

Development of a Procedure for Interior Noise Estimation by Virtual Vehicle Analysis

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ABSTRACT

This paper summarizes the development of a predictive vibro-acoustic full vehicle model of a mid-size sedan and focuses on the engineering analysis procedures used to evaluate the design performance related to engine-induced noise and vibration during the drawing stage of development. The vehicle model is comprised of a mixture of test-based and finite element component models. FRF-based substructuring is used to join these models. The virtual car model is loaded with engine forces resulting from indirect force identification. This force-set includes combustion, inertia, piston slap and crank bearing forces for engine harmonics from the 0.5 to 10th order. Analysis of this model yields predictions of vibration levels for every component and every interface between components, and of interior noise. The objective is to provide the vehicle NVH engineer with the capability to identify major causes for peak noise levels, to set targets and to develop action plans for every component design team. Having such information at any design stage will effectively reduce both the design cycle duration and the number of physical prototypes.

1. Introduction

A long time has passed since demands began for reductions in product development time. Recently, there has also been a desire to increase the speed of automobile development at every stage, from product concept to styling design, and from prototype fabrication to volume production. The most urgent issue for development engineers, in particular, is producing a satisfying match of performance to the product concept while reducing the number of prototype vehicles and prototype lots.

Meanwhile, one of the technical features of the vibro-acoustic performance of a finished vehicle is that this performance is determined by the interrelation of various systems, including the engine, subframe, and body.

Therefore, when altering vibro-acoustic performance, it is crucial to clarify which systems will produce the most effective results. The investigation of methods for achieving vibro-acoustic performance objectives also requires consideration of handling, ride comfort, acceleration performance, collision characteristics, cost, weight, and other such performance objectives that are often in opposition to vibro-acoustic performance. Consequently, it becomes necessary to create many prototype vehicles that have various specifications applied to confirm all of these competing objectives.

A full vehicle model for prediction of interior noise was created to facilitate the management of the various vehicle system characteristics as they relate to vibro-acoustic performance within a shortened development time (Fig. 1).

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This model was constructed from finite element component models combined with component models based on actual measurement data. The validity of this model construction and its prediction accuracy are verified in the first prototype lot. As a result, it was verified that this full vehicle model has the following features:

- The capability to define a vehicle system's contribution to cabin noise
- The capability to predict cabin noise resulting from a component design change

Use of this predictive system enables quantitative study of corrective actions taken on each vehicle system as required to control cabin noise from the initial stage of product development. This makes it possible to formulate action plans that are based on the most efficient direction regarding corrective measures to be taken by the designers of each component. Furthermore, the model enables immediate understanding of the effect outside change requests will have on cabin noise.

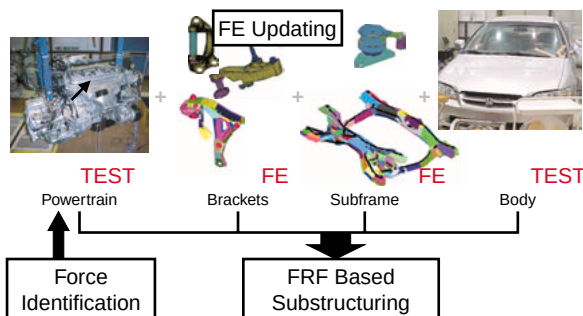


Fig. 1 Virtual vehicle model for estimation of interior noise

This paper describes the virtual vehicle model that was built from the combination of experimental and numerical models, and the verification of the resulting model. It will also show the superiority of this method.

2. Testing and CAE Modeling of Each Component and Construction of the Virtual Vehicle

The following will provide an overview of the modeling techniques used with this method.

The various components are represented by an FRF matrix defined by the degrees of freedom of the inputs, connections, and responses. They are then assembled using FRF-based substructuring.

The relationship of the response vector $\{X_i\}$ to the excitation vector $\{F_j\}$ for the model defined by the FRF matrix $[H_{ij}]$ is defined by the following:

$$\{X_i\} = [H_{ij}]\{F_j\} \quad (1)$$

Substructuring yields the FRF matrix composed of the input force degrees of freedom excited during actual operation and the response degrees of freedom, which are arbitrarily selected. Component combinations are not limited to rigid connections, but can also have connections with an intermediate elastic element that is represented by the commonly used stiffness matrix $[K_s]$. Thus, the structured

transfer function will be in a state that balances force and linearity with respect to displacement.

The experimental model is generated using actual measurement data and is represented by an FRF matrix. Likewise, the FE model is represented by the thus generated FRF matrix. This presupposes that FE mode accuracy was improved by updating the FE model from an experimental modal analysis conducted on the first prototype product.

In addition to the inertia force component calculated from the engine specifications and the combustion force component calculated from the actual measured combustion pressure, the input force is also composed of a residual component that is identified by the inverse matrix method from the actual measured engine vibration.

Fig. 2 presents an overview of the individual components. The vehicle model described in this paper is composed of the powertrain (engine plus transmission), the engine mounting brackets, the engine mountings, the subframe, and the body, for a total of five elements and eleven parts. The powertrain and body are represented by an FRF matrix based on actual measurements. The engine mounting brackets and subframe are defined by an FE model, and the engine mountings are defined by complex stiffness curves that describe the engine mount characteristics.

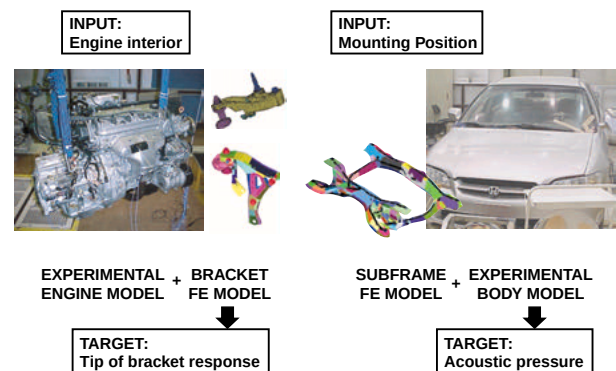


Fig. 2 Individual components of the virtual vehicle model

2.1 Experimental Engine Model

The FRF matrix for the engine model is made up of input forces with 24 degrees of freedom and responses with approximately 150 degrees of freedom. Assuming combustion pressure, piston guidance, and piston slap, the input force directions were set perpendicular to (axial) the cylinder wall and head and perpendicular (axial) to the crank bearing. In order to obtain the transfer function, the pistons, connecting rods, and crankshaft, which are the operational parts inside the engine, were removed and the engine was excited using a minishaker. Fig. 3 shows the position of two minishakers within the cylinder liners.

2.2 Finite Element Model of the Engine Mounting Bracket

The powertrain is attached to the vehicle body by four mounting brackets. The target region for this analysis is the

high frequency band of the second-order to tenth-order components of engine revolution, where there is concern that engine mounting bracket resonance will have a negative effect on the vibro-acoustic performance. For the spot welded stamped shell type bracket, an FE model was generated with shell elements, including a spot weld model. The bracket FE model was tuned using the experimental transfer function and modal parameters obtained from the initial prototype part, increasing the accuracy of the model. This method is also applied to other FE models.

Fig. 4 shows the mode correlation of the final rear bracket FE model with the experimental model. A close correlation was obtained, with MAC values of 0.70–0.95 in at least ten modes.

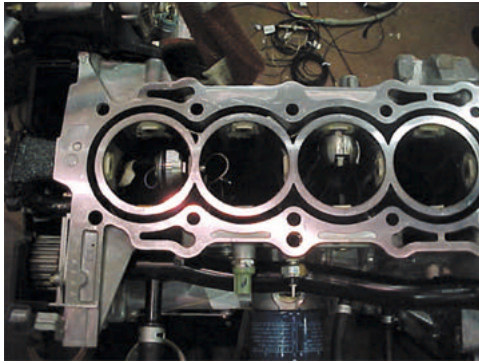


Fig. 3 Minishaker positioning in cylinder liner

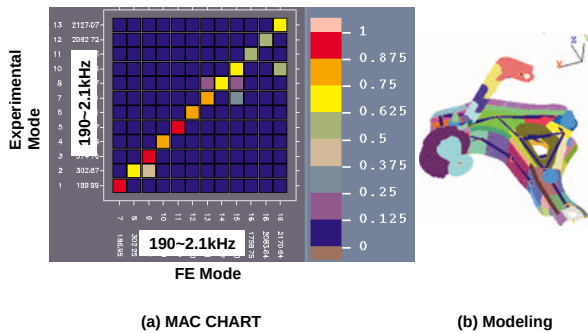


Fig. 4 Final rear bracket FEM and MAC correlation with test model

2.3 Finite Element Model of the Front Subframe

The front subframe supports two of the four mounting brackets and is fastened to the vehicle body by six bolts. The mode correlation for this component model was improved by focusing on the behavior at locations where panels are touching each other between the spot welds.

Fig. 5 shows the mode correlation between the experimental model and the final FE model through 800Hz and above. A close correlation was obtained, with MAC values of about 0.75 or more.

2.4 Test Body Model

The body was defined by a noise transfer function matrix for cabin noise and point mobility corresponding to the degrees of freedom of subframe attachment bolts (six points) and engine mounting (two points).

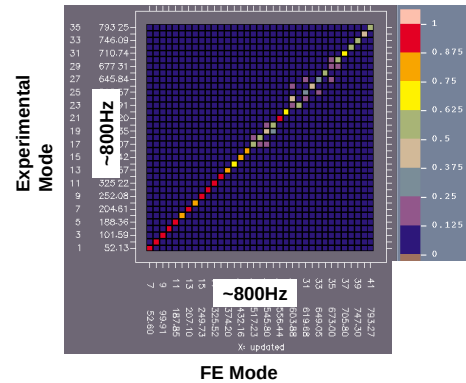


Fig. 5 Correlation chart and final FE mesh of subframe

3. Substructure Model

Two sub-assemblies are made from the various component models. The first assembly is the union of the powertrain and four mounting brackets. The next is the union of the subframe and body. Fig. 6 shows the assembly process and FRF matrix required in the case of connection between the subframe and body.

The point mobility matrices H_{ss} (s: connection degree of freedom) for the various components are connected through the connection stiffness matrix K . The body matrix H_{is} (i: output degree of freedom, s: connection degree of freedom) is the body noise transfer function for the subframe connection degrees of freedom, and the subframe matrix H_{sj} (s: connection degrees of freedom, j: input force degree of freedom) is the transfer function for the responses to body connection degrees of freedom with respect to the input force from front and rear bracket attachment points. The subframe matrix H_{ks} (k: other degrees of freedom) and H_{kj} are the vibration response of points representing the component structure. These are added to the matrices in order to visualize operational deflection shapes following assembly. Similar matrices are also defined for the powerplant and mounting brackets.

The substructure model thus subassembled is verified by actual measurement transfer function. Fig. 7 shows a comparison of the actual measurement and calculated transfer functions for the bracket tip response level with respect to vertical excitation of the crank bearing. The various sharp peaks result from bracket resonance. Fig. 8 shows a comparison of the body noise transfer function from the engine mounting point on the subframe to the driver's left ear as verification of the subassembly of the

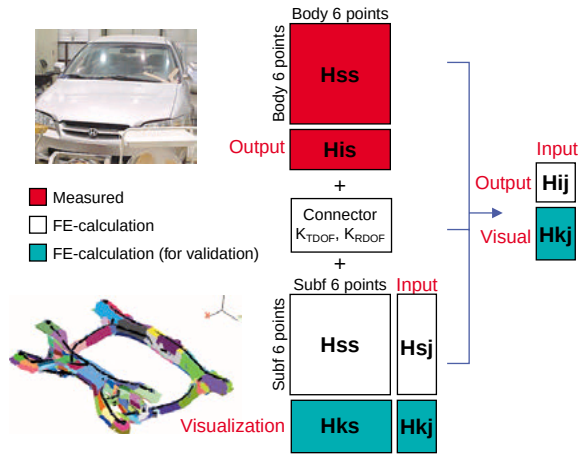


Fig. 6 Substructuring component and assembly matrix for subframe to body model

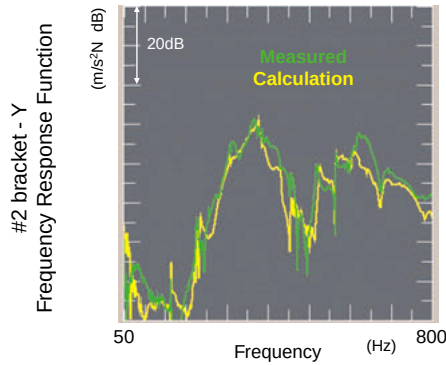


Fig. 7 Measured and predicted FRF between crankbearing force and engine bracket vibration

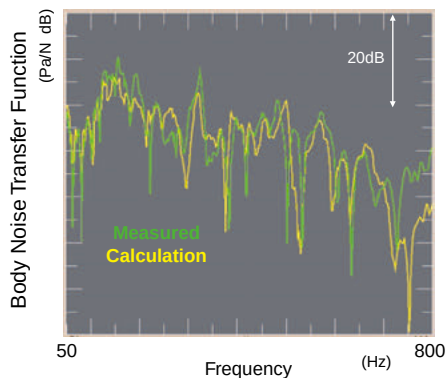


Fig. 8 Measured and predicted body noise transfer function at the engine mounting points on the subframe for the driver's left ear

subframe and the body.

It is also possible to analyze the importance of a single unit mode, in terms of the post-assembly mode, by correlation of the unit mode with the post-assembly mode.

In this paper, the steering system and suspension arms attached to the subframe are not modeled. After analysis of their respective effects during the initial test, the model also made combined use of the FRF matrix with the components that have recognized impact connected to the subframe and the body.

4. Identification of Engine Input

The forces generated inside the engine are calculated by the indirect force identification method [ref2]. This becomes an input condition for the vehicle model assembled by substructuring.

4.1 Direct Identification of Known Forces

First, the inertia force is calculated from the engine specifications (pistons, connecting rods, crankshaft) after synchronizing with the actual measured speed of the engine in operation. Next, the combustion force is derived from actual measurement of the combustion pressure signal. Phase accuracy is assured during analysis of high-order regions by using the camshaft trigger and flywheel timing pulse to convert all measurement signals from time units in order to make them compatible with crankshaft-angle units.

The inertia force and combustion force are calculated in advance. Then the transfer from the crank pin to the crank bearing is defined using the crankshaft FE model, and the force generated by the crank bearing is predicted. The inertia force and combustion force are then joined as the force generated in the crank bearing to create the known force vector. In the present analysis, known force vectors were created for the 0.5 to 10th order components of engine harmonics in 0.5-order increments.

4.2 Identification of Residual Force

Residual force was determined by using equations (2) to (7).

The powertrain vibration response generated by the known force $\{F_{\text{known}}\}$ can be derived by calculating the FRF matrix $[H_{ij}]$ from the input force degrees of freedom to the engine and bracket vibrations in the virtual vehicle (Equation 2).

All of the forces generated during operation are not known. Therefore, the vibration $\{X_{\text{known}}\}$ calculated from the known force above is subtracted from the total vibration generated during actual operation $\{X_{\text{measured}}\}$. The resultant $\{X_{\text{residual}}\}$ is estimated to be the outcome of the remaining residual force $\{F_{\text{residual}}\}$. The components of the known force are removed from the vibration caused during operation (Equation 3). The vibration during operation, which now contains only the components resulting from residual force, and the FRF matrix of the full vehicle model are used in a matrix inversion equation to calculate the residual force (Equation 4). This residual force $\{F_{\text{residual}}\}$ is added to the known force $\{F_{\text{known}}\}$ that was initially estimated, thus yielding the force vector $\{F_{\text{predict}}\}$ of all forces generated for each harmonic component (Equation 5). In order to verify the validity of this predictive process, it is necessary to calculate the engine and bracket vibration $\{X_{\text{predict}}\}$ resulting from all forces generated (Equation 6) and compare this with measured vibration (Equation 7).

$$\{X_{known}\}=[H_{ij}]\{F_{known}\} \quad (2)$$

$$\{X_{residual}\}=\{X_{measured}\}-\{X_{known}\} \quad (3)$$

$$\{F_{residual}\}=[H_{ij}]^{-1}\{X_{residual}\} \quad (4)$$

$$\{F_{predict}\}=\{F_{known}\}+\{F_{residual}\} \quad (5)$$

$$\{X_{predict}\}=[H_{ij}]\{F_{predict}\} \quad (6)$$

$$\{X_{error}\}=\{X_{predict}\}-\{X_{measured}\} \quad (7)$$

4.3 Effects of Matrix Size on the Prediction of Generated Force

The prediction accuracy for force generated during operation is determined by the FRF matrix used when identifying known force and residual force. In order to stabilize the matrix inversion equation, the basic degrees of freedom were set to 90 degrees of freedom for the crank bearing side on the powertrain frame and the cylinder head, while extended degrees of freedom were set for the bracket attachment points (48 degrees of freedom) and for the bracket tips (18 degrees of freedom) as these areas were considered important points for the assembly. Fig. 9 shows the differences in generated force estimates resulting from matrix models of different sizes determined by the basic and extended degrees of freedom. It is apparent that both basic and extended degrees of freedom are necessary in order to stabilize the generated force prediction calculations.

The initial prediction calculation used matrix inversion for the main body of the powertrain alone, excluding brackets. However, comparison of the calculation results for the powertrain with the four brackets attached confirmed that this matrix provided insufficient accuracy. When carrying out force identification prediction calculations in this way, it is essential to have predetermined the FRF matrix required for force identification using the powertrain model that includes the bracket FE.

Fig. 10 shows a comparison of values for 5th order vibration in the bracket tips as measured and as predicted by identification matrices of different types and sizes. During actual operation, damping due to the engine mounts occurs so the prediction results from the model with only the engine and brackets attached formed sharp peaks that were not damped and thus were different from actual measurements. There is a close match with measurement data under the other matrix conditions, which suggested that calculation of residual force using the full vehicle model FRF matrix is necessary to achieve optimum accuracy in prediction calculations.

A simple approximate modeling method for representing the body characteristics of the full vehicle model is to replace the actual measured point mobility with a fixed rigid connection to the ground. The validity of this method for the present analysis has already been verified by the prediction results shown in Fig. 10. This method is suitable for evaluation of engine mounting transfer force or vibration when there is no body model.

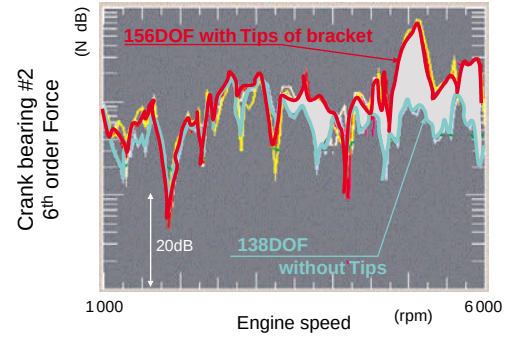


Fig. 9 Crankbearing #2 6th order force estimate for different matrix models

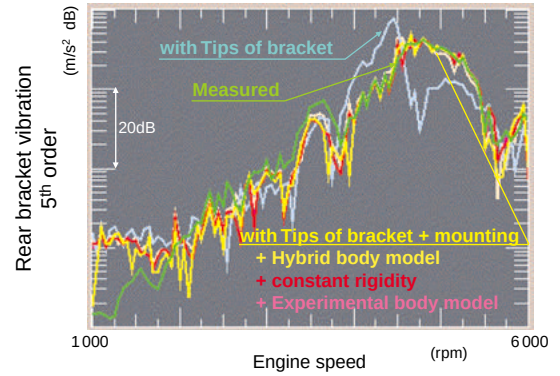


Fig. 10 Measured and predicted 5th order bracket #2 axial vibration for different matrix models

5. Verification of the Virtual Vehicle Model

Application of the identified force to the virtual vehicle yields predictions of vibration from all components and all connections between components, and prediction of cabin noise, and enables evaluation of force transmissibility. Essential matters related to vibro-acoustic performance are vibration levels on the engine side and the body and subframe side, the force transmitted through the mount as a result of internal engine forces, and the noise level in each area of the interior. The accuracy achieved by the present virtual model is described below.

5.1 Vibration in the Engine Side Bracket

Fig. 11 shows a comparison of the actual measured and predicted levels of 6th order vibration in an engine bracket. It was decided to set the FRF matrix used in force identification to specify 156 degrees of freedom in response to the input force with 24 degrees of freedom, making the matrix ratio of the matrix inversion equation 7 or higher. A close correlation was obtained, and similar prediction accuracy was also achieved with other orders of harmonics and other brackets.

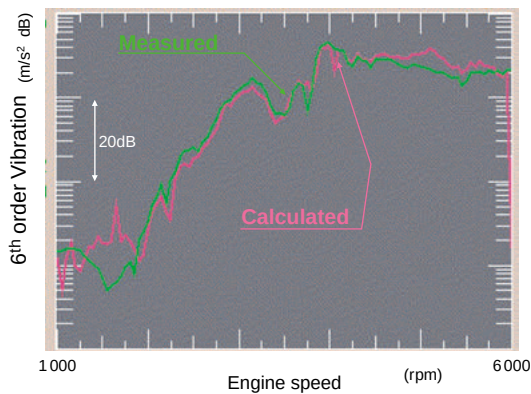


Fig. 11 Measured and predicted vibrations for 6th order bracket #2 axial vibration

5.2 Prediction of Cabin Noise

The ultimate objectives of the virtual vehicle are to predict cabin noise during actual operation and to predict the extent of the changes that will result from structural changes in component levels. Fig. 12 shows a comparison of the calculation results from the virtual model and the actual measurements of 6th order cabin noise. The calculation results show two sets of specifications. The difference is in the range settings for the experimental model and the FE model. Conducting simulations with models defined differently in this way demonstrates the importance of the following parameters to prediction accuracy:

1. Quality and accuracy of the FE component model
2. Range of the FRF matrix used for force identification
3. Reliability of the complex stiffness constant for the engine mounts

The quality of the FE model is increased when the model is constructed using the criteria from the initial prototype part. This can also be revised whenever there are subsequent opportunities to build prototypes. Any FRF matrix that is constructed from test data requires considerable effort in measurement of the previous models of the vehicle body, as well as in the engine bench tests.

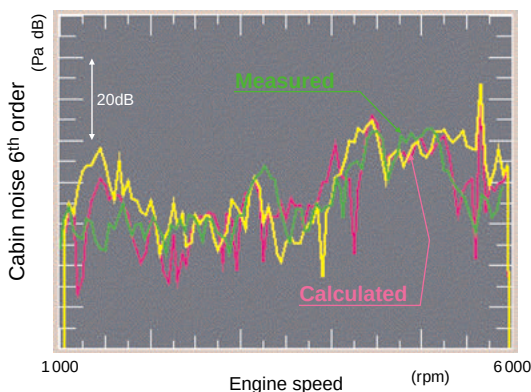


Fig. 12 Measured and predicted 6th order interior noise for 2 virtual car models

6. Application to Product Development

The present method was applied as a development tool in the initial stage of design on the basis of the above verification results.

6.1 Application of New Parts in the Virtual Model

Prototype components (hereafter referred to as A spec) were modeled by FE and incorporated into the component model (prototype engine and body) used as a test base. The resulting full vehicle model was then used for prediction. Fig.13 shows the FE model for the A spec engine mounting brackets and subframe. It was determined in the initial prediction that high noise levels resulted from component characteristics at various engine speeds. Countermeasures were applied in the form of modifications to the FE component model, and the effects were predicted in advance by the virtual vehicle model.

Verification was carried out using actual prototypes after the model predictions were made. Fig. 14 shows a comparison of predicted and measured 6th order vibration

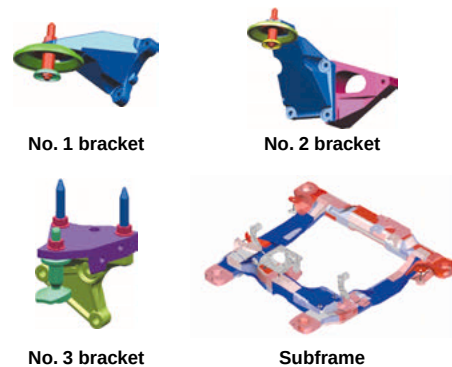


Fig. 13 A-spec parts: engine mounting brackets and front subframe

in the No. 3 engine mounting bracket. It is apparent that the model prediction accuracy is valid. Fig. 15 shows a comparison of predicted and measured body noise transfer function in the subframe. It is clear that the subframe modeling and its assembly in the virtual vehicle, with the exception of one frequency range, were valid. Points that fail to agree with the measurement resulted from structural

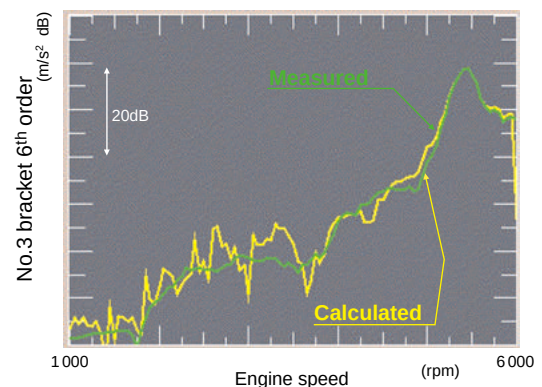


Fig. 14 6th order vibration of the No.3 engine mounting bracket on a 2nd stage prototype vehicle

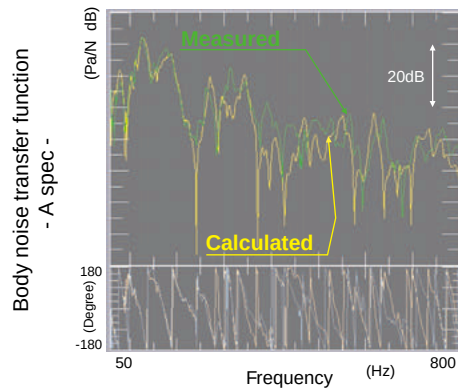


Fig. 15 Measured and predicted body noise transfer function, using improved A-spec subframe in the virtual car model

aspects that were specific to the prototype specifications, and these made up a small proportion of total predicted cabin noise.

6.2 Analysis of the Noise Transfer Path

The contribution of each ultimate transfer path for cabin noise was analyzed in detail.

6.2.1 Force transmissibility

Force transmissibility can be evaluated directly in terms of the frequency response characteristics of unit force inputs by their respective paths. It is possible to calculate the force transmitted through the engine mount during actual operation by using matrix calculation with the identified forces generated inside the engine.

Fig. 16 presents a comparison of the 6th order components of transmitted force. This shows clearly how the force transmitted through the No. 3 engine mount changed due to the change from the base specifications, which were the initial specifications, to the A spec. The transmitted force is shown to be reduced by the new design in nearly all ranges, but a new transmitted force can be seen to peak in the vicinity of 5500 rpm.

In this way it is possible to identify the force, including the effects of structural connections and the total effect from the dynamic actions of the connections between components. Consequently, it is also possible to apply this

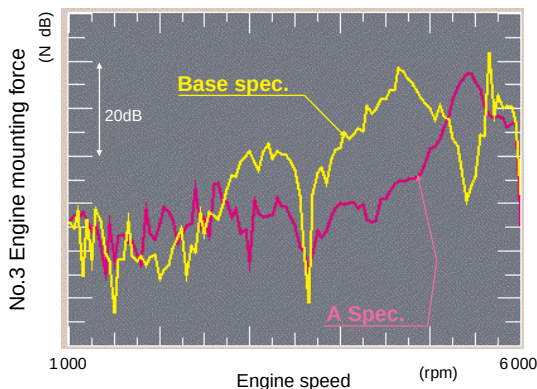


Fig. 16 Calculated force transmitted through #3 engine mounting : virtual vehicle vs. base specification

force as input to the FE model prior to substructuring. Since construction of the substructuring model can be limited to the degrees of freedom of the connection points, the forces at the connection points can be derived from a virtual model that is limited only to the necessary degrees of freedom. Those forces can be applied to the full-size FE model, so that evaluation of all of the degrees of freedom of the FE model can be carried out as a detailed analysis of the model, while at the same time decreasing the time required to construct the model.

6.2.2 Transfer path analysis of the engine mounting

The contribution of the transfer path through each engine mount to cabin noise is analyzed using ordinary path analysis.

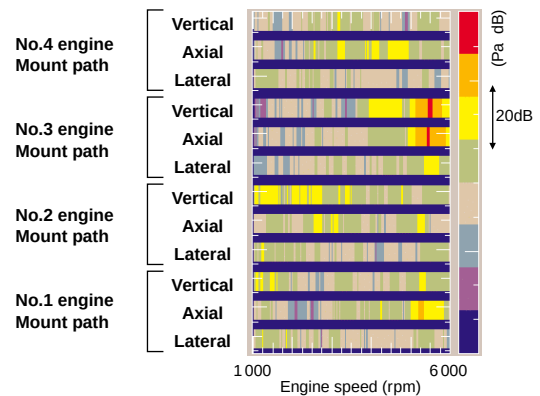


Fig. 17 Engine mount path contribution to interior noise simulation on A-spec parts vehicle

Fig. 17 is a color bar graph that shows these contributions according to engine speed. This graph shows that of the four engine mounts, the No. 3 bracket has a significantly high contribution rate in the 5500 rpm region.

Fig. 18 shows the contribution vector for cabin noise at 5500 rpm. The No. 3 engine mount is the vector that has the highest contribution. This is followed by the contribution from the No. 1 engine mount, which has generally the same phase as the total cabin noise.

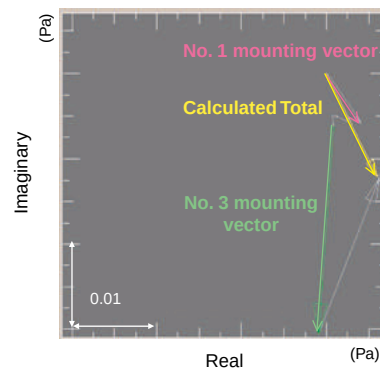


Fig. 18 6th order vector plot of interior noise contributions at 5500rpm on A-spec parts vehicle

6.2.3 Detailed analysis of the transfer path

The force transmitted through each mount is determined by the force generated inside the engine and the transfer characteristics of the engine structure. Conducting analysis of the kind described in the previous section will clarify the contribution of the various independent paths inside the engine. Fig. 19 shows a color map of the contribution from forces inside the engine to the force transmitted through No. 3 engine mount. The interior engine forces follow 24 paths that are axial, lateral, and vertical to the crank bearing and cylinder head.

It was made clear that the predominant contributions are from forces vertical to the crank bearing and to the cylinder

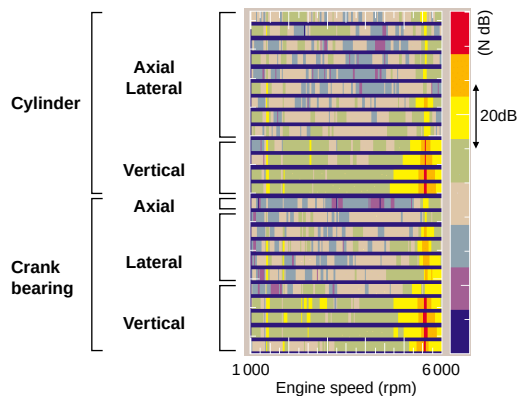


Fig. 19 6th order path contribution from interior engine forces to #3 engine mount force on A-spec parts vehicle

head. However, in the 6th order component, these counteract each other because their phases do not coincide. It is evident that the crank bearing axial force becomes significant at 5500 rpm.

The virtual vehicle model deals with the interactions of all component elements from the engine interior to the cabin interior. Consequently, it also makes it possible to analyze how the various independent paths inside the engine contribute to interior noise.

Fig. 20 shows the contribution of interior engine forces to cabin noise in the form of color bars. The same path definitions as in Fig. 19 are used, but the result includes all component elements of interior noise. It is possible to examine this in detail by using engine mount transfer path analysis to complement the force transmissibility analysis. It is apparent that the largest contributions to 6th order interior noise at 5500 rpm are from the vertical input path through the crank bearing and the vertical input path through the cylinder head, while the next largest contribution is from the axial path through the crank bearing. This shows a similar tendency to that in the contribution analysis of the force transmitted through the No. 3 engine mount. That is, the contribution of interior engine forces to cabin noise (Fig. 20) correlates closely with the contribution of force transmitted through the No. 3 mount (Fig. 19), and it becomes clear that the predominant contribution to cabin noise is from the force transmitted through the No. 3 mount (Fig. 17).

Fig. 21 shows the vector contribution of interior forces to 6th order cabin noise when the engine is operating at

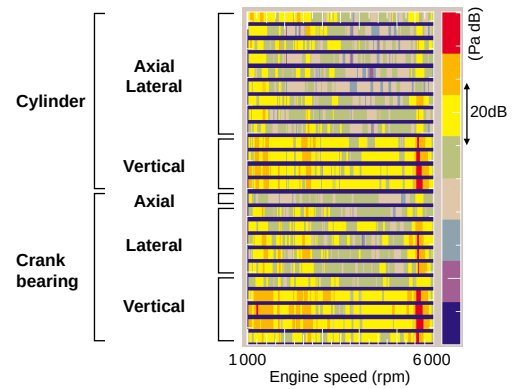


Fig. 20 6th order path contribution from interior engine forces to interior noise on A-spec parts vehicle

5500 rpm. The greatest contributions are from the vertical force through the crank bearing and the vertical force through the cylinder head. These both have generally the same magnitude but opposite phases. It is evident, therefore, that the two paths with the largest contributions are in opposite directions, so that they actually cancel each other out. The next largest contribution is from the axial crank bearing force, and since this is generally in phase with the vector sum, it clearly has a significant effect. The total vector sum also agrees very closely with the actual measured noise vector, verifying the high reliability of the model of the internal engine force transfer path.

This virtual vehicle can be applied to evaluation and analysis of vibro-acoustic design performance at every stage in the design cycle. The force transmitted from the engine to the connections between components, the force

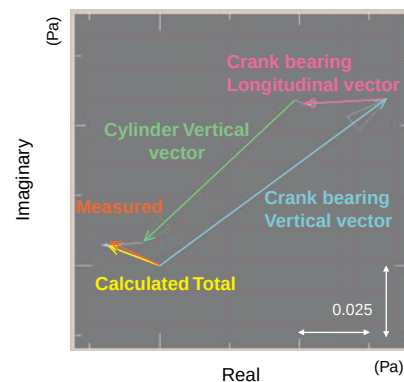


Fig. 21 Vector contribution of interior engine forces to 6th order interior noise at 5500rpm on A-spec parts vehicle

transmitted through the engine mount during operation, the vibration at the mounting bracket tips, the body vibration, and the body noise transfer function are evaluated respectively as system performance elements. Further, the ultimate target of evaluating their respective contribution to cabin noise can be achieved. The effect of the design of individual components on vibro-acoustic phenomena in an actual vehicle can be simulated with ease. Consequently, by continuously expanding and developing this model and synchronizing its use with design changes and prototype lots, it becomes possible to carry out quantitative management of vibro-acoustic objectives, system performance, and component drawings.

7. Conclusion

A method of virtual vehicle model analysis was developed that is suitable for use in decreasing the time required for the vibro-acoustic development process.

- This method makes it possible to evaluate cabin noise targets at every stage of automobile development. Since it allows effects on cabin noise to be taken into consideration at the initial stage of design, it also enables decisions to be made so that design specifications at the product planning stage will comply with cabin noise targets.
- It allows timely feedback to be provided to the component design team on how the deformation shape of subject components during operation, their transfer functions, their path contributions, their forces, and their vibration levels are positioned with regard to the targets.
- It enables path contribution analysis of cabin noise using both the interior engine forces and the forces transmitted through the engine mounts as references.
- When design changes to components are made, use of the virtual vehicle model allows rapid assessment of the changes regardless of whether the change requests are vibro-acoustic related or otherwise.

This method enables verification of contributions to cabin noise and clarification of the underlying factors in failure to meet targets. Consequently, this analysis procedure is effective in facilitating collaboration with component designers in the course of conducting vibro-acoustic management. Use of the procedure makes it possible to give quantitative consideration to product factors such as cost, weight, and strength in conjunction with vibro-acoustic product quality targets, as well as providing an effective means for continuous monitoring of cabin noise objectives.

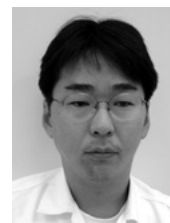
References

- (1) Sakai, T.: Development Procedure for Interior Noise Performance by Virtual Vehicle Refinement, Proceedings of the SAE Noise & Vibration Conference, 2001-01-1538 (2001)
- (2) Sumi, K.: Development of a Hybrid Model for Powerplant Vibration, Proceedings of the SAE Noise & Vibration Conference, 1999-01-1656 (1999)
- (3) Wyckaert, K.: Flexible Substructuring Based on Mixed Finite Element and Experimental Models — A Step Ahead of Transfer Path Analysis, Proceedings of the 14th International Modal Analysis Conference (1996)
- (4) Wyckaert, K.: Hybrid Substructuring for Vibro-acoustical Optimisation — Application to Suspension-Car Body Interaction, Proceedings of the International Noise & Vibration Conference (1997)

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