

Development of the Exhaust System Radiation Noise Simulation Technology Using Exhaust Pulsation

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ABSTRACT

The purpose of this paper is to propose a simulation method using Finite Element Model in order to predict exhaust radiation noise. This newly developed method predicts the radiation noise in a sound field located away from the vibrating surface of structure by coupling calculation between fluid exhaust gas and exhaust parts. By applying measurement values to boundary conditions, accurate radiation noise from the exhaust system is obtained, and the noise causing parts and the mechanism are clarified. This method allows for the investigation of exhaust radiation noise at an early stage in development. Also, the accuracy of measurement values using boundary conditions is improved by introducing semiconductor pressure sensors with 1-MHz sampling and a laser Doppler velocimeter. Furthermore, the measurement results showed the existence of a shock wave during an exhaust pulsation.

1. Introduction

As public consciousness continues to be aroused to environmental issues, there have been increasing demands for quieter automobiles.

Exhaust radiation noise is part of the noise from automobiles. Its main cause is the pulsation of the exhaust gas, which excites vibration of the surfaces of the exhaust piping and muffler. This vibration is finally transmitted as radiation noise.

Meanwhile, another key goal in automobile development is weight reduction, in order to reduce CO₂, but weight reductions tend to exacerbate radiation noise.

It is essential to further develop technologies for simulation and prediction, key components of Computer Aided Engineering (CAE). In the pursuit to balance reduced radiated noise and weight reduction, CAE has been instrumental in improving noise and vibration (NV) reduction in the exhaust system.

Below, the authors describe a method we developed for predicting radiation noise with good precision. It is based on determining the appropriate inputs by measurements taken of the exhaust-pulse. These inputs were then applied to the numerical model to find surface vibrations of the exhaust piping and muffler, yielding radiation noise.

2. Outline of Experimental Technology

2.1. Measurements of Exhaust-pulse

The exhaust-pulse emitted by the engine was the input applied to the numerical model. Since it is presently difficult to directly calculate the exhaust-pulse with any precision, it was measured directly and applied to the numerical model as a boundary condition.

The following were the measured items and procedures:

(1) Flow velocity measurements inside the exhaust piping:

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laser doppler velocimeter (LDV)

- (2) Pressure measurements inside the exhaust pipe: semiconductor pressure sensors, manometer
- (3) Temperature measurements inside the exhaust piping: thermocouple

These measurements were also taken in an anechoic room in order to verify the correlation between them and the radiation noise, as described later. Also, an isolated exhaust system was constructed, as shown in Fig. 1, in order to observe exhaust-pulse due only to the exhaust system. A flexible joint was inserted between the engine outlet and the pipe to eliminate vibration input from the engine.

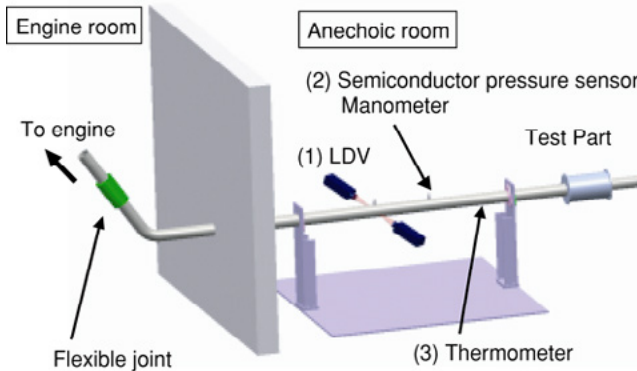


Fig. 1 Exhaust pulsation measurement conditions

2.2. Exhaust-pulse in Pressure Measurements

In order to increase the resolution of the exhaust-pulse waveforms, semiconductor sensors with a highly responsive sampling rates of 1 MHz were used. The results are shown in Fig. 2. Discontinuities are seen in the results; these show the presence of shock waves.

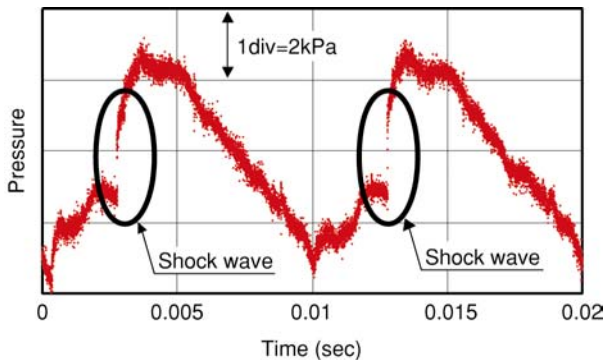


Fig. 2 Pressure of the exhaust pulsation

It was necessary to model the behavior of these shockwaves in order to take precise measurements of the exhaust system phenomena.

2.3. Observations with Shock Wave Generator

For this research, it was only necessary to create a shock wave of a certain level. The shock tube generator shown in Fig. 3 was used to simulate one exhaust process from a single-cylinder of the engine. This apparatus released compressed air from a pressurized tank at a maximum of 1 MPa through a valve down into the exhaust system. The valve was calibrated to mimic the speed of the engine exhaust valves. The chamber behind the shock wave

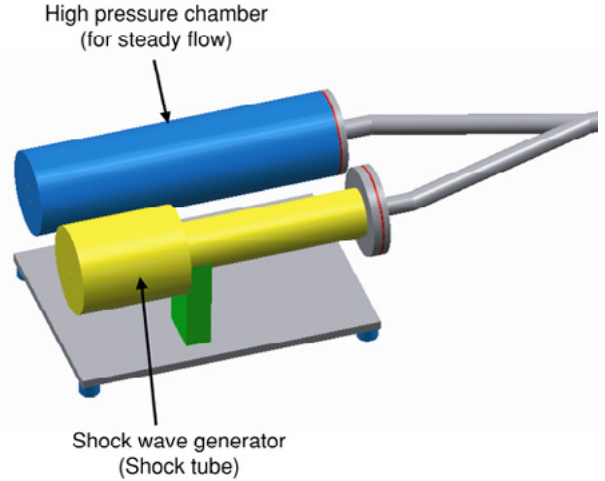


Fig. 3 Shock tube device

generator supplied a constant flow of air to smooth the flow from the compressor and enable measurements of flow velocity with the LDV.

The layout of the entire measurement system is shown in Fig. 4.

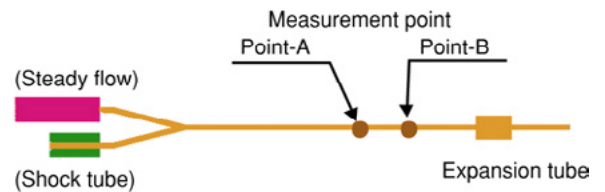


Fig. 4 Block diagram of the whole measurement

The shock wave grows as it proceeds down the exhaust pipe. The Mach number of the shock wave can be calculated by taking pressure measurements at two separated points in the pipe.

The shock wave was discontinuous and contained steady components over a wide spectrum, from low to high frequencies. Therefore, the output signal was processed to remove noise due to sensor resonance and other sources. The pressures were calculated using Eq. (1).

$$\frac{P_2}{P_1} = 1 + \frac{2\gamma}{\gamma+1} (M^2 - 1) \quad (1)$$

(P_1 : Pressure before passage of shock wave P_2 : Pressure after passage of shock wave γ : Ratio of specific heat M : Mach number)

The obtained pressure waveform is shown in Fig. 5.

The flow velocity in the air can be approximated using Eq. (2), assuming that the flow occurs only after passage of the shock wave generated by a traveling wave⁽¹⁾.

Figure 6 compares the pressures observed with the LDV to the approximation.

$$\Delta u = \frac{20.05 \sqrt{T_2} \Delta P}{\gamma \cdot P_2} \quad (2)$$

(T_2 : Temperature after passage of shock wave)

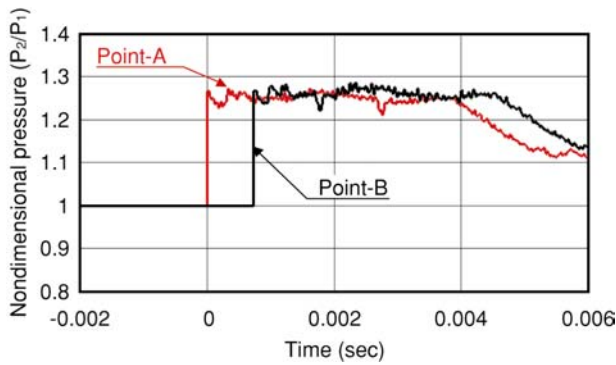


Fig. 5 Pressure wave form

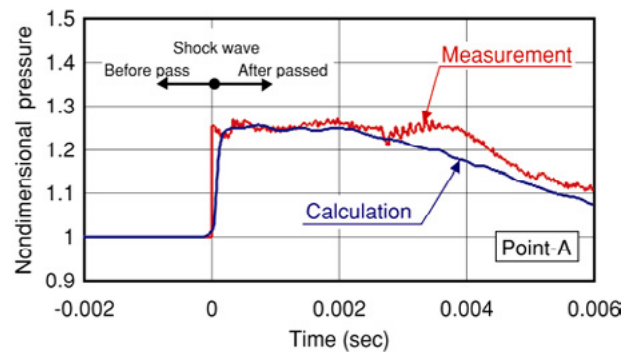


Fig. 7 Pressure comparison

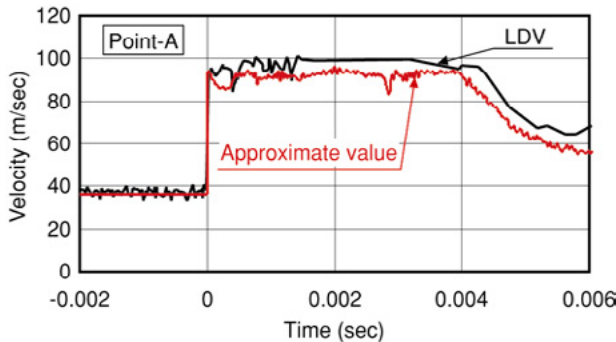


Fig. 6 Velocity comparison between the LDV and approximate value

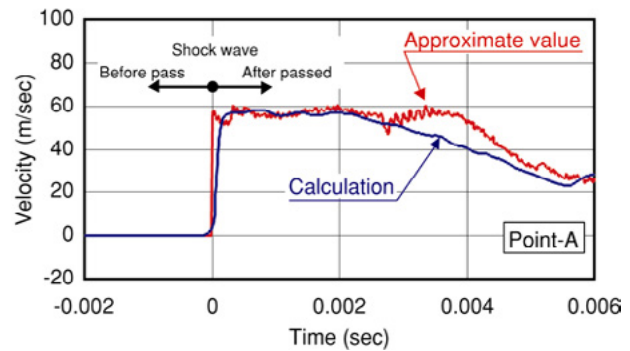


Fig. 8 Velocity comparison

The above methods enabled generation and measurement of shock waves to obtain a correlation with the calculations. It was also found that the calculations were reasonably precise approximations of the measured velocities with the LDV.

3. Outline of Calculation Technologies

3.1. Procedure of Numerical Calculation

RADIOSS-CFD, a package capable of the structural coupling calculations for fluids undergoing shock waves, was used for these calculations. Arbitrary Lagrangian Eulerian (ALE) coupling analysis was used for the fluid and structural system⁽³⁾.

The following section explains the procedure for verifying the correlation between the calculations and the observations.

The authors verified the results of the fluid analysis, structural analysis, and acoustic analysis first; and then the precision of the coupling analysis for the fluid, structural, and acoustic results were verified.

3.2. Fluid Analysis Precision Verification

3.2.1. Verification of shock wave pressure and flow velocity precision of shock wave

An investigation of the measurements in Fig. 4 taken at Point A in the shock wave tube was carried out. The numerical model had about 120,000 elements (mesh size under 2 mm).

Figure 7 shows the waveforms of the calculated and observed pressure waves, and Fig. 8, a comparison of the flow velocities of the calculated and approximated value from the pressure wave.

A steady flow was added to allow measurement of the flow velocity; this was subtracted from the results in the figures stated.

Zero seconds on the time axis of each figure represents the time of the surface of discontinuity in the shock wave. That represents the border between the pressure or velocity at times before passage of a shock wave, on the negative time side, and after, on the positive side.

The calculations showed very low error, within 3% for the maximum pressure ratio and the flow velocity of the surface of discontinuity in the shock wave. The pressure ratio after passage of the shock wave also had a low error.

These results indicated that the shock wave component and the exhaust-pulse component in a single exhaust process could be simulated with sufficient precision.

3.2.2. Exhaust-pulse pressure and flow velocity precision verification

It is possible to construct a complete model of the exhaust system, but at the development stage, in order to save time, components are modeled by themselves, as if cut out of the system.

As shown in Fig. 9, a targeted test piece is placed in the middle of exhaust systems. It is confirmed that either pressure or velocity should be set as outlet boundary condition of the test piece.

In other words, components can be analyzed by themselves in succession down the exhaust stream and the same final result will be calculated as if all the system were analyzed simultaneously.

The physical quantities of the input (exhaust-pulse), were calculated from energy, density and flow velocity because it was a compressible fluid.

The flow velocity could be directly measured with the LDV. The energy and density of the flow was calculated using the measured pressures and temperatures as follows:

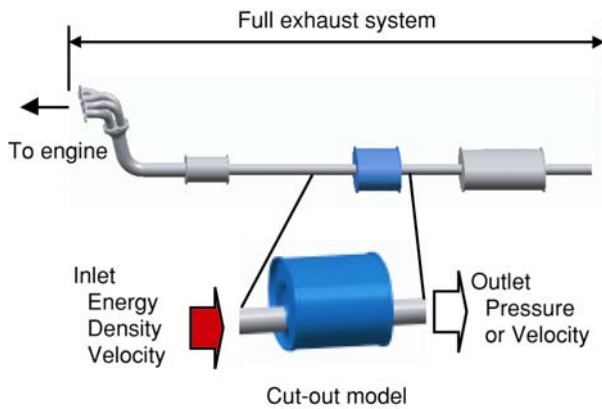


Fig. 9 Calculation model

Generally, the flow was assumed to be isentropic, in which energy exchange is in equilibrium. If pressure and ratio of specific heats are represented by P and γ , respectively, the energy E is calculated with the following equations:

$$E = \frac{P}{\gamma - 1} \quad (3)$$

$$P \cdot \nu^\gamma = \text{Constant} \quad (4)$$

$$P \nu = RT \quad (5)$$

(ν : Specific volume R : Gas constant)

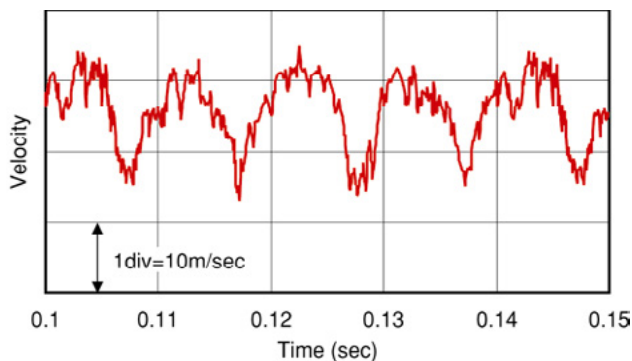


Fig. 10 Measured velocity inside of the exhaust pipe (3,000 rpm WOT)

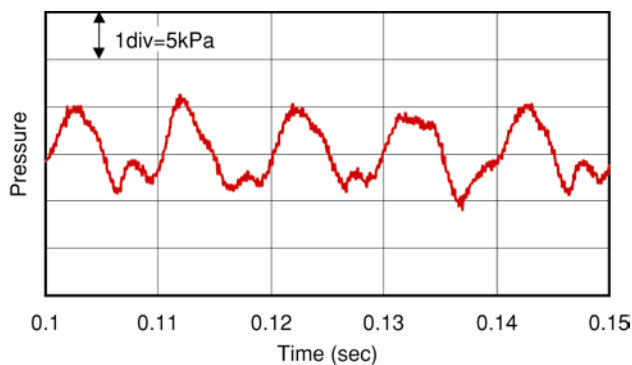


Fig. 11 Measured pressure inside of the exhaust pipe (3,000 rpm WOT)

Equation (6) is derived from Eqs. (4) and (5), and the density was calculated with Eq. (7).

$$P \cdot \left(\frac{RT}{P} \right)^\gamma = \text{Constant} \quad (6)$$

$$\rho = \left(\frac{P}{\text{Constant}} \right)^{\frac{1}{\gamma}} \quad (7)$$

The value of the constant in Eq. (6) was estimated using average values for the measured pressure and temperature. A test piece was placed at the end of the exhaust system in this experiment. The correlation between experimental and numerically predicted conditions was verified using a mass production engine motored with a dynamo. For reference, Fig. 10 and Fig. 11 show the results for flow velocity and pressure, respectively, in the exhaust pipe. The location of measurements here was the entrance to the muffler.

Figure 12 shows a numerical model of a simple muffler.

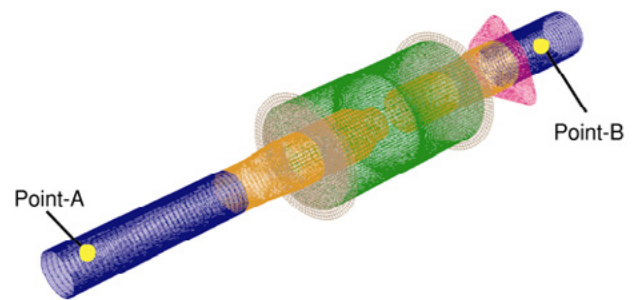


Fig. 12 Simple muffler calculation model

This contains about 450,000 elements (mesh size less than 2 mm). The pressure and temperature were used to find the energy and density, which were used as inputs to this model with the flow velocity.

Figures 13 and 14 show comparisons of the calculated and measured pressure waves upstream and downstream, respectively, of the muffler.

It is clear that the progress of the exhaust-pulse from the mouth of the muffler through the internal structure to the exit is a relatively precise prediction.

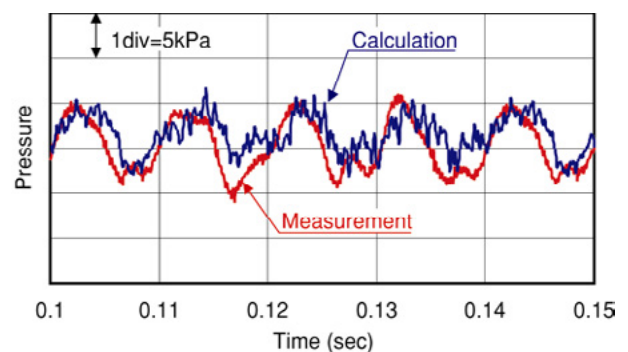


Fig. 13 Comparison of the pressure within the pipe (Point-A)

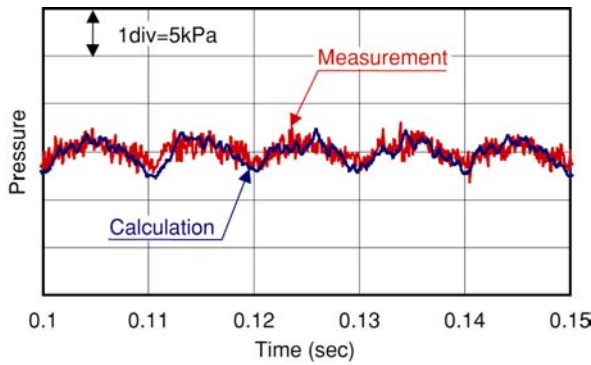


Fig. 14 Comparison of the pressure within the pipe (Point-B)

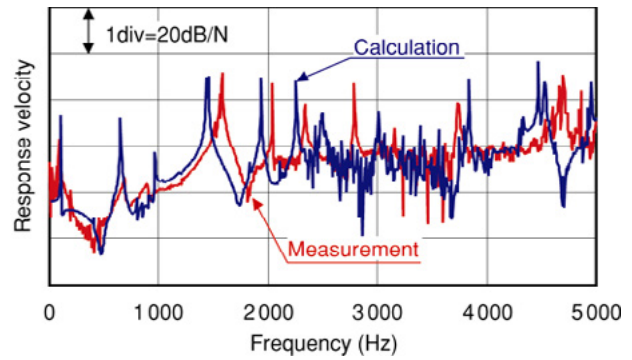


Fig. 16 Comparison of surface vibration on the muffler

3.3. Structural and Acoustic Analysis Precision Verification

Figure 15 shows the FE model of the structure and how measurements were taken.

Hammering excitation, a previously established method, was used for comparison to verify the validity of this model.

The mass of each component in the numerical model was checked; then, all components were combined simply into a model, including the welding beads holding them together. For the comparison examination, the Frequency Response Function (FRF) of the surface, the level of response velocity of the surface vibration on a muffler to a unit load, was calculated. The results of that test are shown in Fig. 16.

Hammering excitation was chosen over acoustic analysis as a method of verifying radiation noise generated from surface vibration of the muffler because it is difficult to carry out an acoustic analysis alone, and since the results of the above structural analysis were so accurate.

The numerical model is shown in Fig. 17. This consists of the model of the interior air within the muffler structure and also a volume of the air surrounding the muffler.

The comparison test used a unit load and the resulting radiated noise measured at 100 mm from the muffler surface. The results are shown in Fig. 18.

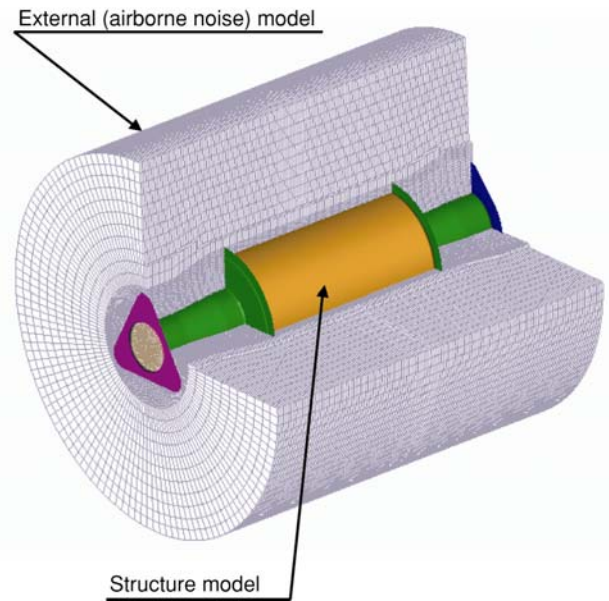


Fig. 17 Airborne noise model

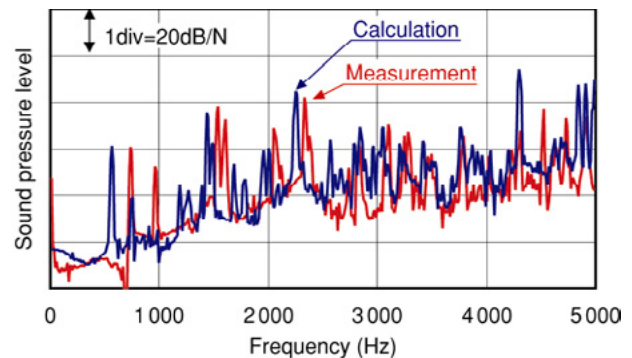


Fig. 18 The result of the radiation noise level

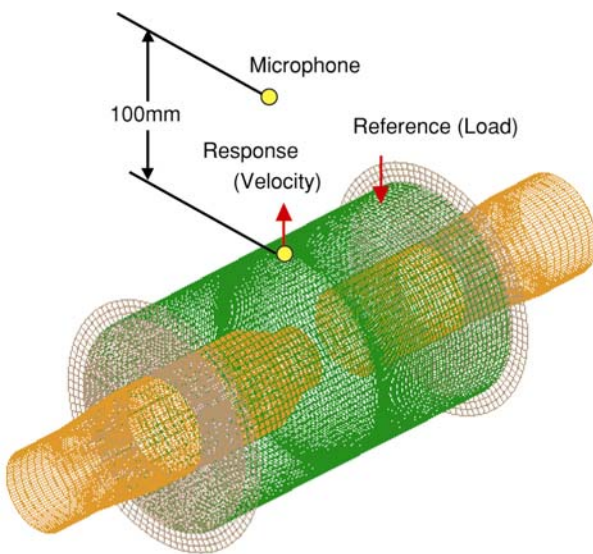


Fig. 15 FE model and measurement condition

In Fig. 16, the peak frequency shows an error of about 70 Hz.

The reason for this was that the model used simple models of the weld beads; if the shapes and material properties of the weld beads had been more accurate, the results would probably have been closer to measured results.

However, this model, with simple welds, still provides a fairly accurate prediction of the response waveform trends. In consideration of the difficulty of modeling the welds correctly, this modeling method seems to have a lot of potential for further development.

As shown by the results in Figs. 16 and 18, simulating the exterior airborne noise and simultaneously taking into account the welding bead stiffness in the structure, a very accurate prediction of radiation noise was achieved using the FE model.

3.4. Exhaust Radiation Noise Precision Verification

Exhaust-pulse from motoring was applied to the obtained boundary conditions and fluid, structural model. Calculations for the fluid conditions and structural vibration in the exhaust pipe and radiation noise at the exterior were executed in the model.

Figure 19 shows the FE model, radiation noise calculations and the location for measurements. The calculated and measured results are in Fig. 20.

It is clear that the predicted exhaust radiation noise matches with the observed noise quite precisely.

These results will allow analysis of many phenomena causing radiation noise, including exhaust-pulse and structure-borne ones.

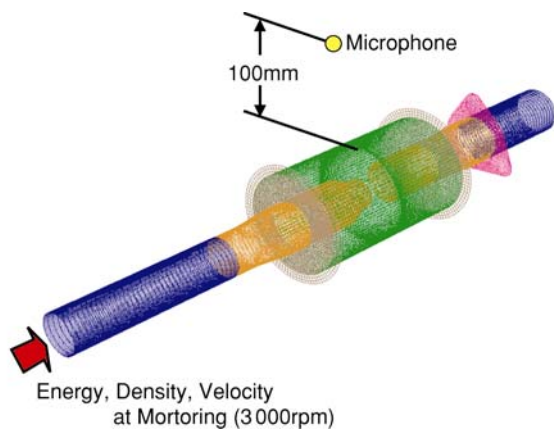


Fig. 19 Radiation noise measurement by the motoring

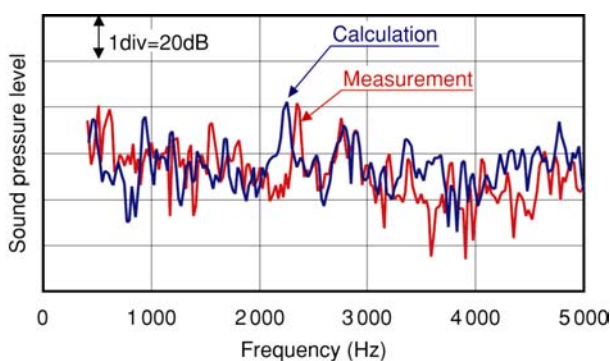


Fig. 20 The result of the radiation noise by the motoring

4. Application to Development

4.1. Example of Analysis of Phenomena

An analysis was carried out using the present method to examine the mechanisms of emission of a 1,400 Hz noise from the simple muffler surface when it is subjected to a

steady flow in flow test.

The calculated and observed surface vibrations at Point-A in the simple muffler shown in Fig. 21 are provided in Fig. 22.

Since the trends of the waveforms of the peak surface vibrations are similar, it follows that the fluid behavior of the input source is being well reproduced.

Therefore, the focus was shifted to the fluid phenomena within the muffler; these were examined with a numerical model. Pressure contours within the muffler are shown in Fig. 23, and the fluctuations of the pressure at Point-B in Fig. 23 are given in Fig. 24.

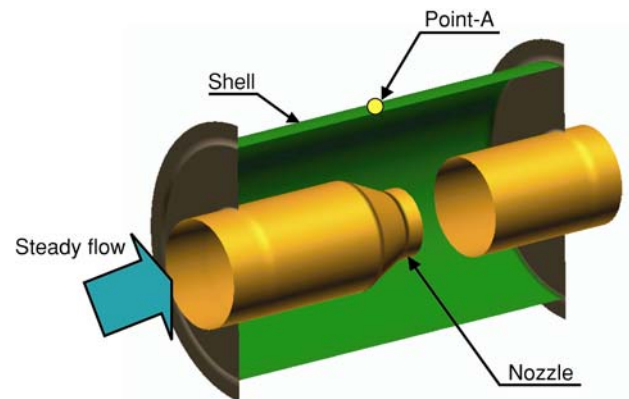


Fig. 21 Inside structure of the simple muffler

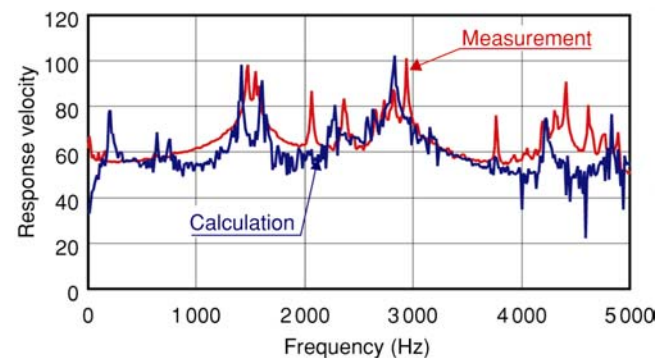


Fig. 22 The result of the surface vibration (Point-A)

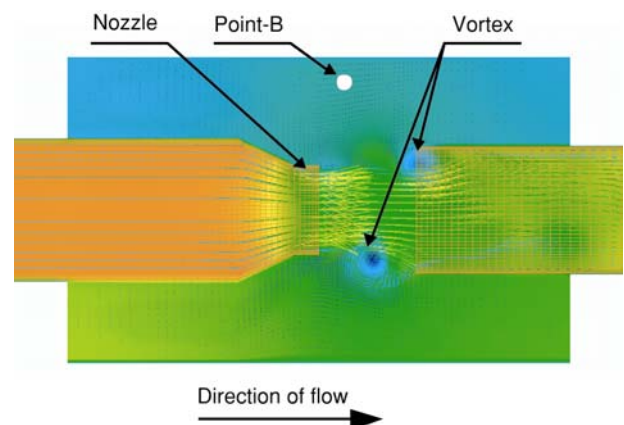


Fig. 23 Pressure contour inside of the muffler

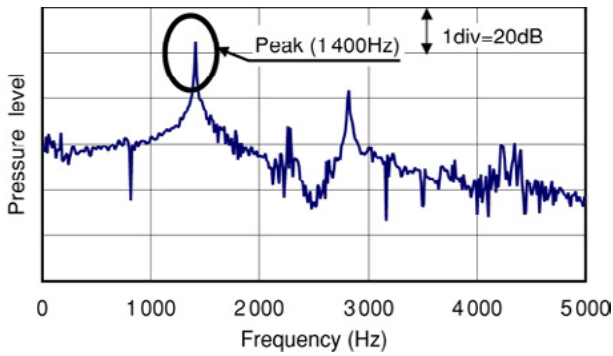


Fig. 24 Pressure inside of the muffler (Point-B)

The results indicate that vortices occur at the narrow end of the funnel portion and disperse in the downstream tube. Furthermore, the rate of pressure fluctuation can be seen here as approximately 1,400 Hz.

Thus, it was determined how to model the fluid behavior inside a muffler, which has been considered difficult to reproduce by experiment, and causative factors in the behavior have been identified.⁽⁴⁾

4.2. Application of Procedure to Reduce Exhaust Radiation noise

It was attempted to reduce the exhaust-pulse with an upstream sub-muffler in order to reduce the radiation noise from an actual main muffler. As shown in Fig. 25, modifying the volume of the expansion chamber of the sub-muffler, making it larger, attenuated the exhaust-pulse. The calculated fluctuations in exhaust-pulse at the exit from the sub-muffler (Point-A) are shown in Fig. 26.

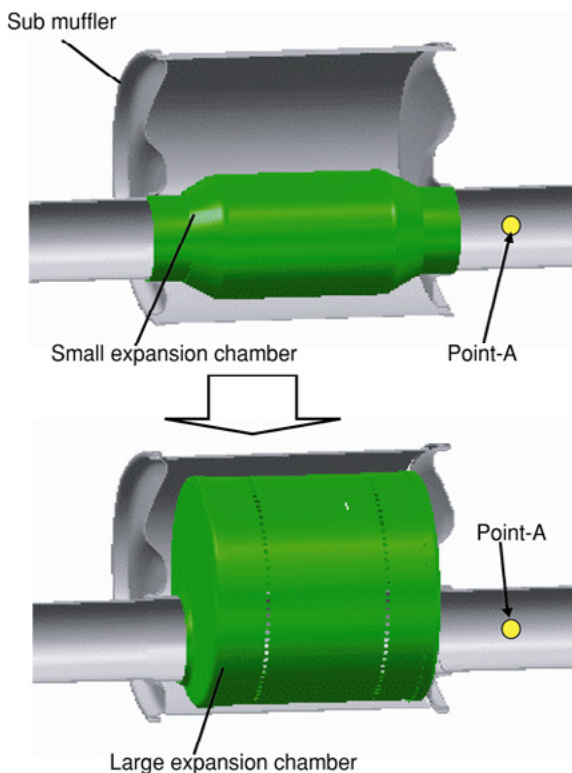


Fig. 25 Expansion chamber inside of the sub muffler

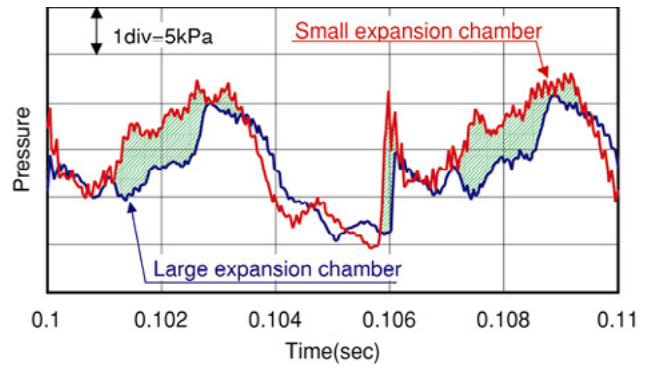


Fig. 26 Comparison of pressure at the sub muffler exit (Point-A)

As indicated by the shaded area in the above graph, the exhaust pressure has been decreased.

A test sub-muffler was constructed on the basis of these results and the radiation noise from the main muffler of a bench-mounted engine was observed. Figure 27 shows the layout of the test including the location of noise measurements, and Fig. 28, the observed results.

The shaded area in Fig. 28 shows how much the sound pressure level was reduced by the addition of the sub muffler.

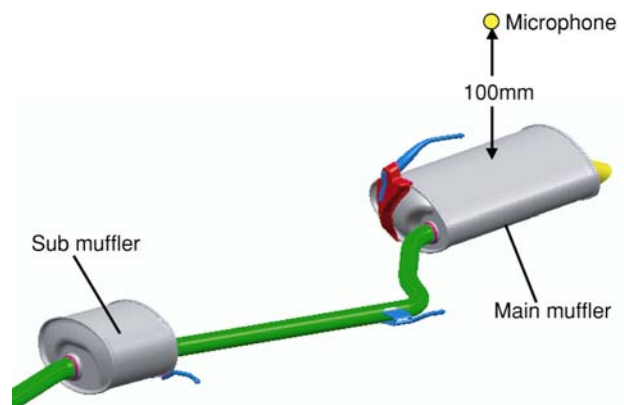


Fig. 27 Measurement of radiation noise from the main muffler

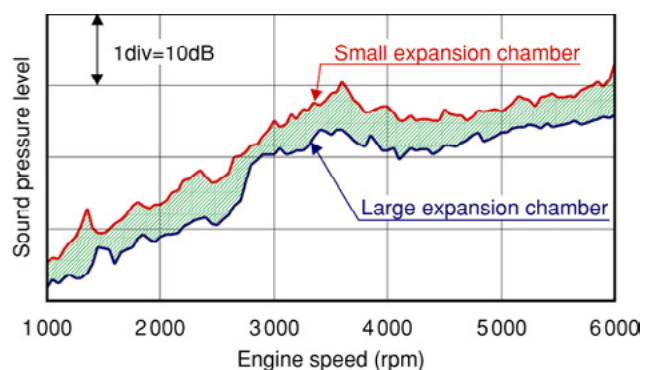


Fig. 28 Comparison of radiation noise from the main muffler

5. Conclusion

A technology for predicting the acoustic energy radiated from the exhaust piping and muffler was developed by the following processes:

- (1) Fluid analysis of the interior of the exhaust system requires accounting for three factors: Exhaust gas energy, density and flow velocity. The energy and density can be calculated using measurements of the pressures and temperatures in the piping, and the velocity can be found with a LDV.
- (2) Structural analysis of the exhaust system was carried out, including verification of the FRF, mode shape, and modeling of welds.
- (3) For acoustic analysis of the exhaust system, a precise prediction of radiation noise was made by creating a FE model which included exterior air surrounding the muffler.

The three processes above made it possible to analyze the complicated phenomena of exhaust radiation noise.

6. Afterword

The tools described above, which allow comparatively quick analysis of noise parameters, from exhaust pipe vibration to radiation noise calculations, were supplied to developers in order to encourage the spread of this technology.

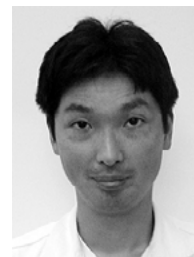
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