

Computer-aided Engineering Technique for the Reduction of Exhaust System Noise and Back Pressure

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ABSTRACT

A technique has been established for predicting the effects of changing cold end parts in an exhaust system on engine output and exhaust discharge noise. This technique enables the sharing of such parameters as velocity and pressure due to the application of flow analysis and acoustic analysis. One-dimensional compressible fluid analysis was used to predict engine output, and a value derived by steady flow analysis using the three-dimensional finite volume method (FVM) was introduced for the effective flow area, which is one of the parameters. Further, it was clarified that exhaust discharge noise is determined by the sum of pulsating noise and flow generated noise. The pulsating noise was predicted by the application of acoustic analysis using the three-dimensional finite element method (FEM). For the FEM input boundary conditions, precision verification was carried out using both pressure fluctuation values derived by compressible fluid analysis and measured values from an actual vehicle, while velocity derivable by steady flow analysis was introduced as a parameter for flow-generated noise prediction. As a result, it was possible to predict the variations of engine output caused by changes in the exhaust system with an accuracy of $\pm 2\%$, and to predict exhaust discharge noise within ± 2 dB. This has made it possible to optimize muffler design by implementing three-dimensional computer-aided design (CAD) modeling to evaluate parts performance from the initial layout investigation stages.

1. Introduction

In conventional exhaust development, one of the main issues has been how to determine specifications so that the high-level balance required between the opposing requirements for high performance engine response and high muffling performance is achieved.

In recent years, there has been an increasing demand for the development of computer-aided Engineering (CAE) technology that can evaluate the function and performance of a 3D CAD model in the early stages of development before prototype parts are made.

This report gives details of the calculation method developed to meet the above needs for both acoustic and flow engineering. The method utilizes the finite element method (FEM) and computational fluid dynamics (CFD), etc.

2. Analysis Method

The analysis flow for exhaust noise analysis and engine output analysis is given in Fig. 1. The details of the

procedure are given in later sections. The aim of this project was to bring the accuracy of analysis up to a level where exhaust system performance can be evaluated without having to create prototype parts. The range for changes to parts handled by this project is highlighted by diagonal lines in Figure 1. These parts are positioned after the catalytic converter in the muffler system and are called the “cold end exhaust”.

Accordingly, both the calculated and actual values for the area immediately behind the catalytic converter were used for the physical values of pressure, velocity, temperature, etc., which are the analysis input conditions. Then, by varying the muffler system characteristics it was possible to study what effects the changes have on exhaust discharge noise, and on engine output.

3. Steady Flow Analysis

The general-purpose VECTIS code was used in the 3D steady flow analysis.

In terms of methods for digitization, the FVM (finite volume method) was employed for the flow parts, and the

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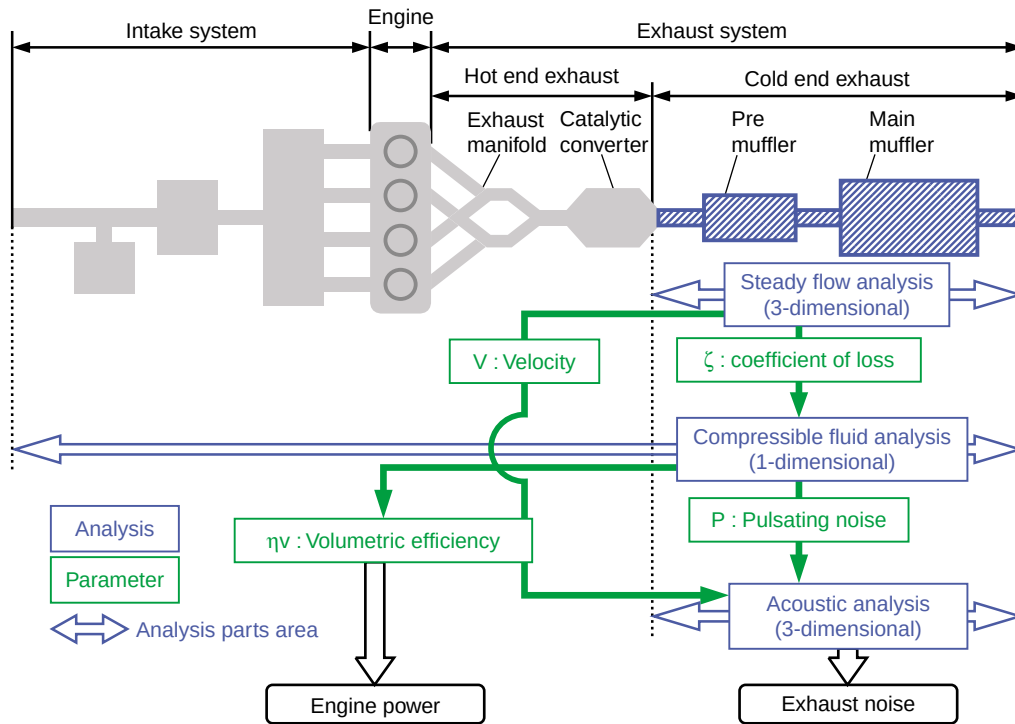


Fig. 1 Flowchart of analysis

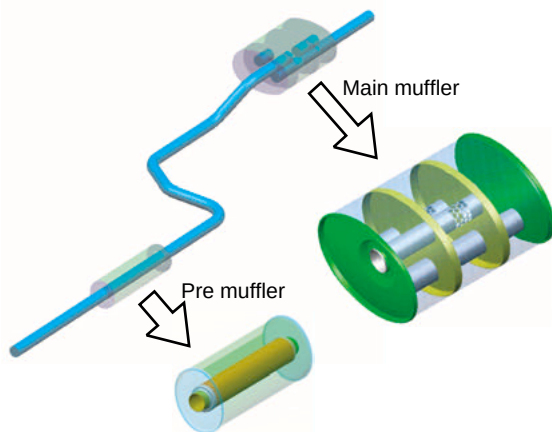


Fig. 2 Steady flow analysis model

SIMPLE method was applied for the velocity and pressure location calculations. The standard $k - \epsilon$ model was used as the turbulence model.

The analysis model used for verifications – an exhaust system with a pre-muffler and main muffler placed along the exhaust pipe – is shown in Fig. 2.

Fig. 3 shows a comparison of the main muffler configurations applied for this verification. Fig. 4 shows a representative FVM mesh model (a) and velocity analysis FVM example (b).

Boundary conditions were set as follows: pressure was provided at the inlet (1 to 6 kPa), the air was released at the outlet, and the temperature was a normal 27°C⁽¹⁾.

$$Q = S \cdot k \sqrt{\frac{2g\Delta P}{\zeta}} \quad (1)$$

Q : steady flow amount
 S : outlet section area
 ΔP : pressure difference
 ζ : loss coefficient
 k : flow rate coefficient

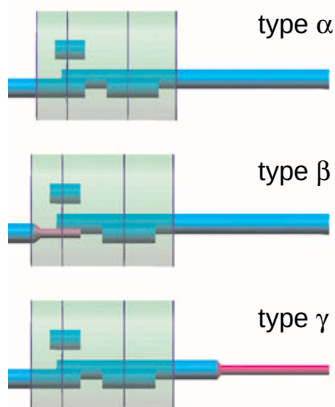


Fig. 3 Comparison model types

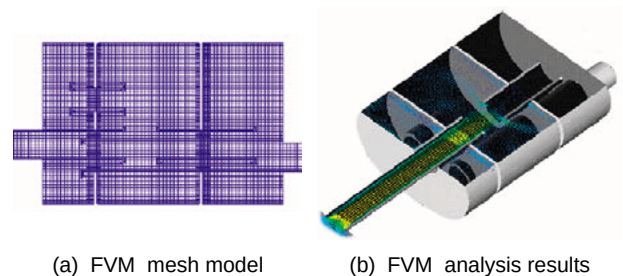


Fig. 4 Example of FVM mesh and analysis results

The following items were kept in mind when improving the accuracy of the steady flow analysis.

- (I) Method for modeling the air release area
- (II) Small shapes (punched holes)
- (III) Method for stabilizing digitalization
- (IV) Dependency of mesh size

Fig. 5 gives the results of comparison between steady flow rate analysis as determined by Equation (1), and the measured values for a flow rate test piece.

There is a correlation between the values, and calculation accuracy shows the favorable result of no more than 2% error.

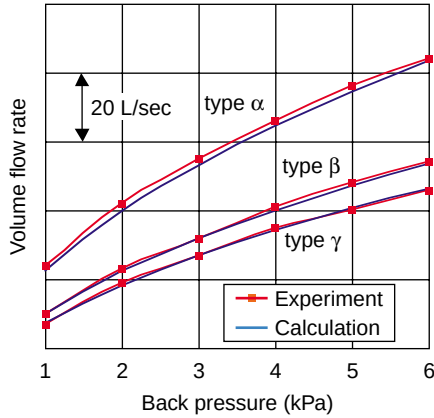


Fig. 5 Comparison of experiment and calculation results

4. Volumetric Efficiency Analysis

1D transient compression flow simulation was used for volumetric efficiency calculations.

Here, flow loss in the muffler can be converted, as shown in Fig. 6, into a single uniform pipe loss. The effective flow area (Z), which is required for volumetric efficiency analysis, was determined using Equation (2) and the flow rate (Q) determined in the aforementioned steady flow analysis.

$$Z = Q \sqrt{\rho / 2\Delta P} \quad (2)$$

Z : effective flow area
 Q : steady flow amount
 ΔP : pressure difference
 ρ : density

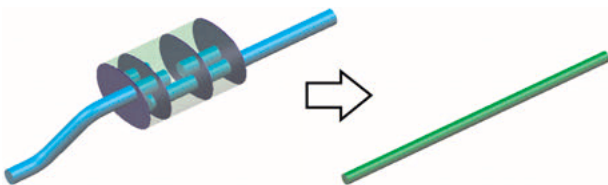


Fig. 6 Conversion of loss coefficients

Also, pipe length and muffler capacity were used for 3D modeling values. Fig. 7 gives an example of one dimensional volumetric efficiency analysis.

$$P_s = 2 \pi \cdot N_e T / (60 \times 1000) \quad (\text{kW}) \quad (3)$$

$$T = k \cdot \eta_i \cdot \eta_m \cdot \eta_v \cdot V_{st} \quad (\text{N} \cdot \text{m}) \quad (4)$$

P_s : engine output
 N_e : rpm
 T : torque
 k : coefficient
 η_i : thermal efficiency
 η_m : mechanical efficiency
 η_v : volumetric efficiency
 V_{st} : cylinder capacity

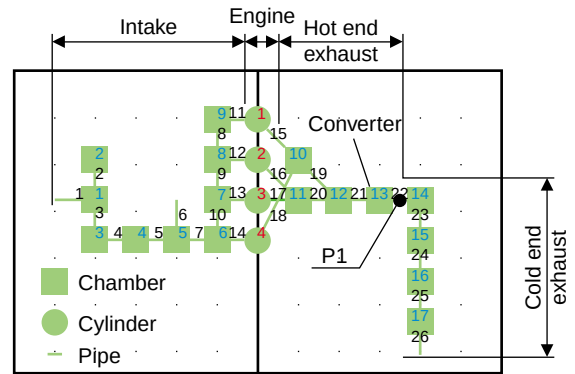


Fig. 7 One dimensional volumetric efficiency model

Fig. 8 shows a comparison between engine output and torque analysis results determined from Equations (3) and (4) and the results of measurements from engine bench testing.

There is a correlation between measured values and calculated values, and error is 2% or less. Thus, analysis accuracy is sufficient.

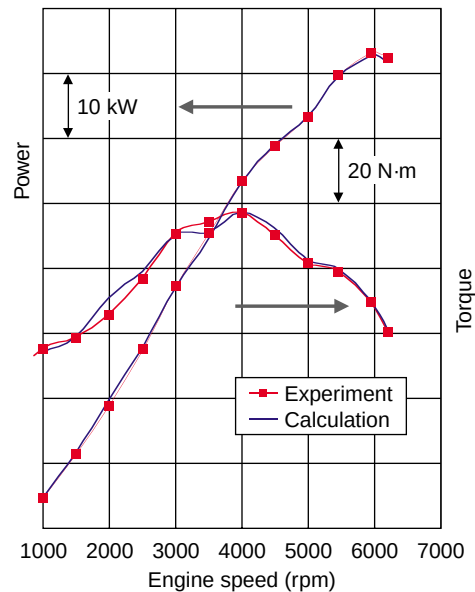


Fig. 8 Comparison of experiment and calculation

5. Exhaust Noise Analysis

5.1 Causes of Exhaust Discharge Noise

Exhaust discharge noise, as shown in Fig. 9, is comprised of pulsating noise (composed of basic order and harmonic components), resonance noise (due to the operation of amplitude within the exhaust pipe), and flow noise (from fluid motion).

A 4-cylinder engine was used in this project, and the effect of changes in muffler characteristics (structure and layout) on exhaust discharge noise pressure was verified through test and calculation results.

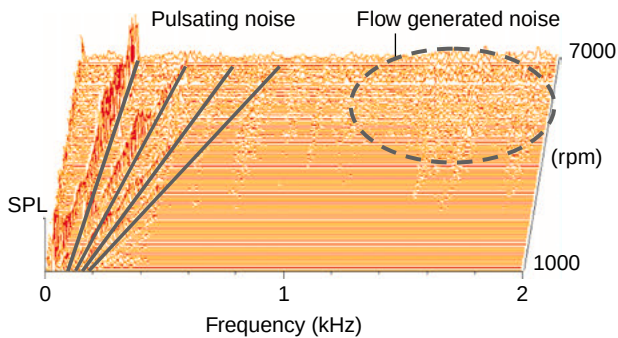


Fig. 9 Exhaust noise components

5.2 Analysis of Attenuation Amount using Acoustic Theory

5.2.1 Attenuation Amount Characteristics

First, verifications were carried out of the acoustic characteristics in a state where exhaust flow factors are eliminated. This was done by comparing actual measurements taken at normal temperature using an acoustic vibration tester and the results of calculations using FEM.

The general-purpose code SYSNOISE was used for the 3D acoustic flow analysis. Fig. 10 shows the analysis model and the corresponding FEM mesh model.

The following items were kept in mind when improving the accuracy of the attenuation analysis.

- (I) Wall surface boundary condition settings
- (II) Sound absorbing material characteristics
- (III) Small shapes (interference hole, gaps with mating parts)
- (IV) Release area boundary conditions
- (V) Dependency of mesh size

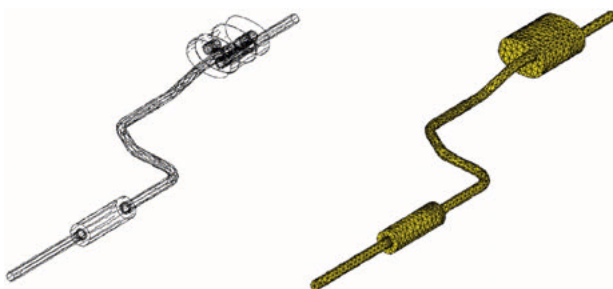


Fig. 10 Analysis model and mesh model

Fig. 11 gives an example of acoustic pressure mode analysis results. Equations (5) and (6) were used for attenuation amount calculations⁽²⁾.

$$X(p) = H_B(p) \cdot H_F(p) \cdot F(p) \quad (5)$$

$$ATT = 20 \text{LOG} \frac{|F(p)|}{|X(p)|} \quad (6)$$

That is, the response $X(p)$ to acoustic vibration input $F(p)$ could be calculated from the exhaust pipe release end transmission function $H_B(p)$, and the transmission function $H_F(p)$ that is determined by multiple physical property values and structures. $H_B(p)$ was calculated using the boundary element method, and $H_F(p)$ using the finite element method.

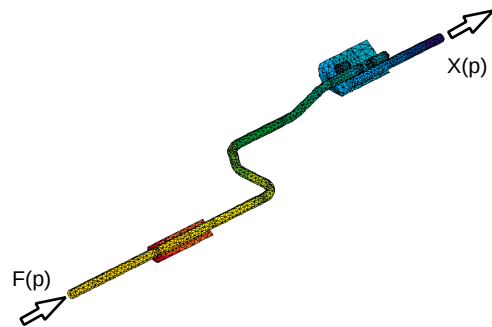


Fig. 11 Acoustic pressure mode (45Hz)

Fig. 12 shows the comparison between the attenuation amount measured using an acoustic vibration tester and the calculated results. The results are extremely accurate, within ± 2 dB, in the 0 to 500 Hz range.

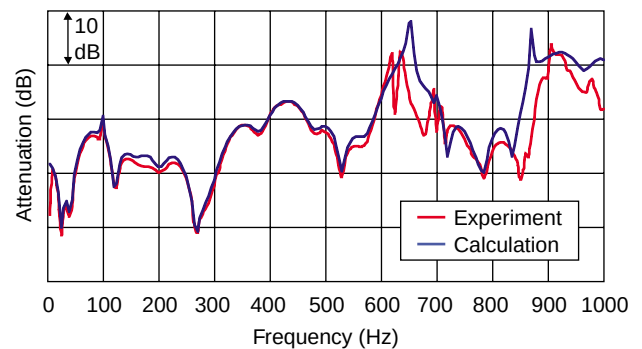
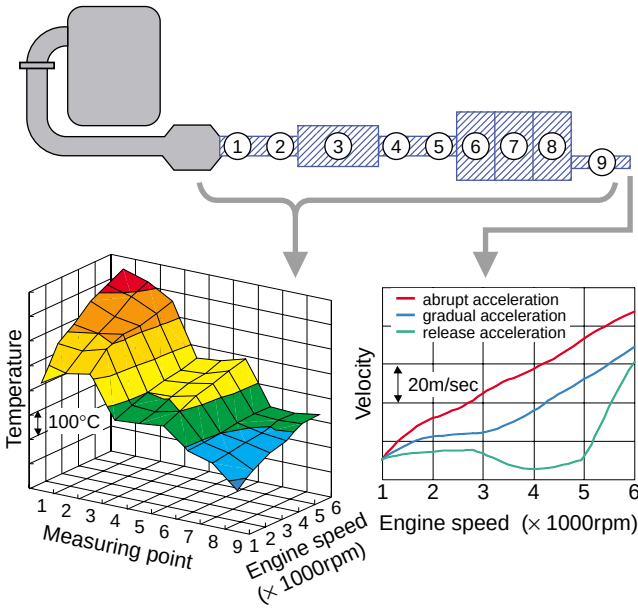


Fig. 12 Comparison of attenuation

5.2.2 Transmission Characteristics with Consideration of Heat and Exhaust Flow

When the engine is operating, the exhaust pipe is under high-pressure exhaust gas flow and high-temperature conditions of use, as shown in Fig. 13. Also, it is necessary to consider velocity and temperature parameters when analyzing pulsating noise.

In order to resolve the transient response problem concerning what kind of effect velocity and temperature have



(a) Temperature of muffler (b) Velocity of tail pipe

Fig. 13 Temperature and velocity measurements

on acoustic characteristics, the authors adopted the following equation of motion (7) ⁽³⁾.

$$[M]\{\ddot{x}(t)\} + [R]\{\dot{x}(t)\} + [K]\{x(t)\} = \{f(\theta^2)\} \quad (7)$$

[M]: rigidity matrix
 [R]: attenuation matrix
 [K]: mass matrix
 {x}: displacement vector
 {f}: vibration force vector

Here the right half of the formula is the velocity function, and the sound velocity and density corresponding to temperature changes are given as variables for each matrix. By introducing the solution of this equation of motion into the attenuation equation, it was possible to calculate the transmission function giving full consideration to velocity and temperature. Verifications were implemented, comparing the above results with measurement results from a vehicle chassis dynamo (Fig. 14).

The patterns used for the accuracy verification are recorded in Table 1.

In Pattern 1, the test vehicle's engine was turned off with the tires rotating in order to minimize temperature dependency. Fig. 15 shows a comparison of transmission function calculation results (2nd order) with changes only to the flow velocity parameter.

In Pattern 2, the test vehicle was in a working engine - throttle off mode in order to minimize the velocity dependency. Fig. 16 shows a comparison of transmission function calculation results (2nd order) with changes only to the temperature parameter.

Extremely high accuracy was obtained for both Patterns 1 and 2. Thus, it was proven that acoustic characteristics under changing temperature or velocity conditions can be determined using Equation (7).

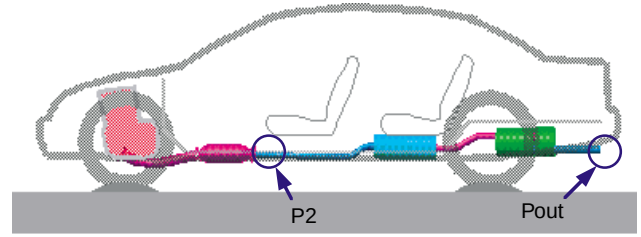


Fig. 14 Chassis dynamo test

Table 1 Comparison patterns

Pattern	Pattern 1	Pattern 2
Vehicle test mode	Motoring	Working engine & throttle off
Calculation condition	Change velocity parameter only	Change temperature parameter only

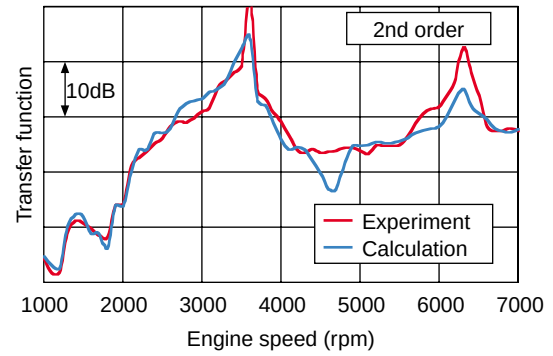


Fig. 15 Comparison of transfer functions

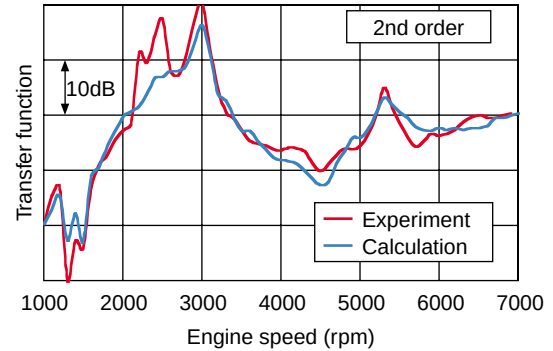


Fig. 16 Comparison of transfer functions

5.3 Pulsating Noise Analysis

In actual exhaust discharge noise analysis, it is necessary to consider both temperature and velocity parameters simultaneously.

A comparative verification of pulsating noise basic order with muffler a used in flow analysis was implemented in the abrupt acceleration mode.

Equation (7) was employed when calculating both temperature and velocity as transient response problems. The results of comparisons with the spectrum calculated from vehicle exhaust noise tests and with 2nd to 8th order pulsating noise are shown in Fig. 17 and Fig. 18, respectively.

The differences in both are greater with the higher frequencies. However, it is possible to obtain sufficiently accurate results when carrying out performance evaluation of parts without making prototype parts.

Using Equations (8) and (9), the exhaust pulsating noise overall value is calculated from two inlet boundary conditions: (1) pressure P1, determined in volumetric efficiency analysis; and, (2) pressure P2, measured in vehicle chassis testing.

$$X(p_n) = H_{v,T}(p_n) \cdot H_L(p) \cdot F(p_n) \quad (8)$$

$$SPL_n = 20 \log \frac{|X(p_n)|}{P_0} \quad (9)$$

$F(p)$: input sound pressure (exhaust source noise)

$X(p)$: response sound pressure (exhaust discharge noise)

$H_{v,T}(p)$: temperature, velocity transmission function

P_0 : minimum audible sound pressure

n : rpm

$H_L(p)$: distance attenuation

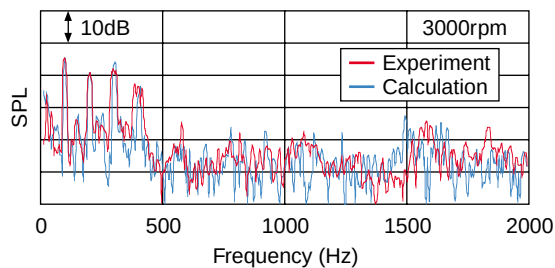


Fig. 17 Comparison of frequency spectrums

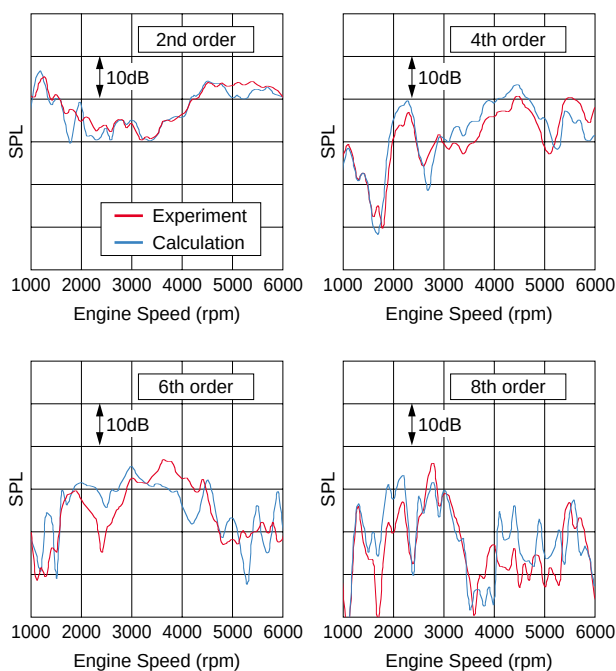


Fig. 18 Comparison of sound pressure levels

Highly accurate analysis results were obtained, with the accuracy of the results for (1) being ± 2 dB and (2) being ± 5 dB, as shown in Fig. 19.

The above are results with a muffler system that considers fully maintaining exhaust pipe diameter and generating a relatively small flow noise. It was thus judged possible to calculate exhaust discharge noise with this sort of exhaust system by adding a spectrum for each engine speed.

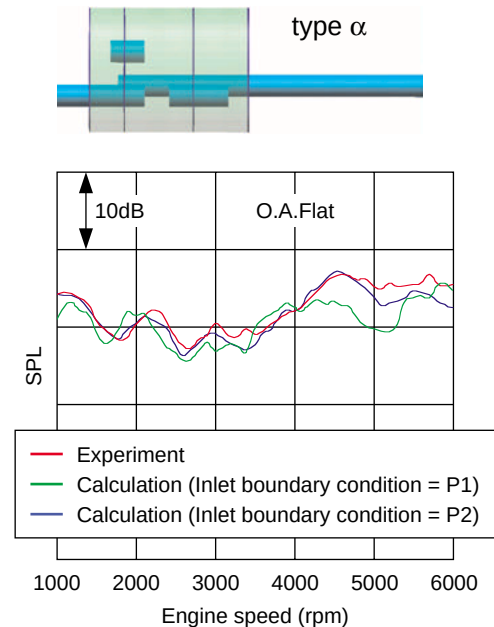


Fig. 19 Comparison of sound pressure levels

5.4 Flow Noise Analysis

With actual exhaust systems (Fig. 24 and 25 show types b and g), it is not uncommon for some portion of the exhaust pipe to be narrow in order to maintain attenuation mainly in the low frequency range. In these cases, it can be seen that the effect of the flow noise on the exhaust discharge noise grows larger as the pipe becomes narrower.

Generally speaking, flow noises are turbulent noises generated from eddies or impacts when high velocity exhaust gas flow passes through the muffler, and jet noises are generated when exhaust gasses are released from the end of the tailpipe. (See Fig. 20.) The frequency of the noise covers a wide range like white noise, but the flow velocity effect is mainly on the high frequency side.

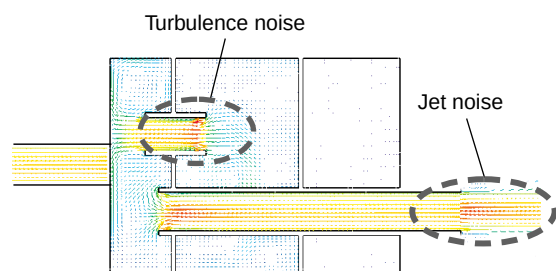


Fig. 20 Exhaust gas flow noise

The authors then made prototypes with narrower internal muffler parts and tailpipes. They then established relationships between engine speed and velocity (Fig. 21) as well as engine speed and exhaust noise (flow noise) without pulsating noise components (Fig. 22) with measurement data, and calculated flow analysis.

From these results it was determined that velocity and flow noise have almost linear characteristics with engine speed. Accordingly, by eliminating the engine speed component, it is possible to judge that velocity and flow noise have a proportional relationship, as expressed in Equation (10) (Fig. 23).

$$SPL = n10\log V + m \quad (10)$$

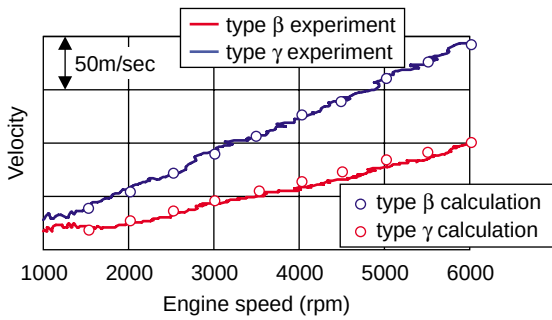


Fig. 21 Relationship between velocity and engine speed

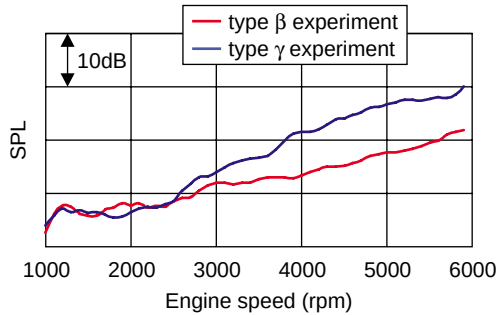


Fig. 22 Relationship between sound pressure level and engine speed

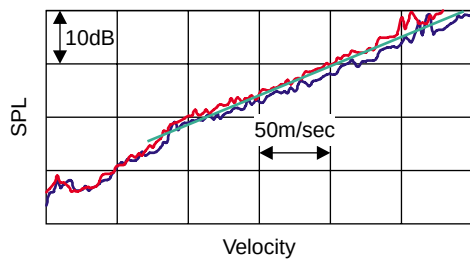


Fig. 23 Relationship between sound pressure level and engine speed

5.5 Exhaust Noise Analysis

Based on the above, exhaust noise can be calculated by adding pulsating noise and flow noise, as shown in Equation (11).

$$SPL = 20\log \frac{|X(p)|}{p_0} + \sum (n10\log V + m) \quad (11)$$

Next, comparing exhaust noise analysis results to exhaust noise measurements for type b and g mufflers as determined from Equation (11), as shown in Fig. 24 and Fig. 25, the authors found that the results were very uniform.

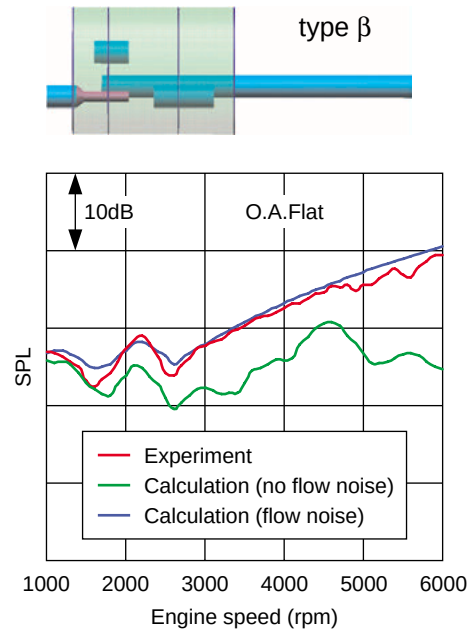


Fig. 24 Comparison of sound pressure levels (type β)

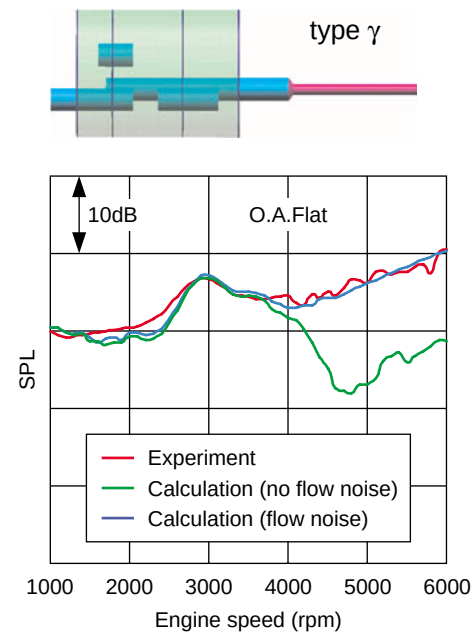


Fig. 25 Comparison of sound pressure levels (type γ)

6. Examples of Applications to Muffler Development

As a result of the methods above, the effects that the cold end parts of the exhaust system have on engine output and exhaust discharge noise have become clear. Further, it is now possible to make optimization studies of muffler design based on analysis of this mechanism.

That is to say, in the prevention of flow noise and back pressure rise, etc. the basic task is to reduce velocity. However, in addition it is very important to prevent flow separation⁽⁴⁾.

Conventionally, it was most effective to make separation visible and take countermeasures to prevent it. However, as shown in Fig. 26, the physical flow phenomena in CFD analysis results are very uniform. Thus, flow noise can be reduced and the flow rate improved by changing the shape of pressure loss areas without actually producing any parts.

Therefore by utilizing the analysis technology described thus far, the authors examined to what degree it would be possible to make the muffler smaller and lighter, and to reduce the exhaust discharge noise.

The assumption here was that the muffler layout within the exhaust system and the back pressure characteristics would remain unchanged.

The results are given in Table 2. To summarize, through the analysis of the mechanisms involved in the exhaust system acoustic characteristics and the fluid characteristics, it is possible to develop a muffler that is small, lightweight, and (as shown in Fig. 27) has a reduced exhaust discharge noise. This technology was applied to the 2001 Civic.

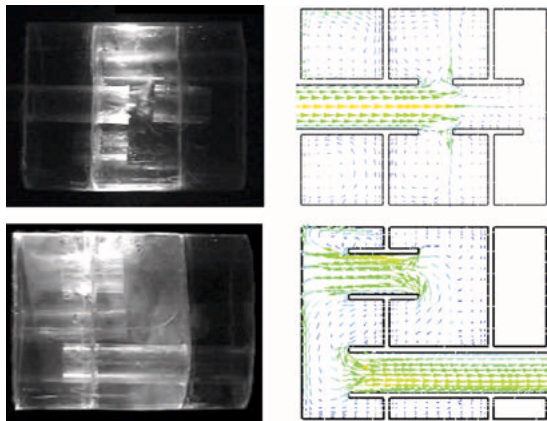


Fig. 26 Comparison of flow visualization and calculation

Table 2 Comparison of exhaust systems

	Base exhaust system	Modified exhaust system
Muffler volume rate	1	0.762
Muffler weight rate	1	0.742
Exhaust noise	Fig.27	

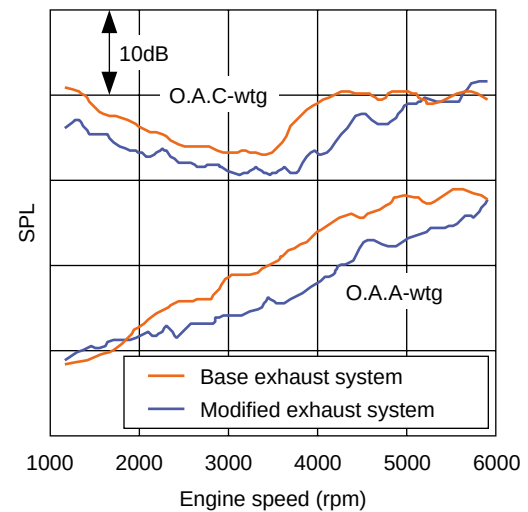


Fig. 27 Comparison of exhaust noise

7. Summary

Aiming for highly accurate predictive technology for exhaust system parts using CAE the following conclusions were reached.

- (1) It is possible to convert even mufflers with complex internal structures into straight pipes with a set diameter, by calculating the loss coefficient from the flow rate determined through 3D steady flow analysis.
- (2) Engine output can be calculated with very high accuracy ($\pm 2\%$) by looking at the muffler as a straight pipe as stated above, and then applying 1D volumetric efficiency analysis.
- (3) In order to improve the accuracy of pulsating noise analysis, it is very important to determine acoustic rigidity, attenuation, and mass matrixes while giving full consideration to temperature parameters, and to define acoustic vibration vectors which introduce velocity.
- (4) It was learned that flow generated noise is composed of turbulence noise generated internally in the muffler, etc., and jet noise generated at the tailpipe outlet. This sound pressure has a proportional relationship to velocity.
- (5) It is possible to calculate exhaust discharge noise with the extremely high accuracy of $\pm 2\text{dB}$, by adding together pulsating noise and flow noise.

Also, it has become possible to develop an exhaust system that has a high-level balance, by analyzing the characteristic factors that affect exhaust discharge noise and engine output.

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