

Study of Radiation Noise for Motorcycle Engines

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ABSTRACT

In reducing the noises emitted by a motorcycle during acceleration, it is very important to reduce the noise radiated from the engine. To reduce engine noise radiation, we have developed an engine-noise-radiation analysis system. In analyzing engine-noise radiation, vibration analysis techniques appropriate for the high frequencies of the engine case and covers, a technique for estimating the excitation force input into the engine, and an analysis technique to assess acoustic output from the measured engine surface vibrations, are required. After overcoming all technical obstacles, our newly-developed system allows assessment of the noise radiation characteristics of motorcycle engines. The above-mentioned analysis method has been applied to reduce noise radiation from an inline four-cylinder motorcycle engine. The reduction of engine noises during acceleration has been confirmed from a modified engine.

1. Introduction

Reduction measures of intake and exhaust noise and drive train system noise in motorcycles have progressed; consequently, the proportion of engine noise in motorcycle noise sources has increased compared to the conventional models. In the future, measures of engine radiation noise reduction are necessary for enhancing quietness. In addition to radiation noises caused by excitation forces from combustion pressure, radiation noise caused by transmission gears is significant because the transmission is united with the crankcase. The frequency range of gear noise is approximately 3 kHz. The vibration mode of the case and cover surface often becomes an issue in the high frequency region.

In order to attenuate the vibration of the case and cover surface, countermeasures have included adding ribs, increasing thickness, and isolating the backside of the cover using a rubber plate pressed by a steel plate.

In conventional acoustic analyses, engine surface vibration velocities were measured using accelerometers and a laser vibrometer. Using general purpose acoustic analysis software, vibration velocity was analyzed to determine which parts have high acoustic radiation energy. The noise contribution rate of each engine component, which had been unknown up to now, can be analyzed in detail. Such analyses, however, have not led to any effective quantitative noise reduction countermeasures. Additionally, the countermeasures were accompanied by increases in weight and costs. For final assessment, it is necessary to quantitatively assess noise radiation by estimating the engine surface vibrations that are not based on actual engine vibration measurements.

As shown in Fig. 1, about 80% of motorcycle engine acoustic energy consists of frequency components more than 1 kHz, while about 50% have frequency components more than 2 kHz. Compared to other noise sources such as intake and exhaust noises, the acoustic energy of a motorcycle engine occupies a very high frequency band.

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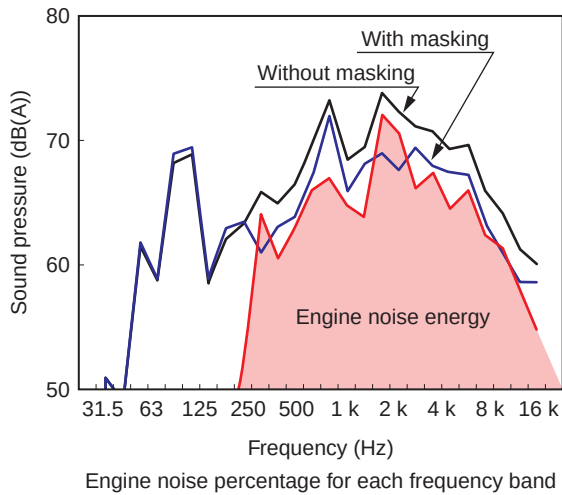


Fig. 1 Engine noise

The modeling method enhancing the analytical accuracy of individual parts and the optimizing method of connector characteristics for each part were applied. The finite element method (FEM) was used to create a structural model for radiation noise analysis. Additionally, it was considered whether the merits of radiation noises could be relatively compared using results that were calculated with the same excitation force applied to two models with modified specifications. As a result, it was found that applying excitation forces with frequency content close to the actual engine frequency characteristic was necessary to compare modified specifications, even in a relative comparison⁽¹⁾. Figure 2 shows the analysis flow of engine noise radiation used in this study.

2. FEM Models of Engine Case and Cover

FEM models of the engine case and cover were generated using the auto meshing functions of CATIA FEM on the CATIA-3D models. An example of the parts group comprising the motorcycle engine is shown in Fig. 3. The model size was determined

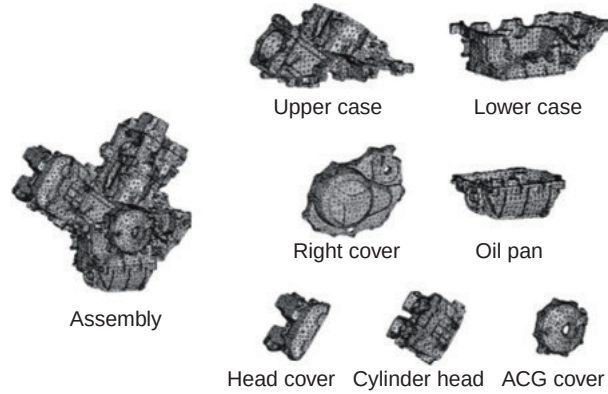


Fig. 3 Parts

(approx. 300,000 nodes) using the calculation resources that were available. The modeling accuracy was verified through the modal analysis of all parts using LMS' Gateway software. With reference to frequency errors, the Modal Assurance Criterion (MAC) value was used as a correlation index of modes. Based on this analysis, the mesh size for each important component and form was determined. For instance, optimization calculations were performed using the Young's modulus of the solid mesh as a parameter to match the calculated results and test results of motorcycle clutch cover. The change in Young's modulus is shown in Fig. 4. This figure suggests that the Young's modulus of the rounded area and the opened area should be reduced. Since the mesh sizes of those areas were found to be large, the overall mesh size and local mesh size were optimized. The change in the transfer function before

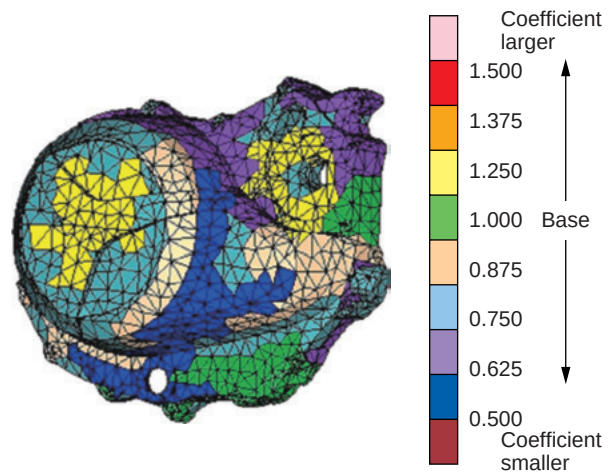


Fig. 4 Coefficient of Young's modulus

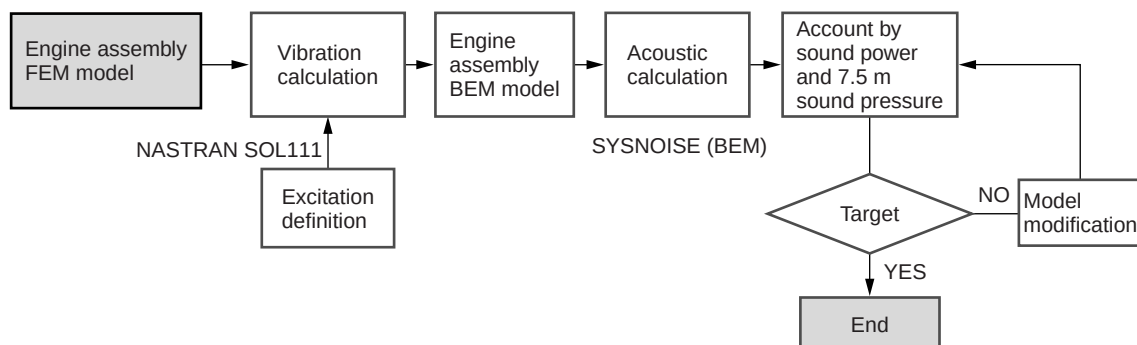


Fig. 2 Analysis flow

and after optimizing the mesh sizes is shown in Fig. 5.

The contact boundary surfaces between each part were calculated previously using GAP analysis. There is also a method of applying springs in parts contacting rigidly⁽²⁾. However, in this method, the meshes of contact boundary surface between each part are not matched. Each part is connected by using the RBE3 elements in NASTRAN at the bolt holes of each part. Each RBE3 element is connected by a six-directional spring element. The spring constant was determined through reference to the modal analysis results of the engine case and cover in their connected conditions. The values are all the same except for the rubber floating part of the cylinder head cover. Figure 6 shows a comparison of the transfer function from calculated results and tested results in the connected condition of the engine case and cover. The MAC value in the connected condition was greater than 0.5 up to a 1.5 kHz. The model accuracy was judged from the level of the transfer function because the mode density significantly increased above 1.5 kHz. Ensuring the accuracy of this analytical model is important because it is also used in identifying excitation forces from the bearing.

3. Estimation of the Engine Surface Acceleration and Excitation Forces from the Bearing during Actual Operation

Conventional techniques for estimating the excitation force acting on the bearing include techniques that estimate forces through the piston and crankshaft from the combustion pressure using EXCITE (AVL Inc.) and estimating the measured stress of bearing. However, the guaranteed accuracy of these techniques is below 1 kHz. Since combustion pressure is not the only cause of engine

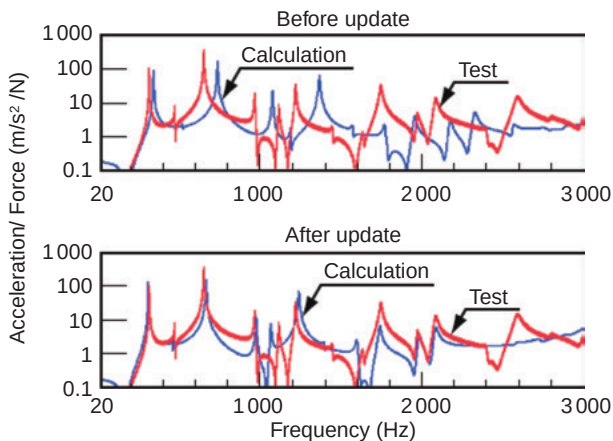


Fig. 5 Transfer function of right cover

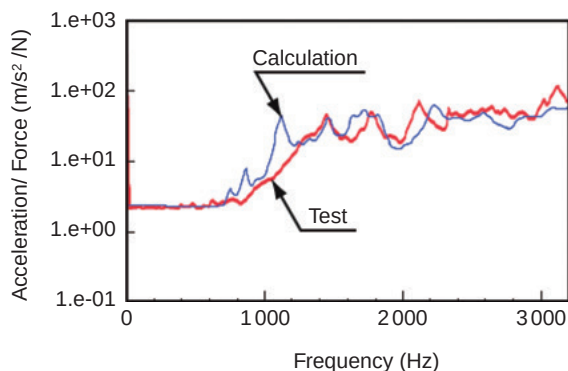


Fig. 6 Transfer function of engine assembly

vibration, these techniques were not used. In this study, vibration measurements were made at multiple points on the actual engine surface. The virtual bearing forces were estimated by the inverse matrix method^{(3), (4)}. The number of bearing excitation points needed to accurately express the vibration of each engine part was studied. It was confirmed that at least three excitation points were needed to accurately express engine behavior. Based experimentally on the number of points that can be measured simultaneously, two virtual bearing excitation points were applied on the crankshaft in addition to two points on the transmission unit. Virtual excitation forces with a total of 12 degrees of freedom were used since four points were taken in three directions. Finally, virtual excitation forces with 12 degrees of freedom were determined using the method of least squares based on the results of simultaneous measurements of 48 degrees of freedom. Figure 7 illustrates the principles. The relationship between transfer function, the excitation force and acceleration response is expressed by Eq. (1). Conversely, the excitation force can also be expressed more clearly as acceleration multiplied by the inverse of the transfer function (Eq. (2)). After determining $12 \times 48 = 576$ transfer functions by structural analysis, 12 excitation forces were determined from accelerations in 48 simultaneous measurements.

$$A = H * F \quad (1)$$

$$F = H^{-1} * A \quad (2)$$

For the inverse matrix calculations, Matlab software (The Math Works, Inc.) was used. Figure 8 shows the estimation technique concept. Only measurement points on the upper and lower cases of engine near the points at which virtual excitation forces act were used. Figure 9 shows an example of the results obtained from the vibration calculations using the excitation forces that were determined. These results are for a V-twin, 1 000-cm³ engine running at approximately 2 800 r/min. It can be observed that the predictions are for a frequency range up to 3 kHz.

4. Results of Verification of Engine Noise Radiation Predictions

From the vibration results found in Section 3, acoustic analysis was performed using a SYSNOISE direct BEM approach. Conversely, engine surface vibration measurements at about 700 points were made during actual operation using a dynamometer. Acoustic analysis was performed using the measured and calculated results on the same model. Calculations could be performed to an accuracy of within 3 dB sound power. Figure 10 shows the results of the analysis verification. Figure 11 shows the comparison of surface sound pressure distributions, expressed as peak sound power. It was determined that both surface sound pressure levels

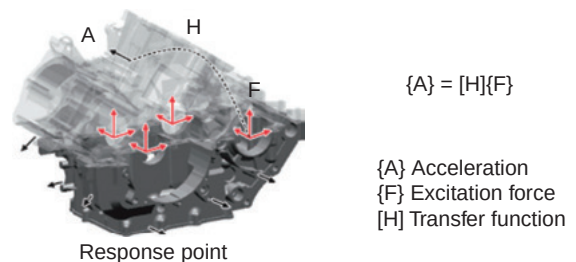
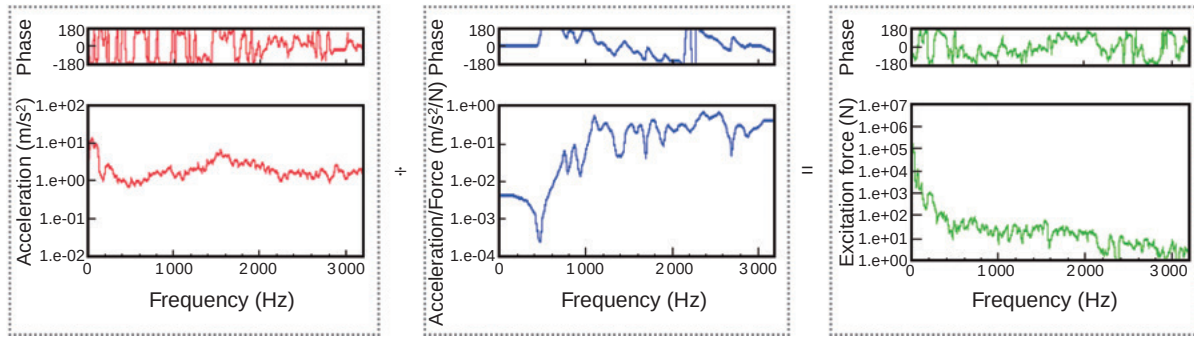


Fig. 7 Definition of excitation force



$$\text{Acceleration (Test)} + \text{Transfer function (CAE)} = \text{Excitation force (Estimation)}$$

Fig. 8 Estimation of excitation force

and distributions were estimated accurately. From the calculations, a significant amount of noise is radiated from the clutch cover and the cam chain chamber.

5. Examples of Actual Motorcycle Applications

A practical application of this analysis method to an in-line four-cylinder 1 000 cm³ engine is shown here. Figure 12 shows the engine 3D model. To identify excitation forces, vibration measurements were made in three directions at 16 points on the upper/lower cases of engine running at approximately 4 900 r/min.

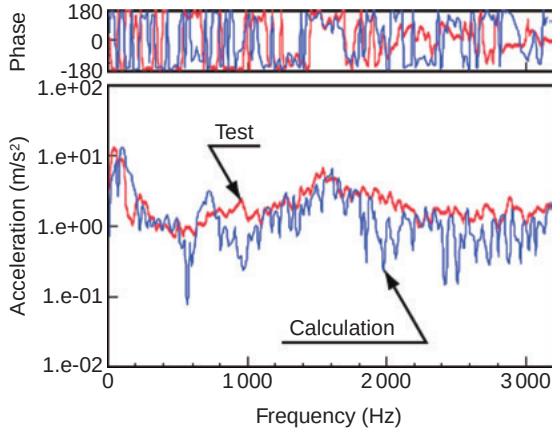


Fig. 9 Comparison of responses between test and calculation

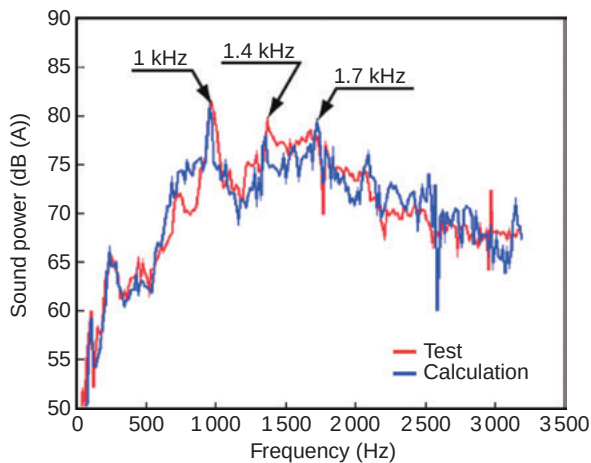


Fig. 10 Comparison of sound powers between test and calculation

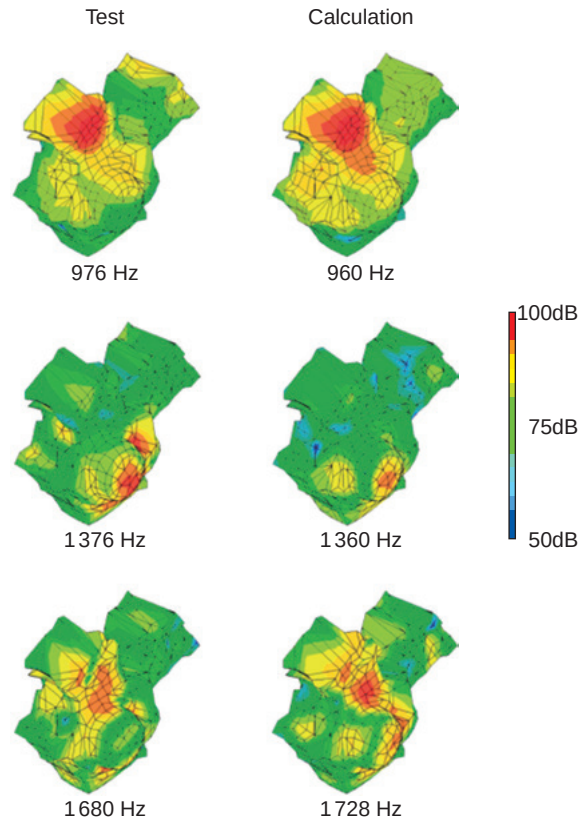


Fig. 11 Comparison of sound pressure distributions between test and calculation

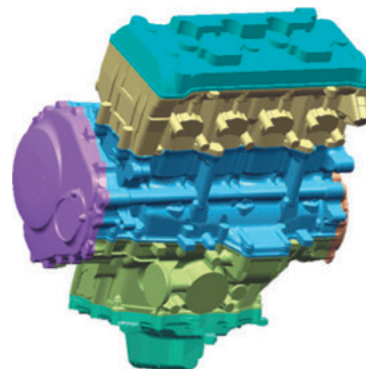


Fig. 12 Engine 3D model

Figure 13 shows the surface acceleration measurements obtained. Figure 14 shows the results of virtual excitation forces that were calculated based on the measured results. As observed from Figs. 13 and 14, the excitation frequencies during gear meshing contributed greatly to vibration, unlike the example of the V-twin 1 000 cm³ engine (Fig. 9). Acoustic analysis was performed for

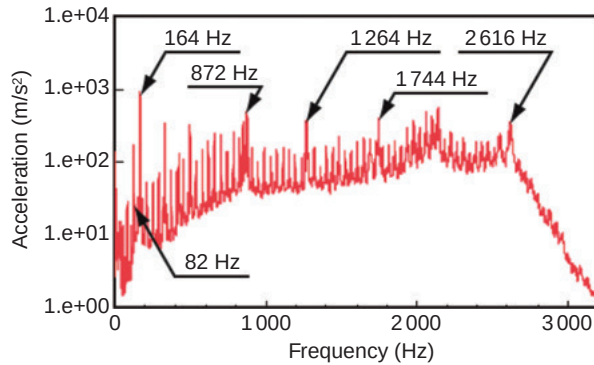


Fig. 13 Measured surface vibration

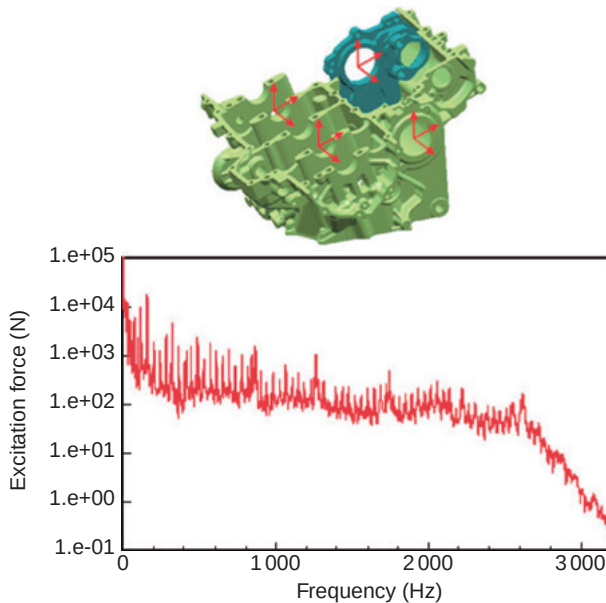


Fig. 14 Estimated excitation force

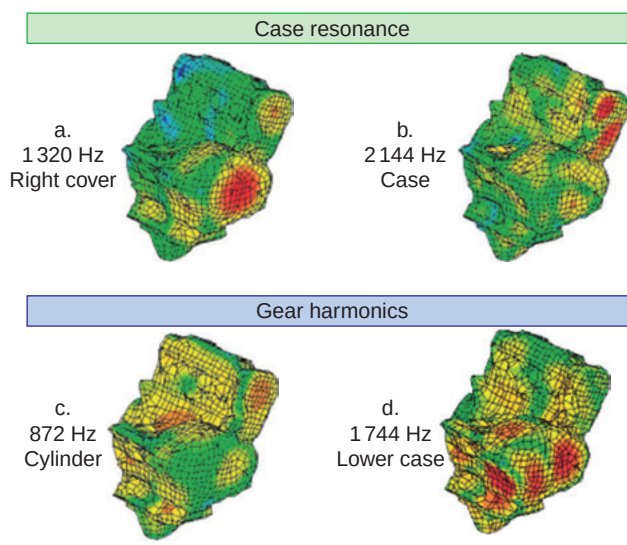


Fig. 15 Calculation result (sound power and surface pressure distribution)

the results mentioned above. Figure 15 shows the surface sound pressure level distribution at peak frequencies of sound power on the engine. This shows that the noise radiation parts vary according to frequencies. Countermeasures for the above were studied (Fig. 16). Ribs were added and board and wall thicknesses were increased. The changes in calculated sound power were plotted as shown in Fig. 17. An approximate 3 dB reduction was obtained in

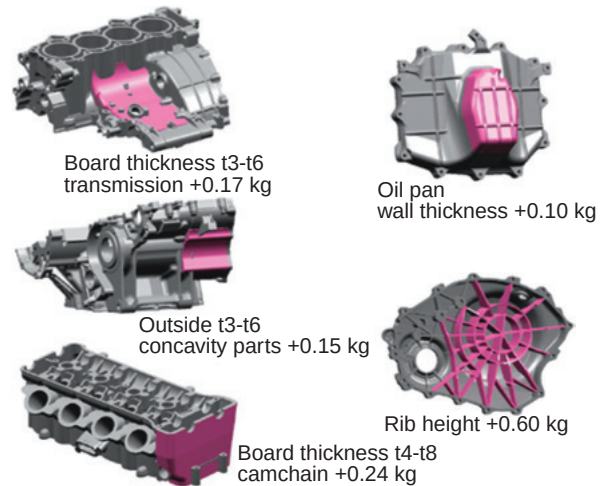


Fig. 16 Modified case and cover

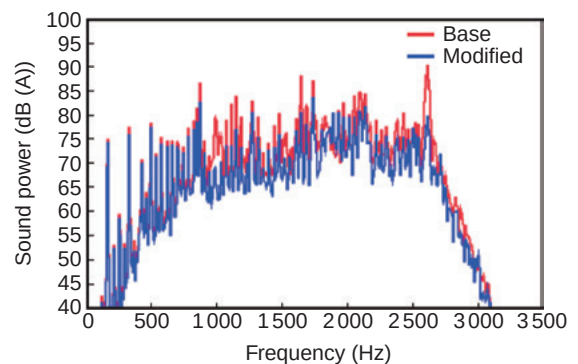
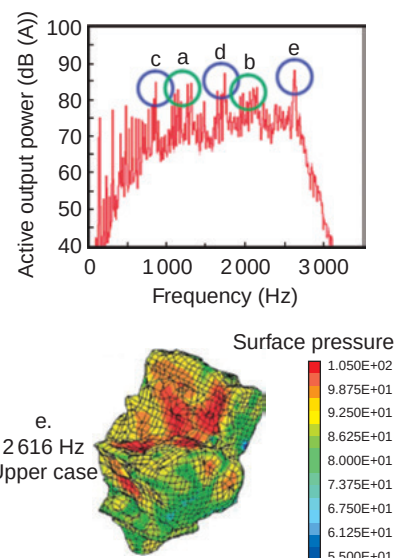


Fig. 17 Comparison of sound powers between base and modified case and cover



the overall sound radiating power. Figure 18 shows the changes in the surface sound pressure distribution before and after adopting the countermeasures. To confirm the effects of these countermeasures practically, a modified engine case and cover was prepared through the use of sand molding. The effect of the countermeasures was confirmed by assembly. Running tests on four different sets of specifications were performed, shown by the circles in Fig. 19. As indicated in Fig. 20, a correlation could be seen between the reduction in calculated sound power and the reduction in the pass-by noise level. The pass-by noise amplitude

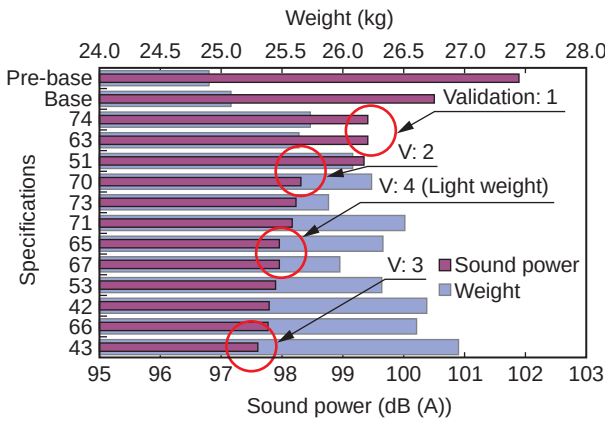


Fig. 19 Sound power versus weight

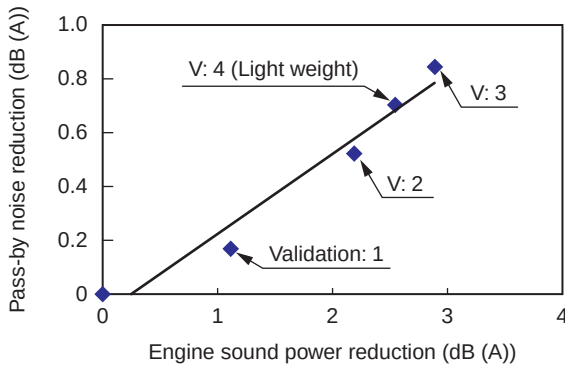


Fig. 20 Effect of engine sound power reduction

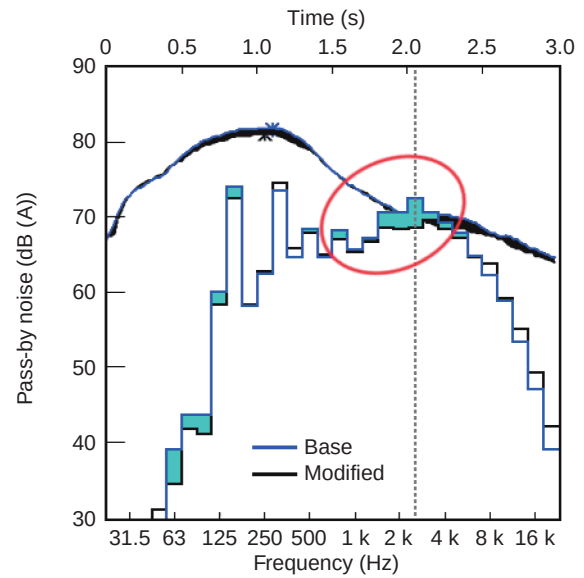


Fig. 21 Pass-by noise

at frequencies caused by engine noise was lowered, as shown in Fig. 21. The pass-by noise was reduced by approximately 0.8 dB. A reasonable result, even when observed from the perspective of the engine's contribution to pass-by noise.

The reduction technique used can be grasped through experience as well as understood through the analyses performed as follows.

- By making sure that there is no coincidence between the resonant frequency of each part and its gear meshing frequency.
- By dispersing the resonant frequencies of various parts such that the peaks do not overlap.
- By increasing the rigidity of each weak point to reduce vibration levels.
- By trying to increase the first resonance frequency as much as possible for gear meshing frequencies below 1 kHz.
- By adding mass for lowering the base response level.

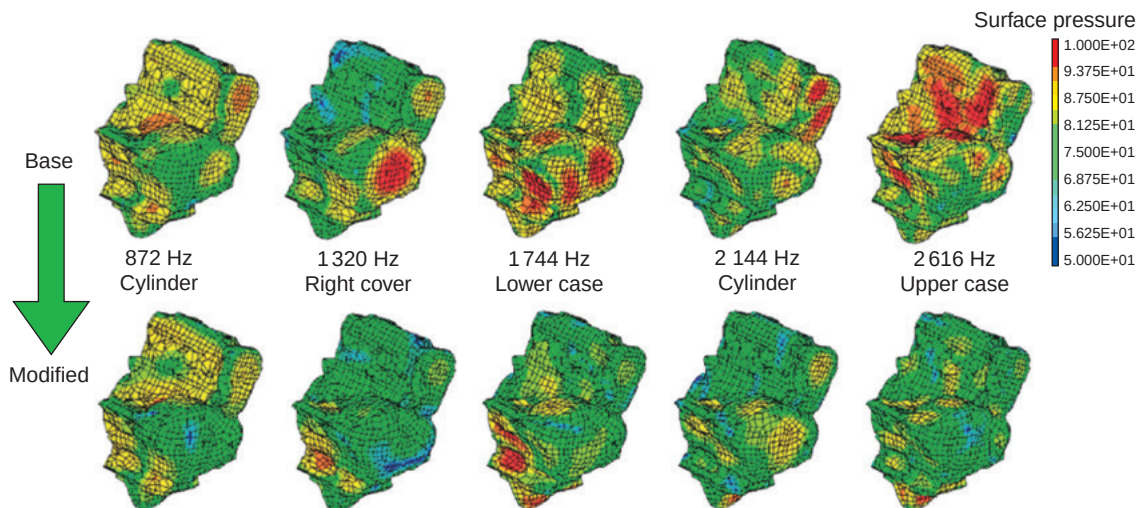


Fig. 18 Comparison of sound pressure distributions between base and modified case and cover

6. Conclusions

- (1) Conventional countermeasures against noise radiation were generally implemented through trial and error by identifying the vibration-producing parts using a stethoscope or accelerometer. However, with the authors' new technique, quantitative judgments on the effect of countermeasures can be made theoretically.
- (2) After preparing modified cases and covers, results verifying pass-by noise reduction were obtained according to the measures adopted. The effectiveness of the accompanying noise radiation analysis was verified.

Simulations were performed using measurement results of bearing forces on the actual engine. In the future, we plan to incorporate parts in the engine, estimate bearing forces that are due to meshing forces from gear transmission errors or combustion pressures, and calculate the noise radiation in the drawing stage.

References

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