

Friction Analysis by Piston Secondary Motion Simulation in Consideration of Cylinder Block Bore Distortion

Kenji SATO*
Makoto ITO**

Kinya FUJII*
Shinsuke KODA*

ABSTRACT

Piston skirt friction has been considered to be related to cylinder block bore distortion and piston secondary motion, but it has been studied almost exclusively in isolation, and its detailed relationship to the functioning of an actual engine has not been clarified. We used computer simulations to reproduce cylinder block bore distortion and piston secondary motion in combination to try to increase our understanding of their relationship with an actual engine. First, calculated results were compared with measured values to ascertain their validity. Next, the technique was applied to the development of an actual engine, and the engine performance was studied along with the establishment of a method to predict friction. The application of the new simulation technique in engine development resulted in increased reliability and a reduction in friction of approximately 2%, demonstrating the effectiveness of the method.

1. Introduction

Reduction of friction is known to be one basic means of increasing engine fuel economy. In particular, reducing friction in the piston area, where about 25% of engine friction occurs⁽¹⁾, is considered an issue. Factors connected with piston friction include distortion of the cylinder block bore and sliding between the liner and the piston skirt due to the behavior of the piston itself. Recently, there have been reports⁽²⁾⁻⁽⁴⁾ of cases in which computer simulation was used to analyze these factors, and such analysis is also being used in engine development. Accordingly, this research was aimed at reconfirming the accuracy of these simulations and at raising the prediction accuracy by combining cylinder block bore distortion and piston secondary motion. This prediction technology was used in actual engine development, and the results obtained are reported here.

2. Research Aims

Cylinder block bore distortion and piston secondary motion have been investigated before, but these investigations have almost always been carried out separately, so it was not possible to clarify the correlation between specification changes and the detailed performance of an actual engine. Also, the piston is required to have various performance specifications during development, so a method that permitted an overall judgment to be made was needed.

Therefore, computer simulation was used to reproduce cylinder block bore distortion and piston secondary motion in combination. Prediction technology was also developed to permit more accurate judgment of friction reduction and seizing reliability when the shape in the vicinity of the piston skirt was changed.

* Tochigi R&D Center

** SANNO TEC Co., Ltd.

3. Calculation Method

3.1. Calculation Flow

Figure 1 shows the calculation flow. ABAQUS was used to perform structural analysis of the distortion and thermal distribution of the cylinder block, and STAR-CD was used to perform computational fluid dynamics⁽⁵⁾ of the thermal heat transfer coefficient distribution in the water jacket to raise the accuracy of the thermal analysis. PISDYN was used to analyze the 3-dimensional behavior of the piston.

3.2. Cylinder Block Distortion Analysis

Figure 2 shows the FEM model used for the cylinder block thermal distribution and distortion analysis. The thermal and distortion analyses are conducted independently of each other. By correlating the results of the thermal analysis to the process following distortion analysis, the distortion when the engine is hot is reproduced.

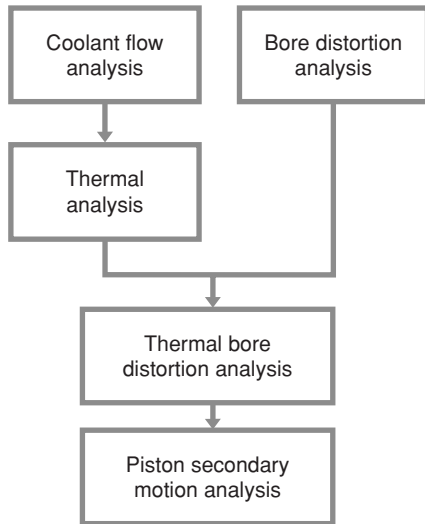


Fig. 1 Calculation flow

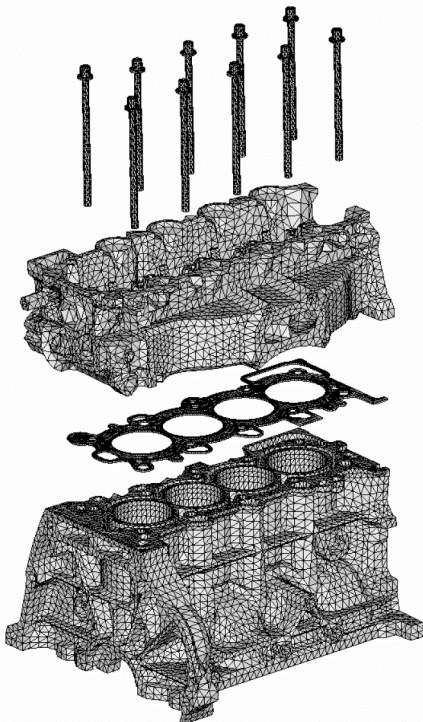


Fig. 2 Cylinder block FEM model

The main input conditions for thermal analysis are the thermal conductivity of each member, the heat transfer coefficient of the surface of the member, and the surrounding temperature. The heat transfer coefficient of the coolant is defined by referring the heat transfer coefficient obtained by computational fluid dynamics to the structural model, as mentioned above. Figure 3 shows the distribution of the heat transfer coefficient obtained from computational fluid dynamics. Regarding other areas, including the liner section, it is necessary to set conditions using measured values or experience, and at present it is desirable to compare these conditions with the measured temperature, and correct them accordingly.

The main input conditions for distortion analysis are Young's modulus, the coefficient of thermal expansion, the assembly bolt force of each member and the spring characteristics of the cylinder head gasket. Regarding Young's modulus and the coefficient of thermal expansion, temperature dependence data is used, while regarding the non-linear spring characteristics of the cylinder head gasket, individual measurement test results are used. These characteristics are shown in Fig. 4.

Figure 5 shows a contour map of the distorted cylinder block when hot.

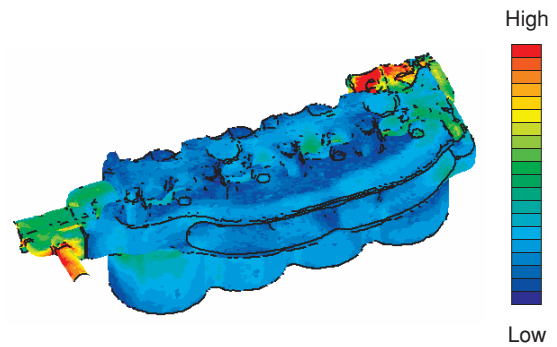


Fig. 3 Distribution of heat transfer coefficient

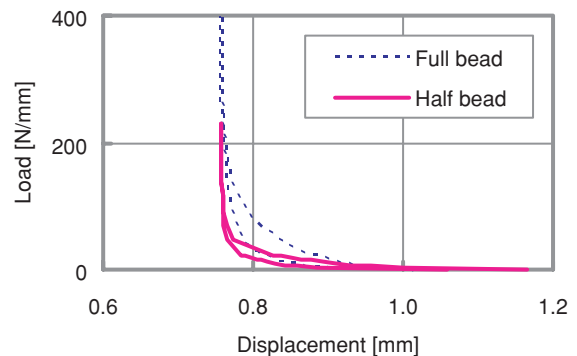


Fig. 4 Gasket characteristic

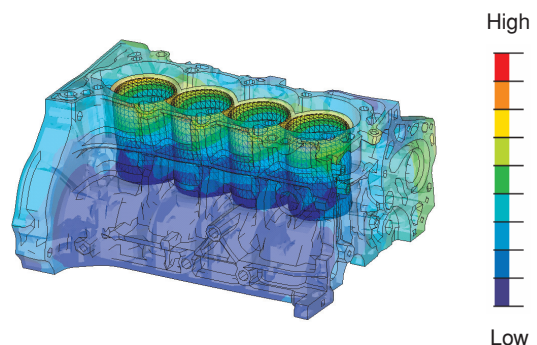


Fig. 5 Cylinder block bore distortion

3.3. 3-dimensional Piston Secondary Motion Analysis

Figure 6 shows an FEM model of a piston. Piston secondary motion analysis is performed using half a cylinder in the program. For the FEM, the piston and cylinder block are prepared, and reduced data is used. Piston temperature is calculated, and a piston model is created, taking into account thermal expansion due to the temperature distribution.

The results of the cylinder block bore distortion analysis are input as numerical data. Oil film characteristics that take into account elasto-hydro dynamics (EHD) are used for the sliding faces of the piston skirt and the liner. The surface roughness R_q is measured and input in order to take into consideration the boundary lubrication condition due to the surface roughness.

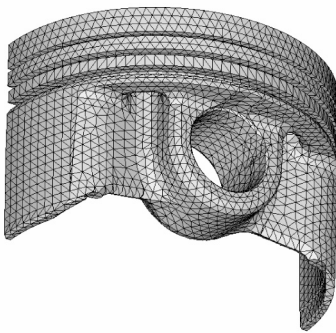


Fig. 6 Piston FEM model

4. Analysis Results and Measurement Results

4.1. Cylinder Block Bore Distortion

Before the distortion analysis results were applied to the performance test of an actual engine, the quality of the results was verified.

Figure 7 indicates cylinder block bore distortion when it is assembled at room temperature, and Fig. 8 indicates cylinder block bore distortion when the cylinder block is heated uniformly in a high temperature chamber. In both cases, the distortion is expressed as a change in diameter in the IN and EX directions. The comparison in a uniformly heated condition is indicated as distortion only corresponding to the temperature change to cancel the error that

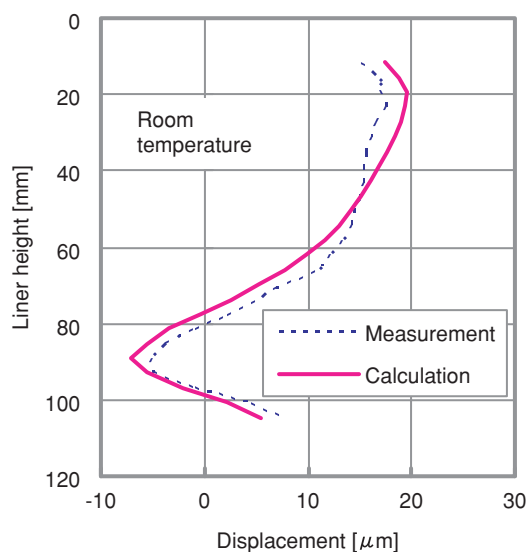


Fig. 7 Bore distortion at room temperature

occurs in the case of room temperature assembly. An inometer was used for measurement, and the comparison was performed using a No.3 bore of an Inline 4-cylinder engine as a representative bore. The results indicate good correlation between analysis and measurement.

Figure 9 shows the temperature distribution measurement points, and Fig. 10 shows a comparison between analysis and measurement. A temperature comparison was performed at two cross-sections located at the liner heights of 9 mm and 89 mm from the upper deck. Thermocouples were used for measurement, and the input conditions were adjusted so that the difference between the analyzed and measured temperatures was within 10 °C.

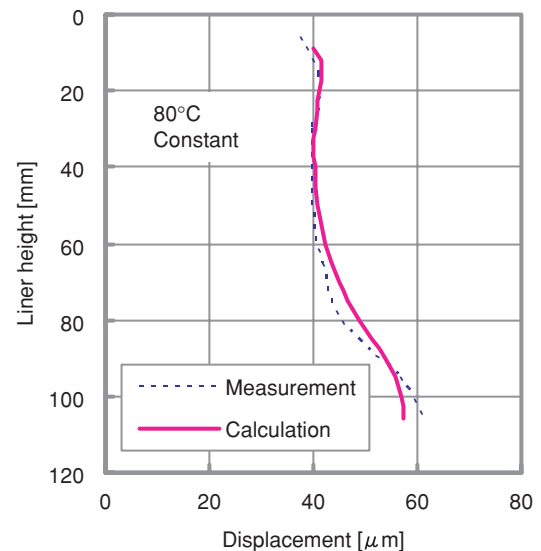


Fig. 8 Bore distortion for temperature difference

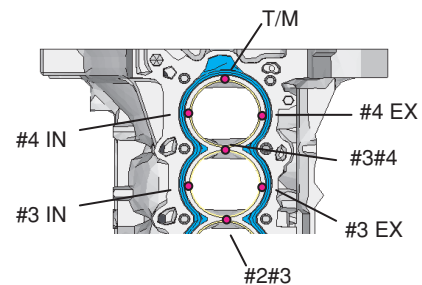


Fig. 9 Temperature measurement point

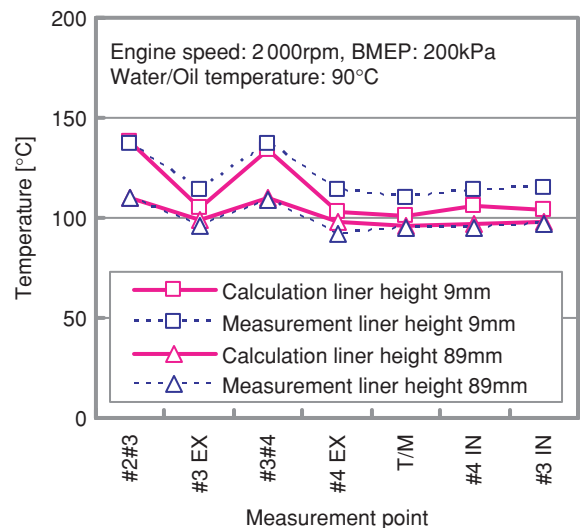


Fig. 10 Temperature distribution of cylinder block

4.2. Piston Secondary Motion

As in the cylinder block bore distortion analysis, a comparison with measured values was carried out to confirm the quality of piston secondary motion analysis. Figure 11 shows a comparison of the analyzed and measured values of translation movement of the piston, and Fig. 12 shows a comparison of the computed and measured values of rotational movement of the piston. Measurement was carried out with gap sensors recessed into the cylinder block. Because of the layout of the sensors, secondary

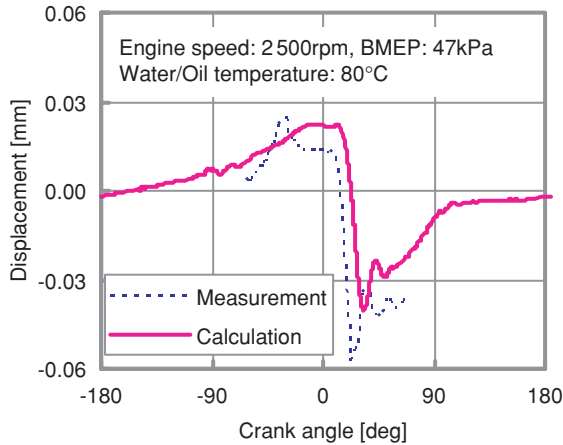


Fig. 11 Translation movement

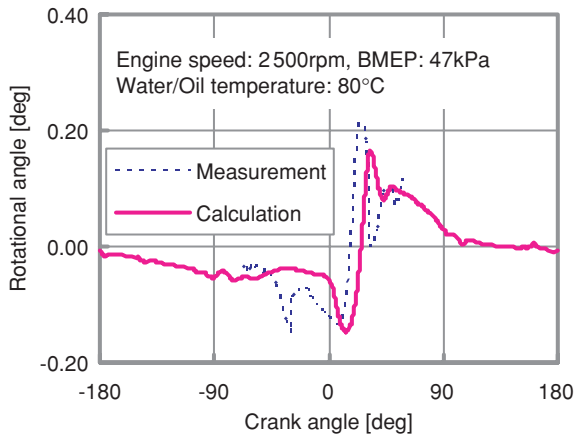


Fig. 12 Rotational movement

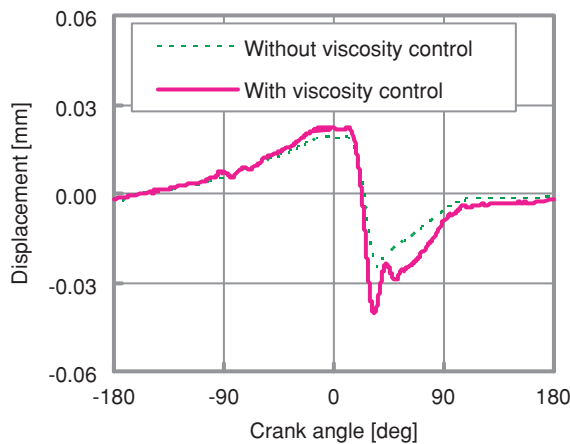
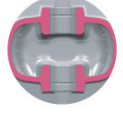
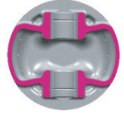


Fig. 13 Comparison of oil viscosity in piston secondary motion

Table1 Specifications of engine and piston

	Previous engine	Newly developed engine
Cylinder configuration	Inline 4-cylinder	Inline 4-cylinder
Engine type	SOHC 2 Valves	SOHC 2 Valves
Ignition system	Dual point sequential ignition	Dual point sequential ignition
Fuel supply system	PGM-FI	PGM-FI
Displacement [cm ³]	1339	1339
Bore x stroke [mm]	73 x 80	73 x 80
Compression ratio	10.8	10.8
Maximum power [kW]	63	70
Maximum torque [Nm]	119	123
Compression height [mm]	26.2	26.2
Skirt width x height [mm]	34 x 29.5	38 x 29.5
Piston weight [g]	207	205
Piston shape		

motion was measured over a range of only about 60 degrees in the vicinity of the TDC.

In the program, the space between the piston skirt and the liner is completely filled with oil, which leads to a tendency to excessive damping. Consequently, in this research, the computed oil viscosity was adjusted, resulting in secondary motion that approximated the measured results. Figure 13 shows the analysis results of the translation movement before and after adjustment of the oil viscosity.

5. Results of Application to an Actual Vehicle

5.1. Basic Specifications of Engine and Piston

As mentioned above, appropriate analytical accuracy was acquired, the effectiveness of changing the piston specification in reducing friction in an engine and increasing reliability was predicted, and the results were compared with the test results.

Table 1 shows the specifications of the engine and piston of a previous engine and those of the newly developed engine. The specifications of the piston were changed twice. In the first change, the shape and material of the piston were changed⁽⁶⁾, and in the second, only the piston form was changed.

Figure 14 shows the stiffness distribution of the piston skirt. Figure 15 shows a comparison of the piston skirt stiffness gradient at the center cross-section of the piston pin. A unit load is assigned to each lattice point of the skirt FEM to obtain the displacement, and the spring value is calculated to express the stiffness. To prevent uneven contact, the development specifications were designed to obtain a softer and more uniform stiffness distribution.

The piston form was first made highly ovalar, aiming at avoiding hard contact along the piston skirt edge. Next, while reducing the hydrodynamic lubrication friction at the oval convexity at the center of the skirt, adjustments were made to prevent the boundary

lubrication friction from increasing.

Within the scope of this study, it was found that the contribution to friction reduction of longitudinal change was not as high as that of the change in ovality.

5.2. Friction Study Results

Table 2 shows a comparison of the analyzed and measured results of the friction reduction rate under motoring conditions subsequent to warm-up. The changes in the shape and material produced a 1.3% reduction in friction, while the change in the piston form produced a 0.7% reduction. Compared to previous specifications, a total friction reduction of 2% was obtained for the engine alone. There was good correlation between analysis and measurement in respect to these values.

Figure 16 shows the pressure distribution for boundary lubrication in the circumferential direction of the piston skirt for

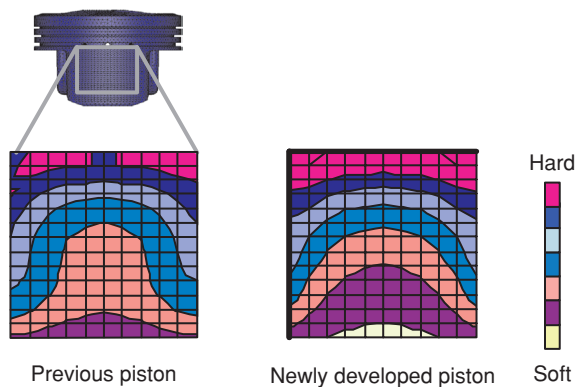


Fig. 14 Piston skirt stiffness

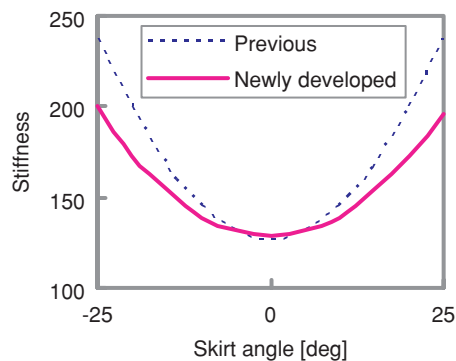


Fig. 15 Piston skirt stiffness in piston pin center section

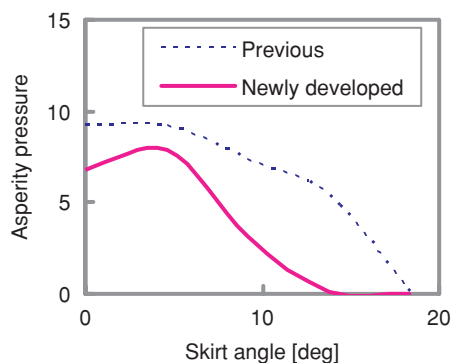


Fig. 16 Asperity pressure

Table 2 Friction reduction rates

	Calculation	Measurement
Piston shape & material	-1.4%	-1.3%
Piston form	-0.7%	-0.7%

the first specification study. The pressure is the integrated value at the cross-section in each circumferential direction. As mentioned above, in the newly developed specifications, the piston stiffness value along the circumferential edge of the piston skirt is low, so the pressure during boundary lubrication is largely reduced along the circumferential edge. From these results, it can be seen that the piston that has a small increase in stiffness along the circumferential edges of the skirt; in other words, it has a more uniform stiffness distribution, and exhibits a lower increase in friction.

Figure 17 shows the contribution to friction reduction of the shape and material (Young's modulus, coefficient of thermal expansion) in the first specification study. The material physical property values were changed in calculations and verified. The contribution due to the abovementioned specification changes were 74% for shape, 22% for Young's modulus and 4% for thermal conductivity.

Figure 18 shows the contribution of the cylinder block bore distortion in the simulation for the second specification change. It can be seen that the friction prediction accuracy is increased by inputting the cylinder block bore distortion.

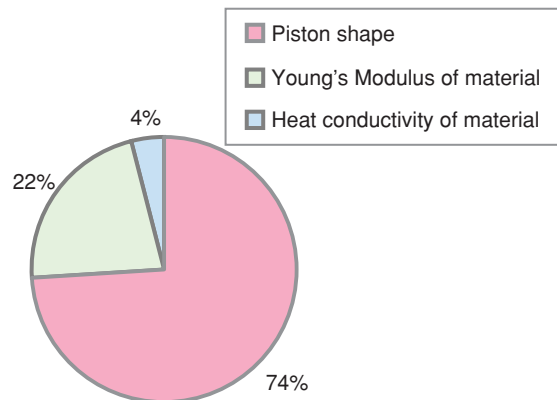


Fig. 17 Influence of piston shape and material

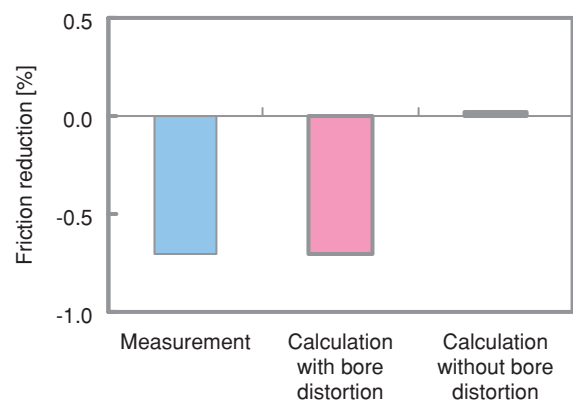


Fig. 18 Influence of bore distortion in friction prediction

5.3. Seizing Study Results

Reliability is evaluated via seizing. The wear load (work rate per contact unit area due to boundary lubrication, face pressure $\times$ velocity) is used as the evaluation index.

Figure 19 shows the comparison results. The study takes into consideration the upper and lower limits of the cylinder block bore distortion based on the trend of the distortion when the temperature of the cylinder block is changed. The error indication in the figure is the wear load at the upper and lower limits of the cylinder block

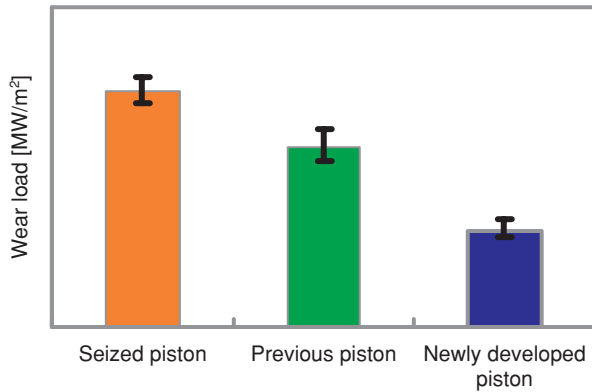


Fig. 19 Comparison of wear load

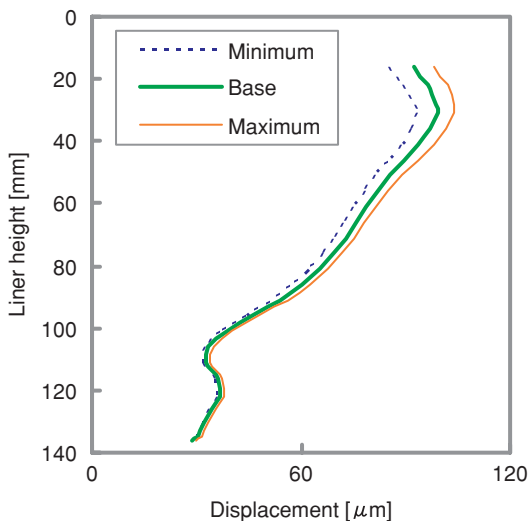


Fig. 20 Bore distortion by temperature difference

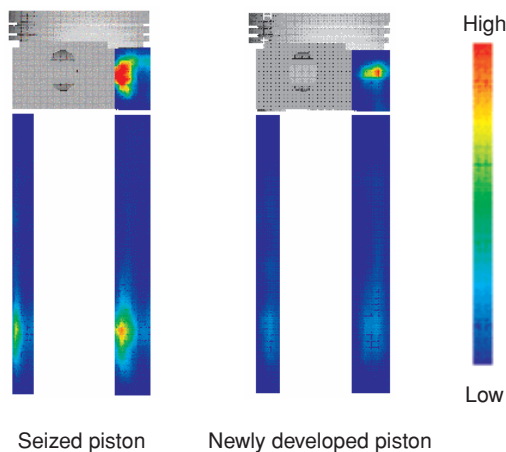


Fig. 21 Distribution of wear load

bore distortion. Figure 20 shows the effect of temperature on the cylinder block bore distortion.

The peak value under previous specifications is lower than that of the specifications under which seizing occurred during an actual engine test, and is even lower in the case of the newly developed specifications.

Figure 21 shows a comparison of the wear load distribution at the piston skirt. It can be seen that the distribution of wear load is narrow. This is because, as in the case of friction, the stiffness of the edge part of the skirt is low, and the ovality of the piston form was increased to prevent a large area of contact on the edge of the piston skirt.

By using the abovementioned specifications, it was possible to complete an actual engine test with no seizing.

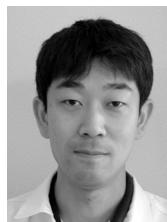
6. Conclusion

- (1) It is possible to carry out a detailed desktop study of friction reduction while judging the trend of durability and reliability.
- (2) A 2% reduction of friction was obtained by using the results of this study in the development of an actual engine.
- (3) Low increase in stiffness at the edge part of the piston skirt and more uniform stiffness distribution have an effect on friction reduction.
- (4) The contribution to friction reduction of the piston under the development specifications was 74% for shape, 22% for Young's modulus and 4% for thermal conductivity.
- (5) The friction reduction effect due to the piston form is 0.7%, and can be increased further by additionally changing the piston form.
- (6) Concerning seizing, an evaluation based on the wear load is possible.
- (7) In the evaluation of friction and seizing, it was necessary to input the cylinder block bore distortion, therefore the evaluation results varied according to the accuracy of the distortion input.

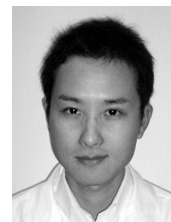
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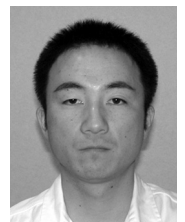
■ Authors ■



Kenji SATO



Kinya FUJII



Makoto ITO



Shinsuke KODA