

Development of Prediction Technology for Resonance Stress of Exhaust System by Considering Nonlinear Properties of Spherical Joints

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ABSTRACT

A highly accurate and versatile prediction technology that provides results equivalent to stress measurement in the vehicle condition was developed. The purpose of this technology was to predict high-cycle resonance stress of the exhaust system in the fully assembled state. The phenomenon of high-cycle vibration of the exhaust system was divided into the two systems of an input system and a response system. The input system was accurately reproduced by considering the multiple degrees of freedom and multiple orders of phase of the engine excitation force transmitted to the exhaust system. The response system was experimentally modeled by conducting standalone tests of the spherical joint and obtaining the stick-slip characteristics. Application of the above enabled accurate prediction for various exhaust system shapes.

1. Introduction

Recently, enhanced fuel economy, emissions and output power performance are being demanded of automobiles. Lighter weight and good heat-up characteristics are demanded of exhaust systems in order to enhance these performances, but at the same time, high reliability is also required.

One issue concerning exhaust system reliability is high-cycle fatigue breakdown due to resonance phenomena induced by the engine excitation force, and designs that take this issue into account are necessary.

In addition, many vehicle types share the same hot-end (parts mainly including the catalyst) from the viewpoint of parts sharing, but the cold-end (parts mainly including the chamber and the silencer) exhaust system variations are tending to increase, and measures to support increased development and test process times are demanded. At the same time there are also demands for shorter development periods, and the development of Computer Aided Engineering (CAE) technology that can accurately predict high-cycle vibratory stress in a short time in the drawing stage is needed.

Previous Finite Element Method (FEM) analysis of exhaust systems in the fully assembled state consisted of

performing frequency response analysis using the FEM CAE software NASTRAN, and comparing these analysis results with the results of stress measurement in the vehicle condition for use in investigating countermeasures.

As an example, Fig. 1 compares the stress results obtained by the previous analysis method with the results of stress measurement in the vehicle condition. While there are noticeable differences in the calculated and measured stress peaks in each mode, the level is such that enables use for investigation of countermeasures.

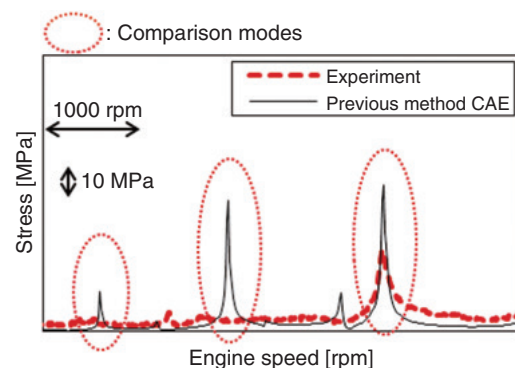


Fig. 1 Comparison of experiment and CAE by previous method

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However, absolute value evaluation of stress is needed to determine the feasibility of specifications in the drawing stage, and increasing this accuracy is an issue.

Three issues for promoting accurate analysis of the exhaust system assembly using FEM are appropriate modeling of couplings between parts, accurate modeling of the engine excitation force, and creation of mesh models that accurately reproduce responses.

In particular, setting appropriate part coupling conditions has been an issue for some time, and especially modeling of the spherical joint that acts to absorb the engine oscillation (Fig. 2). The spherical joint uses a spherical gasket to seal the exhaust gas, and this spherical gasket has characteristics such that the stick state is maintained at small inputs. However, when a certain input or more is applied, the slip state results and the gasket slides with some friction.

As examples, Fig. 3 shows an image of a shaker excitation test for an exhaust system with a spherical joint, and Fig. 4 shows the change in the Frequency Response Function (FRF) due to differences in the excitation input level. In consideration of the input transmission characteristics, the input point was located near the center of gravity of the exhaust pipe used in this study, and the response point used the average value of seven locations

spread over the entire exhaust system to equalize the response effect for comparison.

The results in Fig. 4 confirm that the response level decreases as the input level increases. This indicates that the spherical joint enters the slip state, so the exhaust system is divided and the rigidity drops. The change in the response level is closely related to the stress level, so to increase the stress accuracy it is necessary to construct a model in

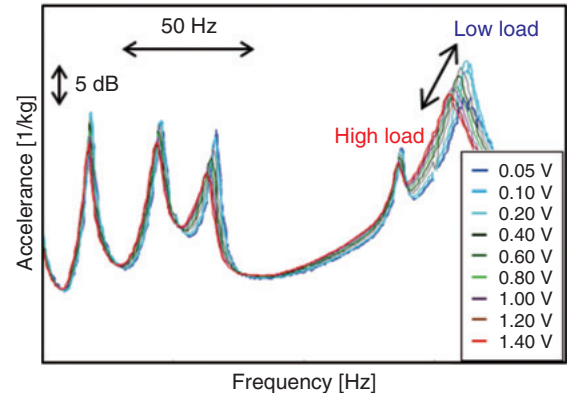


Fig. 4 FRF characteristics under different input conditions

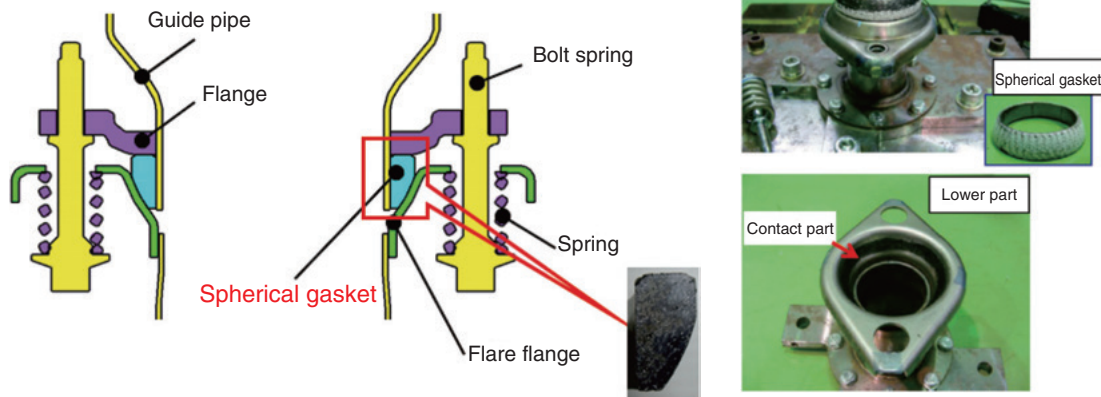


Fig. 2 Section and parts of spherical joint

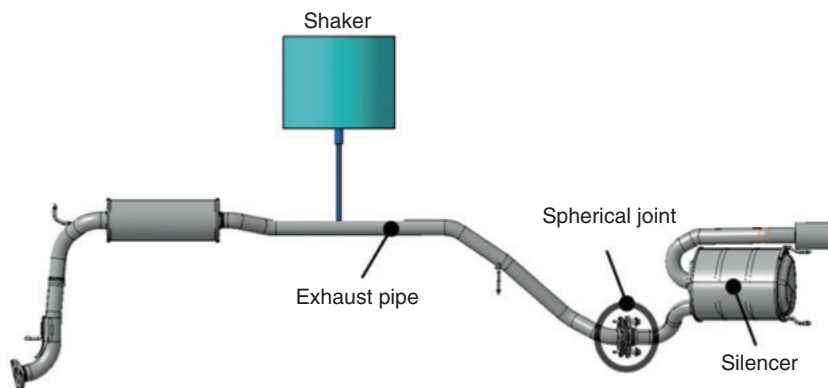


Fig. 3 Shaker test of exhaust system with spherical joint

which the stick and slip behavior of this location changes in accordance with the change in input. To construct this model, friction needs to be considered in the spherical joint model, but the previous analysis method uses the frequency axis, and was unable to take into account the change in the friction behavior over time.

Previous studies have discussed the influence of spherical joint behavior on high-cycle vibration behavior, and modeling has also been attempted⁽¹⁾⁻⁽³⁾. However, there is no known analysis method that takes into account the time-base change in behavior, and the subject frequencies have been limited to relatively low frequencies, and the high-frequency range that is the subject of this study has not been covered.

This paper describes the method of increasing the CAE accuracy and developing stress prediction technology for high-cycle vibratory stress on the exhaust system that is equivalent to stress measurement in the vehicle condition. This is achieved by creating a model that takes into account the nonlinear properties of the spherical joints, modeling the engine excitation force, creating an accurate mesh model, and applying these models to transient response analysis that performs analysis on the time axis.

2. Development of Analysis Method

The parts subject to analysis were the close-coupled converters, exhaust pipe and silencer that comprise the exhaust system in the fully assembled state. This study aimed to develop an analysis method using the simple exhaust system for verification shown in Fig. 5.

Spherical joints are used as the coupling between parts, so an experimental model was defined that considers the friction on the sliding surfaces.

To consider friction it is necessary to understand the changes in overall exhaust system behavior over time, and transient response analysis is given as this analysis method.

However, mesh models have many nodal degrees of freedom, so transient response analysis using all nodal degrees of freedom increases the calculation load. To address this issue, modal reduction was performed on the mesh model to reduce the degrees of freedom, and

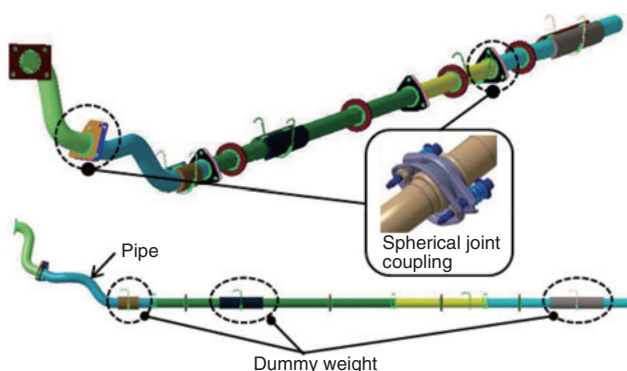


Fig. 5 Simple exhaust system layout for verification

a modally reduced model was assembled using a mode synthesis method with good analysis efficiency. The overall exhaust system behavior was then calculated from the results of dynamic analysis using this assembly model, and the displacement output was applied to a detailed mesh model to obtain the stress.

2.1. Construction of Models

2.1.1. Experimentally modeling the spherical joint

When modeling the spherical joint, it is necessary to consider the two characteristics of stick characteristics during small input, and slip characteristics during large input. These characteristics were obtained by creating a test jig with single pipes welded to a spherical joint, and measuring the rotational stiffness and the friction torque by a shaker excitation test. Figure 6 shows the test equipment used to obtain the friction characteristics of the spherical joint. This study attempted to experimentally model the stick-slip characteristics by performing calculations using an analysis model similar to the test jig, and identifying the spring constant during stick and the friction torque during slip from the test results.

The friction characteristics were obtained by performing a shaker excitation test with two rotational degrees of freedom in the topple direction and one rotational degree of freedom in the direction of rotation around the axis, and obtaining the hysteresis loop. The shaker excitation frequency should be set so that the jig resonance does not influence the friction torque. To avoid this, the FRF resonance frequency was confirmed by a hammering test, and the results (Fig. 7) confirmed that there was no jig resonance at 100 Hz, so that frequency was used for the excitation test.

Figure 8 shows the friction measurement results. Stick characteristics are exhibited during small input, but slip characteristics appear at a certain input or more, creating a hysteresis loop. The slope during small input expresses the gasket stiffness in the stick state, and the hysteresis loop amplitude expresses the friction torque in the slip

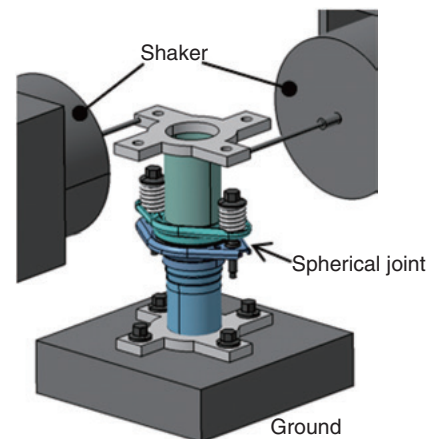


Fig. 6 Test equipment for measurement of friction characteristics

state. These results were used to calculate the stick spring constant and the slip friction value. At the same time, the effect of different inputs on the friction torque was checked, and it was found that the displacement increases as the input increases after slip, but the friction torque indicates a constant value.

Next, Fig. 9 shows an outline of the spherical joint analysis model. In this model, when the input is smaller than the friction torque, the stick model shown on the right side of Fig. 9 does not slide, and only the stick model that expresses the gasket stiffness shown on the left side of Fig. 9 is used. When the input is larger than the friction torque,

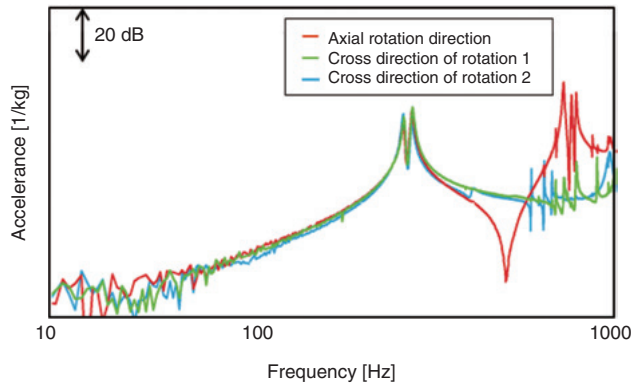


Fig. 7 FRF characteristics of spherical joint

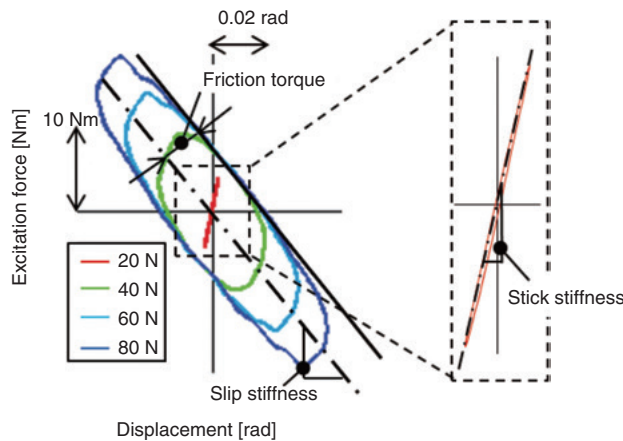


Fig. 8 Method of extracting spherical joint characteristics

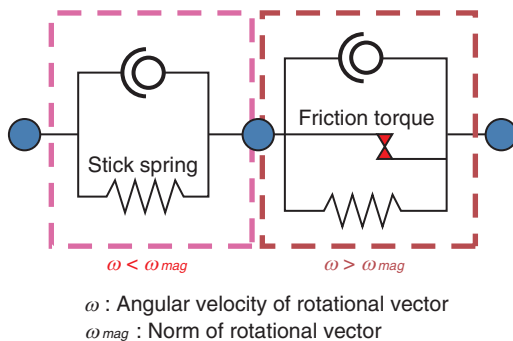


Fig. 9 ADAMS model outline of spherical joint

the slip friction torque is added to the stick model, and sliding occurs. Here, it is necessary to use the total friction force of the three rotational degrees of freedom, so a friction ball model was defined as shown in Fig. 10. The moments of the three rotational degrees of freedom of this model are expressed by the following equations.

$$T_x = T_{x0} \cdot \frac{\omega_x}{\omega_{mag}}$$

$$T_y = T_{y0} \cdot \frac{\omega_y}{\omega_{mag}}$$

$$T_z = T_{z0} \cdot \frac{\omega_z}{\omega_{mag}}$$

T_{x0}, T_{y0}, T_{z0} : Friction around the x, y and z axes

T_x, T_y, T_z : Moment around the x, y and z axes

$\omega_x, \omega_y, \omega_z$: Angular velocity around the x, y and z axes

ω_{mag} : Norm of angular velocity

Figure 11 shows an example comparing results with the results calculated by the mechanics analysis software ADAMS using the above models and the experiment results. Comparison of the change in behavior and the slip torque for different inputs shows that the change in behavior relative to the input and the friction torque width in the

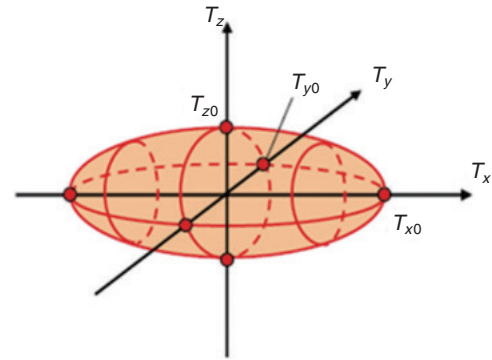


Fig. 10 Friction ball model

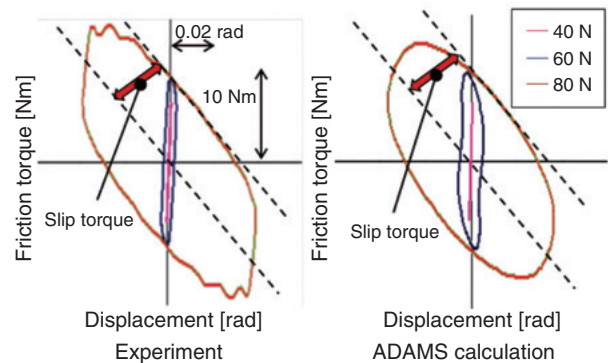


Fig. 11 Comparison of experiment and ADAMS calculation

slip state calculated by this model accurately reproduce the vehicle condition.

2.1.2. Modeling the engine excitation force

The major orders of engine vibration contribute greatly as the main excitation source for high-cycle vibration of the exhaust system. The vibration characteristics transmitted from the engine, which is an excitation source, to the exhaust system are mainly the excitation of the combustion pressure, the excitation of the reciprocating system inertial force, and excitation due to power plant resonance. One method of estimating the excitation force is to calculate the reciprocating system behavior and obtain the power plant behavior using FEM. However, this method takes time for model creation and calculation, so it is not suitable for this analysis. Therefore, this study calculated the input boundary conditions by measuring the excitation force of an actual engine.

To accurately measure the engine excitation force, the acceleration of the exhaust system flange on the engine side was measured using 3-axis accelerometers. However, engine displacement has three translational and three rotational degrees of freedom, so rotational behavior cannot be measured by normal 3-axis accelerometers. Therefore, accelerometers were located at the three flange bolt connection points shown in Fig. 12, simultaneous measurement was performed, and the rigid body motion with phase of each point was estimated with three translational and three rotational degrees of freedom from the measured three points with three translational degrees of freedom using the least squares method.

Normal sensors cannot perform stable measurement due to the effects of noise within the engine bench. To minimize the effects of this noise, accelerometers with charge amplifiers were used to amplify the signal on the sensor side to reduce the effects of noise.

All loads governed by the combustion pressure from idling to the red zone and motoring conditions governed by the inertial force were measured as the operating conditions. The excitation force of major orders that contribute to

exhaust system resonance were calculated from each measured operating condition.

2.1.3. Creating a mesh model of the exhaust system

To accurately reproduce the parts subject to analysis, the mesh model was modeled using shell mesh tetrahedral first order elements, and a mesh size was selected that can adequately reproduce the shape. Reducing the element size increases the accuracy, but the size was adjusted from the viewpoints of shortening the calculation time and reducing the calculation load on the analysis machine. The plate thickness and material characteristics extracted from the drawing values were defined as the physical property values of this mesh model. In addition, to increase the convenience of modeling, the Rigid Bar Element2 (RBE2) was used for welded joint elements and the Rigid Bar Element3 (RBE3) was used for the concentrated mass definition to reduce the number of elements and strive for simple modeling.

As an example of verifying the accuracy of this model, the correlation with hammering test results in the vehicle condition was obtained using Mode Assurance Criterion (MAC). The subject correlation frequency range was up to 450 Hz in the exhaust system assembly. Figure 13 shows the comparison results. These results show that a high correlation is obtained.

2.2. Development of a Dynamic Analysis Method

Transient response analysis using ADAMS was performed as the dynamic analysis method as described below.

Dynamic analysis that takes into account the friction of sliding parts was performed by modally reducing the mesh model and then assembling this reduced model on ADAMS using the spherical joint model. Figure 14 shows the analysis procedure using ADAMS. The assembly resonance characteristics can be accurately reproduced up to the high-frequency range by increasing the upper limit frequency of the modally reduced frequencies of each analyzed part. In consideration of shortening the calculation time and reproducibility of the assembly resonance characteristics,

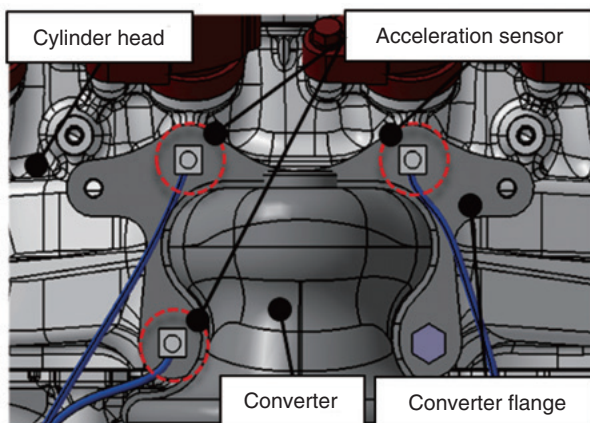


Fig. 12 Acceleration sensor installation locations

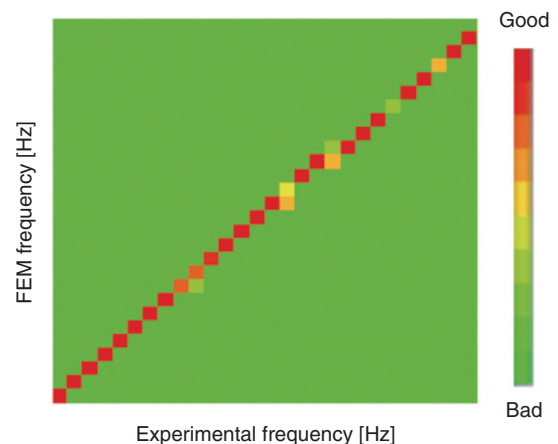


Fig. 13 MAC correlation results

this study adjusted the modally reduced frequencies of the analysis model, and set an upper limit of 1 000 Hz. As an example of verifying the accuracy of the reduced model, assembly eigenvalue analysis was performed separately by NASTRAN in consideration of all nodal degrees of freedom of the mesh model and by ADAMS using the modally reduced model, and the eigenvalue difference was confirmed. Figure 15 shows the comparison results. These results show that the maximum eigenvalue difference was 3.5 Hz, indicating that the reduced model has sufficient accuracy.

This reduced model was assembled on ADAMS using the spherical gasket model as the coupling condition. Figure 16 shows the ADAMS assembly state.

Regarding the input excitation data, waveforms that take into account the displacement excitation and phase of each measured order component were added together on ADAMS and applied as the time-base input waveform.

Stress recovery was performed by NASTRAN transient response analysis based on the displacement output of each

reduced model obtained by transient response analysis performed under the above conditions, and the stress at each evaluation point was evaluated.

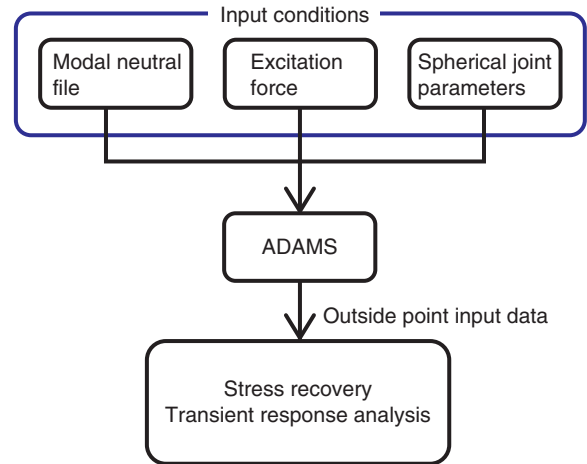


Fig. 14 Analysis procedure using ADAMS

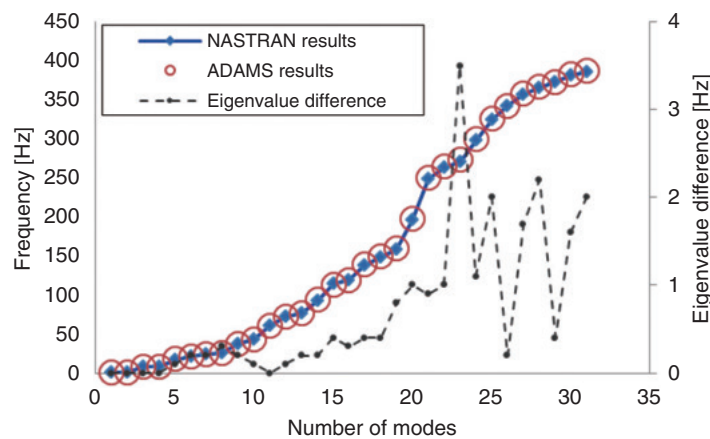


Fig. 15 Comparison of NASTRAN and ADAMS eigenvalue results

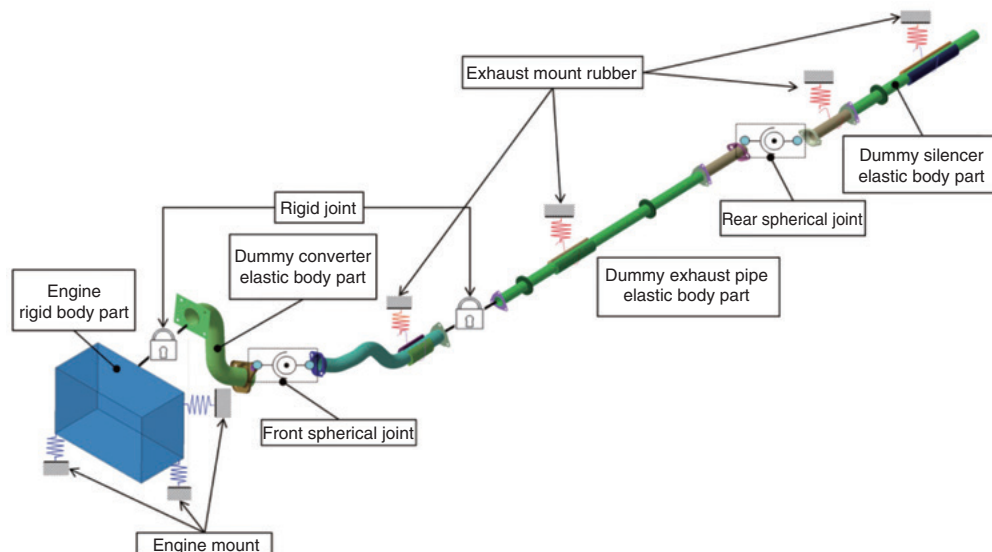


Fig. 16 ADAMS assembly model

3. Verification of Analysis Results

3.1. Verifying the Validity of the Developed Method

The graph shown in Fig. 17 compares the stress results obtained by the previous and new analysis methods and by testing of an actual vehicle as verification of the analysis results. Comparison of the stress peak in each mode shows that in modes where the previous analysis method indicated that high stress was generated, the new analysis method indicates the same values as the actual testing. This shows that force transmission is divided by the spherical joint stick-slip phenomenon, and indicates that the newly developed spherical joint model is a valid model.

3.2. Verifying the Versatility of the Developed Method

The actual stress was measured using the exhaust system of a mass-production vehicle, and the measurement results were compared with the analysis results obtained using the developed method to confirm the versatility of the new analysis method. Figure 18 shows an overall view of the subject exhaust system.

In order to predict high-cycle resonance stress in the drawing stage, the spherical joint characteristics used the newly developed models and parameters. The mesh model was modeled based on 3D-CAD data, and the material properties were defined as specified in the drawings without correcting the stiffness or density. Figure 19 shows graphs comparing the stress results obtained by stress measurement in the actual testing and by the new calculation method.

The results showed a resonance frequency error of 3% and stress error of 20%. This is sufficiently accurate for

practical application as drawing-based prediction results, and indicates the validity and versatility of this analysis method.

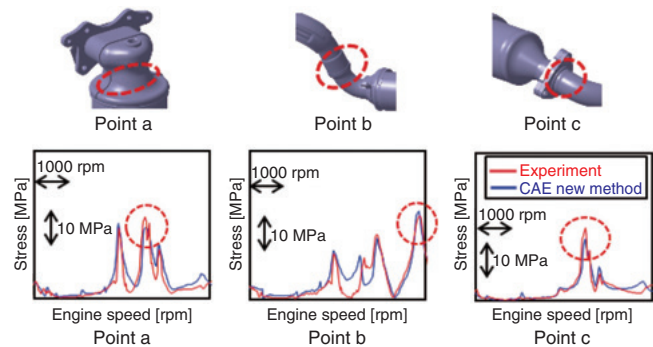


Fig. 19 Comparison of experiment and CAE by new method

4. Conclusion

Technology for accurately predicting the resonance stress of the exhaust system in the engine utility range was developed by the following method.

- (1) A spherical joint model was experimentally modeled by obtaining the stick-slip characteristics of the actual vehicle and reflecting these characteristics to an ADAMS model.
- (2) The input excitation force was accurately measured, and an input model that considers the engine behavior and phase at multiple orders was created and applied as the input conditions for analysis.
- (3) The mesh model was modeled using a mesh size enabling sufficiently accurate analysis, and the correlation was increased by performing MAC correlation with the actual vehicle.

This method enables model-based prediction of the stress on various different exhaust systems, and it is thought that this will contribute greatly to increasing development speeds and reducing costs by reducing the number of prototypes, and also helping to reduce weight. As a future possibility, it is thought that this new method will also enable prediction of exhaust system stress for vehicle types with different spherical joints.

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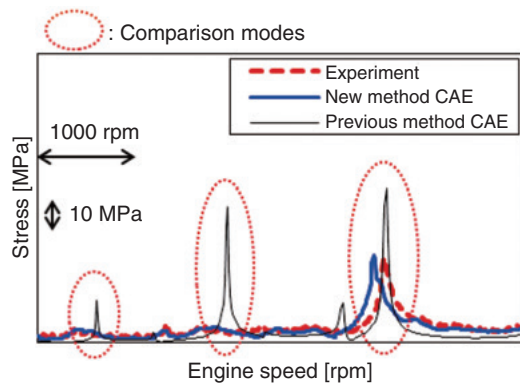


Fig. 17 Simple exhaust system model stress-analysis result

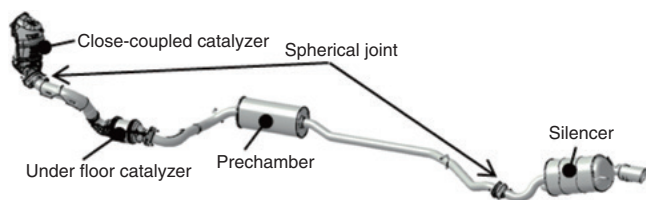


Fig. 18 Exhaust system for verification

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