

Article

Simulation Research on Cylinder Liner Shape and Position Tolerance under Thermo-Mechanical Load

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Abstract: The cylinder liner bears alternating thermal load and mechanical load, and evaluating the cylinder liner deformation is a key issue in the design stage of an engine. In this work, the shape and position tolerance of the cylinder liner to various loads were studied based on the finite element method, the simplex algorithm and the least square method. Firstly, the heat transfer boundary conditions of the cylinder liner were obtained through combustion simulation. Combined with the mechanical load, the transient deformation of the cylinder liner under the thermo-mechanical load was obtained. Subsequently, the out-of-roundness and coaxiality were selected to evaluate the shape and position changes in the cylinder liner. Finally, the transient tolerance analysis of the cylinder liner under alternating thermo-mechanical load was carried out. The results show that the maximum out-of-roundness of the cylinder liner under thermal load, mechanical load and thermos-mechanical load was 15.12, 43.40 and 51.76 μm , respectively. The maximum coaxiality under thermal load, mechanical load and thermos-mechanical load were 6.17, 80.49 and 80.22 μm . The side thrust was more likely to cause uneven deformation of the cylinder liner section, the liner coaxiality was mainly affected by the cylinder burst pressure, and the liner shape tolerance was much more sensitive to the mechanical load than the mechanical load.

Keywords: cylinder liner; simplex method; out-of-roundness; coaxiality

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1. Introduction

The matching relationship between the piston and cylinder liner significantly affects the performance, fuel economy, durability, and emission characteristics of an engine. During the engine operation stage, several thermal and mechanical loads act upon the cylinder liner, causing deviations in its geometric tolerances, such as out-of-roundness and coaxiality. These deviations subsequently affect the dynamic gap between the piston group and the cylinder liner. An appropriate gap distribution is conducive to optimizing the sealing and friction performance of the piston rings, and thereby improving the engine durability. Consequently, elucidating the variation patterns of the cylinder liner's geometric tolerance deviations under coupled thermo-mechanical loads and maintaining them within an acceptable range is paramount for improving the overall performance of diesel engines.

To meet the ever-increasing demands for engine efficiency and stringent emission standards, precision control and deformation analysis of cylinder liners have emerged as critical issues of research in academia and industry. As a standard cylindrical component, the deformation of a cylinder liner can be evaluated by assessing its out-of-roundness and coaxiality deviations. Kaiser et al. [1] proposed a precise definition of out-of-roundness and four widely adopted mathematical expressions for its calculation. Traditional experimental methods, such as micrometry [2], coordinate measuring techniques [3], strain gauges [4], and ultrasonic techniques [5], have been employed to measure the dynamic deformation of cylinder liners. However, these methods are often hindered by high costs and low efficiency.

Finite element analysis (FEA) has emerged as a powerful tool for studying cylinder liner deformation. Mishra et al. [6] utilized the finite element method by applying boundary loads on the piston surface, neglecting friction effects, to analyze the influence of different piston crown structures on piston strength. Menacer et al. [7] proposed a numerical model considering oil ring elastic deformation, dynamic motion, and mixed lubrication friction to investigate the friction forces in the piston ring–cylinder liner contact pair. Li et al. [8] established a geometrical model based on molecular dynamics simulations to study the microscopic frictional behavior of the piston ring–cylinder liner assembly near the top dead center and analyzed the frictional characteristics under varying operating conditions. Hei et al. [9] introduced an improved mixed lubrication model, in which the Fourier equations were adopted to describe cylinder liner deformation. The lubricating oil thermal effects and transport, and the transition from full-film to local dry lubrication conditions were discussed. Gao et al. [10] explored cylinder liner friction and wear mechanisms through FEA. The piston–cylinder liner contact was set as an asymmetric “frictional” behavior and the “enhanced Lagrangian” method was employed to implement cylinder liner friction settings, while the cylinder pressure, thermal coupling, and the influence of cylinder liner deformation were not considered. Liu et al. [11] established a piston ring–cylinder liner friction model incorporating thermodynamic and dynamic characteristics, utilizing FEA to simulate cylinder liner deformation under thermal loads and assembly preloads while comprehensively considering the impact of surface roughness on frictional performance. Chen et al. [12] employed a thermo-structural coupled method for piston assembly simulations to calculate the temperature distribution in the piston and piston rings using the third-kind thermal boundary conditions. Then, the temperature distribution result was imported as boundary conditions to simulate deformation during the working process, but the reliance on empirical heat transfer correlations may lead to inaccuracies in temperature distribution estimation. Ahmad et al. [13] performed FEA to analyze the deformation of cylinder liners made of different materials under the direct coupling of thermal and mechanical loads. Taking Fourier spectra, coaxiality, and main thrust plane bore variation as evaluation indicators, Zhu et al. [14] utilized the FEA to analyze cylinder liner deformation in the pre-tightened state of the diesel engine body–cylinder head assembly, while the cylinder liner’s operating conditions were neglected. Bi et al. [15,16] established fluid–solid coupled models for the coolant flow, deformation, and temperature distribution of the cylinder liner under diesel engine steady-state operating conditions, in which the inner wall heat transfer boundary conditions were obtained experimentally. The effects of thermal stress, bolt preloads, combustion pressure, and piston side thrust forces on cylinder liner deformation behavior were discussed.

However, existing FEA studies primarily focused on steady-state conditions or employed simplified thermal boundary assumption methods to predict cylinder liner deformation, which made it hard to capture the transient and coupled characteristics of cylinder liner deformation during engine operation. The simplified method of mapping obtained thermal boundaries onto FE models with mechanical loads cannot characterize the complex thermodynamic phenomena within the combustion chamber accurately, and may lead to inaccurate predictions of cylinder liner deformation patterns. To accurately predict cylinder liner deformation under coupled thermal and mechanical loads, researchers have adopted various numerical simulation methods. Al-Baghdadi et al. [17] established an FE model to analyze the stress and deformation distribution of a hydrogen fuel internal combustion engine piston under thermo-mechanical loads while neglecting the effect of cylinder liner deformation. Rao et al. [18] considered the influence of cylinder liner thermal deformation, elastic deformation, and vibration characteristics on the frictional dynamic performance of the piston–cylinder liner system and proposed a multi-physics coupled model to realize cylinder liner deformation prediction. Bi et al. [19] established a simulation model on a high-pressure common-rail diesel engine, including the cylinder head, cylinder gasket, and dry cylinder liner, to investigate the deformation characteristics of the cylinder liner under the coupling of cylinder head bolt preloads, gas pressure, piston side thrust force,

and thermo-mechanical effects. Alshwawra et al. [20] studied the influence of cylinder liner deformation on the piston ring–cylinder liner conformity by designing a non-circular cylinder liner profile and proposed an improving cylinder liner roundness method through a non-circular cylinder liner. They also utilized the FE method to investigate the deformation behavior of non-circular cylinder liners under thermal stress, but their work focused solely on the engine's cold state and operating conditions.

Existing research primarily focused on measuring or calculating piston skirt deformation, profile, cylinder liner axial offset, circular cross-section deformation, out-of-roundness, and coaxiality, which were geometric tolerances of the cylinder liner. Although measurement-based methods can obtain cylinder liner deformation data directly, they were hindered by high costs and the inability to measure changes in out-of-roundness and coaxiality under dynamic operating conditions. In contrast, simulation and computational methods showed advantages in predicting cylinder liner shape and positional tolerances at a low cost and with high visualization capabilities. However, the vast majority of studies only analyze single loads, steady-state conditions, or single cross sections, failing to comprehensively describe the deformation process of cylinder liner cross sections over time under dynamic thermo-mechanical loads, which makes it difficult to fully capture the deformation patterns. Therefore, it is necessary to establish a theoretical model for cylinder liner deformation under dynamic thermo-mechanical coupled loads to elucidate its spatiotemporal deformation characteristics.

This study proposed an integrated method that coupled combustion simulation with FEA to elucidate the dynamic deformation patterns of diesel engine cylinder liners under thermo-mechanical loads. The effect of thermo-mechanical loads on the transient geometric tolerances was discussed. Firstly, based on the combustion model, the instantaneous temperature field and convective heat transfer coefficients on the cylinder liner's inner surface were calculated, considering the influence of intake cooling effects and fuel injection characteristics. Subsequently, these thermal boundary conditions were synchronously mapped onto the FE model of the cylinder liner along with the periodic gas pressure loads. The cylinder liner deformation field was simulated at each instant throughout the working cycle. The variations in out-of-roundness and coaxiality at different axial positions and crank angles were revealed. For the cylindrical cylinder liner, to quantify geometric tolerances, the algorithm using the minimum circumscribed circle method and the orthogonal least squares method was proposed to evaluate roundness and coaxiality, respectively. Based on this algorithm, the coordinate data of the nodes on the cylinder liner wall for each cross section were extracted to analyze the spatiotemporal variation patterns of roundness and coaxiality. The effects of thermal loads, mechanical loads, and their coupled effects on geometric tolerances were investigated. This method considered the dynamic characteristics of thermo-mechanical loads, and the limitations of previous studies confined to steady-state or simplified conditions were overcome. The precise prediction of cylinder liner geometric tolerances throughout the engine cycle was achieved, which provided theoretical guidance for diesel engine cylinder liner structural optimization design and fault diagnosis.

2. Definition and Algorithm of Geometric Tolerance

2.1. Definition of Geometric Tolerance

The geometric tolerance is the maximum allowable variation in the shape and position of the elements in the drawing. The parameters such as out-of-roundness and coaxiality were selected to evaluate the deformation of the cylinder liner. The following equations taken from ISO 12181:2011 [21] were employed to evaluate the deformation.

(1) Out-of-roundness

Out-of-roundness is the shape tolerance, which refers to the difference between the largest and smallest radius on a circular cross section. The tolerance zone for out-of-roundness is the area between two concentric circles whose radius difference is the tolerance value on the same normal section. In ISO12181, the roundness error evaluation

methods include the minimum zone circle method (MZC), least square circle method (LSC), minimum circumscribed circle method (MCC) and maximum inscribed circle method (MIC). The MZC method has the highest accuracy and best meets the definition of out-of-roundness, but the calculation method is complicated. The LSC method has low accuracy, but the calculation method is simple. The calculation error of the MIC method and the MCC method is relatively high. The MZC method recommended by the ISO standard was adopted to calculate the degree of out-of-roundness.

The definition of out-of-roundness in the MZC method is shown in Figure 1, where K is the profile curve to be measured, and P is any point within the K curve. A rectangular coordinate system was established. Two concentric circles with P as the center and enclosing the K curve are adopted, and the radius of the inscribed and circumscribed circle are r_1 , and r_2 , respectively. M is any point on the K curve, and the radius difference Δr can be expressed as:

$$\Delta r = \max \left\{ \sqrt{(x_p - x_M)^2 + (y_p - y_M)^2} \right\} - \min \left\{ \sqrt{(x_p - x_M)^2 + (y_p - y_M)^2} \right\} \quad (1)$$

If there is a point P that minimizes Δr , then the radius difference $\min\{\Delta r\}$ is called the out-of-roundness of the section, and point P is the center of the minimum area.

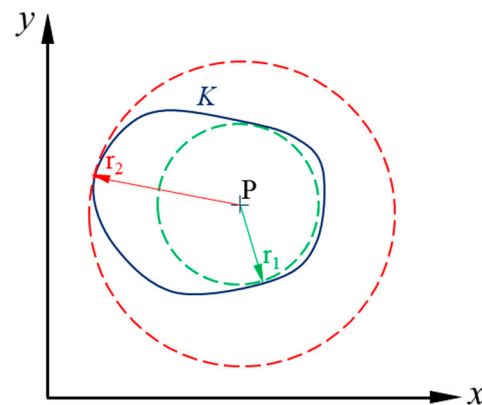


Figure 1. Schematic diagram of out-of-roundness.

(2) Concentricity

Coaxiality is position tolerance. The tolerance zone of coaxiality is the area in the cylindrical surface whose diameter is the tolerance value and is coaxial with the reference axis. The definition of coaxiality is shown in Figure 2.

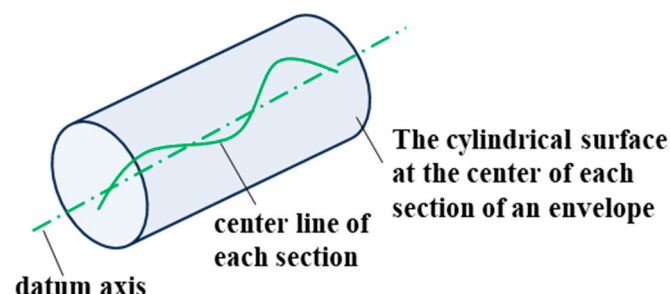


Figure 2. Schematic diagram of concentricity.

The orthogonal least squares evaluation method was used to calculate the coaxiality. Taking the least squares center of each measured contour curve as the reference, the least squares method was used to fit the equivalent axis. The definition of the least squares circle is defined as the minimum value of the sum of the squares of the radius difference between all points on the actual contour and the circle.

The measured elements were sampled at equal intervals in several equidistant sampling sections perpendicular to the axis, the number of sampling points in each section was n , and the least square center of the sampling section was set to $O_j(a_j, b_j)$, then:

$$\begin{aligned} a_j &= \frac{2}{n} \sum_{i=1}^n x_i \\ b_j &= \frac{2}{n} \sum_{i=1}^n y_i \end{aligned} \quad (2)$$

Assuming that the least squares axis passes through point $B(x_0, y_0, 0)$ on the OXY coordinate plane, and one of the direction vectors was $(g, l, 1)$, the axis equation is written as:

$$(x - x_0)/g = (y - y_0)/l = z \quad (3)$$

In the case of discrete sampling, the above formula can be written as:

$$\begin{aligned} (x - x_0)/g &= z_i \\ (y - y_0)/l &= z_i \end{aligned} \quad (4)$$

According to the principle of least square method and orthogonalized processing, four undetermined coefficients of the orthogonal least square method axis were obtained:

$$\begin{aligned} x_0 &= \frac{1}{m} \sum_{j=1}^m a_j \\ y_0 &= \frac{1}{m} \sum_{j=1}^m b_j \\ g &= \sum_{j=1}^m a_j z_j / \sum_{j=1}^m z_j^2 \\ l &= \sum_{j=1}^m b_j z_j / \sum_{j=1}^m z_j^2 \end{aligned} \quad (5)$$

After determining the reference axis, the distance between the reference axis and the least square center $O_j(a_j, b_j, z_j)$ of the inner contour at each sampling section was written as:

$$R_j = \frac{|(O_j - B) \times S|}{|S|} \quad (6)$$

According to the definition of the coaxiality error, the coaxiality error determined by the orthogonal least square method was $D = \max\{2R_j\}$.

2.2. Algorithm Debugging of Cylinder Liner and Position Tolerance

The key for calculating the out-of-roundness was to find the center of the smallest region so that the difference in radius between the outer and inner circles of the envelope curve was minimized. There was no limitation on design variables that could be arbitrarily taken in space. The solving process belonged to the unconstrained minimum value-solving problem in optimization calculations.

The basic iterative format for solving this type of problem was to search forward step by step along the direction in which the value of the objective function decreases until finally finding the optimal solution. The corresponding search direction and step factor were determined at each iteration. The difference between various unconstrained optimization algorithms lies in the construction of search direction and step size factor. The purpose was to find the optimal solution with the smallest number of iteration steps and the fastest speed. It was divided into the indirect method which needs to solve the partial derivative of the objective function value and the direct method which compares the size of the objective function.

The sampling points on the contour curve of the part were discrete points, and the partial derivative cannot be solved. Therefore, the simplex method in the direct algorithm was selected to search for the best circle center position and calculate the out-of-roundness value of each sampling section during a diesel engine working cycle.

The calculation idea of the simplex method was to specify the initial search area, calculate the objective function value of each vertex of the simplex, and sort by the size of the function value. If $f(x_3) < f(x_2) < f(x_1)$, then x_3 was an optional point (x^L), x_2 was a sub-dead pixel (x^G), and x_1 was the dead pixel (x^H). The next new point of the objective function was searched in the opposite direction of the dead pixel. The function value at the new point with the original value was compared, and the new dead pixel was removed. The position of the simplex was constantly updated and the area of the simplex was reduced. The simplex was finally reduced to a point, which was the optimal solution.

For two-dimensional problems, the simplex algorithm was mainly composed of four main steps: reflection, extension, compression, and side length reduction [22].

1. Reflection: Generally speaking, the probability that the optimal solution of the objective function was located at the symmetric side of the worst point was maximized. The reflection point was calculated as:

$$x^R = x^F + \gamma(x^F - x^H) \quad (7)$$

where (x^F) is the midpoint of (x^G) and (x^L). (x^R) is a point on the line of (x^F) and (x^H), and is called the reflection point of (x^H) with respect to (x^F). γ is the reflection coefficient, and was generally taken as 0.5.

The next strategy was determined after calculating $f(x^R)$ and comparing the with the previous $f(x^H)$, $f(x^G)$, and $f(x^L)$.

2. Extension: If $f(x^R) < f(x^L)$, it indicates that the search direction was correct, and the search point can be extended to x^E along the original search path. The extension coefficient α was generally set as 1.5, and the extension point could be written as:

$$x^E = x^F + \alpha(x^R - x^F) \quad (8)$$

If $f(x^E) < f(x^R)$ is extended, it indicates that the forward extension was disconcerting, and (x^R), (x^G), and (x^L) were taken as the new simplexes.

3. Compression: If $f(x^R) > f(x^G)$, the function value at the reflection point was greater than that at the original sub-dead point, meaning that the reflection point was taken too far, and it should be compressed to point (x^S) along the directions (x^R) and (x^F). The compression factor β was set as 0.5

$$x^S = x^F + \beta(x^R - x^F) \quad (9)$$

If $f(x^R) > f(x^H)$, it means that the reflection point was worse than the dead pixel and should be compressed more. The new point should be taken between (x^H) and (x^F) and is written as:

$$x^{S'} = x^F - \beta(x^F - x^H) \quad (10)$$

4. Shorten the side length: When all the points along the search path (x^F) and (x^H) were larger than the original worst point function value, it means that the search along the reflection direction failed. The original simplex should be reduced and the best point was the base point. The initial simplex was reduced by half, and the reduced simplex was used as the starting search area. A new-round search was started according to the above method.

After each iteration constituted a new simplex, the convergence test was carried out. The condition for the termination of the iteration was given as:

$$\left| \frac{f(X^H) - f(X^L)}{f(X^L)} \right| < \varepsilon \quad (11)$$

where ε is the convergence judgment coefficient and was set as 0.000001.

3. Transient Deformation of Cylinder Liner under Thermo-Mechanical Load

3.1. Model Description

The cylinder liner undergoes certain variations during operation due to several factors including its material, operating condition and design. In this work, we focused on the effect of thermal and mechanical load on the overall geometry of the cylinder liner. The transient temperature and deformation analysis on a diesel engine cylinder liner were carried out, and the detailed parameters of the engine are shown in Table 1. The cylinder liner was 170 mm in axial length and 108 mm in diameter. The bushing was installed in the engine block through an interference fit. In the FEA process, the tie constraint was applied between the cylinder bushing and the engine block. The material properties for the cylinder liner are given in Table 2. The rated condition with a moment of 1250 N·m and rotation speed of 2100 r/min was calculated. In the meshing stage of finite element modeling, the Solid 87 element was used; as shown in Figure 3, the mesh numbers for the piston and engine body are 417,711 and 1,287,446, respectively. The element and node number was similar to that in Bi's research [19], which has been proven reliable for the numerical simulation.

Table 1. Main engine parameters.

Parameters	Value
Maximum power	220 kW
Maximum moment	1250 N·m
Number of cylinders	6
Diameter of a cylinder	108 mm
Stroke	136 mm
Compression ratio	18

Table 2. Material properties for cylinder liner.

Elastic Modulus/GPa	Poisson's Ratio	Thermal Conductivity/W·m ⁻¹ ·K ⁻¹	Density/kg·m ⁻³	Specific Heat Capacity/J·kg ⁻¹ ·K ⁻¹	Linear Expansion Coefficient/K ⁻¹
125	0.3	38.7	7300	460	1.22 × 10 ⁻⁵

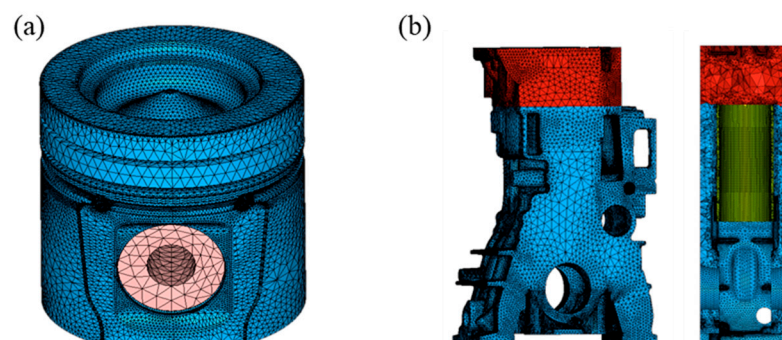


Figure 3. Meshes for (a) piston and (b) engine body.

3.2. Thermal Boundary Conditions on the Piston Top Surface

Usually, the gas temperature and heat transfer coefficient were estimated by empirical formula. The heat transfer process between gas and piston was considered a steady process, and the heat transfer coefficient on the piston surface was given as [23]:

$$h_r = \begin{cases} \frac{2h_m}{1+e^{0.1N^{1.5}}} e^{0.1\left(\frac{r}{25.4}\right)^{1.5}}, & 0 < r < N \\ \frac{2h_m}{1+e^{0.1N^{1.5}}} e^{0.1\left(\frac{2N-r}{25.4}\right)^{1.5}}, & N < r < R \end{cases} \quad (12)$$

where h_r is the heat transfer coefficient for the aimed position, h_m is the mean heat transfer coefficient in a working cycle, r is the distance between the aimed position and piston center, R is the piston radius. It can be seen from the above formula that the heat transfer boundary conditions were the same for the same r value. However, the heat transfer boundary conditions were usually uneven along the circumferential direction of the piston, since the flow conditions caused by the intake and exhaust process were not uniform.

In this work, a numerical combustion model on the commercial code CONVERGE was proposed to reveal the effect of intake, combustion and exhaust on heat transfer boundary conditions of piston and cylinder liner. The grid of the numerical model is shown in Figure 4. The governing equations for the combustion process were the conservation equations of mass, momentum, energy and species. The SAGE solver with a detailed chemistry method was used to describe the combustion process. The standard Droplet Discrete Model was used to simulate the injection process of the diesel fuel. The hybrid model was used to describe the spray atomization and break-up process. The physical properties of fuel were simplified by tetradecane ($C_{14}H_{30}$). The RNG k- ϵ model was used to enclose N-S equations. The model parameters such as intake valve clearance, fuel injection strategy and valve lift curves were set based on real working conditions. The numerical combustion model was verified by comparing the in-cylinder pressure in the numerical model with that in the experimental result, as shown in Figure 5. The in-cylinder pressure obtained from numerical results showed high agreement with the experimental results with a deviation within 1.6%, which proved the accuracy of the numerical model.

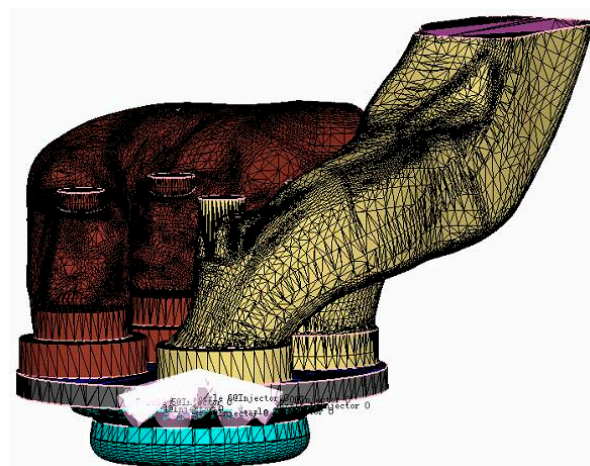


Figure 4. Cylinder grid model.

At the intake valve opening moment, as shown in Figure 6a, the maximum gas temperature on the intake side of the piston top surface was 805 K, and the maximum gas temperature on the exhaust side was 803 K, which appeared near the interface between the intake and exhaust sides. At the intake valve closing moment, as shown in Figure 6b, under the action of low-temperature fresh intake air, the maximum gas temperature on the intake side was decreased to 407 K, and the maximum temperature on the exhaust side was about 450 K. The maximum temperature decrement in the intake and exhaust sides

were 398 and 353 K. It can be considered that the temperature on both sides of the intake and exhaust side was much different due to the influence of the intake cold air, and it was difficult for simple ring division in the empirical formula to meet the temperature field calculation requirements.

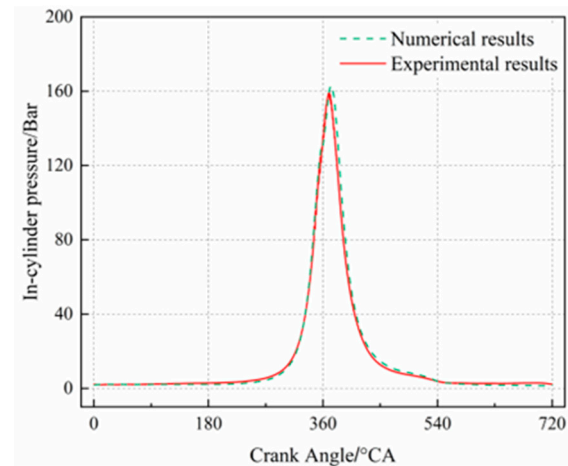


Figure 5. Comparison of in-cylinder pressure obtained from numerical and experimental results.

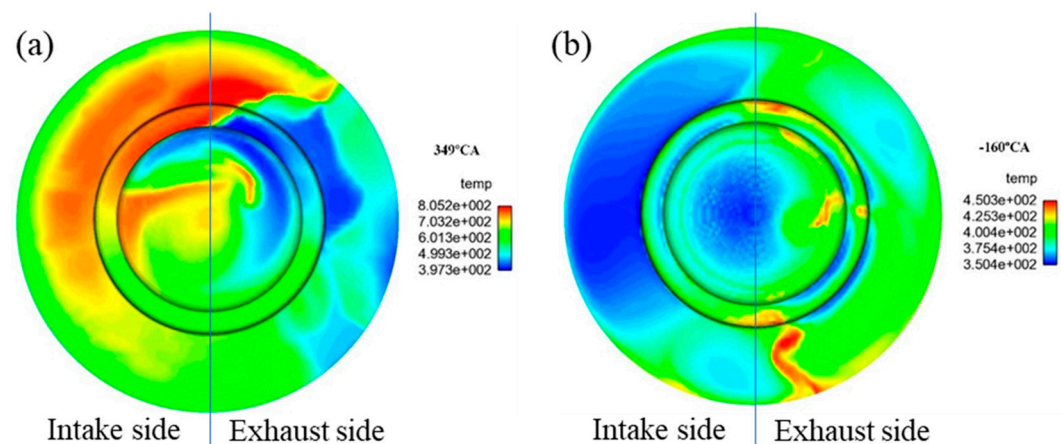


Figure 6. Gas temperature distribution on the piston top surface at (a) the intake valve opening and (b) the closing moment.

The transient gas temperature and heat transfer coefficient obtained from the above numerical simulation was taken as the boundary conditions in the transient thermal analysis of the piston.

3.3. Boundary Conditions for Cylinder Liner

During the working process of the diesel engine, the piston reciprocates in the cylinder liner. The wall surface of the cylinder liner contacts with the gas, the piston, the oil mist in the crankcase. The area and position of the contact surface changed periodically. For a certain moment in the working process, the inner wall of the cylinder liner could be divided into different areas, taking the position of the piston as the reference, as shown in Figure 7. The inner surface of the cylinder liner can be divided into the following three parts from top to bottom:

(1) Working gas contact area: this area was in direct contact with the combustion gas. The gas temperature and heat transfer coefficient were obtained through combustion numerical simulation.

(2) Piston group contact area: this area was in contact with the piston group. The piston group and the cylinder liner were connected by a lubricating oil film, which simplifies the heat transfer of the lubricating oil film into one-dimensional thermal resistance along the radial direction. Based on the above assumptions, the heat transfer boundary between the piston and the cylinder liner can be written as [23]:

$$q_{p-l} = \frac{\lambda_{oil}}{\delta_{oil}}(T_p - T_l) \quad (13)$$

where q_{p-l} is the heat flux between the piston group and cylinder liner, λ_{oil} is the thermal conductivity of the lubricant, δ_{oil} is the thickness of the lubricant, T_p is the piston temperature, and T_l is the cylinder liner temperature. Regarding the thickness of the lubricating oil film, thicknesses of 15 μm and 0.5 mm were set for the ring and skirt lubricating oil film, respectively.

(3) The air convection area of the crankcase: this area was in contact with lubricating oil mist and crankcase air. Referring to the empirical value, the heat transfer coefficient of the crankcase was selected as 200 $\text{W}/\text{m}^2\cdot\text{K}$, and the temperature of the crankcase was selected as 353 K.

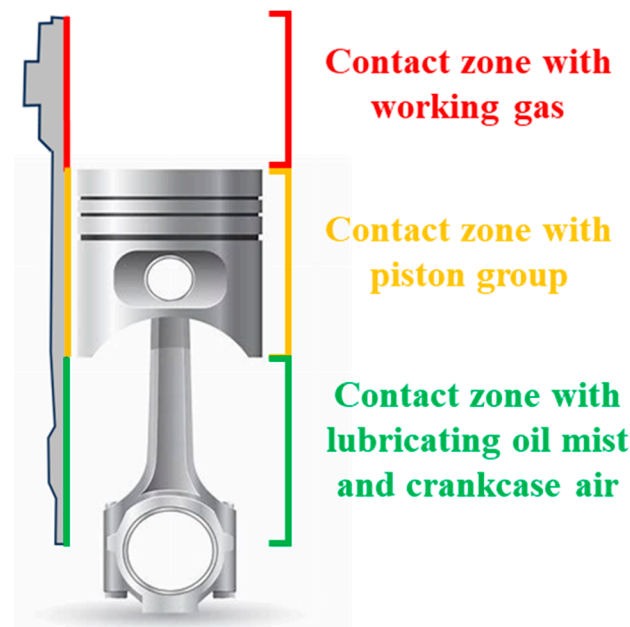


Figure 7. Schematic diagram of the calculation area of the cylinder liner inner surface.

According to the instantaneous gas temperature and heat transfer coefficient in the cylinder obtained by the combustion numerical simulation, combined with the instantaneous position of the piston, the inner surface of the cylinder liner was divided into regions, and the instantaneous thermal boundary conditions were expressed through ANSYS APDL command.

For the mechanical load between the piston group and cylinder liner, the effect of piston lateral thrust force was considered. The detailed calculation procedure was the same as that in Cavalli's research. [24] The lateral thrust force was applied to the piston group contact area.

In the modeling process, the temperature distribution of the engine was calculated firstly based on the temperature distribution; the thermal deformation was then calculated by the finite element method. At the same time, the mechanical deformation was calculated individually, and the modeling method was the same as Cavalli's research [24]. The thermo-mechanical deformation is obtained by superimposing the thermal and mechanical deformations.

3.4. Analysis of Cylinder Liner Deformation

The transient temperature distribution of the cylinder liner is shown in Figure 8. The maximum temperature was 506 K with a fluctuation of 6 K. The temperature distribution of the liner was higher at the top. The temperature was uniform in the circumferential direction and decreased from the top to the bottom. The distribution of temperature was similar to that in Bi's research [19]; Bi attributed the uneven temperature to the role of uneven cooling water jacket flow heat transfer and gas temperature changes during combustion.

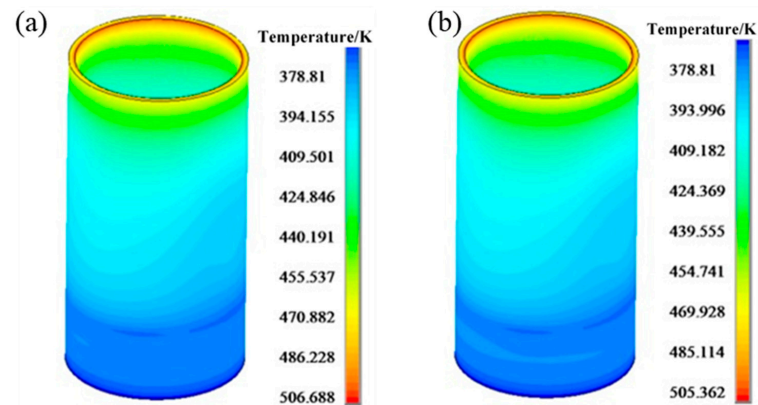


Figure 8. Transient temperature distribution at (a) The highest temperature moment of the piston group (370 °CA) and (b) maximum burst pressure moment (364 °CA).

The deformation of the cylinder liner is shown in Figure 9. The maximum deformation was 69.5 μm , which appeared at the top of the cylinder liner at the maximum burst pressure moment. The deformation amplitude of the cylinder liner was much lower than in Bi's research [19], we determined that the difference was mainly due to differences in engine body construction. The deformation amplitude showed a decreasing trend from the top to the bottom of the cylinder liner. Under the influence of high-temperature gas, serious thermal deformation was observed on the top of the cylinder liner. Although the thermal deformation of the cylinder was constrained by the engine block, the engine block itself will also deform under the thermal load. Moreover, the body near the top dead center area has undergone a relatively high thermal load due to poor cooling water heat dissipation ability, which results in limited constraints on the radial thermal deformation of the cylinder liner by the body. Under the influence of cyclic changes in lateral thrust force, the bottom part of the cylinder liner, which was the load-bearing part with low stiffness, suffered from intermittent large deformation.

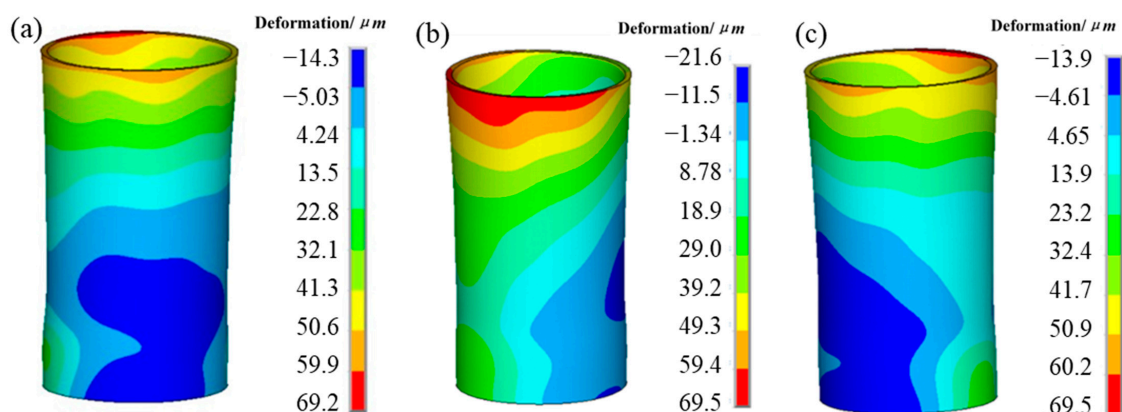


Figure 9. Deformation of the cylinder liner under thermo-mechanical load at (a) -36 °CA (during intake stroke) (b) 4 °CA (Maximum burst pressure moment) (c) 240 °CA (during exhaust stroke).

4. Cylinder Liner Shape and Position Tolerance

The cylinder liner deformation in Figure 8 shows the transient deformation of the cylinder liner. Based on the above results, the shape and position tolerance of the cylinder liner were calculated to further investigate the overall deformation of the cylinder.

4.1. Data Extraction and Processing

The cylinder liner of the diesel engine was divided into 35 sections at equal intervals in the axial direction, and the section near the cylinder head was taken as Section 1. In the circumferential direction of each section, 141 sampling points were taken at equal intervals. The center point of the top surface of the cylinder liner was taken as the coordinate origin O , the radial direction pointing to the anti-thrust side was selected as the positive x -axis, and the axis pointing to the cylinder liner bottom was taken as the positive z -axis. The sampling section of the cylinder liner was divided, and the Cartesian coordinate system was established as shown in Figure 10. For the piston, twelve sampling points were arranged on the piston skirts of the thrust side and the anti-thrust side, respectively, to analyze the deformation behavior of the piston.

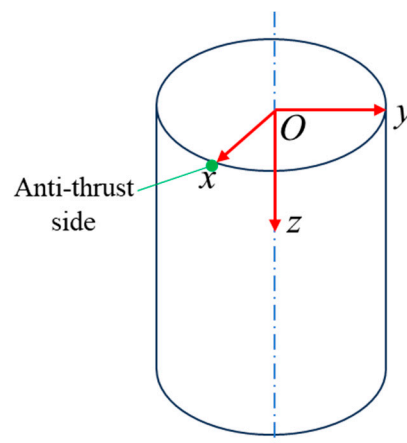


Figure 10. The Cartesian coordinate system for the cylinder liner.

The cylinder liner deformation at each moment in Section 3 was then exported as matrix A of $N \times 6$ and given as:

$$A = \begin{bmatrix} x_1 & y_1 & z_1 & \Delta x_1 & \Delta y_1 & \Delta z_1 \\ x_2 & y_2 & z_2 & \Delta x_2 & \Delta y_2 & \Delta z_2 \\ \dots & \dots & \dots & \dots & \dots & \dots \\ x_{n-1} & y_{n-1} & z_{n-1} & \Delta x_{n-1} & \Delta y_{n-1} & \Delta z_{n-1} \\ x_n & y_n & z_n & \Delta x_n & \Delta y_n & \Delta z_n \end{bmatrix} \quad (14)$$

where (x_n, y_n, z_n) was the original coordinate of sampling point n , and $(\Delta x_n, \Delta y_n, \Delta z_n)$ was the corresponding displacement of point n . The matrix A was then imported into Matlab, and the coordinate of sampling point n after deformation was then given as:

$$B = \begin{bmatrix} x_1 + \Delta x_1 & y_1 + \Delta y_1 & z_1 + \Delta z_1 \\ x_2 + \Delta x_2 & y_2 + \Delta y_2 & z_2 + \Delta z_2 \\ \dots & \dots & \dots \\ x_{n-1} + \Delta x_{n-1} & y_{n-1} + \Delta y_{n-1} & z_{n-1} + \Delta z_{n-1} \\ x_n + \Delta x_n & y_n + \Delta y_n & z_n + \Delta z_n \end{bmatrix} \quad (15)$$

The coordinate matrix B was taken with 4°CA as a load step and then imported into the algorithm to obtain the shape and position tolerance.

4.2. Out-of-Roundness of Cylinder Liner

The out-of-roundness distribution of each sampling section in one working cycle of the diesel engine was solved by an iterative solution of every 4 °CA, as shown in Figure 11.

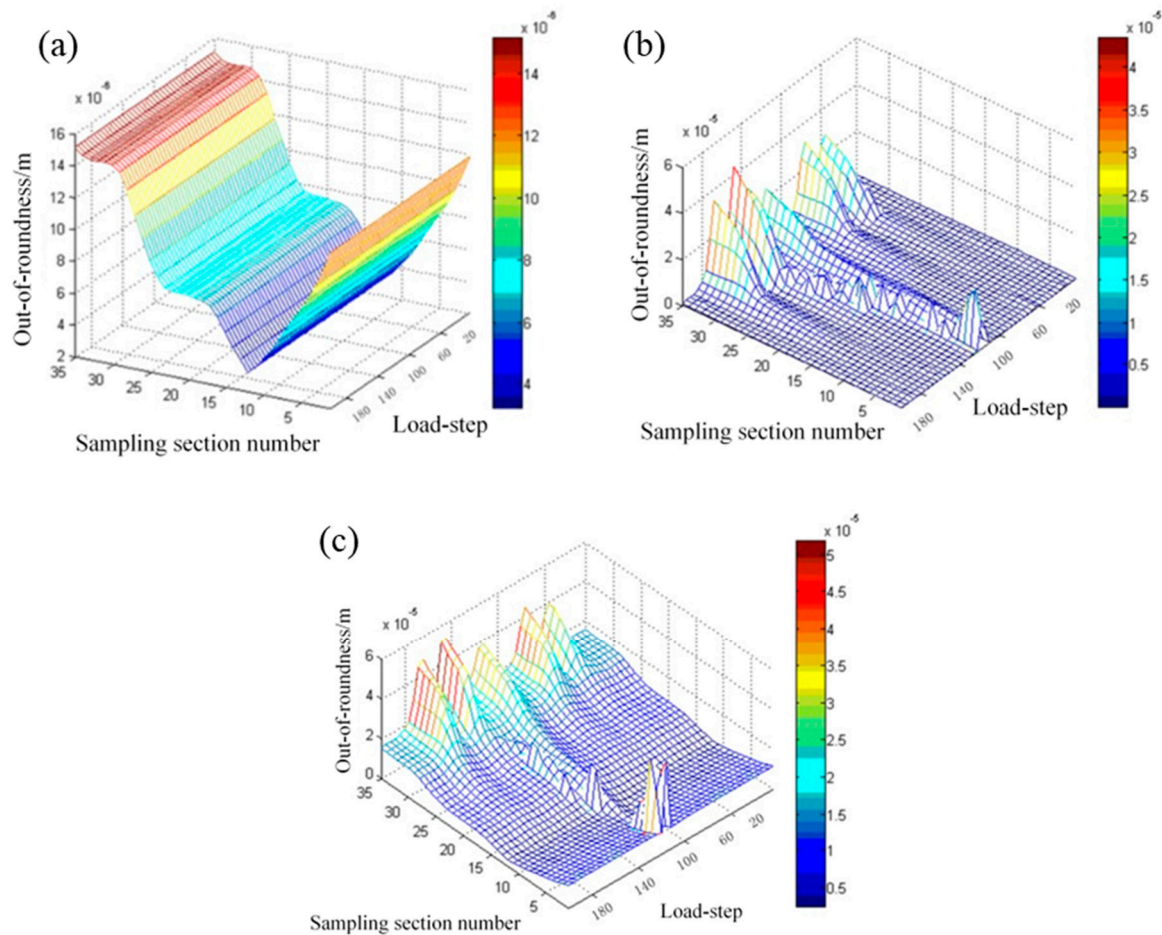


Figure 11. Out-of-roundness of cylinder liner under (a) thermal load, (b) mechanical load and (c) thermo-mechanical load.

As shown in Figure 11a, the maximum out-of-roundness of the cylinder liner under thermal load was 15.12 μm , which appears at 80 °CA in the 35th sampling section. The high-temperature gas was evenly distributed on the inner wall of the cylinder liner, and the thermal stress fluctuated smoothly in the time domain. The out-of-roundness of each sampling section of the cylinder liner did not change much in the time domain. The bottom end of the cylinder liner was in a free state, and the out-of-roundness was slightly higher.

As shown in Figure 11b, the cylinder liner suffered from a maximum out-of-roundness of 43.40 μm under mechanical load, which appeared at 156 °CA in the 35th sampling section. Near 120 °CA, 340 °CA, 380 °CA, 560 °CA, and 680 °CA, the piston moves to the lower half of the cylinder liner, and the out-of-roundness on the thrust side was much larger. Near 364 °CA, the explosion pressure of the cylinder was relatively high, and the extreme value of out-of-roundness appeared in the upper part of the cylinder liner. The lateral thrust force was more likely to cause uneven deformation of the cylinder liner thrust side. The top of the cylinder liner was connected with the cylinder head, so the stiffness was relatively greater. The bottom of the cylinder liner was in a free state, the peak of the out-of-roundness of the cylinder liner appeared on the anti-thrust side in the lower half of the cylinder liner. The lateral thrust force was the main factor affecting the position and extreme value of the cylinder liner out-of-roundness.

As shown in Figure 11c, the maximum out-of-roundness of the cylinder liner under thermo-mechanical load was $51.76 \mu\text{m}$, which occurred at 496°CA in the 35th sampling section. The out-of-roundness of the cylinder liner was larger than that of the cylinder liner under a single thermal or mechanical load, and the deformation of each sampling section was more uneven. The out-of-roundness surface distribution was basically the same as the out-of-roundness surface of the cylinder liner under mechanical load, and the mechanical load was the dominant factor that affected the out-of-roundness of the cylinder liner.

4.3. Cylinder Liner Coaxiality

The offset of the least square center in each sampling section relative to the reference axis under each load step is shown in Figure 12. The sub-figure marked with green, blue and red boxes indicates the x-direction, y-direction and overall offset of the least square center, respectively. The reason for recording the offset of the least squares circle center is that the coaxiality is a scalar, which cannot reflect the offset direction of each sampling section.

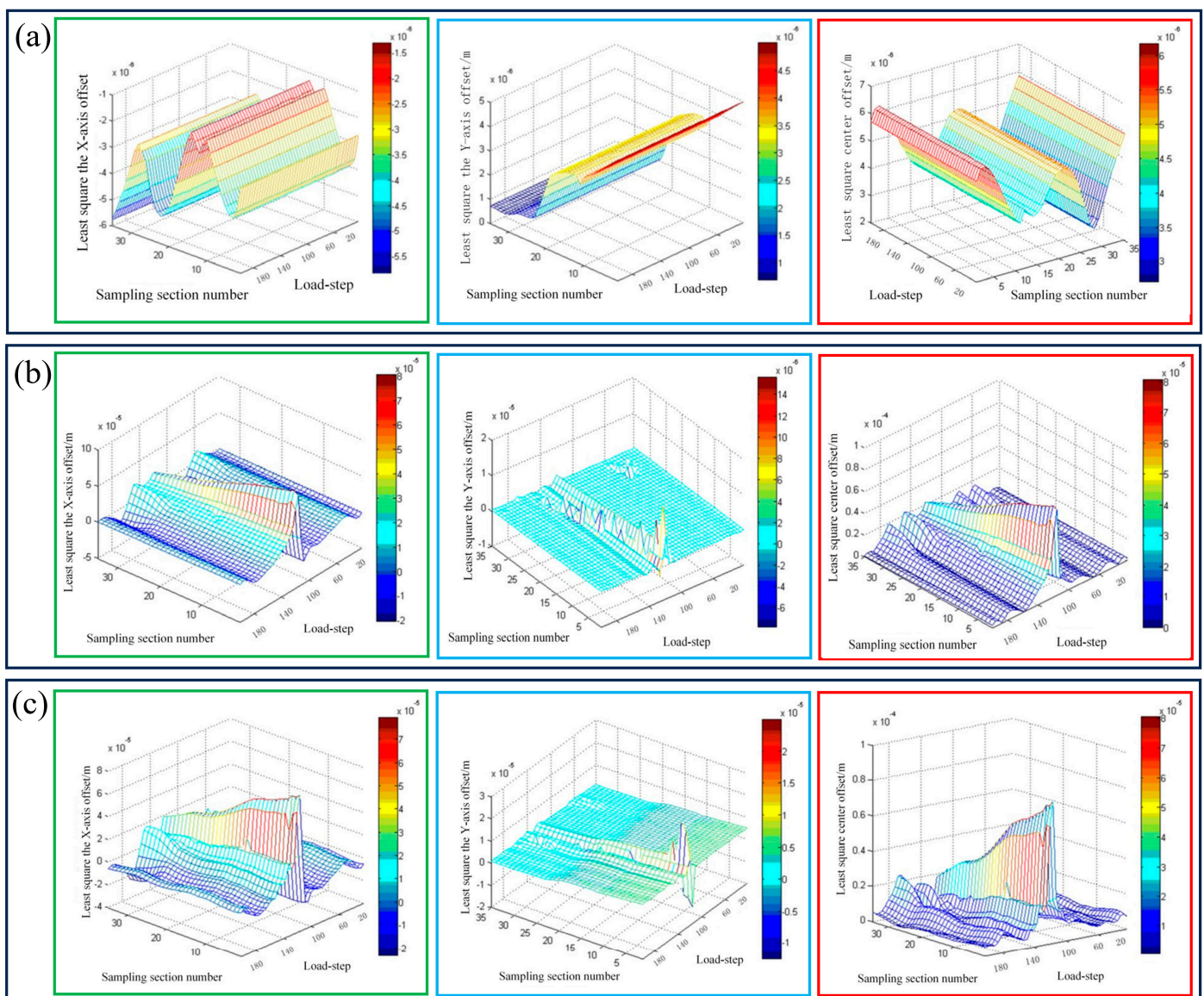


Figure 12. Offset of least square center of the cylinder linersampling section under (a) thermal load, (b) mechanical load and (c) thermo-mechanical load.

For cylinder liners under thermal load, as shown in Figure 12a, the maximum offset of the least squares circle center in the x-direction was $-5.86\text{ }\mu\text{m}$, which appeared at $14\text{ }^{\circ}\text{CA}$ in the 35th sampling section. The maximum offset in the y-direction of the least squares circle center was $4.98\text{ }\mu\text{m}$, which appeared at $352\text{ }^{\circ}\text{CA}$ in the first sampling section. The maximum coaxiality was $6.17\text{ }\mu\text{m}$, appearing at $372\text{ }^{\circ}\text{CA}$, as shown in Figure 13a. The area of the cooling water cavity on the anti-thrust side was larger, the thermal expansion of the cylinder liner was therefore hindered, and each sampling section was offset toward the negative x-direction. The high-temperature gas acted evenly on each sampling section of the cylinder liner, and the thermal stress fluctuated relatively smoothly in the time domain. The offset difference in each sampling section was limited, and the coaxiality value was relatively low, as shown in Figure 13a. According to the temperature distribution results in Figure 8, the temperature distribution along the circumferential on both the top and bottom cross section was uneven, which may result in uneven deformation in the top and bottom cross section. However, the top of the cylinder liner was constrained by the engine body, and the out-of-roundness for the section near the cylinder liner top did not change significantly in the time domain. The bottom of the cylinder liner was in a free-expansion state, and the out-of-roundness in the bottom section was slightly higher.

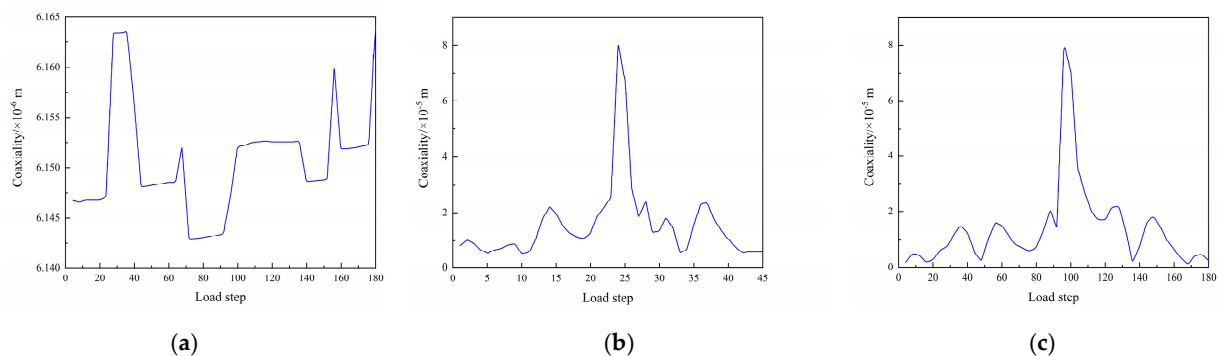


Figure 13. Concentricity of cylinder liner under (a) thermal load, (b) mechanical load and (c) thermo-mechanical load.

For cylinder liners under mechanical load, as shown in Figure 12b, the maximum offset of the least squares circle center in the x-direction was $80.27\text{ }\mu\text{m}$, which appeared at $372\text{ }^{\circ}\text{CA}$ in the second sampling section. The maximum offset in the y-direction was $15.57\text{ }\mu\text{m}$, which appeared at $376\text{ }^{\circ}\text{CA}$ in the second sampling section. The maximum deformation under mechanical load was dominated by x-direction deformation, which was the same as the piston thrust side. The maximum coaxiality was $80.49\text{ }\mu\text{m}$ appeared at $372\text{ }^{\circ}\text{CA}$, as shown in Figure 13b. The moment for maximum deformation appeared at the piston knocking time near the top dead center. We determined that the maximum deformation under mechanical load was mainly influenced by piston knocking. Near the combustion top dead center, the explosion pressure of the cylinder was raised suddenly, which had a greater impact on the cylinder liner and the engine body. The top of the cylinder liner was directly in contact with the explosion gas, so the offset of the sampling section here was larger than that of the lower half of the cylinder liner. The stiffness on the anti-thrust side was lower than that on the thrust side, so the cylinder liner deviated to the anti-thrust side as a whole. The peak value of liner coaxiality was mainly affected by the cylinder explosion pressure.

For cylinder liners under thermo-mechanical load, as shown in Figure 12c, the maximum offset of the least squares circle center in the x-direction was $79.21\text{ }\mu\text{m}$, which appeared at $384\text{ }^{\circ}\text{CA}$ in the first sampling section. The maximum offset in the y-direction was $24.88\text{ }\mu\text{m}$, which appeared at $24\text{ }^{\circ}\text{CA}$ in the third sampling section. The maximum coaxiality was $80.22\text{ }\mu\text{m}$, which appeared at $380\text{ }^{\circ}\text{CA}$. The maximum deformation direction, the deformation direction, and maximum deformation amplitude were the same as that in pure mechanical load; in this situation, we determined that the thermo-mechanical

deformation was dominated by mechanical load. The coaxiality of the cylinder liner was similar to that of the cylinder liner under pure mechanical load. The distribution of the least squares center surface and the coaxiality curve was basically the same as that of the cylinder liner under the mechanical load. The mechanical load dominated the cylinder liner coaxiality. The offset of the sampling section near the cylinder liner head was larger than that of the lower half of the cylinder liner. In addition, the anti-thrust side had lower stiffness than the thrust side, and the cylinder liner deviated toward the anti-thrust side as a whole.

4.4. Piston Deformation Behavior

The deformation of the piston in the radial direction is shown in Figure 14, in which the sub-figure marked with green and yellow boxes indicates the radial deformation on the thrust side and anti-thrust side, respectively. Sampling number 1 indicates the sampling point in the upper part of the piston skirt and sampling number 12 indicates the sampling point near the piston bottom.

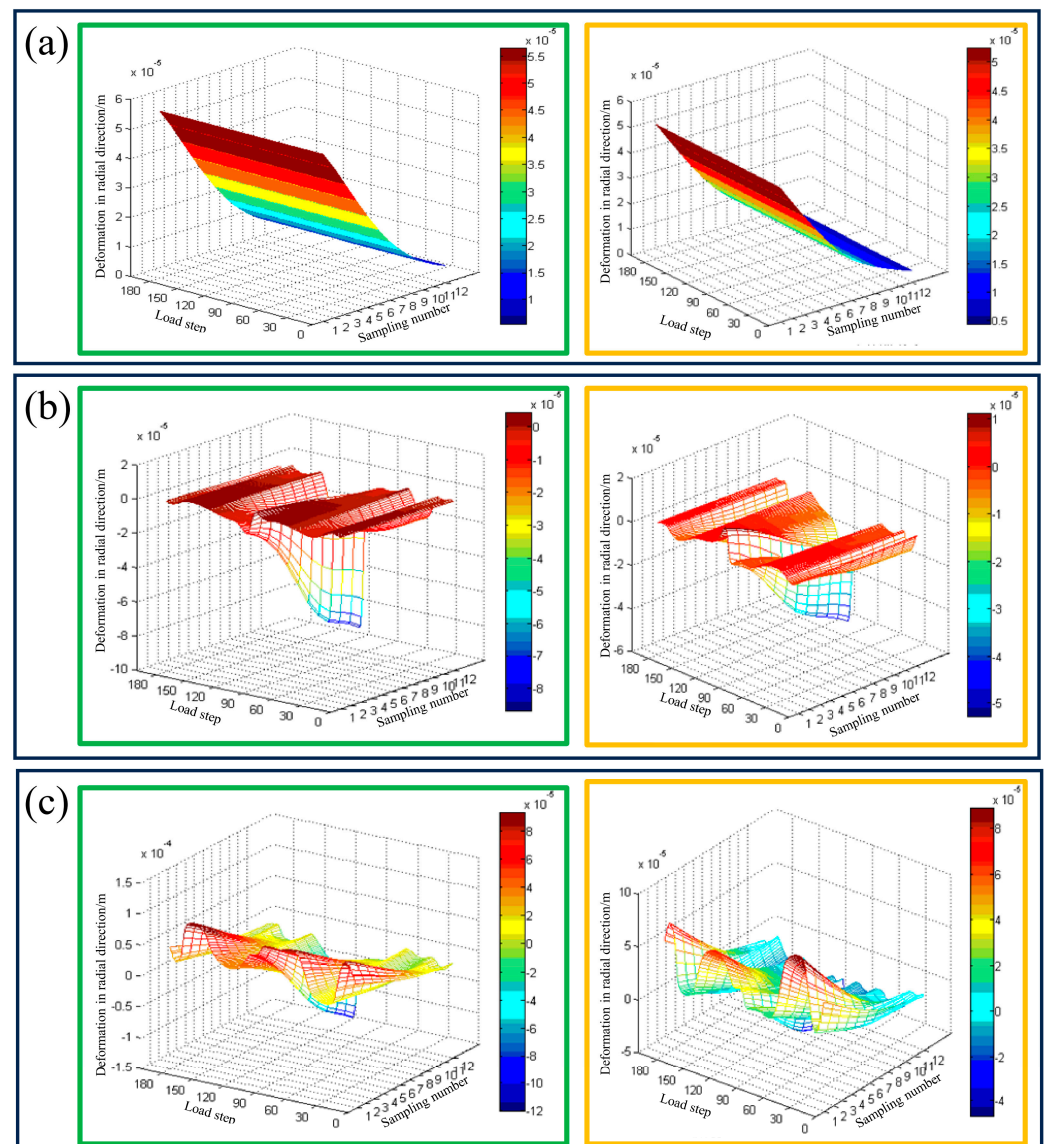


Figure 14. Radial deformation of the piston under (a) thermal load, (b) mechanical load and (c) thermo-mechanical load.

Under pure thermal load, as shown in Figure 14a, the radial deformation on both sides showed a decreasing trend from the top to the bottom of the piston skirt. There was little difference in radial deformation on both sides. The change in radial deformation during the working cycle of the engine was stable. The maximum radial deformation was 56.38 and 52.33 μm on the thrust and anti-thrust side, respectively, which appeared at 340 °CA in sampling point 1. This was due to the fact that sampling point 1 was close to the piston head, and the thermal load was the highest among the sampling points.

Under pure mechanical load, as shown in Figure 14b, the maximum radial deformation on the thrust side was $-68.26 \mu\text{m}$, which appeared at 392 °CA in sampling point 12 (near the bottom of the piston skirt). The deformation at 420 °CA was also relatively high. The maximum radial deformation on the anti-thrust side was $-52.87 \mu\text{m}$, which appeared at 352 °CA in sampling point 12. The deformation at -45 and 600 °CA was also relatively high.

Under pure thermo-mechanical load, as shown in Figure 14c, the maximum positive radial deformation on the thrust side was 92.91 μm , which appeared at 372 °CA in sampling point 1. The maximum negative radial deformation on the thrust side was $-120.11 \mu\text{m}$, which appeared at 392 °CA in sampling point 12. The maximum positive radial deformation on the anti-thrust side was 88.53 μm , which appeared at 116 °CA in sampling point 1. The maximum negative radial deformation on the anti-thrust side was $-46.93 \mu\text{m}$, which appeared at 352 °CA in sampling point 12. Under the action of thermal load, the piston was expanded outward, while under the action of lateral thrust force, the influence of thermal and mechanical loads on the variation in piston deformation was partially offset by each other.

The out-of-roundness may result in a variation in the clearance between the piston group and the cylinder liner. A higher clearance was beneficial for reducing friction work and improving mechanical efficiency, but it can cause an increase in engine gas leakage, thereby reducing thermal efficiency. Therefore, the clearance needs to be controlled within a reasonable range. To minimize the out-of-roundness, the compensation of functional bore distortions [25] and non-circular liner profile [20] could be adopted to improve the working efficiency of diesel engines.

5. Conclusions

In this work, a numerical combustion model was proposed to obtain the thermal boundaries of a cylinder liner and piston. Simulation of the deformation behavior of the piston and cylinder liner under thermo-mechanical load was performed. The out-of-roundness and coaxiality were selected to evaluate the shape and position changes in the cylinder liner. The following conclusions were drawn from the obtained results:

- (1) Under the action of thermal load, the out-of-roundness of each sampling section did not change significantly in the time domain, and the bottom of the cylinder liner was in a free state with a slightly higher out-of-roundness value of 15.12 μm . Under the action of mechanical load, the peak out-of-roundness appeared on the anti-thrust surface near the lower half of the cylinder liner with an out-of-roundness value of 43.40 μm , and the lateral thrust force was more likely to cause uneven deformation of the cylinder liner cross section.
- (2) The maximum out-of-roundness of the cylinder liner under thermo-mechanical load was 51.76 μm , which appeared at 496 °CA in the 35th sampling section; the out-of-roundness of the cylinder liner under thermo-mechanical load was greater than that under pure single load, and the deformation of each sampling section was more uneven. The distribution form of out-of-roundness under thermo-mechanical load was basically consistent with that under pure mechanical load.
- (3) Under the action of thermal load, each sampling section deviated toward the thrust side. The high-temperature gas acts evenly on each sampling section of the cylinder liner. The difference in offset of each sampling section was limited, and the coaxiality value is low with a maximum coaxiality of 6.17 μm . Under the action of mechanical load, the peak value of coaxiality was mainly affected by the cylinder explosion

pressure. The stiffness of the anti-thrust side was lower than the thrust side, and the cylinder liner was deviated to the anti-thrust side as a whole with a maximum coaxiality of 80.49 μm .

- (4) The maximum coaxiality of the cylinder liner under thermo-mechanical load was 80.22 μm , which appears at 380 °CA. The coaxiality was similar to that under a pure single mechanical load. The distribution of the least squares center surface and the coaxiality curve was basically the same as that under pure mechanical load. The mechanical load accounts for the dominant influence on the cylinder liner coaxiality status.
- (5) Under pure thermal load, the radial deformation on both sides showed a decreasing trend from the top to the bottom of the piston skirt. There was little difference in radial deformation on both sides. Under pure mechanical load, the deformation near the bottom of the skirt was the highest due to its low stiffness. Under the action of thermal load, the piston was expanded outward, while under the action of lateral thrust force, the influence of thermal and mechanical loads on the variation in piston deformation was partially offset by each other
- (6) In future work, the design of the shaped cylinder liner's profile based on the deformation results in this work could be carried out to reduce the friction on the piston–cylinder liner, and the work on the effect of the cylinder liner profile on friction could also be carried out.

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Conflicts of Interest: The authors declare no conflict of interest.

Nomenclature

Nomenclature		Greek letters	
r_1	radius of the inscribed circle	ε	convergence judgment coefficient
r_2	radius of the circumscribed circle	λ_{oil}	thermal conductivity of the lubricant
h_r	heat transfer coefficient for the aimed position	δ_{oil}	the thickness of the lubricant
h_m	the mean heat transfer coefficient in a working cycle	Abbreviations	
R	piston radius	FEA	Finite element analysis
r	distance between the aimed position and piston center	MZC	minimum zone circle method
q_{p-l}	heat flux between piston group and cylinder liner	LSC	least square circle method
T_p	piston temperature	MCC	minimum circumscribed circle method
T_l	the cylinder liner temperature	MIC	maximum inscribed circle method

References

- Kaiser, M.; Morin, T. Centers, out-of-roundness measures, and mathematical programming. *Comput. Ind. Eng.* **1994**, *26*, 35–54. [[CrossRef](#)]
- Lazar, V.; Erokhin, M.; Vergazova, Y.G.; Kataev, Y.V.; Gradov, E.A. Improvement of the method of micrometry of cylinder liners. *J. Phys. Conf. Ser.* **2021**, *1889*, 042041. [[CrossRef](#)]
- Ali, S.H.R.; Mohamed, H.H.; Bedewy, M.K. Identifying cylinder liner wear using precise coordinate measurements. *Int. J. Precis. Eng. Manuf.* **2009**, *10*, 19–25. [[CrossRef](#)]
- Bulsara, M.A.; Bhatt, D.V.; Mistry, K.N. Measurement of oil film thickness between piston ring and liner using strain gauge. *Ind. Lubr. Tribol.* **2013**, *65*, 297–304. [[CrossRef](#)]

5. Spencer, A.; Avan, E.Y.; Almqvist, A.; Dwyer-Joyce, R.S.; Larsson, R. An experimental and numerical investigation of frictional losses and film thickness for four cylinder liner variants for a heavy duty diesel engine. *Proc. Inst. Mech. Eng. Part J J. Eng. Tribol.* **2013**, *227*, 1319–1333. [\[CrossRef\]](#)
6. Mishra, P.C.; Roychoudhury, A.; Banerjee, A.; Saha, N.; Das, S.R.; Das, A. Coated Piston Ring Pack and Cylinder Liner Elastodynamics in Correlation to Piston Subsystem Elastohydrodynamic: Through FEA Modelling. *Lubricants* **2023**, *11*, 192. [\[CrossRef\]](#)
7. Menacer, B.; Narayan, S.; Mahroogi, F.O. Studying hydrodynamic friction in the oil ring/cylinder liner contact of an internal combustion engine. *Cogent Eng.* **2023**, *10*, 2289261. [\[CrossRef\]](#)
8. Li, T.; Wu, J.; Ge, C.; Wang, L.; Yu, Y.; Ma, X.; Qiao, Z.; Tang, H. Molecular dynamics investigation on micro-friction behavior of cylinder liner-piston ring assembly. *Tribol. Lett.* **2023**, *71*, 78. [\[CrossRef\]](#)
9. Hei, D.; Zheng, M.; Liu, C.; Jiang, L.; Zhang, Y.; Zhao, X. Study on the frictional properties of the top ring-liner conjunction for different-viscosity lubricant. *Adv. Mech. Eng.* **2023**, *15*, 1–13. [\[CrossRef\]](#)
10. Gao, T.; Chen, H.; Tang, D.; Wang, Y. Inspired by Earthworms and Leeches: The Effects of Cylindrical Pit Arrays on the Performance of Piston-Cylinder Liner Friction Pairs. *Appl. Sci.* **2023**, *13*, 11580. [\[CrossRef\]](#)
11. Liu, N.; Wang, C.; Xia, Q.; Gao, Y.; Liu, P. Simulation on the effect of cylinder liner and piston ring surface roughness on friction performance. *Mech. Ind.* **2022**, *23*, 8. [\[CrossRef\]](#)
12. Chen, T.; Wang, L.; Xu, J.; Gao, T.; Qin, X.; Yang, X.; Cong, Q.; Jin, J.; Liu, C. Effect of Groove Texture on Deformation and Sealing Performance of Engine Piston Ring. *Machines* **2022**, *10*, 1020. [\[CrossRef\]](#)
13. Alshwawra, A.; Abo Swerih, A.; Sakhrieh, A.; Dinkelacker, F. Structural Performance of Additively Manufactured Cylinder Liner—A Numerical Study. *Energies* **2022**, *15*, 8926. [\[CrossRef\]](#)
14. Zhu, X.; Cheng, Y.; Wang, L. Study on Deformation Characteristics of Diesel Engine Cylinder Liner under Different Influencing Factors. *J. Phys. Conf. Ser.* **2021**, *1885*, 032019. [\[CrossRef\]](#)
15. Bi, Y.; Wang, P.; Xiang, R.; Wen, J.; Lei, J.; Shen, L.; Xin, Q. Numerical investigation on the operating characteristics of the cylinder liners of a turbocharged diesel engine. *Sadhana* **2021**, *46*, 150. [\[CrossRef\]](#)
16. Bi, Y.; Wang, P.; Luo, L.; Wang, H.; Xin, Q.; Lei, J.; Shen, L. Analysis of out-of-round deformation of a dry cylinder liner of a non-road high-pressure common-rail diesel engine based on multi-field coupling. *J. Braz. Soc. Mech. Sci. Eng.* **2021**, *43*, 50. [\[CrossRef\]](#)
17. Al-Baghdadi, M.A.R.S.; Ahmed, S.S.; Ghayadh, N.A. Three-dimensional CFD-solid mechanics analysis of the hydrogen internal combustion engine piston subjected to thermomechanical loads. *Int. J. Renew. Energy Dev.* **2023**, *12*, 560–568. [\[CrossRef\]](#)
18. Rao, X.; Sheng, C.; Guo, Z.; Yuan, C. A review of online condition monitoring and maintenance strategy for cylinder liner-piston rings of diesel engines. *Mech. Syst. Signal Process.* **2022**, *165*, 108385. [\[CrossRef\]](#)
19. Bi, Y.; Xu, Y.; Liu, S.; Chen, S.; Shen, L.; Xu, Y. Cylinder liner deformation characteristics based on fourier transform and impact on ring pack sealing and friction power. *J. Braz. Soc. Mech. Sci. Eng.* **2023**, *45*, 242. [\[CrossRef\]](#)
20. Alshwawra, A.; Pasligh, H.; Hansen, H.; Dinkelacker, F. Increasing the roundness of deformed cylinder liner in internal combustion engines by using a non-circular liner profile. *Int. J. Engine Res.* **2019**, *22*, 1214–1221. [\[CrossRef\]](#)
21. ISO 12181–2011; Technical Requirements for the Parameters of Products. Roundness. Part 1. Dictionary and Roundness Parameters. ISO: Geneva, Switzerland, 2011.
22. Nabli, H. An overview on the simplex algorithm. *Appl. Math. Comput.* **2009**, *210*, 479–489. [\[CrossRef\]](#)
23. Liu, B.; Aggarwal, S.K.; Zhang, S.; Tchakam, H.U.; Luo, H. Thermo-mechanical coupling strength analysis of a diesel engine piston based on finite element method. *Int. J. Green Energy* **2024**, *21*, 732–744. [\[CrossRef\]](#)
24. Cavalli, M.; Lavacchielli, G.; Tonelli, R.; Nicoletto, G.; Riva, E. Comparison of analytical and multibody dynamic approaches in the study of a V6 engine piston. *Proc. Inst. Mech. Eng. Part K J. Multi-Body Dyn.* **2017**, *231*, 420–438. [\[CrossRef\]](#)
25. Flores, G.K. *Graded Freeform Machining of Cylinder Bores Using form Honing*; SAE Technical Paper, 2015-01-1725; SAE: Warrendale, PE, USA, 2015. [\[CrossRef\]](#)

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