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Research on Analysis of Friction and Lubrication Characteristics of Piston-Cylinder System in Internal Combustion Engines Using Improved Multi-Layer Thermal Resistance Model

Qian Yichao[1,*], Li Tao[1], Zhang Jie[1]

[1]*Faculty of Mechanical and Electrical Engineering, Kunming University of Science and Technology, Kunming 650500, China*
[*]*Corresponding author: qianyichao@stu.kust.edu.cn*

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Abstract: To analyze the friction and lubrication characteristics of the piston-cylinder system in internal combustion engines using an improved multi-layer thermal resistance model, this study combines the theory of thermal boundary conditions to design a lubrication and friction calculation model, and conducts numerical computations and experimental research. Firstly, a lubrication model for the piston skirt-cylinder liner is proposed, which includes key parameters such as the lubrication model for the piston skirt and the lubricating oil characteristic equation. Secondly, the friction and lubrication theory for the piston ring-cylinder liner is discussed, which includes the leakage model and the piston ring lubrication model. Additionally, the multi-layer thermal resistance model is improved to accurately describe the lubrication of the piston assembly under the thermal boundary conditions of the cylinder liner. Subsequently, experimental research is conducted to investigate the friction and lubrication characteristics of the piston-cylinder system in internal combustion engines under different temperatures. The experiments reveal that: (1) Linear function 1 and sinusoidal function 1 perform well in the piston skirt of the cylinder liner, with lower average friction power consumption and relatively small relative errors of 5% and 6%, respectively. (2) In the experimental study of the friction and lubrication characteristics of the piston ring-cylinder liner, function 2 in the linear function approach has slightly higher average friction power consumption compared to function 1. In the parabolic function approach, function 1 exhibits lower average friction power consumption, while function 2 significantly increases it. In conclusion, this research comprehensively explores the friction and lubrication characteristics of the piston-cylinder system in internal combustion engines by introducing an improved multi-layer thermal resistance model. The aim of this study is to provide theoretical foundations and experimental evidence for further optimizing the design and operation of internal combustion engines.

1. Introduction

Internal combustion engine is a kind of energy conversion equipment that converts the internal energy of fuel combustion into kinetic energy, which has a history of more than 100 years from the 19th century to the present, and has a wide range of applications in transportation, engineering, power generation and other fields. Despite generations of improvements, only 40% of the combustion energy of an internal combustion engine is currently used for power output, and another 10% is caused by friction loss. For example, in a car heating system, heat loss can still be used as a heat source, but friction loss is a non-renewable energy that must be reduced [1].

The emergence of lubricants in this field is highly anticipated. In foreign countries, Ventikos et al. (2022) adopted the energy equation in two-dimensional form and introduced two-dimensional fluid mechanics and elastic flow mechanics models on the basis of two-dimensional fluid mechanics and elastic flow mechanics. After that, different engine opening speeds were set, and the effects of shear heating on the lubrication state under engine starting conditions were explored by measuring the hydrodynamic oil film thickness, viscosity and temperature [2]. Kim et al. (2022) established a hydrodynamic lubrication model of incompressible Newtonian liquid lubricating oil in the mixed lubrication zone by using the finite difference method, and carried out numerical simulation of it by using the two-dimensional energy equation [3]. Mohamed et al. (2022) discussed the variation of engine characteristics at different temperatures, different cooling media and different oil temperatures, and analyzed the test results. It is found that compared with the temperature of the coolant, the temperature change of the lubricating oil has a more significant impact on the performance of the internal combustion engine [4]. In China, Song Haitao (2023) studied the friction dynamics and friction loss of the piston during cold and hot starting of the Mars ignition engine, and the results showed that the friction loss of the piston at low temperature increased by 72% compared with that during hot starting [5]. When simulating piston skirt lubrication, Zhang Weizzheng et al. (2023) conducted simulation and comparison with the thermal deformation of piston skirt as a variable, and found that whether the thermal deformation was considered or not would have a great impact on the oil film thickness and friction loss of piston skirt [6].

To sum up, the current relevant research mainly focuses on the lubrication state and friction loss of the engine under starting conditions, but does not involve the change of lubrication conditions in other engine working conditions. Therefore, the lubrication characteristics of the engine under all working conditions need to be further studied. Based on this, the improved multi-layer thermal resistance model is used to design the lubrication friction calculation model of the cylinder liner piston system of the internal combustion engine, so as to improve the accuracy of the lubrication friction characteristics analysis of the cylinder liner piston system, and deeply understand the lubrication conditions in the engine cylinder and its influence on the performance.

2. Establishment of theoretical basis and lubrication calculation model

2.1 Lubrication friction and thermal boundary condition theory

(1) Piston skirt-liner lubrication friction theory

1) Piston skirt lubrication model

According to the difference of contact surfaces and the different conditions required to form contact, lubrication can be divided into three types, namely hydrodynamic lubrication, mixed lubrication and boundary lubrication. These three lubrication states can be determined by the Stribeck curve [7].

2) Lubricating oil characteristic equation

For liquid lubricating media, temperature and pressure will have a great impact on its characteristics, and the viscosity of the lubricating oil is a measure of the viscosity of the lubricating oil. However, after the temperature rises, the viscosity of the lubricating oil will also become weaker; As the pressure increases, the viscosity of the lubricating oil will increase. Therefore, the Vogel/Barus equation was used to calculate the viscosity of lubricating oil [8].

Oil film thickness is an important index to measure the working state of the friction pair. When the roughness of the contact surface is ignored, the thickness of the oil film between the two contact surfaces is different due to various factors. In addition, since surface roughness, elastic deformation and thermal deformation will also directly affect the actual thickness of the oil film, we need to make comprehensive consideration when calculating the actual thickness of the oil film, using the following formula for calculation:

$$l_{actual} = l_n + \delta_e + \delta_t + \delta_{asp} \tag{1}$$

Shangjiazhong,$l_n$represents the nominal oil film thickness, that is, the vertical distance between different contact surfaces. δ_e stands for elastic deformation. δ_t stands for thermal deformation,δ_{asp}refers to the film thickness due to surface roughness.

The mean value $\overline{l_{actual}}$ of the actual oil film thickness is the mathematical expected value of the actual oil film thickness under normal distribution, and the formula is:

$$\overline{l_{actual}} = E\{l_{actual}\} = \int_{-l}^{\infty} l_{actual} f(\delta_{asp}) d\delta \tag{2}$$

In the above formula,(δ)is the probability function of δ representing the proportion causing rough contact.

4) Contact model

Because the roughness of the contact surface will have a certain impact on the lubrication friction of the contact surface, the lubrication friction will be affected by the surface protrusion height, l_1、l_2, as shown in Figure 1.

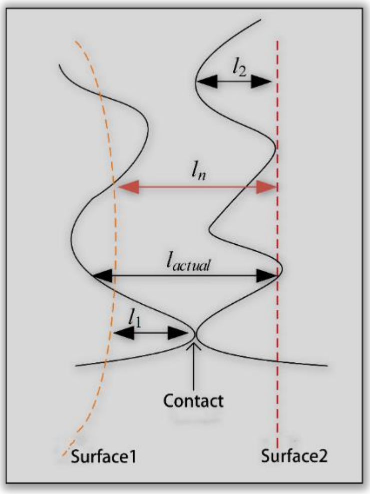


Figure 1: Contact model structure

Figure 1 shows that the protruding heights of micro-convex bodies are different, and their distribution is not regular. Therefore, in the calculation of roughness, it is assumed that the distribution of peak height will obey the Gaussian distribution, so that the standard deviation σ_1、σ_2 of the two contact surfaces can be obtained respectively.Where 1 and 2 refer to different surfaces.

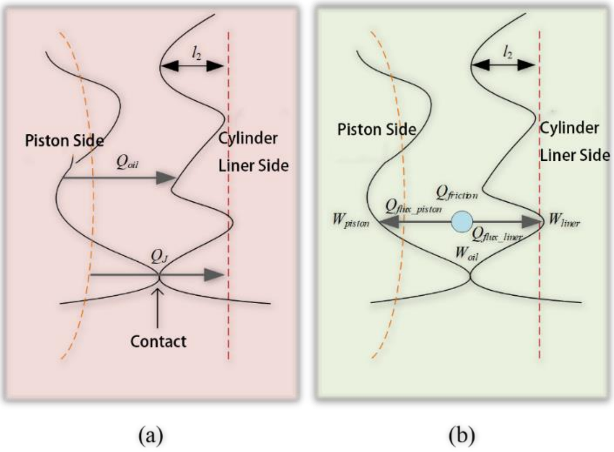
In addition, people also put forward the concept of film thickness ratio, which is defined as the ratio of nominal oil film thickness to comprehensive roughness, namely:

$$H = l_n/\sigma \tag{3}$$

In the above formula, if the film thickness ratio between the two contact surfaces is $H \geq 4$, then the lubrication state can be considered to be in the full film lubrication state. If the film thickness of the two friction surfaces is less than $H < 4$, it is in the mixed lubrication state.

5) Transient heat transfer model

The heat transfer structure diagram and frictional heat transfer diagram between piston cylinder liner structures are shown in Figure 2[9].



(a) Schematic diagram of the transient heat transfer model; (b) Illustration of frictional heat transfer

Figure 2: Transient heat transfer model and friction heat transfer illustration

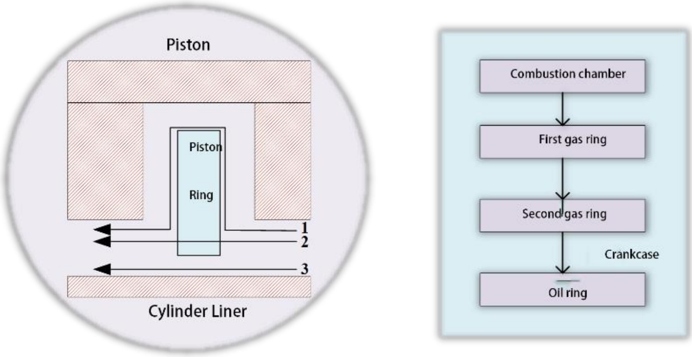
In picture 2,Q_J is the heat transfer when the micro-convex body is in contact,$Q_{oil}$is the heat transferred by the oil film,$W_{piston}$is the real-time temperature of the piston,W_{oil} is the temperature at which the piston creates friction,Q_{flux_piston} refers to the heat of the flux piston, and $Q_{friction}$ refers to the heat generated by the friction of the piston; A refers to the heat generated by the heat conduction layer;Q_{flux_liner} refers to the temperature generated by friction of the heat conduction layer.

(2) piston ring - cylinder liner friction lubrication theory

Through the analysis, it is found that the temperature distribution of oil film has a strong influence on the temperature of cylinder liner. Therefore, it is very meaningful to discuss the lubrication state of piston ring cylinder liner friction pair at different temperatures[10].

1) Air leakage model

Because when the engine is working, the piston ring will be subjected to a lot of burst pressure, and under the joint influence of the burst pressure and piston, the gas will leak from the end gap of the piston ring, the cylinder liner sliding surface of the piston ring and the upper and lower surface of the piston ring, so that the force on the upper and lower surface of the piston ring changes. It is necessary to calculate the pressure during leakage and cracking. The gas leakage model of the piston ring group is shown in Figure 3[11,12].



(a) (b)

(a) Leakage channel; (b) Leakage structure

Figure 3: Piston Ring Group Leakage Principle

In Figure 2, "1" indicates the gas leakage on the upper and lower ends of the piston ring, "2" indicates the gas leakage in the gap between the end of the piston ring, and "3" indicates the gas leakage between the piston ring and the contact surface of the cylinder liner.

2) Piston ring lubrication model

Based on the lubrication conditions of piston skirt, the lubrication model of piston ring cylinder liner friction pair was designed, and the mean Reynolds lubrication equation and micro-convex contact model were incorporated into it [13]. The lubrication equation is shown as follows:

$$\frac{\partial[\varphi_x\cdot(l_{actual}^3/12\eta)\cdot(\partial p/\partial x)]}{\partial x}=\frac{\partial[(\overline{l_{actual}}+\sigma\cdot\varphi_s)\cdot v_{ring}]}{2\partial x}+\frac{\partial(\overline{l_{actual}})}{\partial t} \tag{4}$$

In the above formula, v_{ring} is the axial speed of the piston ring.

(3) The thermal boundary conditions of piston group-cylinder liner lubrication calculation are determined

1) Piston temperature field boundary condition calculation theory

For the setting of temperature boundary conditions on the piston side surface, most of the past studies used a multi-layer thermal resistance model to calculate the temperature boundary of the piston side surface, and regarded the heat transfer path from the piston to the cylinder liner as a one-dimensional flat thermal resistance model [14]. The schematic diagram of the heat transfer path on the surface of the piston side and the structure of the thermal resistance model are shown in Figure 4.

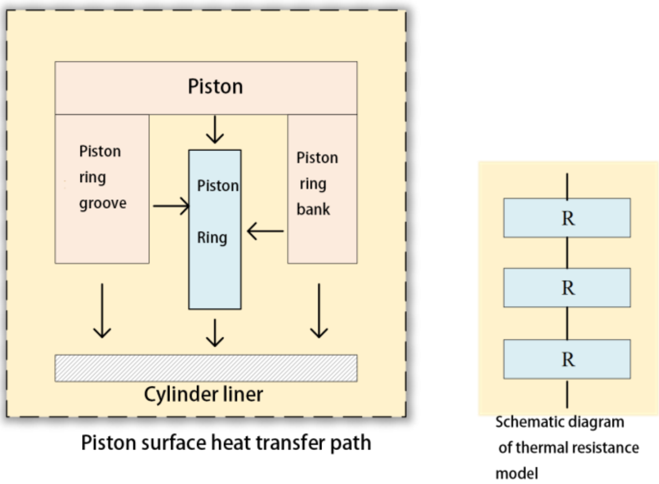


Figure 4: Heat Transfer Path and Thermal Resistance Model Structure on Piston Side Surface

Figure 4 shows that during the heat transfer, the heat from the piston passes through the lubricating oil, the mixture, and finally the cooling water through the piston ring to the cylinder liner. In heat conduction, lubricating oil, gas and piston ring can be regarded as the thermal resistance of heat conduction.

2) Improvement of multi-layer thermal resistance model

For the multilayer plate thermal resistance model, the heat transfer coefficient can be calculated by the thermal resistance, and the thermal resistance value of the multilayer plate heat transfer can be calculated by the following formula:

$$R_{all} = R_1 + R_2 + \cdots + R_n = \frac{\zeta_1}{\lambda_1} + \frac{\zeta_2}{\lambda_2} + \cdots + \frac{\zeta_n}{\lambda_n} \tag{5}$$

Where, ζ refers to the thickness of different heat transfer media, and λ refers to the thermal conductivity of the heat transfer medium.

Previous studies started from the heat conduction path, took the cooling water as the starting point, and used the multi-layer thermal resistance model to inversely derive the heat transfer boundary of the piston ring bank, ring groove and piston skirt [15]. However, due to the important influence of gas heat transfer on liner temperature, the existing water-based layered thermal resistance model does not consider the influence of gas heat transfer and spatial distribution characteristics on liner temperature. The results show that the water temperature and heat transfer coefficient of the cooling water room have obvious non-uniformity in space. Based on this, a new method to solve the heat transfer boundary of piston side wall is proposed. Based on the influence of gas heat transfer, cooling water heat transfer and structural heat transfer, the layered thermal resistance model is adopted to solve the heat transfer boundary of piston side wall, groove and piston side wall.

In addition, if the movement time of a cyclic piston is divided into equal parts, then the cylinder liner temperature corresponding to the same piston position at different times will be different. In Figure 5, the temperature correspondence of the ring bank cylinder liner is shown.

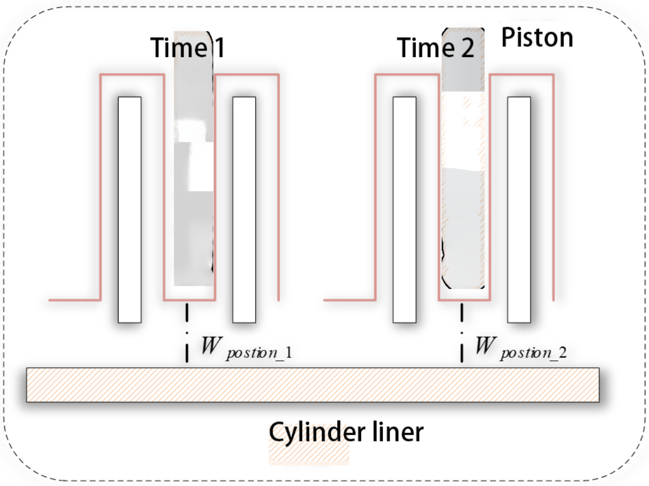


Figure 5: Illustration of temperature distribution on ring-land cylinder liner

Figure 5 shows that at different times, the corresponding position of the piston is not the same, the cylinder liner can be regarded as the track of the piston slide, in this track, the direction of movement of the piston is unique. In addition, under the influence of the crankshaft, the piston will change from up and down to rotating motion, and as the crankshaft rotates, the piston will also move up and down.

2.2 Lubrication calculation of piston group

Because there is a small air gap between the piston ring, piston skirt and cylinder jacket, and the small change of heat flux in the air gap will greatly affect the lubrication performance of the piston ring and the air gap. Therefore, the study intends to reveal the lubrication effect of temperature change in the air gap through numerical simulation of the lubrication performance of the piston ring and the piston skirt.

(1) Calculation model of piston skirt lubrication

In the cylinder liner connecting rod assembly structure, we first set the piston cylinder liner together by setting the contact model and friction coefficient and other indicators. Then, the boundary conditions such as cylinder liner temperature distribution, piston temperature distribution and thermal deformation of cylinder liner piston are substituted into the lubrication model of cylinder liner friction pair for numerical simulation. On this basis, a calculation method based on the lubrication theory of the plunger skirt is proposed, and the parameters required for calculation are shown in Table 1.

Table 1: Lubrication calculation parameter list

piston		Cylinder liner	
Parameter Name	Parameter value	Parameter Name	Parameter value
Skirt height /mm	867	Cylinder liner height /mm	279
Piston skirt roughness /μm	1.6	Cylinder liner roughness /μm	0.5
Piston Young's modulus /GPa	65	Cylinder liner Young's modulus /GPa	129
Piston Poisson ratio	0.26	Cylinder liner Poisson ratio	0.27
Piston density /(kg·m-3)	2692	Cylinder liner density /(kg·m-3)	7395
Piston thermal conductivity/(W·(m·K)-1)	158	Liner thermal conductivity /(W·(m·K)-1)	35
Piston specific heat /(J·(kg·K)-1)	892	Cylinder liner specific heat /(J·(kg·K)-1)	471

(2) Piston ring lubrication calculation model

In this model, three friction pairs, which are composed of two gas rings and one oil ring, are used and analyzed. In addition, we set 10 calculation cycles in the model, the first cycle is preprocessing, the purpose is to provide an initial value for the subsequent calculation, and each calculation result is iterated repeatedly, so that the expected calculation accuracy can be achieved. In addition, the study set a time step of 4 for each crank Angle.

In the calculation, click the option of cylinder liner deformation caused by gas load, and use the analysis method of thin cylindrical shell differential equation to calculate, the formula is as follows:

$$y+\frac{12\cdot y}{r^2\cdot s^2}=\frac{12P\cdot}{E\cdot s^3} \tag{6}$$

In the above formula, E stands for Young's modulus, r stands for cylinder liner radius, s stands for cylinder liner wall thickness, and P stands for gas pressure.

The boundary conditions of the pressure in the cylinder where the piston ring is located are similar to those used in the calculation of piston skirt lubrication. In the temperature distribution of the cylinder liner, the cylinder liner is divided into 20 different heights along the axis, and the average value of each height is taken as the cylinder liner temperature value, and then the new cylinder liner temperature field is calculated on the basis of the original cylinder temperature field, and it is input into the simulated cylinder liner. In the overall parameters of the model, the cylinder diameter is set to 125.5 mm, the crankshaft radius to 78.3 mm, and the connecting rod length to 248mm. Parameters related to piston rings are listed in Table 2.

Table 2: Piston ring parameters

Parameter Name	First ring	second ring	third ring
Mass /g	38.9	39.1	34.1
Center of gravity /mm	1.7	1.5	1.9
Young's modulus N/mm2	160000	90000	90000
Poisson's ratio	0.27	0.3	0.3

2.3 Research on frictional lubrication characteristics of cylinder liner piston system of internal combustion engine at different temperatures

(1) Experimental design

According to the piston skirt lubrication calculation model and piston ring lubrication calculation model designed in the previous section and the relevant parameters set, the simulation experiment of cylinder liner piston system is conducted by using Abaqus simulation software, so as to explore the influence of temperature on the frictional lubrication characteristics of cylinder liner piston system of internal combustion engine.

(2) Cylinder temperature distribution

In the experiment, since the temperature at the top of the turbine liner is mainly affected by gas combustion, and the cooling process in this part is not covered by cooling water, the temperature at the top of the turbine liner is taken as a fixed value, and the average value of the temperature at the top of the cylinder liner is taken. Table 3 shows the temperature distribution of the cylinder liner.

Table 3: Cylinder temperature distribution

Function type	Specific division	equation
Primary function	Primary function 1	$y = 463.2 - 0.35x$
	One time function 2	$y = 463.2 - 0.32x$
Parabolic function	Parabolic function 1	$y = 463.2 - 0.644x + 0.0012x^2$
	Parabolic function 2	$y = 463.2 - 0.019x - 0.00108x^2$
Sine function	Sine function 1	$y = 420.0 + 43.2sin[\pi(x + 134)/268]$
	Sine function 2	$y = 425.5 + 37.7sin[\pi(x + 137.1)/274.2]$

3. Experimental investigation results of frictional lubricity of cylinder liner piston system of internal combustion engine

3.1 Lubrication characteristics analysis of piston skirt at different temperatures

In the piston skirt part, the temperature field arranged in the upper section is used as the temperature boundary of the plunger end to calculate the lubrication of the plunger skirt part. The friction power consumption is analyzed in the results. The results are shown in Figure 6 and Table 4.

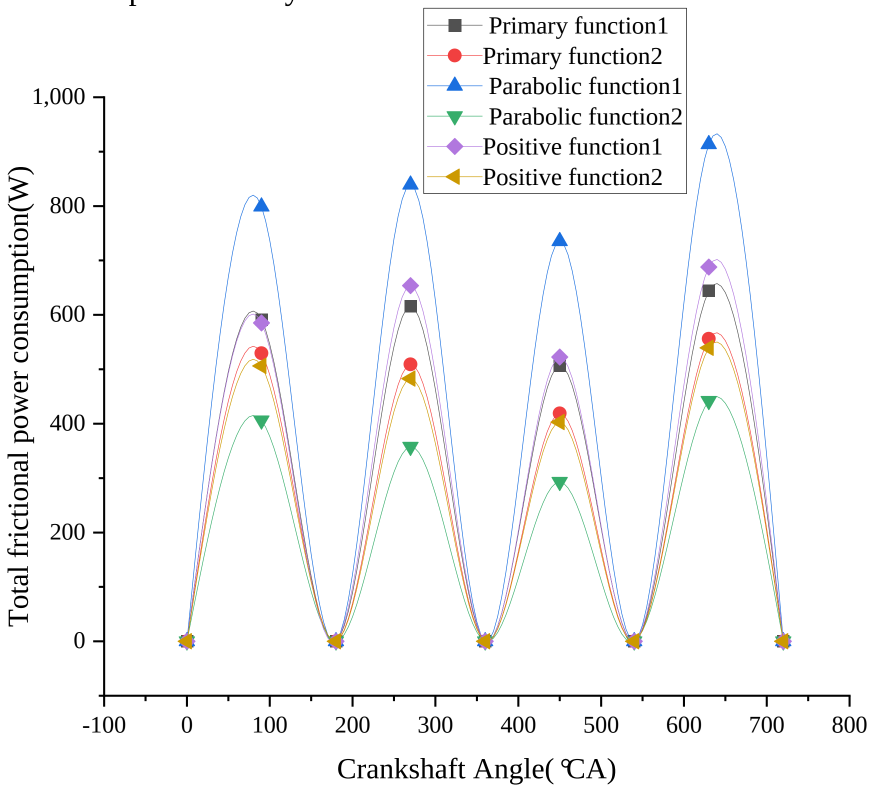


Figure 6: Total friction power consumption under different temperature distributions

Table 4: Friction power loss in piston skirt for different temperature distributions

Temperature distribution scheme	Average friction power consumption (W)	Absolute error	Relative error (%)
Primary function 1	295	14	5%
One time function 2	253	30	11%
Parabolic function 1	391	109	38%
Parabolic function 2	186	97	34%
Sine function 1	298	16	6%
Sine function 2	237	46	16.2%

Figure 6 and Table 4 show that primary function 1 and sine function 1 perform better, with lower average frictional power consumption and relatively small relative errors of 5% and 6% respectively. Parabolic function 1 and parabolic function 2 perform worse, with higher average frictional power consumption and larger relative errors (38% and 34%, respectively). The results show that the choice of temperature distribution scheme has an important effect on the lubrication performance of piston skirt.

3.2 Influence of cylinder liner temperature distribution on piston ring lubrication

Similarly, this experiment is to analyze the frictional lubrication characteristics of cylinder liner piston rings under different temperature distributions, and the experimental results are shown in Figure 7.

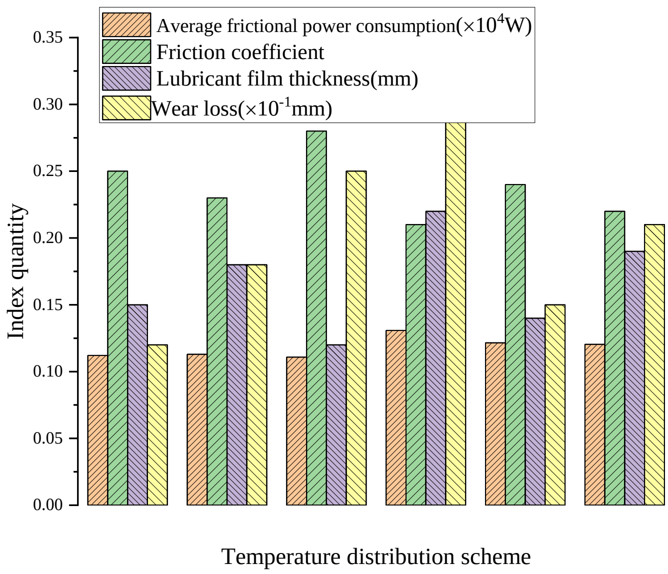


Figure 7: Analysis results of friction and lubrication characteristics of piston ring under different temperature distribution

Figure 7 shows that the average frictional power consumption of function 2 is slightly higher than that of function 1 in the first-order function scheme. In the parabolic function scheme, function 1 has a lower average frictional power consumption, while function 2 increases significantly. In the sinusoidal function scheme, the average frictional power consumption of functions 1 and 2 is relatively high but still acceptable. In terms of friction coefficient, the parabolic function scheme has higher friction coefficient, and the first order function and sine function scheme have lower friction coefficient. There are differences in lubricant film thickness and wear amount. For example, function 1 in the parabolic function scheme has lower lubricant film thickness but higher wear amount.

4. Conclusion and discussion

The research mainly analyzes the frictional lubrication characteristics of the cylinder liner piston system under different temperature distribution schemes by designing calculation models, in which the lubrication performance of the piston skirt and the piston ring are mainly investigated experimentally. The experimental results show that the different temperature distribution scheme will lead to some differences in the lubrication properties of piston skirt and piston ring. Primary function 1 and sine function 1 perform better in terms of frictional power consumption and error rate, while parabolic function 1 and parabolic function 2 perform worse. However, the above research has made some achievements in the analysis of friction and lubrication characteristics of cylinder liner piston system of internal combustion engine, but there are still some shortcomings. For example, the experimental data only contain a limited number of temperature distribution schemes, which cannot cover all possible cases, so the scope of temperature distribution schemes will be expanded in the future to better evaluate and analyze the frictional lubrication characteristics of the cylinder liner piston system of the internal combustion engine.

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