

Development of a Vibration Prediction Technique for Automobile Exhaust Systems

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ABSTRACT

A simulation technique that uses a finite element model for predicting vibration in the exhaust system was developed. The shapes of the flexible tube portions of the flexible joints, or vibration absorption joints, were faithfully modeled and the characteristics of the other parts were assigned numerical values. An accurate model of the ball joints was obtained by assigning numerical values to the dynamic properties of gaskets. The vibration force of the engine, which is the input for this system, was obtained using the sum of the inertial forces of the component parts and the rotational reaction force produced by cylinder combustion pressure. This made it possible to quantitatively predict the forces transmitted to the body from the exhaust hangers and to qualitatively understand exhaust system behavior. This in turn makes it possible to study the optimal locations for the exhaust hangers and the vibration absorption joints in the pre-development stages.

1. Introduction

One of the causes of noise and vibration in the cabin of a vehicle is the engine vibration that is transmitted to the body through the exhaust hangers. This often leads to booming noise, floor vibration, steering vibration and other noise and vibration phenomena that occur at idle.

Conventionally, prototype parts have been repeatedly fabricated during vehicle development to investigate and countermeasure the causes of the excessive noise and vibration (NV) described above in exhaust systems.

However, building and testing countermeasure parts takes considerable time and thereby forestalls the actual resolution of this issue. Thus, the application of simulation as represented by Computer Aided Engineering (CAE) has become a matter of importance.

Therefore, the authors have performed multiple verifications in order to grasp the characteristics and phenomena of each part with the aim of achieving highly accurate simulation technology results that can then be applied to actual development.

This paper reports the results of this verification process. That is, it reports that simulation modeling technology, which reliably simulates reality, has been established.

2. Overview of Calculation Technology

2.1. Modeling the Critical Structures of the Exhaust System

The basic exhaust system body Finite Element Model (FEM) used in this example is shown in Fig. 1.

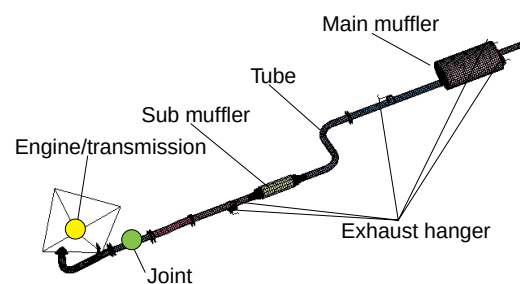


Fig.1 FE model of an exhaust system

All parts, whose shapes could be modeled, such as the exhaust system pipes and the mufflers, were modeled using shell elements.

The modeling know-how required to obtain the characteristics of the vibration absorption joints was reflected.

As the frequency range used in this research project is low for the engine and transmission, they were hypothesized to be within the rigid body range and the characteristics of the rigid bodies were expressed as a mass point.

2.2. Modeling of the Composite Joint Structure

The vibration absorption joints (flexible joints and ball joints) are the portion of the exhaust system that reduces overall exhaust system input and influences the vibration mode.

These joints are comprised of multiple parts. In this method, the mechanism of vibration absorption for each

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component part was analyzed to gain an understanding of the dynamic characteristics of the part. The obtained characteristics were then combined to create a model.

The vibration analysis was performed by verifying the directional characteristics of each joint's displacement.

(1) Flexible Joints

The component parts that comprise the flexible joints are shown in Fig. 2.

Test methods and restraining conditions as shown in Fig. 3 were used to analyze the characteristics of the flexible joints. The directions of displacement analyzed were both the expansion and contraction direction as well as the flex direction.

The frequency response function (FRF), which is the acceleration response level for each unit of load on each of the component parts, is shown in Fig. 4 for the expansion and contraction direction.

It is clear from the FRF peak in Fig. 4 that the flexible tube has dominant characteristics and that the other component parts are merely adding spring, attenuation and related characteristics.

The model of the dominant flexible tubing component was constructed by faithfully reproducing its actual shape. Modeling the shapes of the other component parts is difficult and thus the characteristics of these parts were expressed using numerical values.

A model was then built based on these results. Fig. 5 shows a comparison of the FRF for the actual measurements and for the calculations.

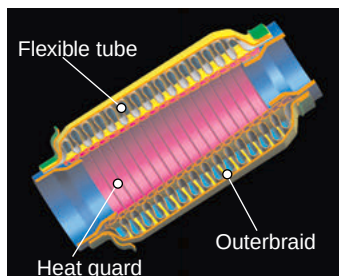


Fig.2 Flexible Joint

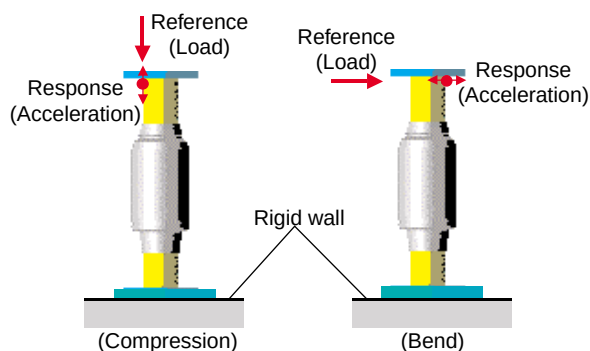


Fig.3 Test part for analyzing dynamic characteristics of the flexible joint phenomenon

(2) Ball Joints

Fig. 6 shows the parts that comprise the ball joints.

There are fewer parts comprising the ball joints than the flexible joints and, based on the behavior of the joints, the rotational (dynamic) spring is dominant. Thus, testing was performed on the dynamic spring characteristics focusing on the ball rotational center.

Fig. 7 shows the test method and restriction conditions and Fig. 8 shows the results of the tests.

As is clear from Fig. 8, the dynamic stiffness of the ball gasket and the dynamic stiffness for the entire ball joint are nearly equivalent.

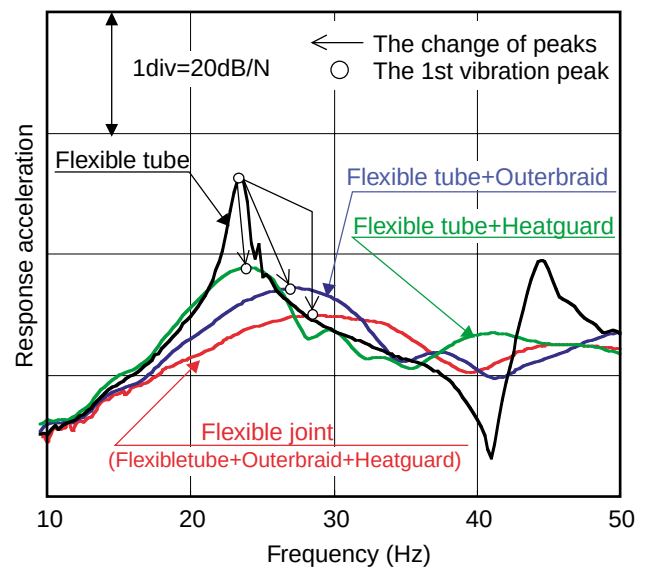


Fig.4 Measured FRF of flexible joints

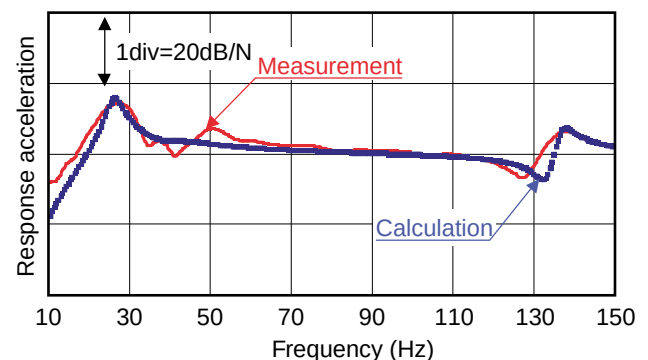


Fig.5 Comparison of FRF of the flexible joint

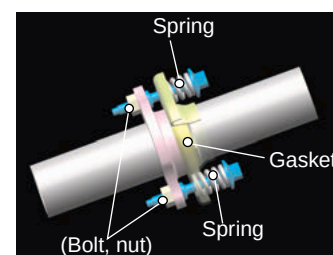


Fig.6 Ball Joint

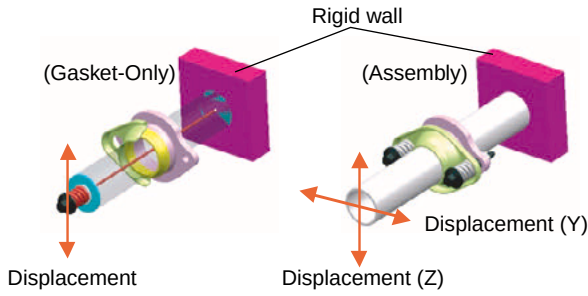


Fig.7 Test part for analyzing the mechanisms of the ball joint phenomenon

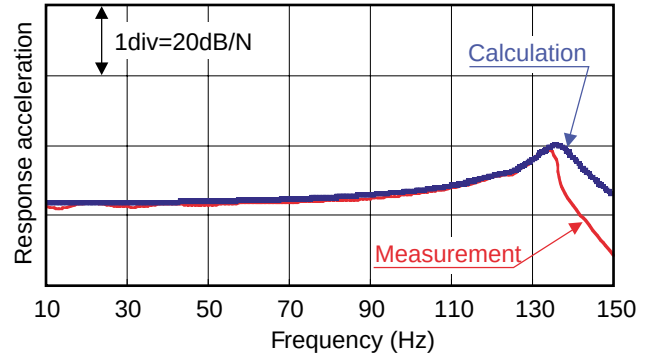


Fig.10 Comparison of FRF of the ball joint

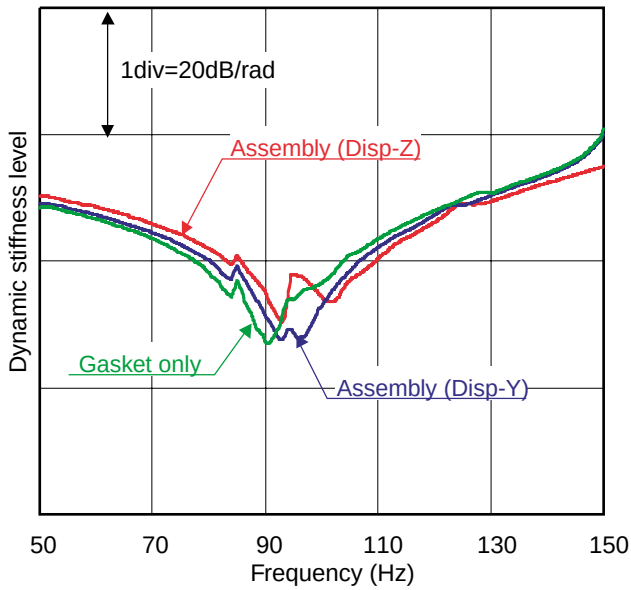


Fig.8 Measured dynamic stiffness of ball joints

Next, the test results of the FRF of the actual measurements and the calculations are discussed.

To assure consistent results, actual measurements were performed using the set-up shown in Fig. 9.

A highly accurate model was created through precise input of the spring characteristics of a ball gasket. Fig. 10 shows the results of the actual measurements and the calculated model.

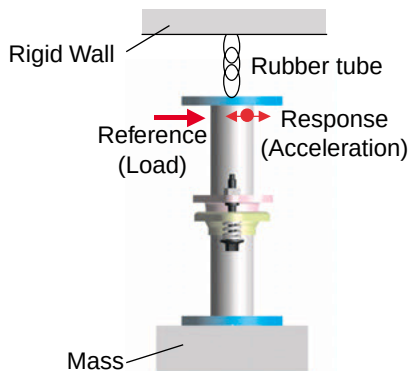


Fig.9 Test part for verifying correlation

As described above, the characteristics of each component part comprising the vibration absorption joints were understood through actual measurements. The characteristics of the parts were substituted into a calculation model. Further, a highly accurate model was obtained through the combination of these characteristics.

2.3 Examination and Identification Methods for Engine Modeling Application

The engine and transmission are critical environmental conditions for the exhaust system. We treated these areas as rigid bodies to identify the rigid characteristics from the FRF of the actual measurements.

The following describes the concept used to identify the rigid body concepts.

In a mass model of a spring with a one-degree of freedom system, with the frequency range above the resonant frequency, a rigid body movement frequency exists where the inertia level is constant. This level is determined by the mass value.⁽⁴⁾

This one-degree of freedom system is an expanded method in which there are a total of 6-degrees of freedom - 3 in the translational direction and 3 in the rotational direction.

If the six-degrees of freedom mass matrix characteristic is $[M]$ when the input force $[F]$ is applied, then the movement formula is:

$$[F] = [M][\ddot{x}]$$

In the equation above, $[F]$ is the input level and $\ddot{x}$ is the vibration acceleration response level; if these two are known, they lead to $[M]$, the answer to the linear equation system. Furthermore, if the coordinates of the input and response are known, it is then possible to attain $[M_c]$, the center of gravity by replacing the coordinates around the reference points, and calculating $[M]$ for these reference points.

2.4 Verifying the Accuracy of Calculations

Bench measurements were taken to verify correlation with the FEM.

Eigenvalues, natural frequency correlation coefficients, and FRF comparisons were used in the correlation verification. At least 120 FRFs were measured for the exhaust system in Fig. 1.

Also, a target of at least 0.8 was set for the native mode correlation coefficients (MAC).

Fig. 11 shows the calculation model accuracy verification flow.

Using the generated model, the engine and the exhaust system installed in a vehicle were reproduced to verify the FRF characteristics.

Vibrations were input from a shaker at the rear of the exhaust system, as shown in Fig. 12, for the measurement. Then, the acceleration levels at representative hanger points A and B were measured. Two vibration absorption joints were used in the measurements: a flexible joint and a ball joint.

The results of FRF verification on the exhaust system fitted with a flexible joint are shown in Fig. 13.

In the same manner, the results of FRF verifications on an exhaust system fitted with a ball joint are shown in Fig. 14.

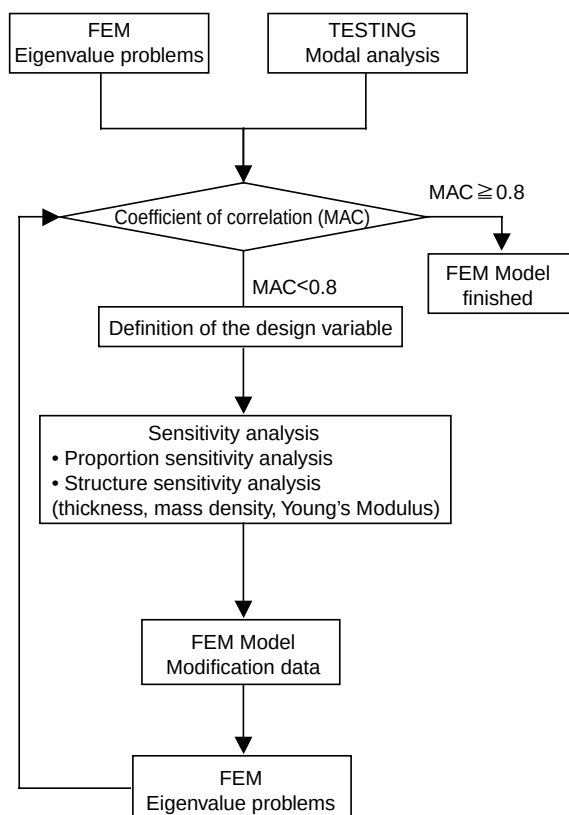


Fig.11 Flow chart for verifying correlation of accuracy

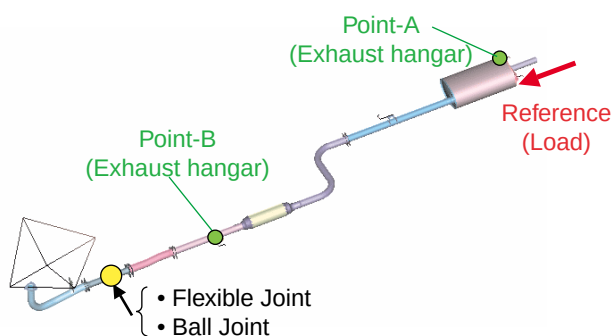


Fig.12 FE model of an exhaust system

Looking at Figs. 13 and 14, it is apparent that resonance peak level and frequency are both accurate.

Thus, based on the above, the behavior of the entire exhaust system including the engine could be quantitatively predicted.

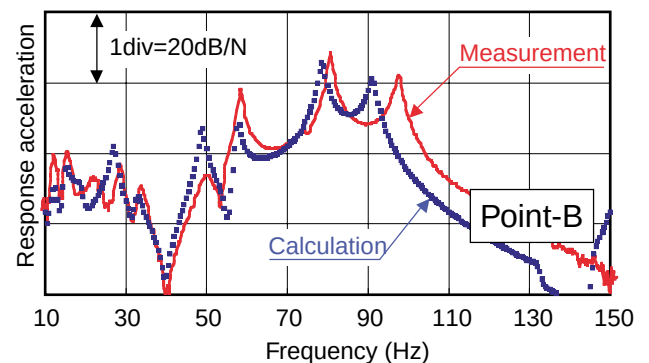
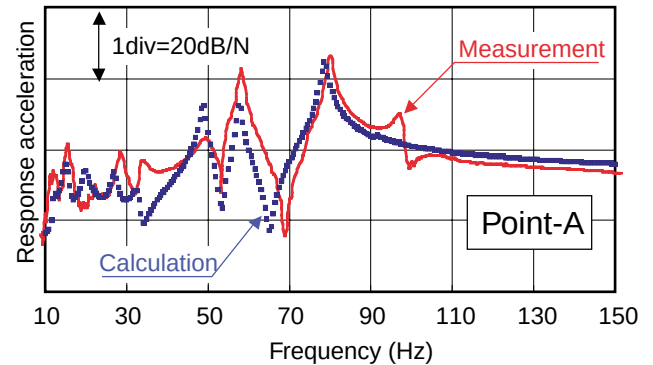


Fig.13 FRF comparison of point A and B in the flexible joint

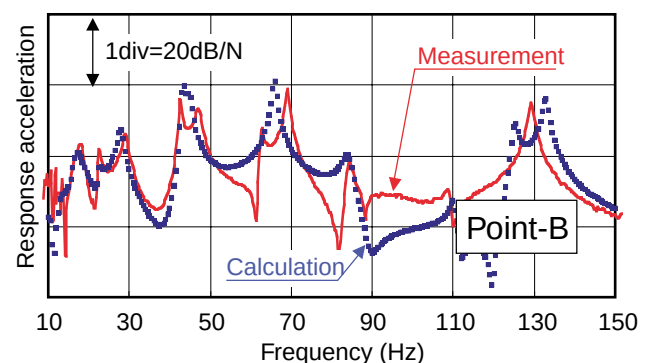
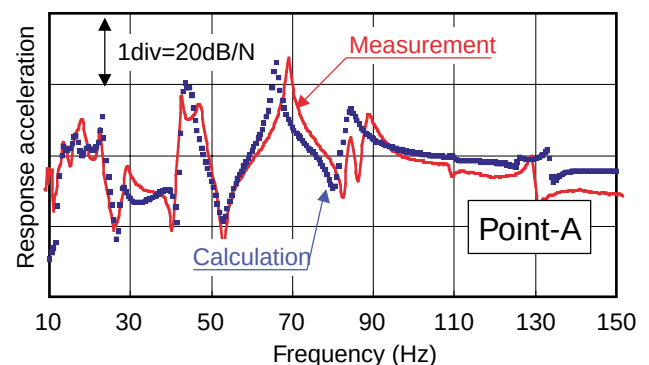


Fig.14 FRF comparison of point A and B in the ball joint

3. Engine Vibratory Force Calculation Method

Engine vibratory force was calculated by applying a model wherein all the parts, including the piston, connecting rod and cranks, were considered rigid bodies. ^{(2), (3)}

That is, the overall engine vibratory force was derived by calculating the total of the inertial force of both the reciprocating motion of the piston and the small end of the connecting rod and the rotary movement of the crank and the big end of the connecting rod, and the rotational reaction force of the crankshaft caused by the cylinder combustion pressure as determined for each cylinder, and finally by compensating for the phase differences between the cylinders.

The cylinder combustion pressure used was the average of the pressure data taken during test multiple cycles.

The following is a brief description of the equation used to calculate the inertial force and the rotational reaction force.

If the piston displacement from the piston top dead center is (S) in Fig. 15, then

$$S = r(1 - \cos \theta) + \frac{1}{4}r\rho(1 - \cos 2\theta)$$

where,

$$\rho = \frac{r}{l}$$

Acceleration would then be

$$\frac{d^2S}{dt^2} = r\omega^2(\cos \theta + \rho \cos 2\theta)$$

Therefore, if the mass of the reciprocating portion comprised of the piston and the small end of the connecting rod is M_1 , then the vertical inertial force of the reciprocating portion is derived using the following equation.

$$F_{1z} = -M_1r\omega^2(\cos \theta + \rho \cos 2\theta)$$

Next, if the mass of the rotating portion comprised of the crank and the large end of the connecting rod is M_2 , then the vertical inertial force and the lateral inertial force of the rotating portion would be found using the following equations.

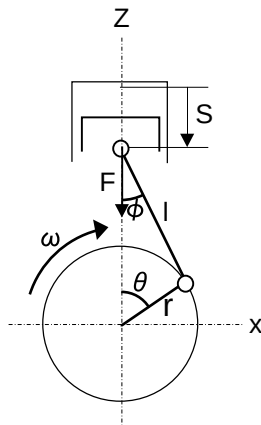


Fig.15

$$F_{2z} = M_2r\omega^2 \sin \theta$$

$$F_{2x} = M_2r\omega^2 \cos \theta$$

Crank rotational reaction force is derived using the following equation, when the sum of the cylinder combustion pressure and the vertical inertial force for the reciprocating portion as determined above is defined as F.

$$F_c = F \cdot \frac{\sin \theta}{\sqrt{\frac{1}{\rho^2} - \sin^2 \theta}} \cdot (r \cos \theta + l \sqrt{1 - \rho^2 \sin^2 \theta})$$

Fig. 16 shows the results of the actual measurements and calculations of the vibration acceleration of the engine mounts caused by the firing of the engine.

We compared the response at points A and B in the exhaust model shown in Fig.12 when the engine vibratory

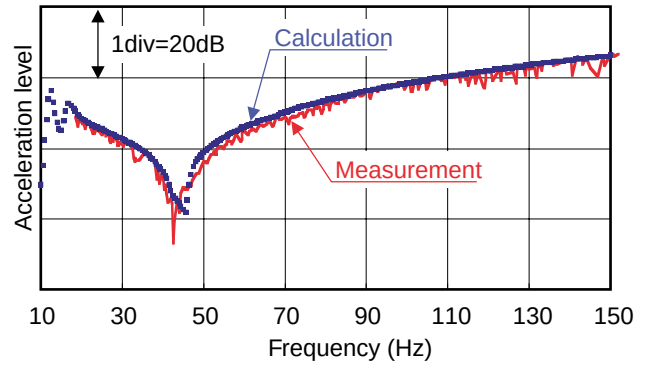


Fig.16 Response comparison of the engine mount during firing

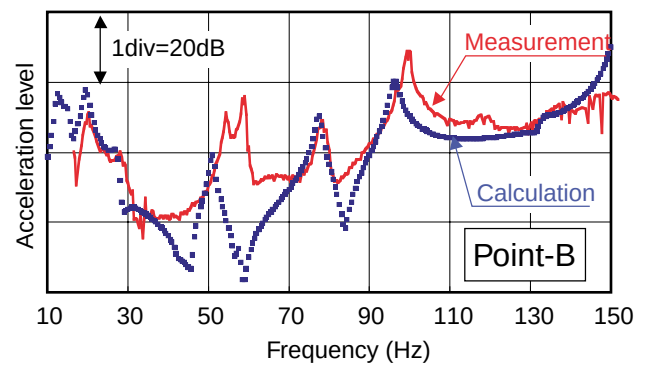
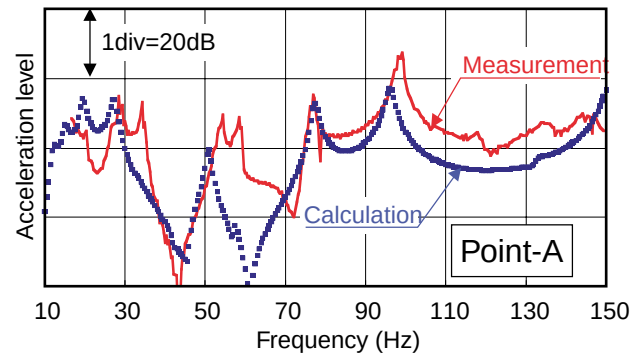


Fig.17 Response comparison of point A and B with a flexible joint during firing

force was applied with the response measured during firing.

The results of the exhaust system acceleration fitted with a flexible joint are shown in Fig. 17.

In the same manner, the results of the exhaust system acceleration with a ball joint are shown in Fig. 18.

From the results of Figs. 17 and 18, we see that the trend of the waveforms, the resonance peak levels, and the frequencies are accurate.

Therefore, vibration transmitted from the exhaust system hanger to the body can be obtained as absolute values, and judgment is rendered possible.

4. Application to Exhaust System Vibration Reduction Methods

Next, we will discuss an example wherein this method is applied to an FE model of the exhaust system, as shown in Fig. 19.

Here, we investigated the optimum positioning of the exhaust system hangers - the points at which the exhaust system vibration is transmitted to the body.

In this method, positions that could easily be nodes were determined based on multiple exhaust system mode shapes; the hanger positions were then shifted to those positions. Fig. 20 shows the results of calculations of the exhaust system acceleration at the hangers of the main muffler with input from the rear of the exhaust system. Here, it is possible to infer a reduction of the response levels at the exhaust system hanger points, seen in Fig. 20, and therefore we were able to verify the effectiveness of this predictive technology.

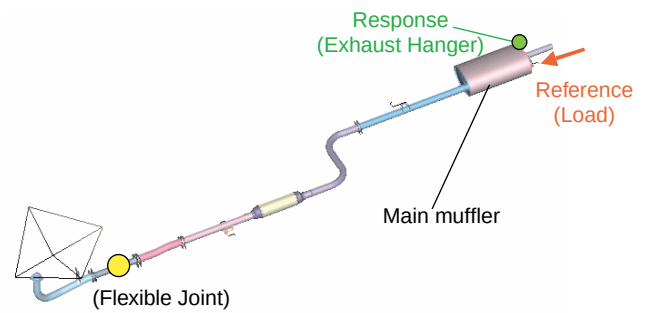


Fig.19 Structural model of an exhaust system

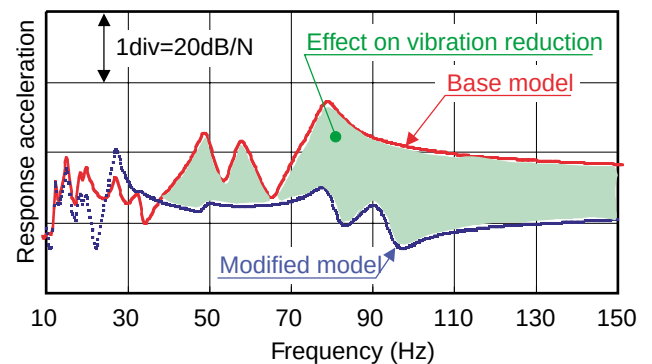


Fig.20 Response comparison at the hanger points

5. Conclusions

A highly reliable exhaust model that conforms to the real world was constructed using the following methods:

- (1) A faithful model of the flexible tube shape was created, as the flexible tube is the dominant characteristic of a flexible joint. Also, numerical values were assigned to the dynamic characteristics of the other component parts.
- (2) As the ball joint characteristics are basically determined by the ball gasket characteristics, the dynamic characteristics are assigned numerical values.
- (3) An engine area model was constructed by treating the engine and the transmission as rigid bodies and then obtaining the center of gravity mass matrix characteristics using the FRF and the measurement positions from the tests.
- (4) Engine vibratory force was derived by considering the phase differences between the individual cylinders based on the inertial force, caused by the reciprocating and rotary motions of the component parts for each cylinder, and the rotational reaction force, caused by the actual cylinder combustion pressure.

As a result, analysis of the phenomena of the entire exhaust system, including the engine, could be performed due to the highly accurate FE model of the exhaust system and the modeling of the engine area; the area that sets the limit conditions for the system. Further, by applying rigid body engine vibratory force, we were able to predict the

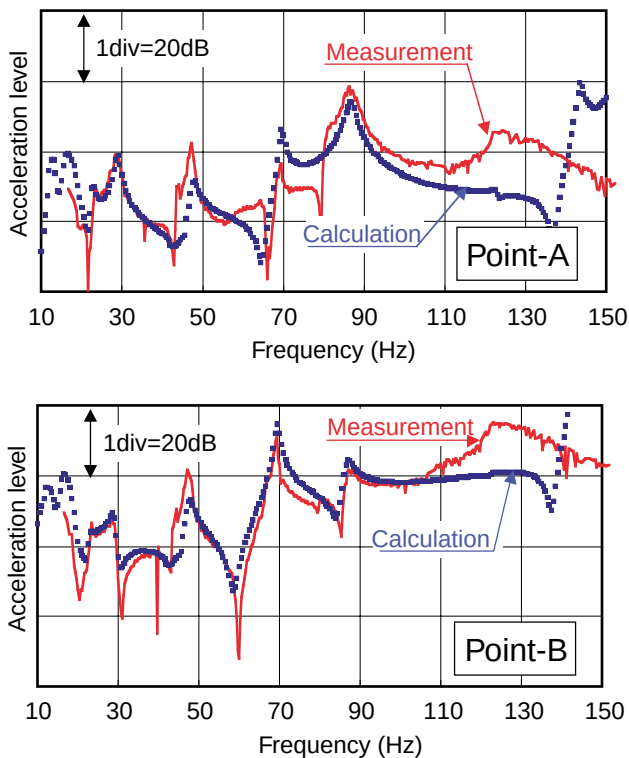


Fig.18 Response comparison of point A and B with a ball joint during firing

exhaust vibration that affects the noise and vibration in an evaluation vehicle with enough accuracy for real world application.

6. Prospects

Prediction engineering tools and engine vibratory force derivation tools that enable coherent analyses from eigenvalue analysis to sensitivity and structural analysis were provided to development engineers for effective use in the calculation and design stages of development that come before prototype builds.

The technologies were applied to actual vehicles so that exhaust system vibration could be reduced, thus contributing to improvements in automobile noise and vibration performance.

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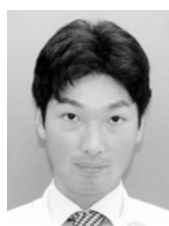
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