistance with the material being used. Although the temperature gradient across the ring is small, the temperature
increase from the room temperature, when the ring is installed in the piston cylinder assembly, to the hot operation condition is significant. This large temperature change does act as a thermal load that can cause ring volumetric thermal expansion. Thus, the temperature influenced ring free shape must be studied when analyzing the ring–cylinder wall contact.

Figure 19 shows the ring at room temperature condition (blue mesh with shade) as well as the ring after thermal expansion (red mesh). It clearly shows that the ring grows in the radial direction. This means the ring radius of curvature increases under thermal load. The growth along the circumferential direction is not as significant as in the radial direction. However, the ring curvature length increases as well.

As for the room temperature condition (cold), the ring free shape normalized radius at high temperature operation condition (hot) is also plotted, for comparison. The plot can be found in Fig. 20. The maximum radial growth for the ring is 0.33 mm. It is obvious the free shape has changed significantly with the temperature effect.

The ring–cylinder liner contact was also analyzed for the temperature influenced free-shape ring. Cylinder bore thermal deformation is not considered in this study, as the emphasis is on thermal load influence on the ring only. As the ring O.D. curvature changes significantly as shown in Fig. 20, the constraint force pattern is expected to be different from the condition without thermal load. These constraint forces and separation gap sizes at the upper and lower edges along the ring circumference with temperature influence can be found in Fig. 21. The constraint force pattern along the ring circumference shows that separation gaps also exist at the ring tips. However, these gaps have been significantly reduced. The constraint loads at the locations about ±30 deg and ±120 deg away from the ring back are very small compared to those at other locations. Slight differences in contact force can be found between the upper edge and the lower edge at the same circumferential location. This results from the temperature difference across the ring cross section. The higher temperature at the ring inner upper corner and the lower temperature at the outer lower corner introduce a twist torque that tends to twist the ring negatively. The higher constraint force at the upper corner and lower force at the lower corner form another moment that balances the temperature induced twist moment. Since the temperature gradient is low, the twist of ring does not actually occur. The separation gap at the ring tips also shows a difference between the upper and lower edges, with the upper gaps slightly smaller than the lower ones.

The magnitude of the constraint forces and the separation gaps for the deformed ring with temperature influence are shown in Table 3. It is also worthwhile to study how the constraint force pattern changes between the cold ring and temperature influenced hot ring. Figure 22 shows the constraint forces at the upper and lower edges along the ring circumference. At the cold condition, the constraint forces at the upper and lower edges are identical at the same circumferential location as explained previously in the Results and Discussion section. Only one bar plot is needed to represent the constraint forces at either edge at cold condition. The constraint forces all increase with different magnitudes except for the one at around 120 deg away from the ring back with the consideration of the temperature compensation. The location of the maximum constraint force shows agreement as at the ring back.

Conclusions

Piston ring and cylinder bore interactions are complex 3D phenomena. A finite element ring–cylinder liner contact model was developed using eight-node hexahedral elements to simulate this complex interaction. The model, verified by experiment, is capable of predicting the contact between the ring and the cylinder liner. Temperature compensation was found to have a significant impact on this ring–cylinder liner interaction as the constraint forces change along the ring circumference, as well as the separation gap size at the ring tips. The ring with the thermal load also grows in length resulting in a smaller end gap that needs to be considered in engine ring pack design. The FEA model can be employed in the design process to evaluate piston rings for various applications.

The interaction between the ring and the cylinder liner has a significant impact on engine sealing, ring dynamics, ring pack friction, wear, etc. Most existing analytical tools are two-dimensional models, considering the ring tension and dynamic behavior as all the same throughout the ring periphery. Thus, these models are not able to simulate the variation along the ring circumference. Some three-dimensional models consider this circumferential variance; however, they assume the ring conforms to the cylinder bore perfectly at every periphery location. The present study provides a basis to integrate the ring–bore interaction in a three-dimensional manner. Along with the 3D ring dynamics, the lubrication, friction, and wear can all be modeled in a more accurate way to study the internal combustion engine power cylinder system.

Acknowledgment

We acknowledge the support of EcoMotors, Cummins Engine Company, and Mid-Michigan Research for supporting this effort. The authors also appreciate the technical support from Tom Stuecken and the help from Alan Seery of Michigan State University.
References


