Influence of piston-bore clearance on second motion characteristics of piston and skirt wear

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Abstract. The piston dynamic simulation was carried out for the wear of the piston skirt after the test of gasoline engine. In addition, the second motion and wear are studied for the improvement proposal of different cylinder clearances. Results show when the original cylinder clearance is 0.0325–0.0575 mm, the minimum operating clearance of the piston skirt is smaller; it is not good for the formation of lubrication oil film, which causes wear on the thrust side and anti-thrust side of piston skirt. When the cylinder clearance was increased to 0.0400–0.0650 mm, the wear load on the thrust side and anti-thrust side of piston skirt was reduced by 68.3% and 68.1%, respectively, the effect of improved wear is better and the result coincides well with engine test result.

Keywords: Piston-bore clearance / piston dynamics / second motion / wear load

1 Introduction

A piston of an internal combustion engine moves up and down inside the engine cylinder under the action of gas pressure, inertia force, side pressure and so on. Apart from this, piston also oscillates and moves laterally which is called the second motion. Compared with reciprocating linear motion, second motion is very tiny but has an essential impact on the wear of piston [1–7]. There are various factors influencing the mechanism of piston wear, such as piston-bore clearance, pin hole eccentricity, crankshaft eccentricity, center of mass of piston, piston deformation and design of shape line and so on [8–13]. To be more exact, crankshaft eccentricity affects the relative swing of the skirt and the head at the moment of changing over during reciprocating movement of the piston; piston deformation usually influences the contact between the piston and the related parts once engine starts working; piston-bore clearance not only affects the amplitude of the second motion, but also plays a key role in the friction and the wear between piston and cylinder [14–19].

This paper aims at studies on the mechanism of piston wear. The piston test of gasoline engine was carried out and the wear condition of the skirt after the test was then analyzed. By using the dynamic analysis and simulation software for pistons “Pisdyn” of Ricardo Company, dynamic calculations were carried out with different piston-bore clearances. With the dynamic simulation calculation of different cylinder clearance conditions, the optimized scheme as well as the influence law of piston-bore clearance on second motion characteristics and on the wear of gasoline engine were finally obtained.

2 Governing equation of piston secondary motion

The force of piston when moving in cylinder is shown in Figure 1. In this figure, the cylinder axis is defined as X-axis with the direction toward the top of the cylinder positive. Y-axis is defined as the direction perpendicular to both the pin hole axis and the direction pointing to the secondary thrust side is positive. With hypothesis that the gas pressure in cylinder is $F_{\text{gas}}$ and its action line passes through the center line of the cylinder, the gas pressure will produce the torque $M_{\text{gas}}$ that revolves the piston around the piston pin in the case of a piston pin hole bias.

Piston is considered to move only in the plan of the main thrust side and the subthrust side. According to the principle of dynamic force balance when the piston moves in the cylinder, the dynamic equation of piston could be obtained as follows:

In the direction of X

$$m_{e} \ddot{x}_{e} = \sum F_x = \sum_i (F_{cx_i} - F_{g cx} - F_{g as} - F_{r x} + F_{p incx} - F_{linkx})$$

(1)
In the direction of Y:

\[ m_c \ddot{y}_c = \sum_i F_{cy} - F_{gc_y} - F_{ry} + F_{pin_c_y} - F_{link_y} \]  

Revolve around the piston pin axis:

\[ \Theta_c \ddot{K}_c = \sum_i M = M_c + M_{ge} + M_{gas} + M_r + M_{pc} \]  

In the formulas above, \( m_c \) is the piston mass, \( \Theta_c \) is the moment of inertia of piston around piston pin axis, \( F_{gc} \) is the mass force of piston, \( F_{gas} \) is the gas pressure in cylinder, \( F_r \) is the contact force between piston ring and piston, \( F_{ci} \) is the contact force in cross section \( i \) of the main thrust side and the subthrust side, \( F_{pin} \) is the applied force on piston pin, \( F_{link} \) is the applied force on the little end of connecting rod, \( M_{pc} \) is the friction moment of piston and piston pin.

When piston does linear reciprocating motion in cylinder bore, movement of sliding along the wall could lead to shear effect. Moreover, extrusion effect is supposed to be produced at both the TDC and BDC during the transverse movement of piston. And all these appear to be the fluid lubrication characteristic similar to a sliding bearing. After taking the influence of roughness and waviness between piston skirt and the lubricating surface of cylinder wall into consideration, the 2-D Average Reynolds Equation for fluid lubrication on rough contact surface between cylinder bore and piston, modified according to Patir and Cheng, was applied and described in formula below:

\[
\frac{\partial}{\partial x} \left( \Phi_x h^3 \frac{\partial p_h}{\partial x} \right) + \frac{\partial}{\partial y} \left( \Phi_y h^3 \frac{\partial p_h}{\partial y} \right) = 6 \mu U \left( \frac{\partial \Xi_T}{\partial y} + \Phi_x \frac{\partial \Phi_x}{\partial y} \right) + 12 \mu \frac{\partial \Xi_T}{\partial t}
\]

where \( p_h \) is the average fluid pressure, \( t \) is the time, \( h \) is the nominal film thickness, \( \Xi_T \) is the average of actual film thickness, \( \mu \) is the lubricating oil viscosity, \( U \) is the velocity of piston, \( \Phi_x, \Phi_y \) are the pressure flow factors, \( s \) is the root mean square value of surface roughness between cylinder bore and piston.

3 The main boundary conditions for the calculation of piston dynamics

3.1 The basic parameters of the engine

This paper mainly takes the piston of a four cylinder gasoline engine as the object of study and the main parameters of this engine are shown in Table 1.

3.2 Thermal deformation of piston skirt

Piston contacts directly with high temperature gas under working conditions. As a result, the temperature gradient could be produced inside the piston result in the thermal deformation which could be calculated in finite element simulation. In order to ensure the accuracy of the simulation calculation, the temperature of the piston should be tested by the hardness plug before simulation. The test values obtained were then employed in calibrating and validating the simulation of temperature field. The arrangement of the test points of the piston temperature test is shown in Figure 2. In this figure, 1 is the central position of piston combustion chamber, 2–5 represents the surrounding location of combustion chamber, 6–9 is the position of fire shore and 10–11 is the medial side of pin hole.

| Items                  | Parameters |
|-----------------------|------------|
| Number of cylinders   | 4          |
| Bore (mm)             | 76.5       |
| Stroke (mm)           | 90         |
| Calibrated power (kW) | 82         |
| Calibrated speed (r min\(^{-1}\)) | 5500   |
The test results of the piston hardness plug of each cylinder as well as the test results of the average temperature of each test point are shown in Figure 3. It can be seen from the test results that the highest temperature of the piston is 241°C which appears at the central position of combustion chamber. The temperature of both surrounding location and fire shore is 211°C approximately and that of medial side of pin hole is 185°C.

The temperature field of the piston and the heat deformation of the skirt are calculated by the iterative calculation of numerical simulation by ANSYS software which are shown as in Figures 4 and 5. It is obvious that the thermal deformation of the top end of the skirt is 0.132 mm and that of the lowest end of the skirt is 0.092 mm. In addition, the gradient of thermal deformation increases when closing to the top end of the skirt. Similarly, the gradient of thermal deformation decreases when closing to the lowest end of the skirt.

### 3.3 Piston stiffness characteristics

Piston stiffness is used to indicate the resistance to elastic deformation when a certain part of piston is under the action of external load. And the stiffness of the piston skirt is a key factor that should be taken into consideration in kinetic analysis. The stiffness of piston could be quantified by the amount of elastic deformation. Figure 6 illustrates
the elastic deformation obtained by finite element calculation in the direction of 0 °C of piston. As per the figure, skirt stiffness declines from top to bottom and a better flexibility could be found in location below the pin hole which is an advantage for reducing the percussion of the piston in changing-over.

4 Dynamic calculation results and test verification

4.1 Calculation scheme

The initial piston-bore clearance in this investigation was case 1; cylinder clearance is 0.0325–0.0575 mm. As the skirt wear appeared after the loading test, the piston-bore clearance was optimized as shown in Table 2.

According to the boundary conditions obtained, the piston hydrodynamic lubrication model was established combining to the geometric and numerical model parameters of piston, cylinder liner, link, piston pin and so on. Relative calculations were also carried out for three cases under condition of full speed and full load.

4.2 Results

4.2.1 Piston motion angle and radial displacement

Figure 7 shows the swing angle of the piston around the piston pin received by calculations while Figure 8 demonstrates the radial displacement curves of piston in which 0 ° is the top dead center. It can be seen that when crankshaft angle was 12°, the maximum swing angle of piston in case 1 was 0.112°; the piston swing angle in case 2 increased by 58.9% compared to that in case 1 and reached the maximum which was 0.178°; the swing angle in case 3 was 0.133°. As per the graph 8, when crankshaft angle was 128°, the maximum of radial displacement of piston in case 1 was 0.039 mm; the radial displacement of piston in case 2 rose by 33.3% compared to that in case 1 and arrived the maximum which was 0.052 mm; the maximum of radial displacement in case 3 was 0.043 mm. Therefore, the swing angle of second motion as well as the radial displacement of piston increase with the piston-bore clearance and the rise is extremely obvious.

4.2.2 Piston knocking kinetic energy

Piston knocking kinetic energy is an important parameter used to evaluate the engine NVH indirectly. Indeed, the peak values as well as the total kinetic energy in the cycle of the knocking kinetic energy of the piston to the cylinder liner are all expected to be weak. Total kinetic energy is composed of axial motion kinetic energy of piston $E_a$, radial motion kinetic energy $E_t$ and oscillating kinetic energy $E_r$ and is expressed in following equations:

$$ E = E_a + E_t + E_r $$

$$ E_a = \frac{Mv^2}{2} $$

$$ E_t = \frac{Mu^2}{2} $$

$$ E_r = \frac{Mw^2}{2} $$

where $M$ is the weight of piston assembly, $v$ is the axial velocity, $u$ is the radial velocity and $w$ is the swing angle of piston.

Figure 9 shows the knocking kinetic energy curves of piston. The peak value of knocking kinetic energy of piston

| Case   | Large point diameter (mm) | Cylinder clearance (mm) |
|--------|---------------------------|-------------------------|
| Case 1 | 74.465 ± 0.0075           | 0.0325–0.0575           |
| Case 2 | 74.440 ± 0.009            | 0.0560–0.0840           |
| Case 3 | 74.4575 ± 0.0075          | 0.0400–0.0650           |
in case 1 is 8.0 N mm; the peak value of knocking kinetic energy of piston in case 2 is 22.3 N mm, increasing by 1.8 times compared with that in case 1; the peak value of knocking kinetic energy also rises and reaches 11.4 N mm. Apparently, the increase of piston-bore clearance results in the obvious increase of piston knock which is severe for the gasoline engine noise.

4.2.3 Contact and wear of piston skirt

The wear condition of the piston skirt surface is another important index to assess the rationality of piston design. The wear load can be used to express the wear of piston surface. The value of wear load is supposed to be affected not only by contact pressure but also by the running speed of the piston in the cylinder liner which could be indicated by the product of the contact pressure and the running speed of piston \( V \) expressed as follow.

\[
W = P_c \cdot V \quad (9)
\]

In this formula, \( W \) is the wear load in N m/s or W/m², \( P_c \) is the contact pressure and \( V \) is the running speed of piston including the first-order motion and the velocity of second motion.

The final wear of piston skirt is actually formed by a cyclic accumulation process. While the engine was tested under condition of full speed and full load, a vertical score on the inside of the hole of the cylinder was found through endoscopy during the stop routine check of engine. The score is shown in Figure 10.

After the disassembly of the engine, there was obvious wear found on the main and the secondary thrust sides of the four pistons in case 1 as shown in Figure 11.

It was obvious that there were continuous and varying different degrees of wear from top to bottom on the main and the secondary thrust sides of piston skirt. By detecting the contour of the line and the knife depth, it was proved that the knife depth at the upper end of skirt changed from 0.010–0.011 mm before test to 0.002–0.003 mm; similarly, the knife depth at the lower end of skirt changed from 0.012–0.013 mm before test to 0.008–0.010 mm; coatings and knife depth in other wear zones were 0 and the wear depth was between 0.015 and 0.024 mm.

When the piston moves in the cylinder liner, the clearance between them varies all the time. Figure 12 illustrates the minimum clearance of skirt in three cases obtained by dynamic simulation calculation of piston. It can be seen from the chart that the minimum operating clearance of the piston in case 1 is about 4.6 \( \mu \text{m} \) in which situation the oil film lubrication cannot be well formed. By contrast, the cold state clearance was enlarged in cases 2 and 3 so as to enlarge the minimum clearance of skirt which facilitates the formation of lubricating oil film of piston skirt.

The calculation result of cumulative wear load of piston skirt in case 1 is shown in Figure 13. As per the graph, the wear load of the main and the secondary thrust sides of piston are 88.589 MW/m² and 80.268 MW/m², respectively. It is predicted that obvious wear is more likely to appear
in the middle part of the skirt on both thrust sides which is confirmed by the piston wear condition obtained in case 1 in the process of engine bench test and shown in Figure 11.

Figures 14 and 15 show the calculation results on cumulative wear load of piston skirt in case 2 and case 3, respectively. Actually, the results of the main and the secondary thrust sides in case 2 decreased by 63.9% and 57.7% respectively compared with that in case 1; similarly, the results in case 3 decreased by 68.3% and 68.1% correspondingly. The exact values of wear load is shown as in Table 3.

Therefore, the wear load of skirt would be significantly reduced after the cold state clearance was enlarged. Indeed, the clearance of the piston in the cylinder liner is more reasonable in preventing the excessive pressure destroying the lubricating oil film. As a result, the liquid dynamic lubrication of piston skirt is favorable.

After the enlargement of the piston-bore clearance in both cases 2 and 3, the wear loads of skirt were all reduced so as to reduce the risk of skirt wear. According to a series of analysis, although the increase of piston clearance in case 2 is more important than that in case 3, the reduction of the contact pressure as well as the wear load is not obvious. However, the amplitude of the second motion and the knocking kinetic energy increased greatly. Therefore, case 2 is considered to be the best improving scheme and this evaluation was further verified in the second engine bench test. Figure 16 shows the fitting results of assembled piston in case 3 after running under full speed and load. Apparently, the piston skirt fits well after the test and no wear was found.

5 Conclusion

The piston-bore clearance is one of the most important factors affecting the wear characteristics. Through research, conclusions are as follows:

1. Increasing clearance between piston and cylinder liner results in the increase of the swing angle, the radial displacement and the knocking kinetic energy of piston second motion. Thus, a reasonable clearance could ensure the smooth performance of piston motion and reduce the knocking of the piston to the cylinder liner.

2. According to the piston dynamics calculation, the initial clearance in case 1 is small resulting in a tiny oil film gap and a greater wear load. After the enlargement of the piston-bore clearance in case 3, the wear load of the main and the secondary thrust sides in case 3 decreased by 68.3% and 68.1% respectively compared with that in case 1 which greatly improves the wear of the piston skirt. This result is further verified in the engine bench test.

3. The optimum selection of clearance between piston and cylinder liner should be determined taking consideration of the smooth performance of second motion, the wear

\[
\text{Table 3. Wear load result.}
\]

| Items | Main thrust side | Secondary thrust side |
|-------|-----------------|-----------------------|
| Case 1 | 88.589          | 80.268                |
| Case 2 | 24.622          | 22.582                |
| Case 3 | 28.122          | 25.582                |

Fig. 13. Wear load on the main and secondary thrust side of the piston.

Fig. 14. Wear load on the main and secondary thrust side of the piston.

Fig. 15. Wear load on the main and secondary thrust side of the piston.

Fig. 16. Piston pictures after the test.
and tear, the scoring, the knock and so on. At the same time, good hydrodynamic lubrication of piston skirt should also be guaranteed as well as other performance requirements.

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