

Piston Slap-noise Prediction Using 3-dimensional Piston Secondary Motion Simulation

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ABSTRACT

A method of predicting piston slap noise was developed. Conventionally, piston slap noise was evaluated using moving energy or vibration of cylinder liner obtained by simulating the piston secondary motion. However, prediction accuracy and reliability were lower. This new method focuses on cylinder liner vibration, which has a high correlation with piston slap noise. Increasing the accuracy of predicting the cylinder liner thermal deformation, the piston and cylinder block dynamic stiffness model, and the 3-dimensional piston motion simulation program through oil film model modification enabled cylinder liner vibration to be predicted highly accurately. As a result, it was confirmed that piston slap noise could be predicted with sufficient accuracy for all practical purposes based on a comparison with the measured values for an actual engine under running conditions. By applying this method to a newly developed engine, the piston specifications could be determined at the initial stage of development.

1. Introduction

The importance of piston design in reciprocating engines is increasing due to requests in recent years for better environmental performance and comfort and overall drivability in automobiles. Amidst this matter, there have been many reported cases of analyses using piston secondary motion simulations to increase durability and reduce friction in the piston design^{(1) - (4)}. Various approaches using secondary motion simulations have also been applied to piston slap noise^{(5) - (7)}. In all of these studies, piston slap noise has been predicted based on the motion energy or cylinder liner vibration that is considered to correlate closely to slapping noise. However, the results have been limited to a qualitative range and the accuracy cannot be considered adequate from a practical viewpoint. A more accurate method for predicting piston slap noise is needed because the prediction of durability, friction, and piston slap noise during the initial stage of design can reduce the hours involved in testing and enhance the efficiency of design.

2. Outline of Prediction Method

2.1. Correlation between Piston Slap Noise and Cylinder Liner Vibration

Several kinds of piston slap noise are known to occur due to different kinds of secondary motion depending on the operating condition of the engine. In this research, these different forms of noise are stated collectively as slap noise. Particularly, two kinds of slap noise are covered here: slap noise that occurs at the anti-thrust side during no-load idling and slap noise that occurs at the thrust side during low-load at low-speed operation.

Figure 1 shows an example. It indicates the correlation between the relative change of cylinder liner vibration and subjective evaluation of piston slap noise under the conditions of different piston specifications and temperatures in a running engine. The cylinder liner vibration of each specification is defined as 0 dB at a borderline subjective rating. The subjective evaluation is a numerical value obtained by averaging the evaluation values assigned by several experts.

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From this graph, it is judged that there is a correlation between the change in subjective evaluation of piston slap noise and the relative change in the cylinder liner vibration and that the change in the subjective evaluation of piston slap noise is accurately predictable by the cylinder liner vibration.

2.2. Calculation Flow

Figure 2 shows the calculation flow. When predicting piston secondary motion, thermal deformation has a large effect on the clearance. Consequently, it is important to accurately predict the thermal deformation of the cylinder liner and piston when piston slap noise occurs.

A general purpose fluid analysis program was used to analyze the thermal distribution inside the water jacket,

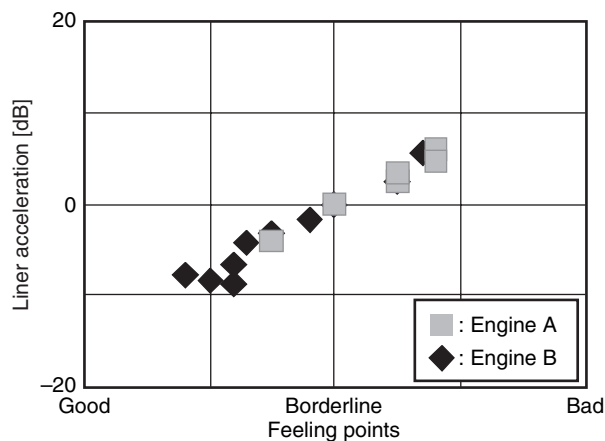


Fig. 1 Correlation of cylinder liner acceleration and feeling points

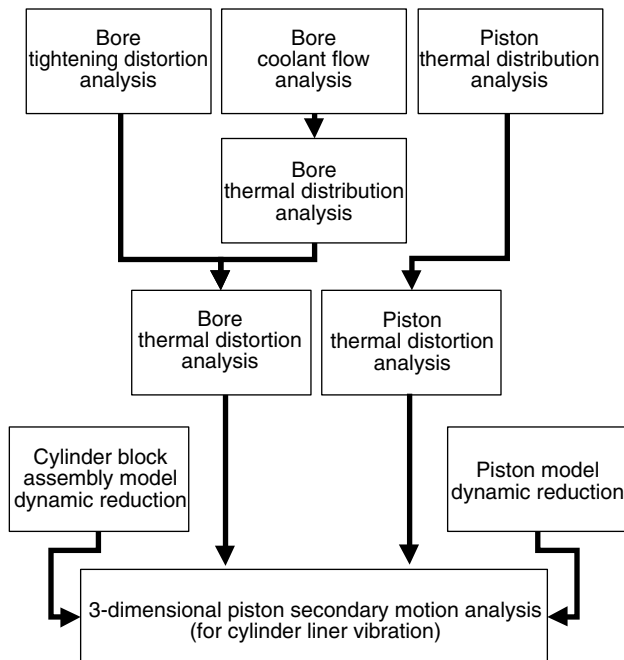


Fig. 2 Analysis flow

and a general purpose non-linear structural analysis program was used to analyze a tightening distortion of the cylinder block, enabling prediction of the thermal deformation of the cylinder liner during the low-temperature warm-up stage.

The thermal deformation of the piston is predicted by analyzing the thermal distribution taking the heat input from the combustion chamber into account.

Simultaneously, an FEM model of the cylinder block and piston, which were used in the previous thermal analysis for an accurate prediction of the cylinder liner vibration after the piston collision, is obtained by dynamic reduction in consideration of the vibration characteristics.

The cylinder liner vibration is accurately predicted by these models and a 3-dimensional piston secondary motion program.

3. Prediction of Thermal Deformation and Verification of Prediction

3.1. Cylinder Liner Thermal Deformation Analysis

Cylinder liner deformation in the engine running condition was predicted by combining the tightening distortion analysis and thermal deformation analysis based on the thermal distribution calculation using the heat transfer coefficient distribution with the coolant flow⁽⁸⁾. Figure 3 shows the FEM model of the cylinder block used in the analysis.

Figure 4 shows the results of a comparison between the temperature values obtained from the thermal distribution analysis performed with this model and measured cylinder liner temperature values obtained under

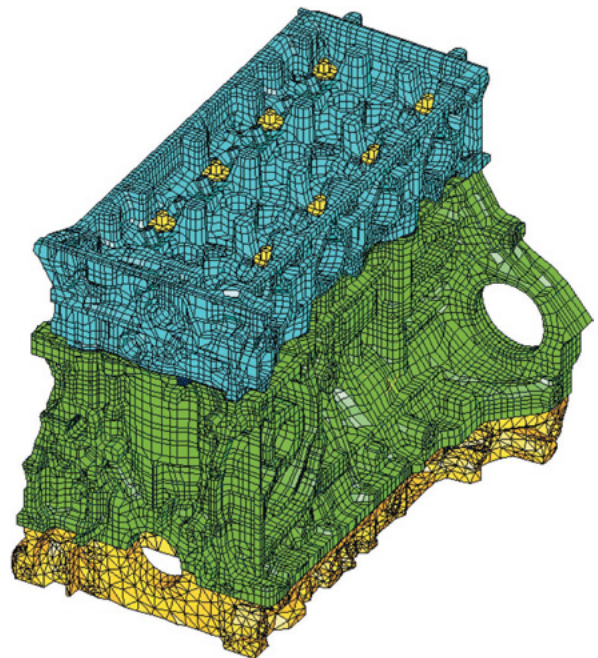


Fig. 3 Cylinder block FEM model

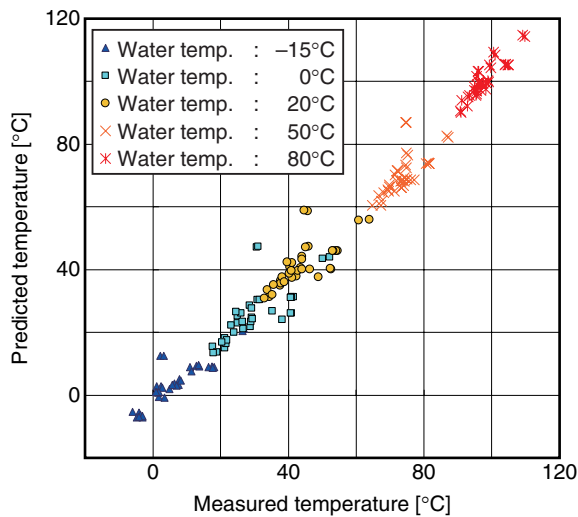


Fig. 4 Verification results for thermal distribution of cylinder block

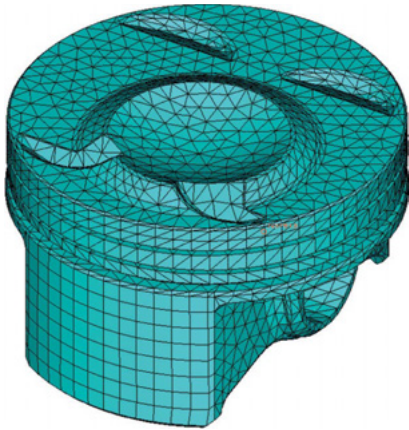


Fig. 5 Piston FEM model

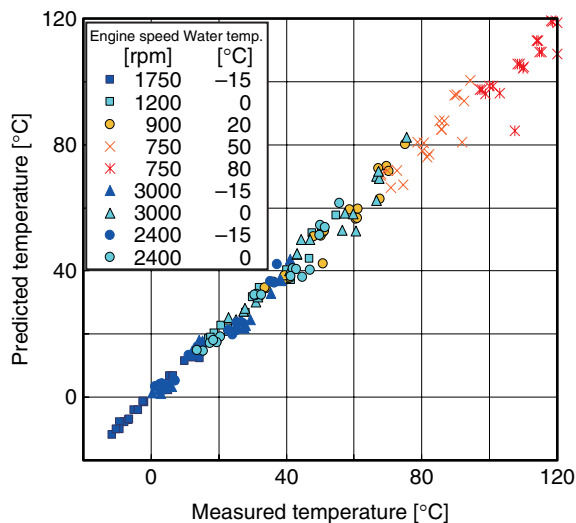


Fig. 6 Verification results for thermal distribution of piston

engine running conditions. Temperature deviations between the predicted and measured values obtained immediately after a cold start, during warm-up, and after warm-up were $\pm 17^\circ\text{C}$ in the analysis, while the standard temperature deviation was 5.8°C , and the obtained predicted temperatures were reliably accurate. Based on these analysis results, thermal deformation analysis is performed.

3.2. Piston Thermal Deformation Analysis

The piston deformation under engine running conditions was analyzed using a database of past analyses. The heat input value in the vicinity of the combustion chamber and boundary conditions were used as input conditions for the thermal distribution analysis. Figure 5 shows the FEM model of the piston used in the analysis.

Figure 6 shows a comparison between the predicted value of the thermal distribution analysis results and the piston temperature measurement value under engine running conditions. The temperature deviations at the measurement points were within 10°C , and the temperature prediction accuracy was judged adequate for the piston as well. Therefore the thermal deformation analysis was performed using the results of the thermal distribution analysis.

4. Three-dimensional Piston Secondary Motion Simulation

4.1. Reduction Model

In the 3-dimensional piston secondary motion program used in this research, the cylinder block model and piston model used in the thermal deformation analysis in Figs. 3 and 5 were reduced by the dynamic reduction method to enhance the accuracy of the dynamic stiffness model for a more accurate prediction of the cylinder liner vibration. Figure 7 shows the results of the verified accuracy of the cylinder block model.

The frequency response function was measured by taking the top part of the inside wall of the cylinder liner

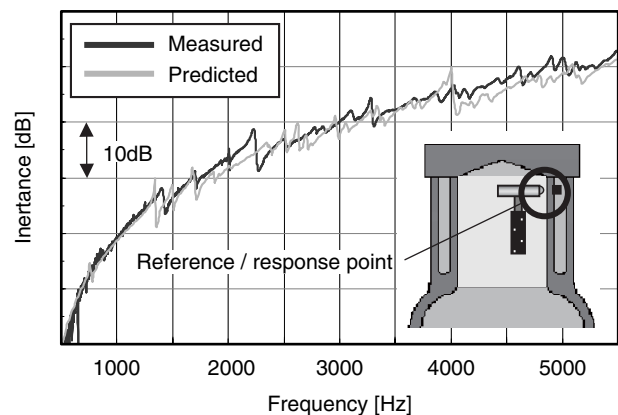


Fig. 7 FRF verification results of cylinder block

as an excitation point and the point for measuring the cylinder liner vibration in an actual engine as the response point. The same points were used as the excitation and response points for the frequency response analysis using the FEM model.

As a result, the predicted values were located nearly on the same line as the measured values, so it is judged that the model had sufficient accuracy as a cylinder liner vibration analysis model.

4.2. Oil Film Model

In the 3-dimensional piston secondary motion simulation program, the condition with the space between the piston skirt and cylinder liner completely filled with oil was represented by a model because the oil film characteristics incorporating elasto hydrodynamics (EHD) are applied to the sliding faces of the piston skirt and cylinder liner. However, in an actual engine, a mechanism that forces lubrication between the piston skirt and cylinder liner does not exist, and the actual lubricated condition is thought to result in air bubbles. For this reason, a model with air bubbles in the oil between the skirt and the cylinder liner was introduced as shown in Fig. 8⁽⁹⁾ to correct the oil film pressure due to the inclusion of air bubbles.

4.3. Piston Secondary Motion Analysis

After the analysis using the results in 3.1 and 3.2 and the model, the predicted values of the piston secondary motion and the measured values for a running engine were compared.

Figure 9 shows an outline of the device used for measuring the piston secondary motion. The secondary motion was measured by a method in which four pairs of gap sensors were mounted in the cylinder block and the lateral and rotational displacements of the piston calculated by using the outputs of two pairs of these sensors. Figure 10 shows the results of a comparison of the predicted and measured values of the lateral displacement of the piston secondary motion under the condition where the slap noise is pronounced (no-load idling at 2400 rpm). Figure 11 shows the results of a

comparison of the predicted and measured values of the rotational displacement of the piston secondary motion under the same conditions.

Comparing the predicted and measured values confirmed that the predicted values of the piston secondary motion accurately reproduced the measured values and taking air bubbles in the oil film into account enhanced the accuracy.

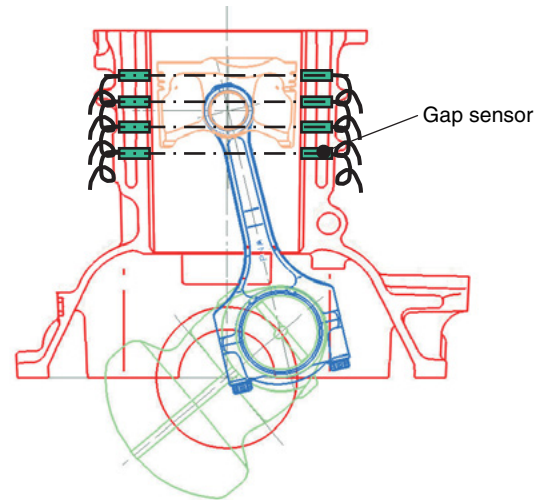


Fig. 9 Piston motion measurement method

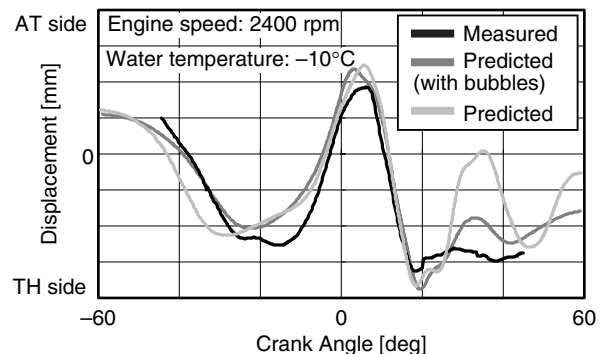


Fig. 10 Verification results for lateral displacement of piston

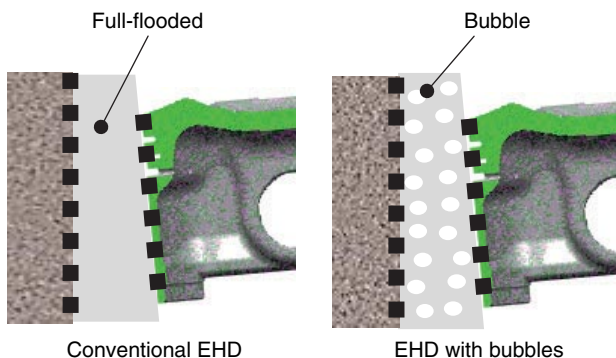


Fig. 8 Conception of bubble model

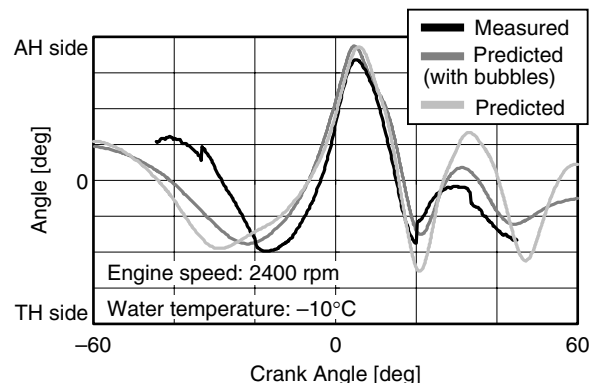


Fig. 11 Verification results for rocking angle of piston

Table 1 Specifications of engine and piston

Items	Specifications
Cylinder configuration	Inline 4-cylinder
Valve train	DOHC 4 valves
Crank offset length [mm]	0
Ignition system	Single point ignition
Fuel supply system	PGM-FI
Displacement [cm ³]	1998
Bore × stroke [mm]	86 × 86
Compression ratio	9.8
Maximum power [kW]	118
Maximum torque [Nm]	191
Compression height [mm]	26.2
Skirt width × height [mm]	34 × 29.5
Piston weight [g]	310

Table 2 Variations among pistons

	BASE	A	B	C	D
Pin offset [mm]	0.8	1.3	0.3	0.8	0.8
Skirt form	Base	Base	Base	Thin	Thick

4.4. Verification Results

As mentioned previously, it was confirmed that the prediction accuracy of the piston secondary motion was increased by enhancing the prediction accuracy of the cylinder block and piston thermal deformation, enhancing the accuracy of the dynamic stiffness model, and modifying the oil film model. This method is used to compare the cylinder liner vibration change with respect to a change in the piston specifications with the measured values of the cylinder liner vibration under engine running conditions.

Table 1 shows the specifications of the test engine and the pistons, and Table 2 shows the specifications changed for the verified piston of each specification.

The object of analysis was slap noise at the anti-thrust side under no-load idling around 2400 rpm where the difference among specifications is prominent in the measurement with the running engine.

The cylinder liner vibration was measured by installing a small acceleration sensor inside the water jacket of the engine, on the outside wall of the cylinder liner (15 mm below the gasket face in the thrust direction).

Figure 12 is a graph showing the relative change of the cylinder liner vibration obtained from the above measurement, taking the level of the base piston as 0. This

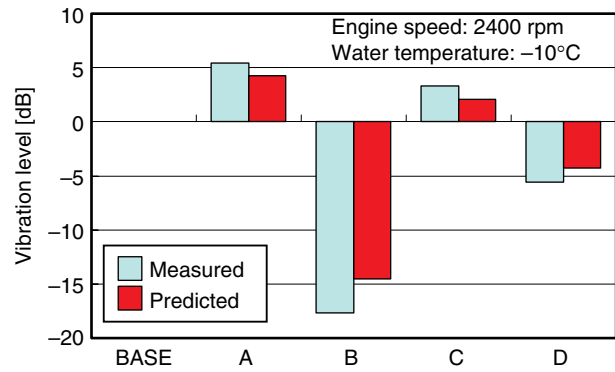


Fig. 12 Verification results of cylinder liner accelerations

graph shows a reliable prediction accuracy for all specifications, although the analysis results tended to be slightly lower.

5. Application to Actual Engine

Using this prediction method, a study of the skirt profile in the simulation was performed for a newly developed engine.

Table 3 shows the specifications of the newly developed engine and piston.

Piston slap noise is generated due to different secondary motions resulting from different engine operation conditions as mentioned previously. For this reason, when determining specifications, it is necessary to simultaneously reduce slap noise for these different regions.

Table 3 Specifications of newly developed engine and piston

Items	Specifications
Cylinder configuration	Inline 4-cylinder
Valve train	SOHC 2 valves
Crank offset length [mm]	14
Ignition system	Dual point sequential ignition
Fuel supply system	PGM-FI
Displacement [cm ³]	1339
Bore × stroke [mm]	73 × 80
Compression ratio	10.8
Maximum power [kW]	70
Maximum torque [Nm]	123
Compression height [mm]	26.2
Skirt width × height [mm]	38 × 29.5
Piston weight [g]	205

Therefore, it has to be confirmed that slap noise is not enhanced under a number of operation situations even when the skirt profile is changed.

The barrel form and ovality of the skirt profile were changed from the base piston shape.

Figure 13 shows the analysis results of the cylinder liner vibration at the anti-thrust side during 3000 rpm no-load idling and at the thrust side during 1500 rpm low-load operation for four kinds of pistons with modified skirt profiles. As mentioned previously, the graph

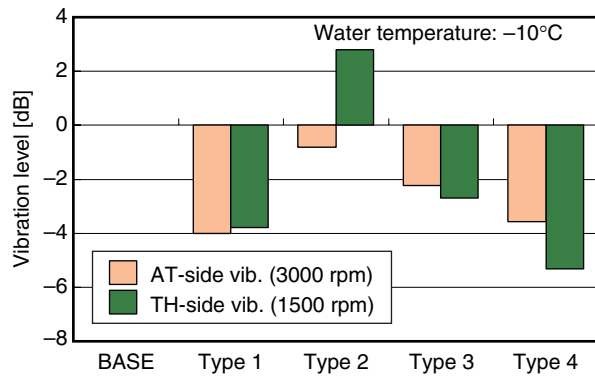


Fig. 13 Comparison of predicted cylinder liner acceleration (TH side and AT side)

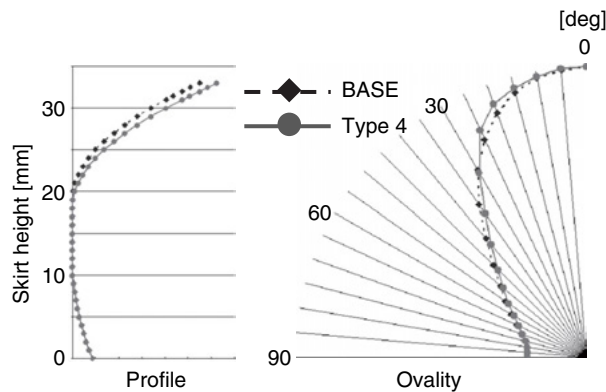


Fig. 14 Skirt profile

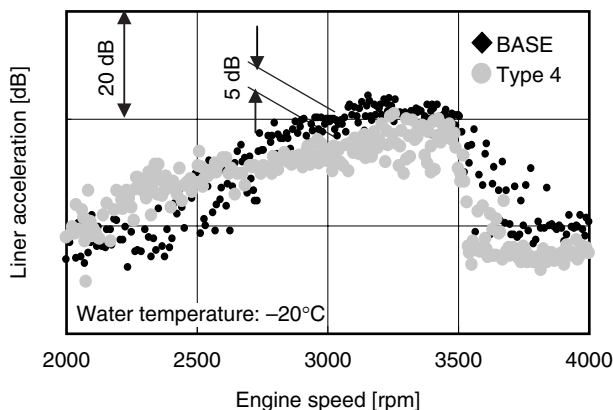


Fig. 15 Cylinder liner acceleration (BASE versus Type 4)

indicates the relative change when the cylinder liner vibration level of the base piston is defined as 0.

This graph shows the reduction of piston slap noise at both the thrust and anti-thrust sides would be predicted because the vibration levels of the Type 1 and Type 4 forms were reduced by about 4 to 6 dB.

At the same time, friction and scuffing was predicted⁽⁴⁾. Of the proposed specifications, an increase in scuffing and friction was not predicted with Type 4. Therefore, Type 4 was actually created and verified using a running engine.

Figure 14 shows an outline of the base and Type 4 skirt profiles.

By checking piston slap noise in a running engine, it was confirmed that as predicted, the cylinder liner vibration was about 5 dB lower than that of the base piston, and slap noise was also reduced in the case of the subjective evaluation.

Figure 15 shows the measured results of base and Type 4 cylinder liner acceleration on the anti-thrust side. This confirms cylinder liner vibration acceleration is reduced in the region between 2800 and 3200 rpm where piston slap noise was prominent.

6. Conclusions

- (1) In the region where piston slap noise occurs, the relative prediction of piston slap noise was performed by a reliable and accurate prediction of the cylinder liner vibration.
- (2) The accuracy of piston slap noise prediction using cylinder liner vibration obtained by analysis was verified and its effectiveness confirmed by measurement under actual engine running conditions.
- (3) This method was used for a newly developed engine to predict the reduction of piston slap noise due to the change in the piston form with the optimum form proposed. As a result, the specifications were determined in the early development stage, contributing to a reduction of hours involved in testing.

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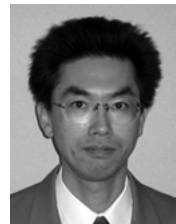
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