

Development of Trimmed Body Model for Noise and Vibration

Nobuo AIZAWA*

Fumihiko IDE*

Yasuhiko KAGIYAMA*

ABSTRACT

Modeling Technology for a trimmed body that can predict noise and vibration with a short computation time and high accuracy was developed. In order to reduce the total degrees of freedom, parts that exert only a small influence on the acoustic characteristics were simplified from shell elements to concentrated mass models. The acoustic significance of trim parts was determined both experimentally and computationally. Also, Automated Multi-Level Substructuring (AMLS) was applied resulting in a significant reduction in computation time. Finally, the trimmed body prediction results agreed closely with experimental data.

1. Introduction

In recent years, improved CAE prediction estimation is being demanded during development in the prediction of noise and vibration in automobiles. Since this requires detailed modeling, the degree of freedom (hereafter referred to as DOFs) of FE models has increased significantly. At the same time, analysis frequency ranges have also expanded leading to an increase in the computation time. The goal of this development is to reduce the computation time while maintaining model accuracy.

Typically, three methods are available for reducing the computation time. The first method is to combine experimental and FE analyses⁽¹⁾. This method makes use of a model with fewer DOFs. In this case, the accuracy of the FE model is supplemented by adjusting stiffness and mass to improve correlation with the experiment. Its application in the development stage is difficult because prototypes typically do not exist. The second method makes use of Component Mode Synthesis (CMS) for converting the FE model to the modal domain by which DOFs not necessary for analysis are eliminated⁽²⁾. This method requires considerable time since the detailed FE model must be converted to the modal domain. Furthermore, the sub-structuring of the model into modal models requires high-level knowledge and technique. In this research a third method, AMLS was used for reducing the DOFs and the computation time. In addition

to using AMLS, many components were reduced to concentrated mass elements if accuracy was not compromised. In this study, the accuracy of AMLS (the newly proposed computation method) was verified and the reduction of computation time was quantified.

2. Reducing the Computation Time using AMLS

AMLS is an eigenvalue computation method developed by Bennighof⁽³⁾. The constrained mode method⁽⁴⁾, which is

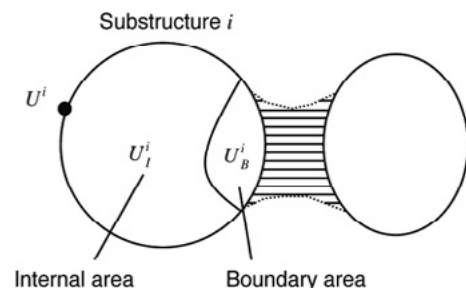


Fig. 1 Definition of substructure

* Tochigi R&D Center

the basis of AMLS, is described here. As shown in Fig. 1, the entire structure is divided into substructures and each substructure is classified into boundary and internal areas. Thus, the displacement U^i of the substructure i is classified into the displacement of the internal area U_I^i , and displacement of the boundary area, U_B^i . The displacement of the internal area is expressed as given below, by the displacement of boundary area and modal matrix of the internal area.

$$\{U^i\} = [T_B^i]\{U_B^i\} + [\Phi_I^i]\{\xi_I^i\} \quad (1)$$

The first term in Eq. (1) expresses the deformation of the internal area due to displacement of the boundary area, and is a static condensation matrix according to Guyan. The second term expresses the deformation from the mode shape of the internal structure when the boundary area is restrained. It is defined to improve the accuracy of the static condensation in the first term and obtained from Eq. (2).

$$([K_I^i] - \omega^2 [M_I^i])\{\Phi_I^i\} = 0 \quad (2)$$

The displacement of substructure i can be expressed as below using Eq. (1) and (2).

$$\{U^i\} = \begin{Bmatrix} U_B^i \\ U_I^i \end{Bmatrix} = \begin{bmatrix} I & 0 \\ T_B^i & \Phi_I^i \end{bmatrix} \begin{Bmatrix} U_B^i \\ \xi_I^i \end{Bmatrix} = [A^i] \begin{Bmatrix} U_B^i \\ \xi_I^i \end{Bmatrix} \quad (3)$$

$[A^i]$ represents the conversion matrix by the mode synthesis method. After determining $[A^i]$ from Eq. (3), and pre and post multiplying with the eigenvalue equation of the entire structure, the eigenvalue equation of the entire mode system can be obtained.

$$([\hat{K}] - \omega^2 [\hat{M}])\{\hat{\Phi}\} = 0 \quad (4)$$

Substructure stiffness matrices with several thousand to several tens of thousands of steps are automatically generated by AMLS. As shown in Eq. (2), the substructure matrix is converted to modal DOFs by eigenvalue analysis. As a result, the DOFs of the entire model can be reduced, thereby achieving high-speed computation. Since multiple substructures are used, high-speed processing is also enabled by parallel computation.

Figure 2 shows the flow of computations using AMLS. Special settings are not necessary and the user needs to add

only a one-line description to the NASTRAN execution card for running AMLS. Furthermore, since the computed results are output in NASTRAN format, new techniques need not be acquired.

3. Reducing the Degrees of Freedom in the Modeling Method and Maintaining Accuracy

Figure 3 shows the components of the trimmed model and the ratio of DOFs. The DOFs of the model should be reduced to speed up the computation time. On the other hand, if detailed modeling is performed to enhance the accuracy, the DOFs will increase.

If the DOFs are reduced by simplifying the metal structural parts including the Body in White (hereafter referred to as BIW) and door panels, the accuracy of the model cannot be maintained. Therefore, the model should be constructed very accurately according to the drawings. Accordingly, the element size for modeling metallic parts (such as BIW) was approximately 15 mm maximum, where fillets with radius greater than 5 mm were also represented

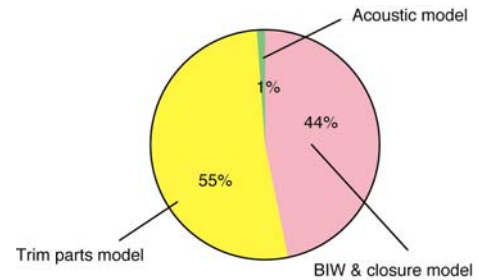


Fig. 3 Degree-of-freedom ratio

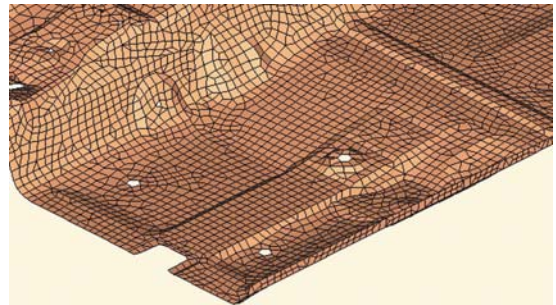


Fig. 4 Floor panel model

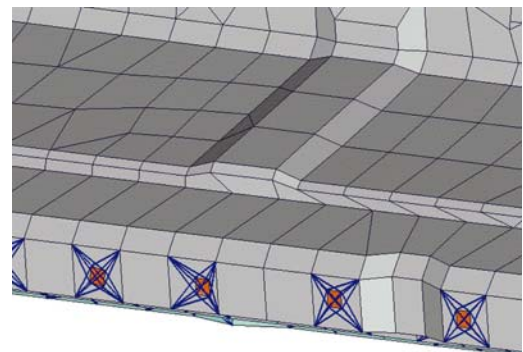


Fig. 5 FE model for Spot welding

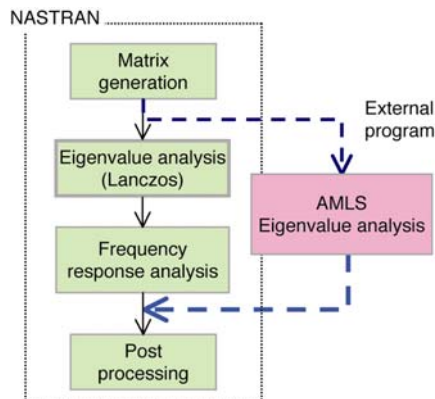


Fig. 2 AMLS process

(Fig. 4). The diameter of spot welds was also considered important for model accuracy⁽⁵⁾. These were represented using HEXA and RBE3 elements of MSC NASTRAN (Fig. 5).

In considering the number of DOFs needed to model the Acoustic Cavity accurately, it was confirmed that the

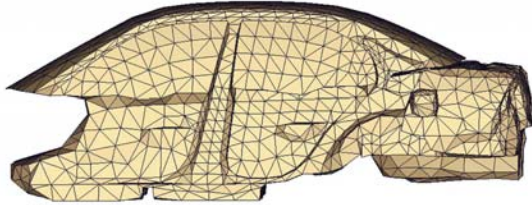


Fig. 6 Acoustic model for cavity

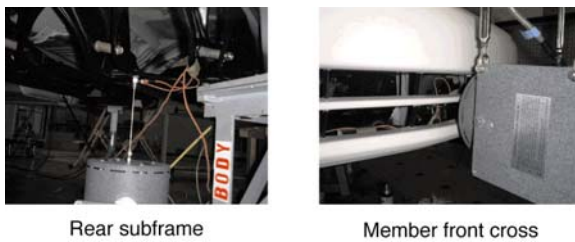


Fig. 7 Excitation points

frequencies and mode shapes of the cavity did not vary appreciably even if the passenger cabin interior surface was reproduced in great detail. Therefore, the element size of the acoustic model was taken to be less than $1/8^{\text{th}}$ of the wavelength of the maximum analysis frequency of interest, thus minimizing the total DOFs for the trimmed body model (Fig. 6).

The effect on model accuracy of reducing each trimmed part from a structural mesh to a concentrated mass was determined. Therefore, where accuracy was not compromised, trim parts were represented as concentrated mass.

3.1. Modeling of Trimmed Parts for Reducing the Total Degrees of Freedom

If all the trimmed parts are expressed by concentrated masses, the accuracy deteriorates. If insignificant parts are modeled in detail, the DOFs increase, leading to an increase in the computation time. Therefore, the effect of each trimmed part on the total acoustic response was examined. The trimmed parts were divided into groups that could be modeled by mass only and groups that had to be modeled structurally. The former were replaced by mass models to reduce the DOFs. This procedure is described below.

- (1) The trimmed parts were removed individually in an experiment. For parts where the acoustic frequency response function changed, a detailed shell element model seemed to be required, and the remaining parts were modeled using concentrated mass models.

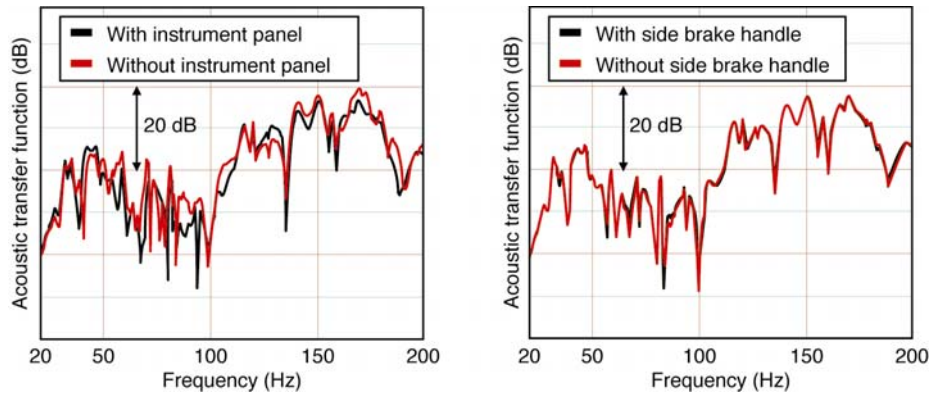


Fig. 8 Influence of trim parts

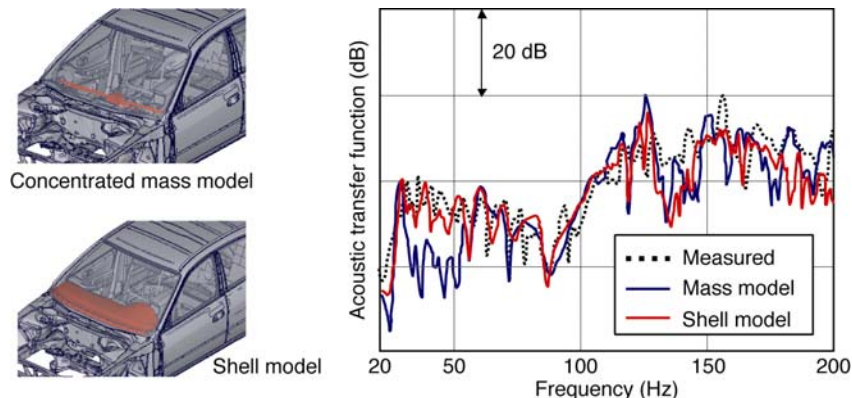


Fig. 9 Influence of instrument panel model

(2) Next, for the sensitive parts in step (1) detailed and concentrated mass FE models were calculated and compared. If the difference in acoustic response was small, a concentrated mass model was used.

3.2. Trimmed Part Removal Experiment

The experimental excitation for inducing trimmed part vibration was as follows: forces were input in the vertical direction of the front and rear sub-frames, and in the longitudinal direction of the member front cross. The sound pressure at the front and rear seats and in the trunk was measured (Fig. 7).

All trimmed parts were removed individually, the changes in acoustic response function before and after removal were compared, and priority was determined based on the degree of the changes. Figure 8 shows the frequency response function of the instrument panel with maximum change in acoustic characteristics, and the parking brake with minimum change in acoustic characteristics.

3.3. Simplification of Trimmed Parts Model

Trimmed parts judged to have a big influence on the acoustic characteristics in the experiment were modeled in detail using shell elements. Then the shell element model was replaced by a concentrated mass model, and the effects of the simplified model were verified. Figure 9 shows the results for the instrument panel, which had the largest effect on acoustic characteristics. It was confirmed that the correlation with the experiment deteriorated in the range of 30 to 60 Hz when the shell element model was replaced by a concentrated mass model. This indicates that a detailed model is required to ensure accuracy in this frequency band. Next, the replacement of the shell element model of the roof lining by concentrated mass model was studied. Figure 10 shows the results. Since the acoustic characteristics did not exhibit appreciable change, the roof lining was considered to be adequately represented by the concentrated mass model. By similar verification, parts with small effect on the acoustic characteristics were simplified, and the DOFs of the entire trimmed model were reduced by 23%.

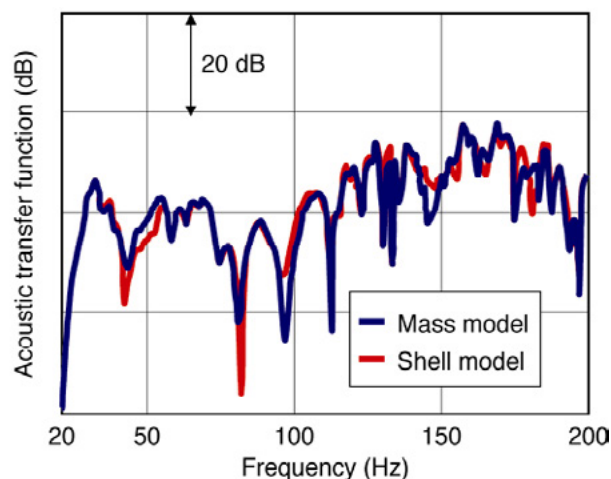
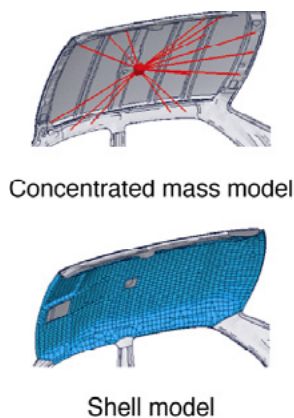


Fig. 10 Influence of roof lining model

3.4. Accuracy of Trimmed Body Model

Figure 11 shows the trimmed model consisting of a mix of parts modeled by shell elements and by concentrated mass based on the modeling decision process described earlier. The accuracy of the complete trimmed model was verified by comparing the frequency response function with the experiment. Specifically, the acoustic response due to vertical input on the front sub-frame was calculated. Figure 12 shows the results of the verification. The difference in level between the FE model and experimental values was within ± 5 dB up to 200 Hz. It was confirmed that the trimmed model had practical accuracy that could be used during development for the prediction of noise and vibration.

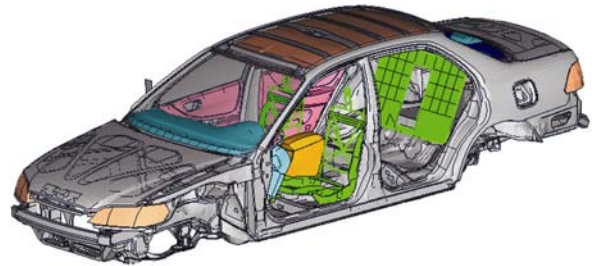


Fig. 11 Trimmed body model

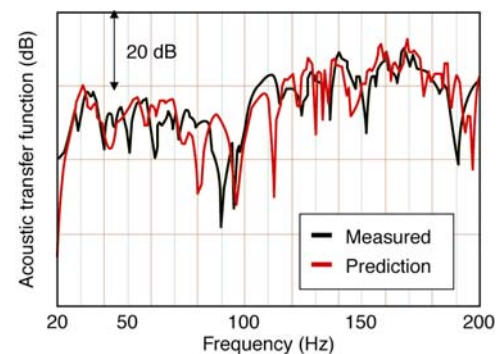


Fig. 12 Verification for trimmed body

4. Verification of Computation Speed

Frequency response analysis was carried out up to 300 Hz to verify the computation speed. The computation times were compared for cases where the DOFs of trimmed parts were reduced, were not reduced, and when AMLS was used. Figure 13 shows the computation times. The computation time was shortened by 20% when the DOFs of the trimmed parts were reduced. A further reduction in time of 66% was achieved when AMLS was used. Thus, the effect of reducing the DOFs of the trimmed body and the effect of using AMLS were substantiated.

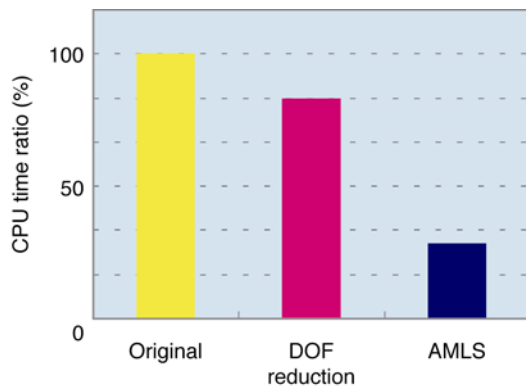


Fig. 13 CPU time ratio

5. Conclusion

The results of studies on reducing the DOFs of trimmed parts and computation methods led to the following conclusions:

- (1) It was found that treating some of the trimmed parts as concentrated mass elements was an effective and practical method of modeling the trimmed body with high accuracy and reduced computation time.
- (2) The effectiveness of this modeling method for reducing the DOFs of trimmed parts and the reduction in computation time of eigenvalue analysis using AMLS were verified.

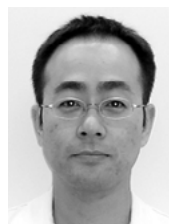
By application of the modeling technique mentioned above, the development in the prediction of noise and vibration by CAE could be carried out at a higher efficiency especially with regard to accuracy and time.

References

- (1) Kimura, F., Shimizu, H.: Parametric studies on correlation with experiments, Digital Engineering, p. 25, (2002) (in Japanese)
- (2) Society of Automotive Engineers: Jidoushagijutu Handobukku Kiso Riron Hen, p. 247 (1991) (in Japanese)
- (3) Bennighof, J.K., Kaplan, M.F., Muller, M.B.: Extending the frequency response capabilities of automated multi-level substructuring, SDM Paper AIAA-2000-1574
- (4) Nagamatsu, A.: Constrained Mode Synthesis, sub-structure synthesis, p. 103-104 (1991) (in Japanese)

- (5) Heiserer, D., Sielaff, J., Chargin, M.: High performance, process oriented, weld spot approach, 1st MSC Worldwide Automotive User Conference, (1999)

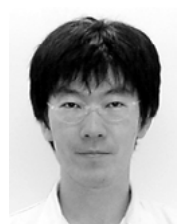
■ Authors ■



Nobuo AIZAWA



Fumihiko IDE



Yasuhiko KAGIYAMA