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Investigation of Pressure Distribution and Frictional Heat on Self-Lubricated Piston Rings in Reciprocating Compressors

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ABSTRACT

Piston ring sealing is of great importance in oil-free reciprocating compressors for high pressure application. Pressure distribution and frictional heat on piston rings have been investigated with the aim of finding the key factors influencing the life time of piston rings. A mathematical model simulating the unsteady flow within the gaps of piston rings was established to present the pressure distribution and disclose the mechanism of the uneven abrasion of different rings. Meanwhile, the test rig was built to validate the mathematic model, on which the pressure distributions between the piston rings were measured using pressure sensors. The results showed that the severe non-uniformity of the pressure distribution between piston rings was suggested to be the root cause of premature failure of the sealing rings. The pressure distributions between the rings were always dramatically non-uniform with the same ring cut, and the first ring afforded more than 75% of the total pressure difference. Then the method to equalize the pressure difference through each ring was proposed by re-distributing the cutting size of each ring, and it was also validated experimentally. Another factor that influenced the life time of piston ring was the high temperature caused by the frictional heat, so this paper also investigated the frictional heat generation and conduction between piston rings and cylinder by the Finite Element Method.

Key Words: Reciprocating Compressor, Piston Ring, Pressure Distribution, Frictional Heat

1. INTRODUCTION

Piston ring sealing was the key technology of high-pressure oil-free piston compressors. In order to seal high pressure difference, several piston rings were used on a piston. In ideal condition, the pressure difference on each ring should be the same, so that the frictions and wears on each ring were also similar and the life of the entire piston rings group could be the longest. However, the actual life of the rings was much shorter than that in ideal condition, and generally some certain rings were damaged firstly. The reason for this was that the loads on the certain rings were much greater, and this was caused by the non-uniform pressure distributions. Different authors did not make consistent conclusion about the pressure distributions between piston rings. Yu (2000) suggested that the pressure difference on the first piston ring was the largest, and then it decreased along the latter rings, and the first three rings afforded most of the pressure differences. Altukhov (1966) considered that, firstly, the pressure difference on the last ring was the least, so its wear was the least; and secondly, the sealing effect depended mainly on the number of piston rings on a piston, and not on the quality of the seal. Shishkin and Lando (1973) held that the pressure drops on the first two rings were commensurate with those on the last two rings, while the pressure drops on rings in the middle part of the piston seal were 20-30% of them. Milovanano and Budanov (1990) thought that the first ring bore most pressure difference in two rings sealing system. While Liu and Yu (1986) pointed out that in a compressor with three sealing piston rings the first and third rings were liable to wear, which was because the pressure differences on them were greater than the second one. Comparing these research literatures, the reason for different conclusions was found that there were great differences in their test conditions, especially, some investigations were conducted when the piston was static, which had essential difference with the actual condition. Therefore, it was necessary to

investigate the pressure distributions between piston rings under various pressure differences or ratios, which could provide the basis for predicting the life of piston rings.

Wherever friction occurs, mechanical energy is transformed into heat. The maximum surface temperature associated with this heating can have an important influence on the tribological behavior of the contacting components. There are few literatures concerning about the frictional heat between piston rings and cylinder in compressors, but Bai et al. (2005) investigated the frictional heat between piston rings and cylinder as well as its influence on the complete model simulation research on heat transfer of internal combustion engine. Bos and Moes (1995) described a numerical algorithm to solve the steady state heat partitioning and the associated flash temperatures for arbitrary shaped contacts by matching the surface temperatures of the two contacting solids at all points inside the contact area. While Vick and Furey (2001) developed another theoretical solution for the temperature rise due to sliding contact between surfaces with multiple, interacting asperities and to use this solution to examine the effects of the important contact area and system parameters.

A mathematic model to simulate the unsteady flow in the gaps of the piston rings was established in the paper, which could reveal the mechanism as well as the regulation of non-uniform pressure distribution. The experiment was conducted to validate the simulation, and the actual pressure distribution was obtained. The temperature distribution fields of the cylinder, piston and rings were also obtained by the FEM.

2. INVESTIGATION OF THE PRESSURE DISTRIBUTION

2.1 Theoretical Modeling

The flow in the leakage gaps of the piston rings can be seen as unsteady flow in several series expansion chambers, which reciprocate periodically. The mathematical model should be established to predict the gas flow in piston rings group and get the pressure distribution between them, which could reveal mechanism for quick invalidation of the piston rings in reciprocating compressors.

There are three possible paths of the gas leakage through the piston rings: gaps between rings and internal surface of the cylinder; gaps between rings and lateral face of the piston ring groove; cuts of the rings. In ideal condition, the piston rings attach the internal surface of the cylinder and lateral face of the piston ring groove closely, so the first two leakage paths are not considered in the simulation. The flow through the cuts of piston rings is presumed as one dimensional compressible isentropic flow through a convergent- divergent nozzle. The simplified model is shown in Fig. 1, where p_0 is the pressure in the compression chamber and also the pressure before the first ring; p_i is the pressure in the i th volume between rings, and also the pressure before the $i+1$ th ring; p_4 is the atmospheric pressure.

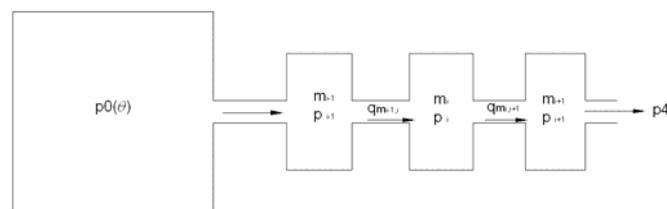


Fig. 1 Mathematical model of the pressure distribution

Therefore, the gas leakage through the cut of the i th piston ring can be written as

$$q_{m_i} = \alpha A_i \rho_{i+1} v = \alpha A_i \frac{p_{i+1}}{T_{i+1}} \sqrt{\frac{2\kappa}{R(\kappa-1)}} \times \sqrt{T_i \left[1 - \left(\frac{p_{i+1}}{p_i} \right)^{\frac{\kappa-1}{\kappa}} \right]} \quad (1)$$

where, α is the flow coefficient, κ is the isentropic exponent, R is the gas constant, and A_i represents the leakage area of the i th volume.

Generally, the maximum speed of the gas equals the sound speed, even though the pressure ratio exceeds the critical pressure ratio, the gas speed will not increase any more, so, the maximum leakage rate is

$$q_{m_{\max}} = \alpha A p_i \sqrt{\frac{1}{RT}} \sqrt{\kappa \left(\frac{2}{\kappa+1}\right)^{\frac{\kappa+1}{\kappa-1}}} \quad (2)$$

The gas leakage through rings depends on the pressure and temperature in the cylinder and the pressure distribution in the volumes between the rings. On the assumption that the pressure in the cylinder and volumes between rings is steady temporarily in a very short time, the flow through the rings can be seen as quasi-steady flow thus it is convenient to identify the relation between the pressure in the volumes and crank angle. In addition, some other assumptions are proposed: the working process in the cylinder is isentropic; kinetic energy is ignored in the volumes between the rings; all the geometry parameters of the piston rings keep constant, as well as the flow coefficient through the cuts of rings; the heat transfer between the gas in the volumes and all the contacting surfaces is negligible.

Each volume between the rings is considered as the control volume, of which the mass conservation equation and the energy conservation equation can be established, and these equations are the basis for building the mathematical model of the pressure distribution between the piston rings. Then the discussions are displayed according to different situations. When $p_{i-1} > p_i > p_{i+1}$, the gas flow direction is $m_{i-1} \rightarrow m_i \rightarrow m_{i+1}$, then the gas mass in the i th volume is

$$dq_{m_i} = (q_{m_{i-1}} - q_{m_{i+1}}) \frac{d\theta}{\omega} \quad (3)$$

and the pressure in the volumes can be written as

$$dp_i = \frac{RT_i}{V_i \omega} (q_{m_{i-1}} - q_{m_{i+1}}) d\theta \quad (4)$$

if the gas speed reaches the sound speed, the formula should be changed relatively. There are three other conditions, which include: $p_{i-1} < p_i < p_{i+1}$, $p_{i-1} < p_i < p_{i+1}$, $p_{i-1} > p_i < p_{i+1}$, and the control equations can be established respectively. For a group with four piston rings as shown in Fig. 1, there are three volumes between them, so these equations become simultaneous equations and it can be solved with the fourth-order Runge-Kutta method, of which the convergence condition is $f(0) \approx f(2\pi)$. From the mathematical simulation, the pressure in each volume between the rings could be calculated. Meanwhile, the gas leakage rate through each ring could be obtained. The pressure distribution could be improved through changing some design parameters, such as the numbers of rings, the size of ring cuts, or the volumes between rings. Therefore, the pressure distribution can be uniformed.

2.2 Experimental Setup

In order to get the dynamic pressure distributions between the piston rings, the piston rings must be reciprocated, and the pressure difference on them must change periodically, so the optimal method is modifying an existing reciprocating compressor, and it is crucial to record the dynamic pressure of gas in the volume between piston rings. Therefore, a reciprocating compressor was refitted, and special cylinder as well as piston was fabricated.

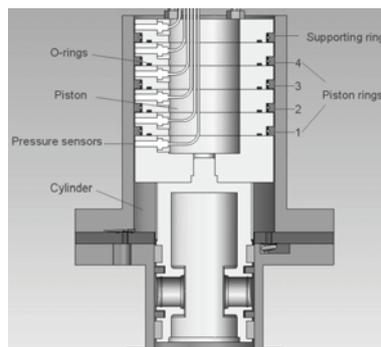


Fig. 2 Schematic diagram of the test device

The schematic diagram of test device is shown in Fig. 2. The piston was composed of several pieces of piston flake, which was annular and the external surface was ladderlike. Two piston flakes could compose a complete piston ring groove, and an o-ring was used for the sealing of them. The integrated piston composed by piston flakes was hollow, so the dynamic pressure sensors that measured pressure between piston rings could be installed into the piston and

the wire of sensors could be lead out from the inner of it. The piston rings group contained four piston rings as well as one support ring. The piston rings were defined as the first, second, third and fourth ring from high pressure to low pressure, which was shown in Fig. 2, and there was an angle difference of 180° between the cuts of two adjacent rings.

2.3 Results and Discussion on Pressure Distribution

The dynamic pressure distributions under various conditions were obtained, and the results showed that the pressure distributions were extremely non-uniform and the first piston ring afforded most of the pressure difference. All the pressure distributions measured coincided well with the calculated ones. When the suction pressure was 0.1 MPa and the pressure ratio was about 5, the test was carried out. The experimental results were shown as follows.

Figure 3 shows the measured pressure distribution as the cut is 3 mm, where p_0 , p_1 , p_2 , p_3 , p_4 respectively represented the pressure before the first, second, third, fourth ring and the supporting ring. In order to simulate the actual situation, the pressure measured in the cylinder was sent to the pressure before the first ring in simulation, which meant p_0 in the calculated results equaled to p_0 in the experimental results. So the calculated results comparing with the measured ones in Fig. 3 are shown in Fig. 4. As shown in Figures 3 and 4, most part (more than 90%) of the pressure difference was on the first ring. In the measured results, this phenomenon was even more obvious. The reason for this was that when the suction pressure was atmospheric pressure, the discharge pressure could not be very high. At the same time, the cuts of piston rings were small, so the first ring could provide good sealing for the low pressure. As a result, only very little gas leaked from the first ring, and that could not make the following rings work. Therefore, the first ring afforded almost all the loads. In this way, it wore heavily and would be invalidated fast. When the former ring was invalidated, the latter one began to take it over, and thus the rings were invalidated one by one, finally the entire piston rings group failed.

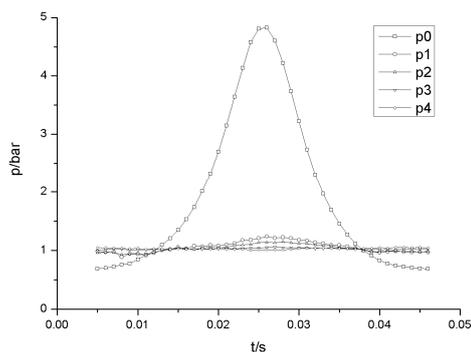


Fig. 3 Measured pressure distributions when the cut was 3 mm

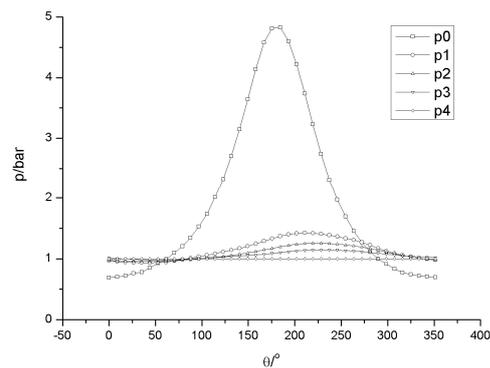


Fig. 4 Calculated pressure distributions when the cut was 3 mm

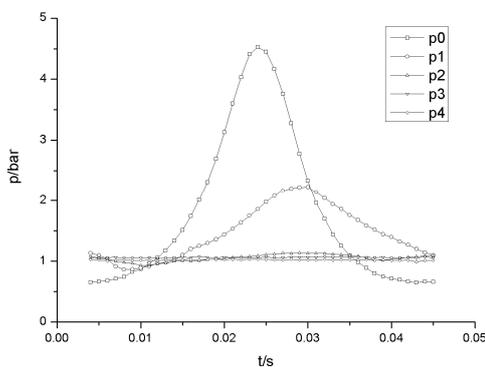


Fig. 5 Measured pressure distributions when the cuts were not the same

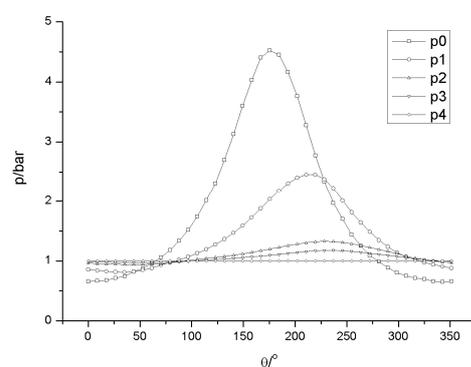


Fig. 6 Calculated pressure distributions when the cuts were not the same

In order to improve the pressure distribution, the rings with the cuts of different sizes could be arranged on the piston, which increased the gas leaking from the former rings and made the latter rings afford part of the pressure

difference. So the pressure difference is uniform as shown in Fig. 5. The rings' cuts were 3 mm, 2 mm, 1 mm, 1 mm from high pressure to low pressure. From Fig. 5, it was found that on this condition the first ring afforded about 60% of the total pressure difference, while the second one afforded about 40%, and the third ring also had the trend of affording pressure, but because the pressure before the second ring was low, the gas leaked from it could not make the third ring work efficiently. So it could be seen that this method had effect of pressure distribution uniformization, but had to be further improved.

Figure 6 shows the calculated results comparing with the measured ones in Fig. 5. Generally, the two were agreed well, but there were still some differences between them. In the calculated results, the third and fourth rings afforded more pressure difference than that in the measured ones. The reason for this was also that in the experiment, the gas leakage from the second ring was so little that the pressures before the third and fourth rings were too small to make them work. While in the simulation, it was not considered whether the pressure before the third or fourth ring could make it work, as long as there was pressure before it, it would work and afford the pressure difference.

3. INVESTIGATION OF THE FRICTIONAL HEAT CONDUCTION

3.1 Governing Equations of Heat Transfer

When the piston rings move with the piston, the friction appears between them and the cylinder. All the mechanical energy is supposed to transform into heat, so the frictional heat is

$$Q = fpv \tag{5}$$

Where, f is the frictional factor, p is the pressure on piston rings, v is the velocity of piston rings.

The frictional heat is distributed into the piston rings and cylinder, and the relationship between them is

$$\frac{Q_1}{Q_2} = \sqrt{\frac{\lambda_1 \rho_1 c_{p1}}{\lambda_2 \rho_2 c_{p2}}} \tag{6}$$

Where, the subscript 1 represents piston rings, and 2 is the cylinder, λ is the thermal conductivity, ρ represents the density, and c is the specific heat.

Then the heat conducts in piston rings and cylinder individually, and the differential equation for heat conduction is

$$\rho c \left(\frac{\partial T}{\partial t} + V_x \frac{\partial T}{\partial x} + V_y \frac{\partial T}{\partial y} + V_z \frac{\partial T}{\partial z} \right) = \ddot{q} + \frac{\partial}{\partial x} \left(\lambda_{xx} \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left(\lambda_{yy} \frac{\partial T}{\partial y} \right) + \frac{\partial}{\partial z} \left(\lambda_{zz} \frac{\partial T}{\partial z} \right) \tag{7}$$

Where, T is the temperature and t is the time. V is the velocity for mass transport of heat, and \ddot{q} is heat generation rate per unit volume.

Integrated over the volume of an element, Equation (7) could transform to the equivalent integral form:

$$\begin{aligned} & \int_{vol} \left(\rho c \delta T \left(\frac{\partial T}{\partial t} + \{V\}^T \{L\} T \right) + \{L\}^T \delta T ([D] \{L\} T) \right) d(vol) \\ & = \int_{S_2} \delta T q^* d(S_2) + \int_{S_3} \delta T h_f (T_B - T) d(S_3) + \int_{vol} \delta T \ddot{q} d(vol) \end{aligned} \tag{8}$$

Where, vol is volume of the element, δT is an allowable virtual temperature, $\{V\}$ is the velocity vector for mass transport of heat, and $\{L\}$ is the vector operator, $[D]$ is the conductivity matrix,

$$\{V\} = \begin{Bmatrix} V_x \\ V_x \\ V_x \end{Bmatrix} \quad \{L\} = \begin{Bmatrix} \frac{\partial}{\partial x} \\ \frac{\partial}{\partial y} \\ \frac{\partial}{\partial z} \end{Bmatrix} \quad [D] = \begin{bmatrix} \lambda_{xx} & 0 & 0 \\ 0 & \lambda_{yy} & 0 \\ 0 & 0 & \lambda_{zz} \end{bmatrix}$$

S_2 is the surface specified heat flows acting over and S_3 is the surface specified convection acting over, q^* is the specified heat flow, h_f is the film coefficient, and T_B is bulk temperature of the adjacent fluid.

The variable T is allowed to vary in both space and time, so this dependency is separated as:

$$T = \{N\}^T \{T_e\} \tag{9}$$

Where {N} is the element shape functions and {Te} is the nodal temperature vector of element. Supposed [B] = {L}^T{N}, so Equation (8) may be reduced to:

$$\begin{aligned} & \rho \int_{vol} c \{N\} \{N\}^T d(vol) \{T_e\} + \rho \int_{vol} c \{N\} \{V\}^T [B] d(vol) \{T_e\} + \rho \int_{vol} [B]^T [D] [B] d(vol) \{T_e\} \\ & = \int_{S_2} \{N\} q^* d(S_2) + \int_{S_3} T_B h_f \{N\} d(S_3) - \int_{S_3} h_f \{N\} \{N\}^T \{T_e\} d(S_3) + \int_{vol} q \{\ddot{N}\} d(vol) \end{aligned} \tag{10}$$

Equation (10) may be rewritten in matrix form as:

$$[C_e^t] \{\dot{T}_e\} + ([K_e^{tm}] + [K_e^{tb}] + [K_e^{tc}]) \{T_e\} = \{Q_e^f\} + \{Q_e^c\} + \{Q_e^g\} \tag{11}$$

Where:

$$[C_e^t] = \rho \int_{vol} c \{N\} \{N\}^T d(vol) \text{ -----element specific heat (thermal damping) matrix}$$

$$[K_e^{tm}] = \rho \int_{vol} c \{N\} \{V\}^T [B] d(vol) \text{ -----element mass transport conductivity matrix}$$

$$[K_e^{tb}] = \int_{vol} [B]^T [D] [B] d(vol) \text{ -----element diffusion conductivity matrix}$$

$$[K_e^{tc}] = \int_{S_3} h_f \{N\} \{N\}^T d(S_3) \text{ -----element convection surface conductivity matrix}$$

$$\{Q_e^f\} = \int_{S_2} \{N\} q^* d(S_2) \text{ -----element mass flux vector}$$

$$\{Q_e^c\} = \int_{S_3} T_B h_f \{N\} d(S_3) \text{ -----element convection surface heat flow vector}$$

$$\{Q_e^g\} = \int_{vol} \ddot{q} \{N\} d(vol) \text{ -----element heat generation load}$$

So the general equation could be written in matrix form as:

$$[C] \{\dot{T}\} + [K] \{T\} = \{Q\} \tag{12}$$

3.2 Finite Element Model

Using the Finite Element Method (FEM), a 2-dimension model for piston rings and cylinder was established as shown in Fig. 7. It was half of the cross area of cylinder, piston and piston rings. There were 4 piston rings and the piston rings contacted with the inner surface of the cylinder, and reciprocated with the piston.

Natural convection existed between the outer surface of the cylinder and the surroundings, as well as between the inner surface of the cylinder and the high-temperature compressed gas. While forced convection occurred between the high-temperature compressed gas and the piston as well as the rings. There were heat conduction between the cylinder and piston rings, and between piston rings and piston. Pressure was loaded on each piston ring.

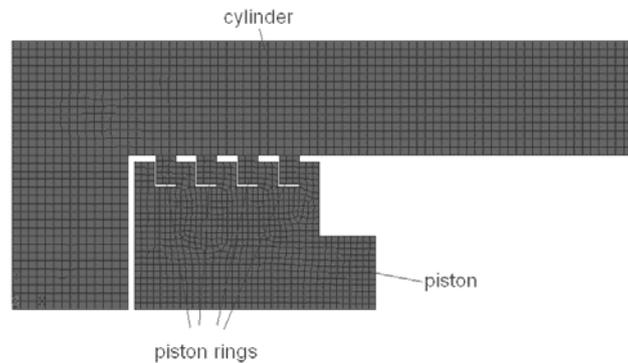


Fig. 7 Finite element model for piston rings and cylinder

3.3 Results and Discussion

The temperature distribution field of the cylinder and piston rings could be obtained by the FEM. Fig. 8 shows the integral temperature field, while Fig. 9 and 10 show the temperature fields of the cylinder and piston respectively.

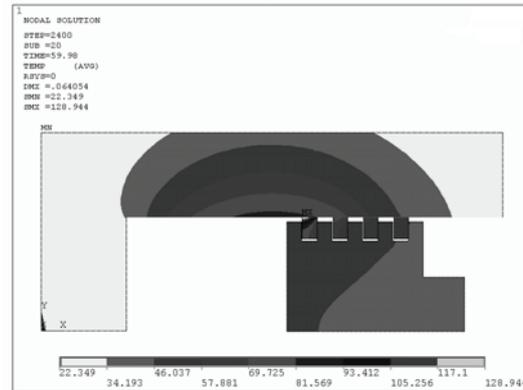


Fig. 8 Temperature field of the cylinder, piston and rings

As shown in Fig. 8, the highest temperature occurred on the frictional surface of the piston ring near high-temperature gas, which was called the first ring. The reason for this was that most of the pressure difference was loaded on the first ring, as analyzed in Section 2. So compared with other rings, it was more possible that the first ring was damaged by high temperature. Considering that it also afforded the most pressure difference, the first ring was the easiest to fail. The highest temperature of other rings occurred not on the frictional surface, but on the surface near the high-temperature gas. It demonstrated that the temperature rise of the last three rings was mainly caused by the high-temperature gas that leaked through the rings.

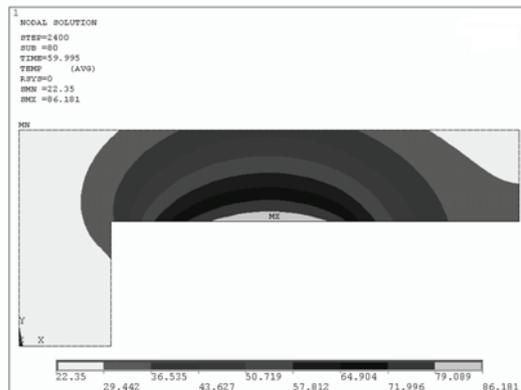


Fig. 9 Temperature field of the cylinder

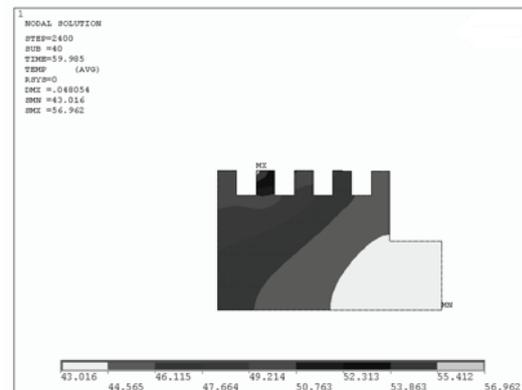


Fig. 10 Temperature field of the piston

As shown in Fig. 9, the highest temperature of cylinder was on the area that the first ring slid over. Because of the good conductivity of the cylinder's material, heat transferred from the frictional surface to the outer one quickly and then convected with surroundings. The highest temperature of piston could be seen on the lateral face of the first piston ring groove in Fig. 10, which contacted with the first ring. This temperature rise was caused by two factors, one was the heat conducted from the first ring, and the other was heat convected from the high-temperature gas.

Through varying the parameters, the important factors that influenced the temperature field could be obtained. For instance, the cooling conditions outside the cylinder could vary by changing the film coefficient between the cylinder and surroundings, so the temperatures under different cooling conditions could be investigated. Also the temperature with different piston ring materials could be calculated and compared. So the method to enhance the heat transfer and reduce the temperature could be presented. This work is undergoing, as well as the experimental verification.

4. CONCLUSIONS

A mathematic model for simulating the unsteady flow in the gaps of the piston rings had been developed, and the mechanism as well as the regulation of non-uniform pressure distribution was revealed. Experiment was carried out to validate the mathematic model, and the experimental results showed good agreement with the calculated ones. The temperature distribution fields of the cylinder, piston and rings were investigated by the FEM. There are some conclusions:

- In the working process of the piston rings group, the first ring afforded most part of the total pressure differences, so it wore severely and was invalidated quickly, which affected the life of entire piston rings group; Using the piston rings with cuts of different sizes could improve the pressure distributions efficiently, which could make them uniform and prolong the life of entire piston rings group.
- The highest temperature occurred on the frictional surface of the first piston ring because of the most pressure difference it subjected to, while the temperature rise of the other rings was mainly caused by the high-temperature gas that leaked through the rings; the method to enhance the heat transfer and reduce the temperature could be obtained and further investigation is been doing.

NOMENCLATURE

A	area	(m ²)
c	specific heat	(J/(kg·K))
f	friction factor	(-)
h_f	film coefficient	(W/(m ² ·K))
p	pressure	(MPa)
Q	frictional heat	(W/m ²)
q_*	leakage flow rate	(kg/s)
q	specified heat flow	(W/m ²)
\ddot{q}	heat generation rate	(W/m ³)
R	gas constant	(J/(kg·K))
T	temperature	(K)
t	time	(s)
v	velocity	(m/s)
α	flow coefficient	(-)
κ	isentropic exponent	(-)
λ	thermal conductivity	(W/(m·K))
ρ	density	(kg/m ³)

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