The Comparison of the BEM and FEM Techniques for the Interior Noise Analysis of the Passenger Car

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ABSTRACT

The structural and acoustic analysis were performed for a passenger car to reduce interior noise in the frequency range of 30 Hz to 200 Hz. Structure-borne noise due to both road-induced vibration and engine vibration were considered by using the BEM and FEM techniques. The vehicle design cycle was shortened and acoustic quality was improved because that the design changes were recommended before the prototype vehicles were tested.

The coupling analysis between the vibration of the vehicle structure and acoustics of the interior cavity was performed by using the MSC.Nastran. Also acoustic optimization was performed by using the optimization module of MSC.Nastran.

MSC.Nastran was used in the structural analysis to calculate the vibration of the interior cavity surfaces. These vibrations were then used in a boundary element acoustic code SYSNOISE to calculate the interior sound levels. The effective panel contribution analysis was performed to efficiently identify surface structure locations where vibrations must be reduced to minimize noise.

The results and procedure of acoustic analysis such as engine noise, road noise and noise transfer function by using the both BEM and FEM techniques were compared and summarized.

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1. INTRODUCTION

The several analysis techniques for the vehicle development process were developed and applied from the concept design stage as increasing the demands of customer for the improving noise and vibration performance.[1,2] The noise analysis of the passenger car can be divided into engine induced noise analysis and road induced noise analysis transmitted by tire and chassis system. Also the analysis can be classified into structure borne noise and air borne noise based on the transfer path and frequency range.

Structure borne noise is produced by vibration of vehicle panel which is induced by engine excitation or road input. Air borne noise is induced by transmission of noise from engine room or intake system and exhaust muffler system and also depends on the exterior style. The frequency range of air borne noise is usually high frequency and can be controlled by proper addition of absorbing material and insulation material. For the case of structure borne noise, the frequency range is below 200 Hz and it is more effective to reduce the structure borne noise by modification of the body structure.

The studies of the noise analysis in DAEWOO motor company were initiate early 1990. Some applications for the passenger vehicle were performed by using the boundary element method such as SYSNOISE[3] from LMS and COMET/ACOUSTICS[4] from AAC. Also the acoustic analysis of the finite element method by using the FSI[6] from SDRC and MSC.Nastran[7] were also performed. Various design recommendations were made to reduce interior noise levels at critical frequencies that were identified by the analysis.

Structure-borne noise analysis in the frequency range of 30 to 200 Hz due to road and engine input were considered in this paper.[8,9] The analysis allowed changes to be implemented in the design before prototype vehicles were built, for shortening the vehicle design cycle and improving acoustic quality. The application examples for the noise reduction of the vehicle were summarized and the comparison of the noise analysis method was outlined in this paper.

2. NOISE ANALYSIS

2-1. ANALYSIS MODEL FOR THE NOISE ANALYSIS

The accurate finite element model of the vehicle is needed to predict accurate results for the noise analysis. The interested frequency range of analysis must be determined before the NVH analysis. The frequency range is depends on each NVH status and application and also affects the size of mesh, the accuracy of the analysis and time frame of the analysis considering the performance of machine and characteristics of the software.

The BIW maintains most of the vehicle stiffness and also affects interior noise. So the global modes are important for the low frequency vibration and local modes are main parameters for the interior noise. The BIW was modeled detailed to represent both global mode and local modes. The detailed finite element model of the B.I.W was constructed by using the linear thin shell elements that were represented each body panels. These panels were connected by rigid elements to simulate the spot welds. The correlation was performed to validate modal properties. The trimmed body was modeled by adding the windshield and trim parts to the B.I.W. by considering the effect of mass and stiffness.

The chassis system model was consisted of several subsystems such as powertrain, front suspension, subframe, rear suspension, and exhaust pipe system. The flexible components were modeled by shell or solid elements and the rigid element and beam element were used to represent the stiff components. The spring element were used to represent bush and springs. The mechanical motions of the joints were modeled by using the multi-point constraint equation and the rigid element and the end release code of the rigid element was modified.

There are two kind of interior cavity models. The one is the shell model represented the cavity surface for the boundary element method. The other is the solid model represented the volume of the cavity for the purpose of finite element coupling method. The Pseudo Elements are needed for the finite element coupling method to reduce the solving time. Figure.1-2 shows the vehicle model and cavity model used in the noise analysis.

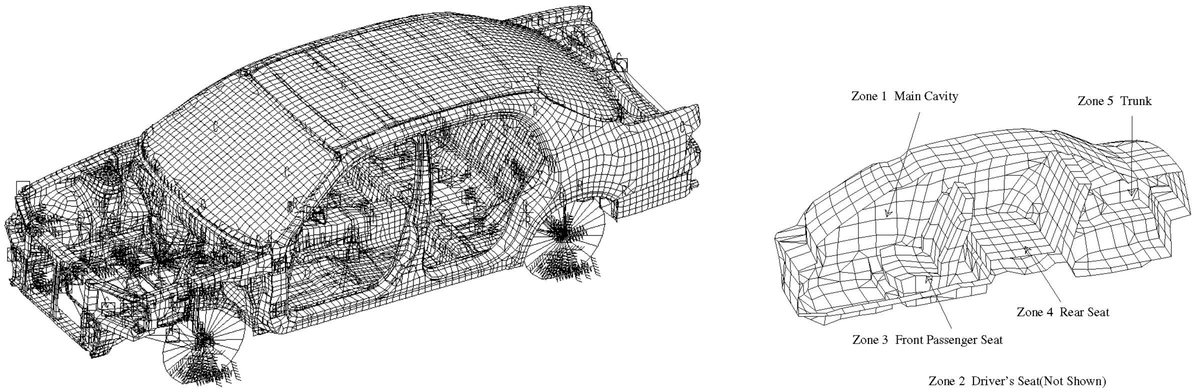

Figure 1. Passenger Vehicle Model & BEM Cavity Model

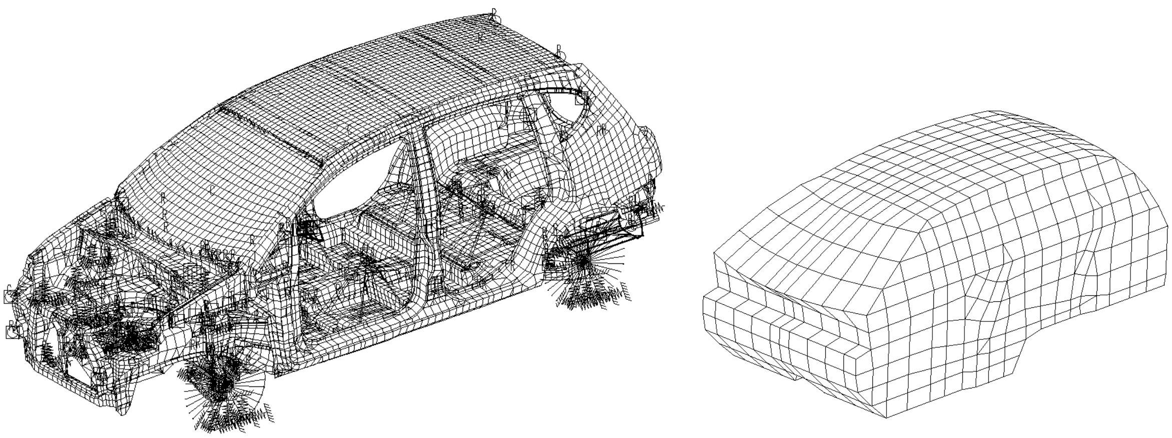

Figure 2. MPV Vehicle Model & FEM Cavity Model

2-2. NOISE ANALYSIS OF THE BOUNDARY ELEMENT METHOD

The structure vibration modes more affect to interior acoustic modes than the air affects to vehicle structure. So the one-way coupling method was used to perform the noise analysis of the boundary element method. To perform the coupling analysis, the forced response analysis for the vehicle was performed by using the MSC.Nastran. After the vibration of the vehicle panels was converted to velocity boundary condition, the interior noise was calculated by using the boundary element method considering the absorbing material. The SYSNOISE(9) from LMS and Comet/Acoustics(10) from AAC were usually used to perform the acoustic analysis.

The acoustic sensitivity analysis for the body is defined as the interior sound pressure due to unit dynamic force applied to the body. The acoustic sensitivity analysis was performed to find the transfer characteristics at main chassis mount and engine mount areas.

The effective panel contribution analysis were then used to further investigate design changes such as location of deadener or insulation that would directly result in interior sound pressure level reduction. The interior noise was reduced combining the design changes of body panels and location and thickness changes of the insulation and deadener based analysis results. Figure.3 shows example of the interior noise reduction.

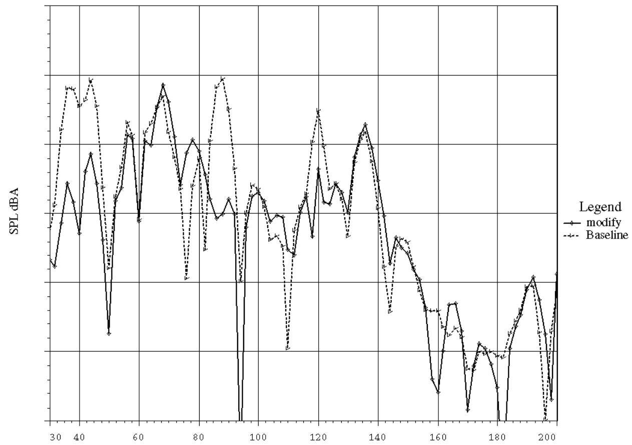

Figure 3. SPL of Road Noise Analysis

2-3. NOISE ANALYSIS OF THE FINITE ELEMENT METHOD

2-3-1. NOISE EFFECT FOR THE CHANGING OF BODY STIFFNESS

The mass of MPV style body was reduced maintaining the global stiffness by using the design sensitivity analysis and strain energy distribution. To find the effect of interior noise of the vehicle adapted the mass reduction body, the engine noise analysis was performed. The coupling analysis was performed using the finite element method by MSC.Nastran V.70. The acoustic cavity was modeled by three dimensional elements including the seats. The response locations were selected driver's ear location and Figure.4 shows the results of analysis. The effects of the interior noise for the changing of body stiffness was that the overall response was similar. But, for the case of adapting the light BIW, the response was slightly high in the range from 1500 rpm to 1800 rpm.

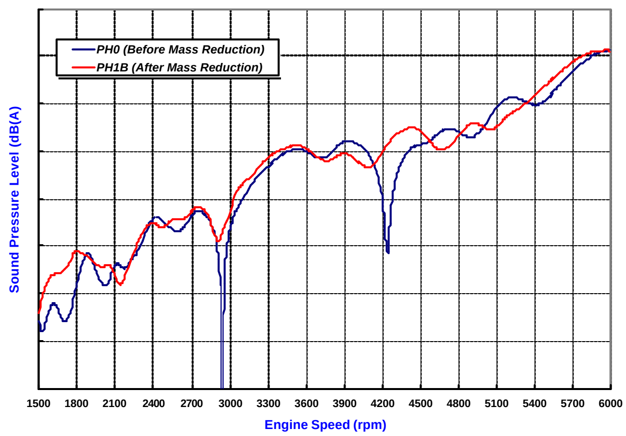

Figure 4. SPL of Engine Noise Analysis for Body Stiffness Change

2-3-2. NOISE EFFECT FOR THE APPLICATION OF THE CRADLE BUSH

The application of the bush isolators on the front cross member rear side was investigated to improve the performance of vibration of the vehicle. The road noise analysis and engine noise analysis were performed to find the effect of bush application by using the finite element coupling techniques. Figure.5 shows the results of the engine noise analysis. Reviewing the results, for the case of application of bush isolator, the response of the noise level was higher in the frequency range from 50 Hz to 70 Hz, and from 105 Hz to 125 Hz. But the response of the noise level was lower in the frequency range from 70 Hz to 105 Hz and above 150 Hz.

The road noise analysis was also performed by using the same method and same model. Figure.6 shows the results of the road analysis. For the case of application of bush isolator, the response of the noise level was higher in the frequency range from 60 Hz to 105 Hz. But the response of the noise level was lower in the frequency range above 105 Hz. Also the response level was similar in the frequency range below 60 Hz.

The results shows that if the bush was applied on the front cross member, the noise response would reduced in frequency above 120 Hz and results including vibration and noise would similar in frequency below 120 Hz.

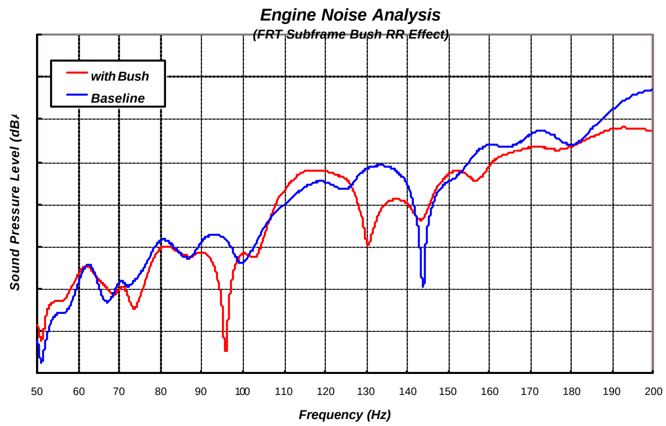

Figure 5. SPL of Engine Noise Analysis for Subframe Bush Effect

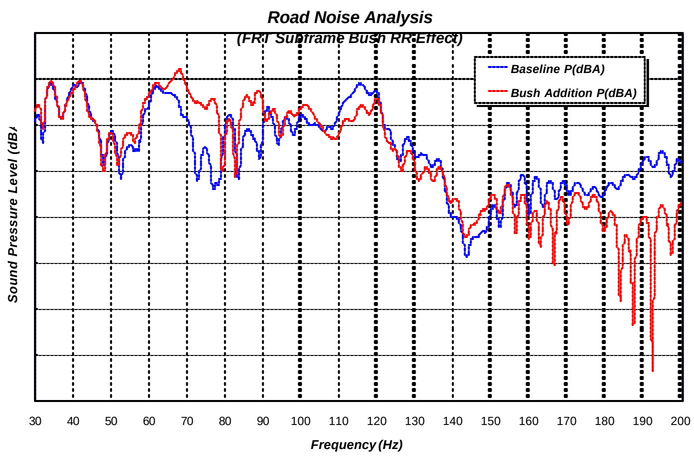

Figure 6. SPL of Road Noise Analysis for Subframe Bush Effect

2-3-3. NOISE TRANSFER FUNCTION ANALYSIS

The noise analysis was performed to identify the frequency range of peaks and the transfer path of vibration and noise on the chassis mount area in the vehicle. The noise transfer functions, mode shapes of the vehicle structure, mode shapes of the acoustic cavity were also calculated. The main chassis mount area to body such as front suspension, rear spring, rear damper, and rear axle damping bush and also engine mounts were selected to performed the analysis of the noise transfer function. The unit forces were excited at the chassis mount position in three directions and the driver's ear location was selected for the position of the response. The finite element method was used for the noise analysis of the coupling technique.

Reviewing the result curves of the noise transfer function on the main area of chassis mount, there were peaks near the frequency range 63 Hz, 105-112 Hz, 155 Hz in the averaging response curve. The main reason of these peaks was coupling effect between the structure modes of the vehicle and acoustic modes of the cavity. The peak in the frequency range from 90 Hz to 130 Hz in the maximum envelope curve of noise transfer function was excited from left side rear damper. Also the main source of peak in the frequency range from 160 Hz to 190 Hz was right side rear spring. Figure 7 shows the result curves of the noise transfer function on the main area of chassis mount. Figure 8 shows the curves of average and curve of maximum envelope.

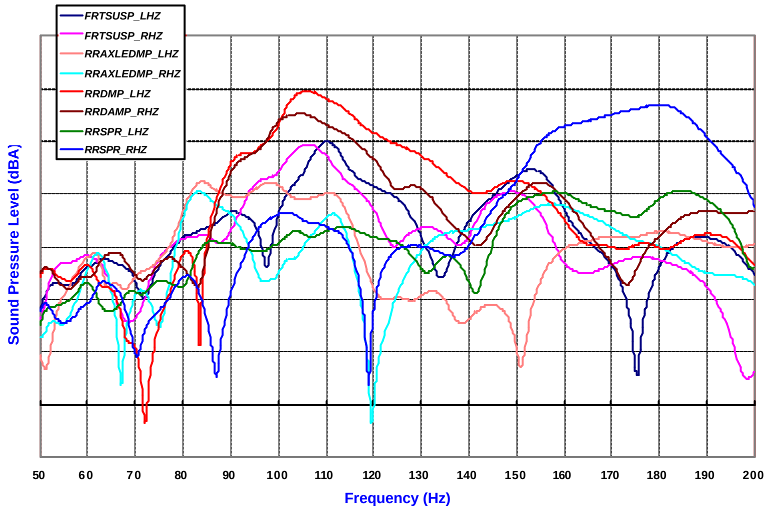

Figure 7. SPL of Noise Transfer Function Analysis

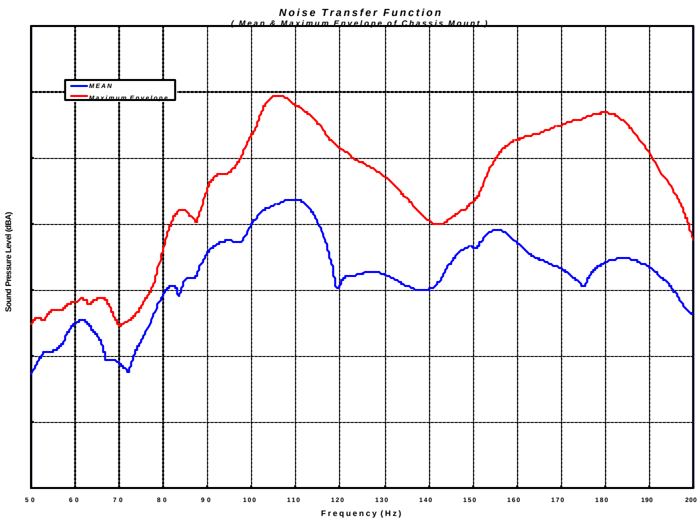

Figure 8. Mean & Max. of Noise Transfer Function Analysis

To identify the noise and vibration path, the mode shapes of the vehicle in specific frequency range were studied. The area of design modifications which could affect the noise level were found and the relations between the noise level and acoustic modes of the cavity were identified. In the early design stage, the noise path and relation of structure vibration and acoustic characteristic were found.

2-3-4. APPLICATION OF THE ACOUSTIC OPTIMIZATION

The coupling analysis between the vehicle structure and acoustic cavity were performed by using the coupling technique of finite element method to reduce the road noise effectively. Also the concept of the modal participation factor that indicates modal contribution to response point was applied.

To apply the optimization technique for the acoustic analysis, the panel thickness and stiffness of the bush which would affect the noise level were selected as design variables. The noise level was reduced in frequency range from 62 Hz to 112 Hz compared to noise level of baseline by using the acoustic optimization. To validate the results of acoustic optimization, the noise analysis was performed again applying the resultant panel thickness. The limit of panel thickness for the optimization was assigned 10% for the practical design application. The case of applying update panel thickness which have positive sensitivity, The response level was reduced in the frequency range 62 Hz and 112 Hz.

The mode shapes of rear floor were identified to effective area for the interior noise based on the results of the modal participation factor and mode shapes. To simulate the effect of reduction of interior noise, the noise analysis was performed by adding the bead near the rear floor. The interior noise was reduced in frequency range from 50 Hz to 70 Hz and above 160 Hz. The control of specific structure modes by addition of bead was effective for the noise reduction.

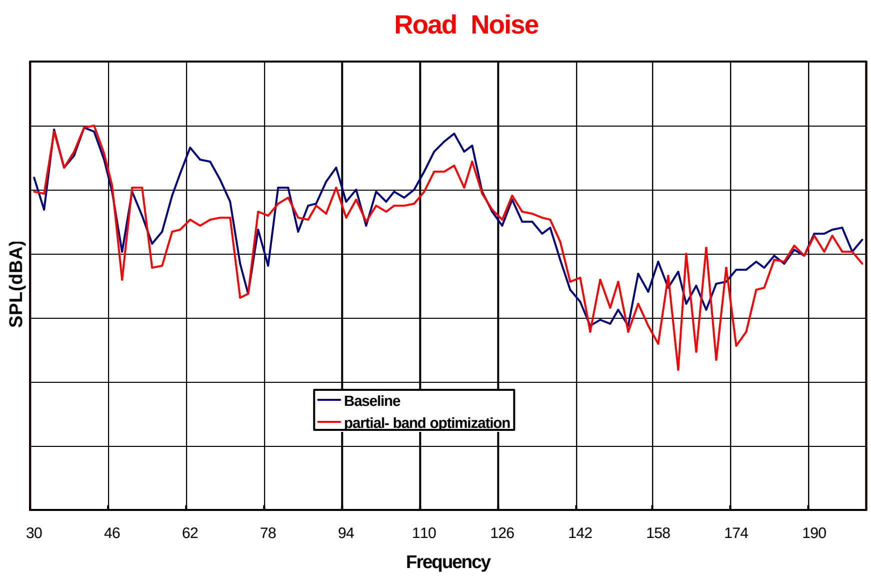

Figure 9. SPL of Road Noise Analysis for Optimizatoin

2-4. COMPARISION OF THE BEM AND FEM METHOD

The road noise analysis was performed by using the same MPV vehicle model to compare the results of boundary element method and finite element method. Figure.10 shows the results of sound pressure level at driver's ear location by using the BEM and FEM techniques for the road noise analysis and the results of noise levels were similar.

Comparing the two noise analysis method, the boundary element method was more convenient for the modeling. In the finite element method, three dimensional mesh was needed and also pseudo elements were needed. The optimization analysis of finding the effective location of insulation was proper to boundary element method and radiation noise analysis was difficult in the finite element method. In the finite element method, the analysis time could be reduced and vibration and noise analysis could be performed by using the one program.

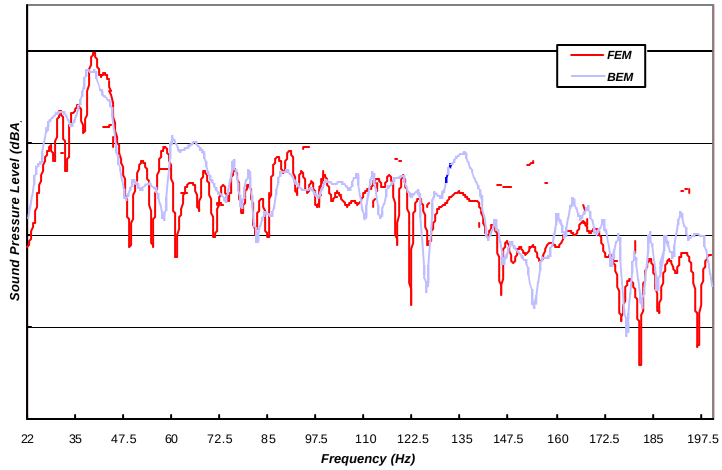

Figure 10. SPL of Road Noise Analysis(FEM & BEM)

3. CONCLUSIONS

The procedure of noise analysis to reduce the structure-borne noise was established by using the BEM and FEM techniques. So the design cycle of vehicle development was shortened and acoustic quality was improved

For the boundary element method, the vehicle vibration was calculated to get the velocity boundary condition on the cavity surface and the effective panel contribution analysis was performed to efficiently identify the locations of structure surface where vibrations must be reduced to minimize noise.

The coupling analysis between the vibration of the vehicle structure and acoustics of the interior cavity was performed by using the MSC.Nastran. Also the acoustic optimization was performed by using the optimization module of MSC.Nastran.

The results and procedure of acoustic analysis by using the both BEM and FEM techniques were compared and summarized to apply proper method. In the future, the construction of database for the acoustic materials based on correlation of test data was needed to get the accurate results

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