

Experimental Method Extracting Dominant Acoustic Mode Shapes for Automotive Interior Acoustic Field Coupled with the Body Structure

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ABSTRACT

For a numerical model of vibro-acoustic coupling analysis, such as a vehicle noise and vibration, both structural and acoustical dynamic characteristics are necessary to replicate the physical phenomenon. The accuracy of the analysis is not enough for substituting a prototype phase with a digital phase in the product development phases. One of the reasons is the difficulty of addressing the interior acoustical characteristics due to the complexity of the acoustical transfer paths, which are a duct and a small hole of trim parts in a vehicle. Those complex features affect on the nodal locations and the body coupling surface of acoustic mode shapes. In order to improve the accuracy of the analysis, the physical mechanisms of those features need to be extracted from experimental testing.

The accuracy of the vibro-acoustic coupled system model for the low frequency range depends on how accurately modal characteristics are represented at the input, output, and the structure-acoustic coupling surface. Therefore, this study focus on extracting the detailed acoustic mode shapes on the coupling surface for the improvement of the model accuracy. The non-linear least square method as the one utilized in the previous study was applied to the new test data sets of an actual vehicle.

In the previous study, it had one remaining issue, which was how to extract acoustic mode shapes in the frequency range of higher damping and higher acoustic modal densities. In order to solve this issue, the number of acoustic excitation was increased considering acoustic mode shapes. The eight loudspeakers were utilized as an acoustical excitation to excite acoustic modes evenly in the acoustic interior dimensions for higher frequency.

With the results of this testing, the acoustic modes of an actual vehicle with heavy damping were accurately extracted as the complex mode shapes of no phase lag between nodes, which looks similar the un-coupled normal modes without rotation in animation up to 200Hz. The synthesized FRFs were replicated well with only the extracted several dominant acoustic mode shapes.

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INTRODUCTION

A finite element (FE) model has being applied to analyze the performance of the noise, vibration and harshness (NVH) in the passenger vehicle development process. Generally, the vehicle FE model, a vibro-acoustic system, consists of the

structural model, which has a couple of million degree of freedom (DOF), and the interior acoustic field model, which has a couple of hundred thousand DOF. The number of the coupled modes, which are the dynamic characteristics of the structural model coupled with the interior acoustic field model, of the vehicle FE model exists a couple of thousands



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in a frequency range of 0-200 Hz. For an accurate prediction of the NVH performance, it is necessary to create the model having precise dynamic characteristics of both the structure and the acoustic.

The previous literature has mainly focused on the accuracy of the structural vibration of the model comparing FE model results with experimental test results. In most of the reports, it was discussed about the comparisons of the frequency response functions (FRF), the comparisons of the operating deflection shapes, and the comparisons of mode shapes with Modal Assurance Criteria (MAC). Each of these has some issues to validate the accuracy of the FE model. First of all, since FRF is a relative robust measurement, it is suited characteristic to compare with the FE model results. However, in order to validate the model accuracy, the comparisons of FRF are not enough, especially for the model having more than a couple of million DOF, because it only gives the magnitude of amplitude and phase. Secondary, for the comparisons of the mode shapes between FE model and the test, it has a technical issue to extract the mode shapes from the actual vehicle by conducting experimental modal analysis for higher frequency. Thirdly, for FE model issue, it has lack of exactness to compare the vibration of structural FE model with the one of the actual vehicle consisting of the structure and the acoustic. Since an actual vehicle has an essential issue that the dynamic characteristics of the vehicle show a variation, the way to validate the model has no right answer from a viewpoint of the predictive precision of the sound pressure.

On the other hand, much less attention has been paid for the accuracy of the dynamic characteristics of the interior acoustic field model. Since the number of the acoustic modes is much less than the one of structural modes, it has possibility to validate the accuracy of the acoustic model with the use of mode shapes of the interior acoustic field in the frequency range of interest, 0-200 Hz. In the previous studies⁽¹⁶⁾, it was confirmed that the accuracy of the noise transfer function (NTF), from input force on the structure to output at the driver's ear position, was dramatically improved by the use of the acoustic FE model that the accuracy of the acoustic mode shapes was improved. Therefore, it is worth of the study to improve the accuracy of the acoustic model.

This paper explained the way to extract the detailed interior acoustic field mode shapes from an actual vehicle in the range of 0-200Hz with the use of the example of experimental test conducted for SUV type vehicle.

2. EXPERIMENTAL MODAL ANALYSIS

An investigation was undertaken to see if the same procedure as the experimental acoustic modes could be extracted by the same method as the experimental structural modes.

2.1. Measurements

FRFs of sound pressure were measured at the surface and inside of the interior trimmed parts excited by four speakers simultaneously. The measured points were set at each 0.2 meter or 0.3 meter interval and then the one for SUV was set for approximately 350 points. The experimental test image and the measurements are shown in [Picture.1](#) and [Figure.1](#), respectively.

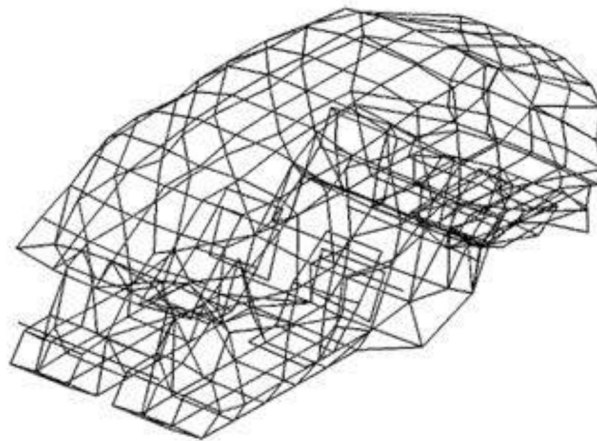


Figure 1. Measurement points



Picture1. Photograph of the experimental setup (1)

[Figure.2](#) shows the speaker locations for SUV. At least one of the speakers is set to excite the 1st mode of Back & Forth direction, the one of Lateral direction, and the one of Vertical direction. This setting follows the way to select the driving point, which is selected for the location of the maximum amplitude of the modes, for structural excitation. A spherical speaker, 0.2m in diameter, made in-house that incorporated six speaker units was used as the sound source. It is confirmed that the influence to dynamic characteristics of the interior acoustic field due to the placement of these

speakers and the characteristics of Omni-direction up to 1000Hz could be ignored.



Picture2. Photograph of the experimental setup (2)

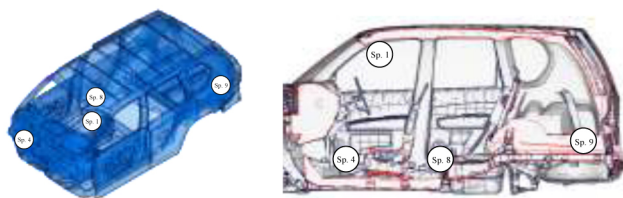


Figure 2. Layout of 4 sound sources

One of the measured FRFs of the sound pressure is shown in Figure.3. The driving point, No. 1, of Figure.3 is located close to the front window of the driver side. The response location, No. 2, is located close to the center of the rear left door. Generally, FRF of the sound pressure, $H(\omega)$, is defined as follow:

$$H(\omega) = P(\omega) / Q(\omega) \quad (1)$$

where $P(\omega)$ and $Q(\omega)$ are the sound pressure and the volume acceleration of acoustic excitation, respectively.

In this test, the input voltage to the speaker was used as the reference signal of the FRFs. This input voltage was converted to volume acceleration. A calibration was checked by the reciprocity of the structural excitation with electromagnetic shaker on the automotive structure to the driver's ear position, where the one of the loudspeaker was located and acoustical excitation vice versa. For the acoustic modal analysis, the reciprocity of those eight loudspeakers was also checked. The acoustical responses were measured by means of a roving array of microphones. The number of acoustical responses was approximately 350 for each acoustical excitation. As shown in Figure.3, the system has high damping and gradual phase change because there are no clear peaks in the FRF of the sound pressure.

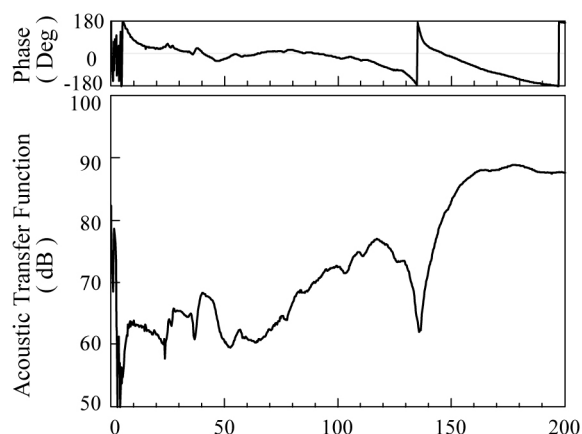


Figure 3. Measured acoustic transfer function

2.2. Driving Point Measurements

Though the driving point measurement was measured or estimated to calculate FRF, there are only few reports about the treatment of the driving point measurements for experimental acoustic modal analysis^(8,16). The speaker as a sound source of the test has the characteristic proportional to one over the distance between sound source and response since it can be considered as enough small pulsating sphere in the frequency range of interest. Therefore, when the sound pressure response at the speaker surface is utilized as the driving point measurements, it gives rise to the issue that the dynamic characteristics of the system could not be seen in the data due to high value of the response relative to the others. One of the measurements is shown in Figure.4. In this study, the experimental acoustic modal analysis was conducted with no driving point measurements for the above issues.

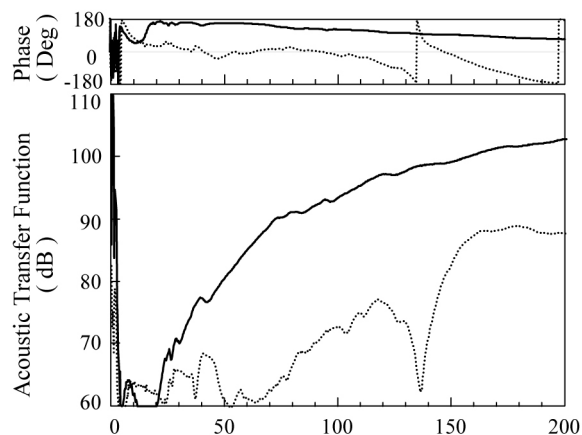


Figure 4. Comparison of acoustic transfer functions
(Dotted line : point 100mm away from Speaker, Solid line : point close to Surface of Speaker)