A Study on Optimal Design of Panel Shape of a Body Structure for Reduction of Interior Noise

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Key Words : Optimization, Panel Shape, Transfer Path Analysis(TPA), Panel Contribution Analysis(PCA), Vibro-Acoustic Coupling Analysis, Structure-Borne Noise, Noise Transfer Function(NTF)

ABSTRACT

This paper presents an optimal design process using beads on a body panel to improve interior noise of a passenger vehicle. Except modification of structural members, it is difficult to find effective countermeasures that can work for the intermediate frequency range from 100 Hz to 300 Hz which lies between the booming and low medium frequency. In this study, it is a major goal to find additional counter-measures for this intermediate frequency range by performing optimal design of beads on body panels. The proposed method for design optimization consists of 4 sub-steps, that is, a) problem definition, b) cause analysis, c) countermeasure development and d) validation. The objective function is minimization of interior noise level. The major design variables are the geometrical shape of a bead and combination of beads on the critical panels. Sensitivity analysis and optimization are performed according to the predefined process for an optimal design. It is verified that the proposed design decreases the level of noise transfer function above 5 dB.

1. Introduction

Reduction of interior noise level in a passenger vehicle is one of important quality criteria fpr development of a new automobile. Therefore, many engineers for NVH (Noise, Vibration and Harshness) eager to achieve a good acoustic quality, which leads to customer satisfaction.

Interior noise of a passenger car consists of structure-borne and air-borne noise. Especially, the structure-borne noise results from vibration of thin body panels which encompass the acoustic cavity, and the panels are excited by external forces from engine, road, etc. Therefore, it is necessary to decrease excitation forces, to increase attenuation level of mounting bushes and to improve dynamic characteristics of a body structure.(1)

In order to improve dynamic characteristics of a body structure, structural countermeasures, which are reinforcement of a structural member, increase of panel stiffness and attenuation of a body panel, can be applied. On the one hand, non-structural counter-measures, such as anti-vibration pad, insulation, composite materials, and so on, are available.

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Major researches relevant to interior noise may be categorized into two groups in terms of countermeasures. The first one is modification or optimal design of a structural member(2~5). The second one is the optimal design of the anti-vibration pad(6~8). Therefore it is necessary to make a research regarding shape design of a thin body panel, insulation, composite material, etc.

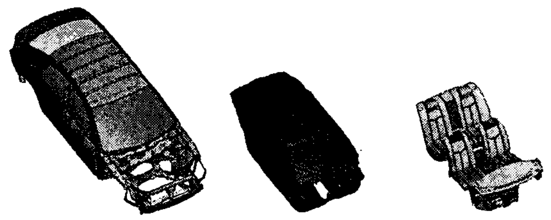
Among these countermeasures, body panels surrounding the acoustic cavity are usually made out of thin sheet metal, which has a very low bending stiffness. So, it is a common necessity to increase the stiffness of these panels by either introducing ribs or beads.(7) In the recent researches related to body panels, T. Onsay et al(9) and K. J. Min et al(10) performed the studies on acoustic characteristics with respect to a panel which has a simple pattern of a bead.

In order to improve booming noise from 70 Hz to 150 Hz, reinforcement of structural members is mainly considered. For low medium frequency range from 200 Hz to 500 Hz, modification of dynamic characteristics of panels using anti-vibration pads is usually applied. By the way, it is difficult to find effective countermeasures which can work for the frequency range from 100 Hz to 300 Hz. This frequency range lies between the booming and low medium frequency. Hence, it is required to develop countermeasures for the intermediate frequency range.

For this purpose, it is necessary to develop the countermeasures by performing optimal design of beads on body panels.

2. Vibro-acoustic coupling analysis

A study for interior noise improvement in the design stage is introduced. This study had been done with body panel modification using a vibro-acoustic coupling analysis using finite element method of a trimmed body. Considering frequency range of concern and compatibility of modeling technique, the FE model can be classified into low frequency model up to 200 Hz and medium frequency model from 200 Hz to 800 Hz. Figure 1 shows the FE model for low frequency model and the corresponding cavity and trim model.



(a) Structural model (b) Cavity model (c) Trim model

Fig. 1 Finite element models for vibro-acoustic coupling analysis of a trimmed body

Trimmed body, which consists of a BIW structure and interior trim parts like cockpit module and opening parts such as door, hood and trunk lid, is an important object in improving the interior noise of a vehicle.

If there is no acoustic excitation load in a cavity of a trimmed body, and then sound pressure in a point r in the cavity, $\mathbf{p}(r)$ created by excitation of a structural load $\mathbf{f}_s$ is [11]

$$\mathbf{p}(r) = \mathbf{H}_{pf}\,\mathbf{f}_s \qquad (1)$$

The equation of motion for a vibro-acoustically coupled system can be represented by a matrix form. Applying modal transformation, the noise transfer function $\mathbf{H}_{pf}$ can be expressed as

$$\mathbf{H}_{pf} = -\omega^2\,\mathbf{\Phi}_a\,\mathbf{Z}_a\mathbf{C}\mathbf{Z}_s\mathbf{\Phi}_s^T, \qquad (2)$$

$$\mathbf{Z}_s = [\mathbf{K}_s + j\omega\mathbf{D}_s - \omega^2\mathbf{M}_s]^{-1}$$

$$\mathbf{Z}_a = [\mathbf{K}_a + j\omega\mathbf{D}_a - \omega^2\mathbf{M}_a]^{-1}$$

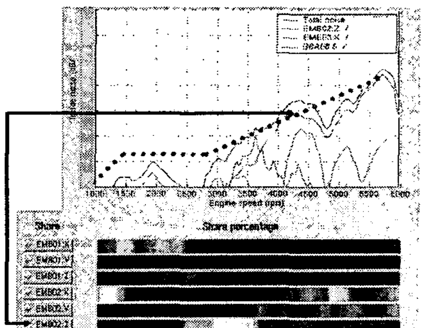
where $\mathbf{\Phi}_s$ is the uncoupled, undamped structural modes and $\mathbf{\Phi}_a$ is the uncoupled, undamped, rigid-wall acoustic modes. $\mathbf{M}_s$, $\mathbf{K}_s$ and $\mathbf{D}_s$ are respectively structural mass, stiffness and damping matrices, and $\mathbf{M}_a$, $\mathbf{K}_a$ and $\mathbf{D}_a$ are respectively acoustic mass, stiffness and damping matrices. Also $\mathbf{C}$ is a structural-acoustic modal coupling matrix. ω is excitation frequency.

3. Process for design optimization

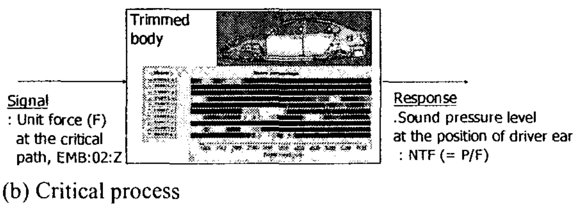
3.1 Problem definition

When a customer operates a vehicle, side effects are induced that are not expected by the customer. The interior noise is also an example of side effects. It is needed to define the critical process in the overall process in order to make an effective design proposal and reduce the side effect.

In the early stage of development, the overall process generating interior noise need to be analyzed to identify the critical processes by using a computational technology, such as TPA (Transfer Path Analysis)[12,13]. Figure 2.a shows that the concerned interior noise(black line) is structure borne one, which is mainly induced by the engine input force applied to the engine mounts, EMB:02:Z (blue line). Figure 2.b reveals the critical process with respect to the critical transfer path.



(a) Booming noise and TPA vs. RPM



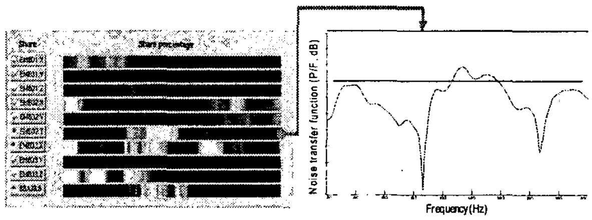
(b) Critical process

Fig. 2 Problem definition of the concerned interior noise

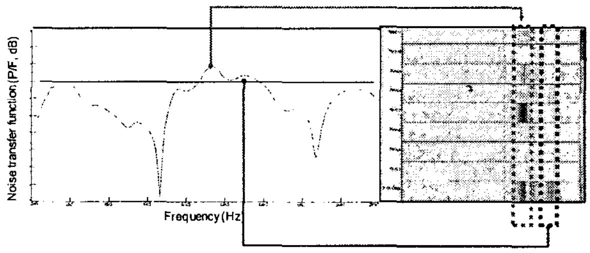
3.2 Cause analysis

Reviewing excitation forces, attenuation of mounting bushes and NTFs with respect to the selected transfer paths, the transfer path that NTF cannot satisfy the pre-defined design target is identified.

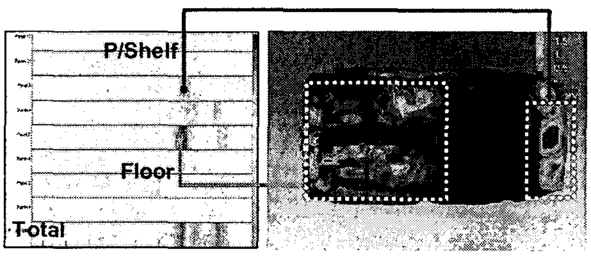
Figure 3.a shows contribution[14] of major body panels such as floor, roof, dash, parcel shelf, etc. with respect to the NTF. It is clear that the floor and the parcel shelf domineer the NTF at the first peak, and the floor and the roof have an impact on the second peak. Contour of panel contribution at the first peak is displayed in Figure 3.b.



(a) NTF with respect to the critical path



(b) Panel contribution with respect to the NTF



(b) Contour of panel contribution

Fig. 3 Panel contribution analysis with respect to the 1st peak in the critical NTF

Comparing panel vibration with panel contribution about the first peak of the NTF, it is found that panel contribution and panel vibration have similar distribution in case of the parcel shelf. Meanwhile, distribution of panel contribution is different from one of panel vibration and is local in case of the floor.

If some countermeasures are developed based on the vibration analysis, whole panel vibration on the floor need to be decreased. After understanding the panel contribution, local one needs to be controlled. Therefore, in order to achieve improvement at the first peak of NTF, it is more effective way to improve vibration of the local panel where contribution is larger than other one.

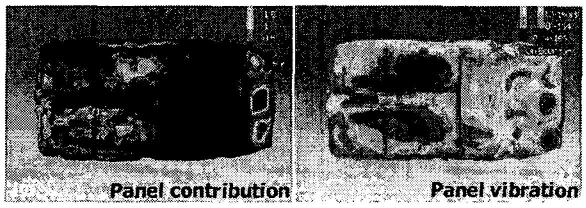


Fig. 4 Comparison of panel vibration and panel contribution

3.3 Counter-measure development

The floor consists of one big panel having many different kinds of bead and structural members. Especially, this panel gives lots of contribution to interior noise because the size is larger and excited easier than other panels.

In order to improve booming noise from 70 Hz to 150 Hz, reinforcement of structural members is mainly considered. On the other hand, for low medium frequency range from 200 Hz to 500 Hz, modification of dynamic characteristics of panels using anti-vibration pads is usually adopted. By the way, it is difficult to find effective countermeasures that can work for the intermediate frequency range from 100 Hz to 300 Hz between the booming and low medium one.

In this study, it is one of goals to find effective counter-measures for the intermediate frequency range. For this purpose, it is necessary to develop the counter-measures by performing optimal design of beads on panels. Especially, it will be applied to the floor which is concerned in the previous step, that is, cause analysis.

Figure 5 depicts the critical process diagram that consists of signal, response, noise factor and control factor. This study will make a deterministic design without noise factors. To this aim, shape and pattern of beads on the floor will be used as control factors.

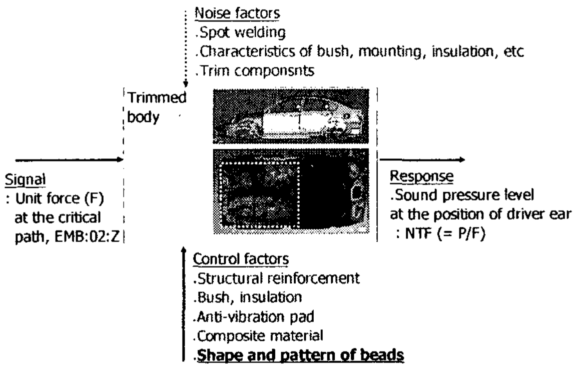


Fig. 5 Critical process with control and noise factors

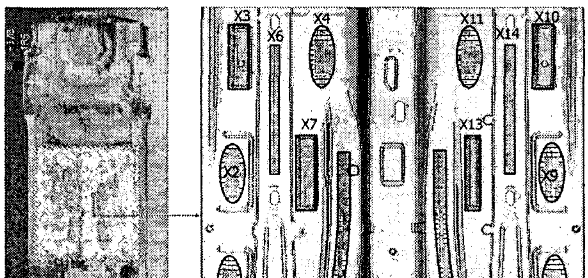
The proposed design will be accomplished with 2 phases, that is, sensitivity analysis and minimization of the critical NTF. In the first phase, sensitivity of totally 14 beads on the floor is considered and more sensitive beads will be selected. In the next one, an optimal design using the selected beads will minimize the value corresponding to the peak in the critical NTF. The description and level of control factors according to the stages are displayed in Figure 6. Each control factor is geometrically represented in Figure 6.c.

(a) 1st phase for sensitivity analysis

Control factors		Level 1	Level 2
Xi, i=1, 14	Depth of bead	0.0 mm	10.0 mm

(b) 2nd phase for improvement of the critical NTF

Control factors		Level 1	Level 2	Level 3
Yi, i=1, 7	Depth of bead	0.0 mm	5.0 mm	10.0 mm

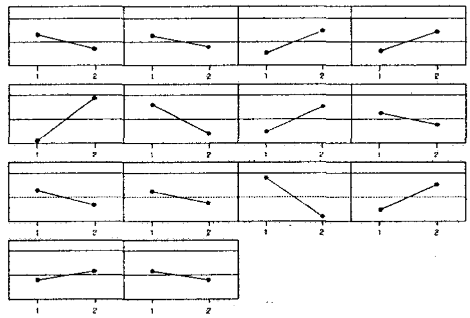


(c) Geometric representation of the control factors

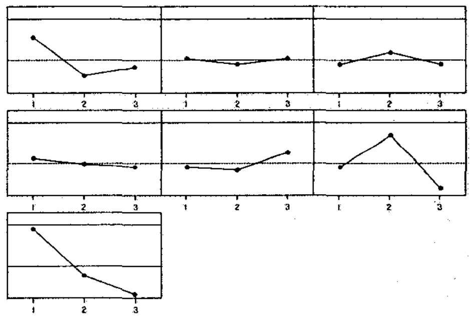
*The lateral and longitudinal dimensions of the beads are proportional to the depth

Fig. 6 Control factors

In the first phase, an orthogonal array $L_{32}(2^{14})$ is adopted for virtual tests using the computational model for vibro-acoustic coupling analysis which is shown in Figure 1. After analyzing calculation results according to the Taguchi's law, the response table as shown in Figure 8.a can be archived. Based on the response table, sensitivity of control factors can be understood. More sensitive 7 factors are selected as the control factors for the second stage. In the second phase, an orthogonal array $L_{27}(3^{13})$ is adopted for virtual tests using the same model. Following the same calculation process with the one of the 1st stage, we can obtain the response table which is shown in Figure 8.b. According to the response table, the optimal design can be achieved.



(a) Response table of the 1st phase



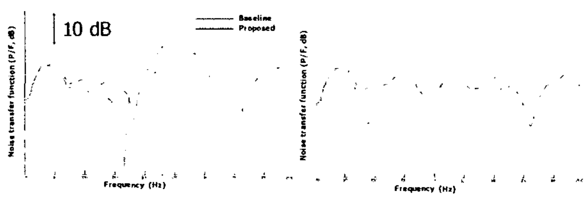
(b) Response table of the 2nd phase

Fig. 7 Response tables

3.4 Validation

Pre-validation of the proposed design can be predicted by using the computational method. Figure 8 shows the critical NTFs from the baseline and the proposed design. It is foreseen that the proposed design decreases the level of NTF above 5 dB.

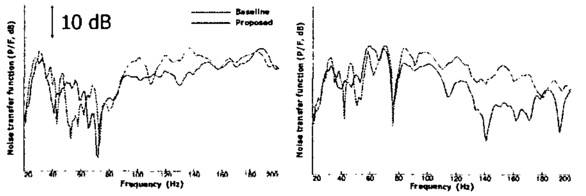
Validation of the proposed design should be performed by use of a real proto vehicle. Figure 9 shows reduction of the critical NTFs. Similar to the result of the pre-validation, it is verified that the proposed design decreases the level of NTF above 5 dB.



(a) Driver ear (b) Rear passenger ear

* Baseline design (red) vs. Proposed one (blue)

Fig. 8 Pre-validation of the critical NTFs



(a) Driver ear (b) Rear passenger ear

* Baseline design (red) vs. Proposed one (blue)

Fig. 9 Validation of the critical NTFs

4. Conclusions

In this study, an efficient process for improvement of interior noise has been introduced. The major factors of this study can be summarized as follows:

1) The proposed method for design optimization consists of 4 sub-steps, that is, a) problem definition, b) cause analysis, c) countermeasure development and d) validation. Performing design optimization through this systematic process, it is clear that the proposed process is more efficient to find the effective design under practical design complexity.

2) In reducing structure-borne noise level, it may be a usual way to reduce vibration level of the critical panels surrounding the cavity. However, it can result in an over-design or side effects. This study has proposed a more efficient approach. The proposed method is to directly control the sound pressure level in the cavity and thus it can efficiently consider the coupling effect of the vibration and acoustics during design optimization.

3) One of goals is to develop countermeasures for the intermediate frequency range using beads on a body panel. Based on the validation result, reduction of NTF by above 5 dB, it is verified that the bead can be effective for the concerned frequency range.

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References

(1) J. Jans, K. Wyckaert, M. Brughmans, M. Kienert, H. Van der Auweraer, S. Donders and R. Hadjit, 2006, "Reducing Body Development Time by Integrating NVH and Durability Analysis from the Start", SAE 2006-01-1228, 2006 SAE World Congress.

(2) A. Nomura, S. Murakami, H. Kuroda and K. Tanaka, 2003, "Improvement of BIW NVH characteristics Using a Concurrent Design Optimization Approach, SAE 2003-01-1596, SAE Noise and Vibration Conference & Exposition.

(3) K. C. Kim, I. H. Choi and C. M. Kim, 2005, "A Study on the Development Process of a Body with High Stiffness", SAE 2005-01-2464, SAE 2005 Noise and Vibration Conference and Exhibition

(4) H. Seino , K. Kimura and Y. Kamada, 2000, "Booming Noise Analysis of Passenger Car using Integrated Approach of CAT/CAE", F2000H186, Seoul 2000 FISITA World Automotive Congress June 12-15, 2000, Seoul, Korea.

(5) S. D. Gogate and A. E. Duncan, 2001, ""Digital Prototype" Simulations to Achieve Vehicle Level NVH Targets in the Presence of Uncertainties", SAE 2001-01-1529, SAE Noise and Vibration Conference & Exposition.

(6) H. S. Kim and M. H. Sung, 2000, "Analysis of Damping Effect of the Stiffness of the Body Panel". KSAE NO. 2000-03-3034, pp. 21 ~ 26.

(7) S. Subramanian, R. Surampudi, K. R. Thomson and S. Vallurupalli, 2003, "Optimization of Damping Treatment for Structure Borne Noise Reduction", SAE 2003-01-1592, Noise & Vibration Conference and Exhibition.

(8) D. Caprioli, C. Bertolini and C. Gaudino, 2001, "Improved NVH Performance Via Genetic Optimization of Damping and Shape of Vehicle Panels", SAE 2005-01-2329, SAE 2005 Noise and Vibration Conference and Exhibition.

(9) T. Onsay, A. Akanda and G. Goetchius, 1999, "Vibro-Acoustic Behavior of Bead-Stiffened Flat Panels: FEA, SEA, and Experimental Analysis", SAE 1999-01-1698, Noise and Vibration Conference & Exposition

(10) K. J. Min, C. W. Kang, K. H. Seo and H. J. Youn, 2003, "The Experimental Study on the Body Panel Shape to Minimize the Weight of the Damping Material", SAE 2003-01-1715, Noise & Vibration Conference and Exhibition.

(11) D. C. Park, 1996, "A Study on the Interior Noise Reduction of Passenger Cars using the Sensitivity Analysis and Optimization Technique for Structural-Acoustic Coupling System", Ph. D. Thesis, Seoul University.

(12) B. Dong, G. M. Goetchius, M. Balasubramanian, A. E. Duncan and S. D. Gogate, 1999, "Process to Achieve NVH Goals: Subsystem Targets via "Digital Prototype" Simulations", SAE 1999-01-1692, SAE Noise and Vibration Conference & Exposition.

(13) Y. K. Kim, B. K. Bae, Y. H. Kim, K. J. Kim and M. G. Kim, 1997, "Transfer Path Identification of Road Noise; Using Multiple Coherence Function and Relative Acceleration", KSAE NO. 97370072, Vol. 5-4, pp. 84~92.

(14) Y. S. Jin, C. M. Kim, K. H. Ko, S. G. Heo, H. S. Kook and J. S. Sa, 2003, " A Study on the interior analysis technique of a passenger car using the panel contribution analysis", KSAE 03-F0107, pp. 707~712.