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IMPROVEMENT OF POWERTRAIN CHARACTERISTICS USING HYBRID CAE TECHNOLOGY

Kai-Hong Lai*, Che-Hung Yang

CARTEC, China Motor Corporation
49, Shio Tsai Rd., Yang Mei, Taoyuan 326, Taiwan, R.O.C.
942094@china-motor.com

Abstract

In recent vehicle development, the noise and vibration reduction is an important design index to fit customer expectation. It is also well understood that the vehicle sound is not only about "sound pressure" but also about "sound quality". Powertrain system vibration can result in an increase in vehicle interior noise when excited by engine loads through the vehicle body. To evaluate the powertrain system vibration, the finite element method was applied to calculate the normal modes of the powertrain system. In this paper, Hybrid CAE technology combined numerical and experimental analysis is applied for the passenger car to improve the NVH characteristics caused by normal mode of powertrain system. The target to enhance the performance of noise and vibration is to improve the sound quality. Using Hybrid analysis technology, the difference between the experimental frequencies and analysis frequencies are about 3%. Updating CAE model can be applied to modify and optimize the characteristics of powertrain system. In order to reduce the powertrain vibration, the powertrain reinforcement was designed to increase the stiffness of powertrain system. The vibration level of first bending and torsion mode will affect the interior sound quality directly. From the result of modal analysis, the natural frequencies of first bending and torsion mode were increased significantly. The run-up testing was used to verify the effect of the powertrain reinforcement. From this test result, the vibration magnitude of 3^{rd} to 6^{th} order were decreased, especially for 4^{th} and 6^{th} order. In addition, the Index of sound quality was also become better than original condition.

INTRODUCTION

Recently, CAE method is used very commonly for predicting static and dynamics behavior of vehicles. But many properties of the physical structure are very difficult to measure. The wrong parameters will lead to do the inaccurate dynamic calculations. Therefore, realistic vehicle testing is necessary procedure to identify dynamic analysis of CAE method. The purpose of the experimental modal analysis (EMA) is to provide mode shapes, natural frequencies, and damping. Using the sensitivity analysis and modal updating, the experimental results can be used to update the CAE model until the natural frequencies from the CAE agree with natural frequency from experiment. This modified model could be used to predict the problems during the process of new vehicle design and find out the countermeasure. Then, we can save time and cost more efficiently.

In the literatures, a lot of research work discussed about the analytical and numerical solutions. Recently, material properties of engineering structures were obtained by using the inverse method. Shi et al. [1] used the inverse method to find the material properties of sandwich structure.

But for some complicated structures, it is harder to build up an effective model for inverse method. Although the effect of material properties can be revealed by experiments, it is rather time consuming to design and manufacture new experimental facility for each case. This is the reason why the hybrid method [2,3] could be adopted to determine the equivalent material properties and boundary conditions.

Building up a numerical analysis, the material properties and boundary conditions are the major components. Different parameters of boundary conditions and material properties (P_j) make the natural frequencies and mode shapes different. It is natural that the experimental model usually differs from the numerical model in many parameters. Some parameters influence the system response significantly. The change of parameters, δP_j, corresponds to the change of system response, δR_i, can be expressed as the sensitivity (S_{ij})[4], i.e.

$$S_{ij} = \left[\frac{\delta R_i}{\delta P_j} \right] \tag{1}$$

where the subscript i=1, …, N and j=1, …, M.

By applying the hybrid method to proceed the optimization analysis, the difference between the estimation and experimental results can be kept below 4% and the CAE model can be applied for further calculation. In this paper, the modification model was added the powertrain reinforcement to restrain the relative deformation between engine and transmission. As expectation, powertrain reinforcement was succeeded to decrease the deformation of engine and transmission.

ANALYSIS MODEL

As for the finite element analysis, the commercially available FEM package MSC.Patran and Nastran software was employed to build up the numerical model, calculate the normal modes and to achieve the optimization analysis. The whole model contains 14778 shell elements, 7322 solid elements and 269 MPC. For the optimization case, the tuning parameters were Young's modulus (E), thickness (h) and Poisson ratio (ν). However, the sensitivity of Poisson ratio was low, the E and h were taken as the tuning parameters. By tuning the material properties, the difference between the FEA and EMA results can be reduced. In this paper, the number of iterations for tuning is six for optimization analysis.

TEST PROCEDURE

There are two test jobs in this paper, including modal testing and run-up testing. For the modal testing, there are eighteen sampling points for the powertrain model. The powertrain was fixed via engine mount to the fixture. This powertrain was excited by a hammer and the vibration signal was measured by the accelerometer.

For the other case, the range of run-up testing was from 1500 to 6000 rpm. The measurement points were placed in the central of the front and rear seat, and near the height of ear. After order analysis, the sound pressure results were shown in waterfall graphs and help us to judge the performance of reinforcement.

DISCUSSIONS

For our initial CAE model, the natural frequencies of bending and torsion mode were 131.7 Hz and 216.9 Hz (Shown in Figure 1(a) and (b)). There are significant differences between modal analysis and modal testing. The difference is about 60%. Therefore, for the optimization process, the 1st bending mode of EMA was taken as object frequency. And the frequency of 1st bending and 1st torsion mode was taken as reference frequencies. The EMA frequency of 1st bending mode is 216.9 Hz. So, the range of object frequency was set from 210 to 220 Hz. At the end of optimization, the 1st bending mode of FE model was became 210.3 Hz. The tuning frequencies of CAE model were listed as table 1. Examine the optimization results, the difference of bending mode between EMA and FEA was reduced to 3.2 %, and the difference of torsion mode was also reduced to 0.3%. We will base on this optimized CAE model to do the further calculation and discussion.

1. After proceed the optimization analysis, the natural frequency of modal analysis and modal testing were become more similar. The natural frequency of bending and torsion mode is 210.3 Hz and 339 Hz. Compare the natural frequency of CAE results and the waterfall of run-up testing (shown in Figure 2), we can found that the 6th order of frequency spectrum affects the sound quality significantly. The 3rd, 4th and 5th order also affect the response of the interior noise.
2. From the assembly conditions of the reinforcement and powertrain, the function of the reinforcement was used to restrain the deformation of the powertrain. Therefore, Figure 1(a) and (b) indicated the natural frequencies of bending and torsion mode were increasing from 210 to 276 Hz and 339 to 403 Hz after adding the reinforcement on the powertrain. The frequencies of bending and torsion mode have 30% and 18% increasing. Compare with the frequency response function (FRF) of without and with reinforcement condition, the acceleration of engine mount and rear roll stopper have improve significantly (Shown in Figure 3).
3. As depicted in Figure 4, without reinforcement, there are bigger noise level in 4000~4400 rpm (2nd ~5th order) and 2800~4400 rpm (6th order). For the case of adding reinforcement on powertrain, the vibration magnitude of 3rd to 6th order

were decreased, especially for 4th and 6th order. In addition, the index of sound quality was also become better than original condition.

CONCLUSIONS

In order to reduce the powertrain vibration, the powertrain reinforcement was designed to increase the stiffness of powertrain system. The vibration level of first bending and torsion mode will affect the interior sound quality directly. Updating CAE model can be applied to modify and optimize the characteristics of powertrain system. Using hybrid analysis technology, the difference between the experimental frequencies and analysis frequencies are about 4%. This method also can help us to propose the countermeasure without further testing. From this, we can save the testing cost to enhance the marketing competitiveness of vehicle.

ACKNOWLEDGEMENTS

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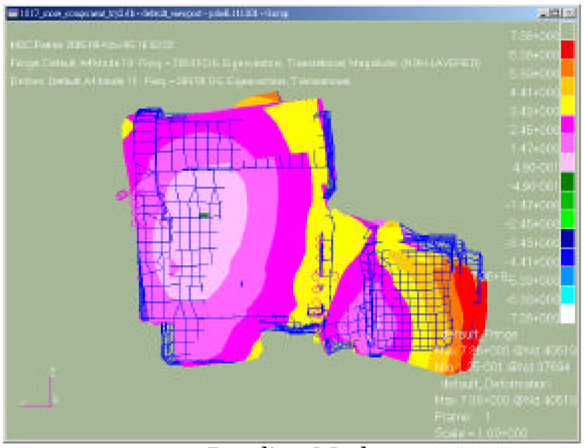
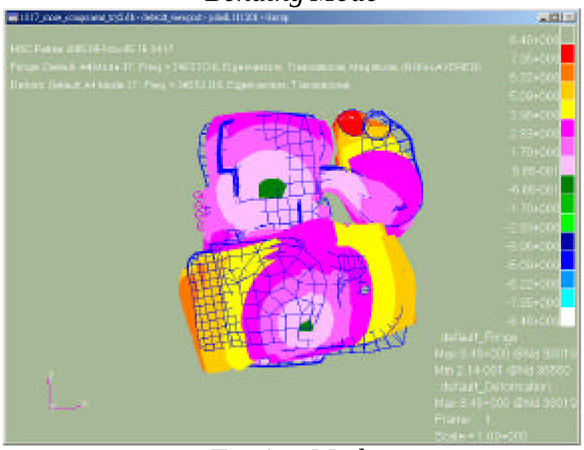
Table 1 List of the natural frequencies

	Without Reinforcement			With Reinforcement
	Modal Testing	Before Tune	After Tune	
Bending Mode	216.9 Hz	131.7 Hz	210.3 Hz	276.6 Hz
Torsion Mode	342.2 Hz	216.9 Hz	339.2 Hz	403.88 Hz

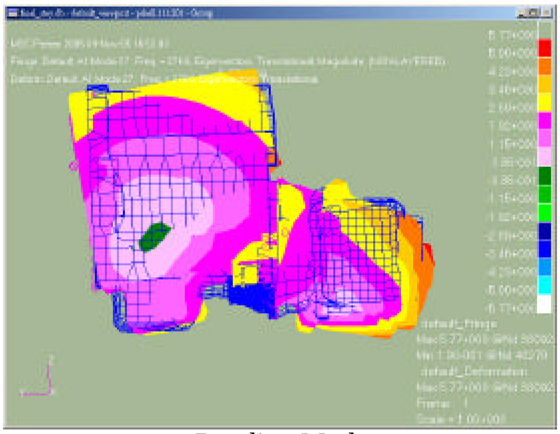


Bending Mode

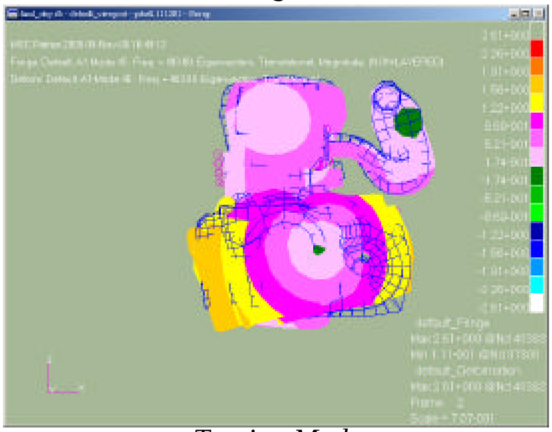


Torsion Mode

Figure 1(a) Mode shapes of the Powertrain with reinforcement



Bending Mode



Torsion Mode

Figure 1(b) Mode shapes of the Powertrain without reinforcement

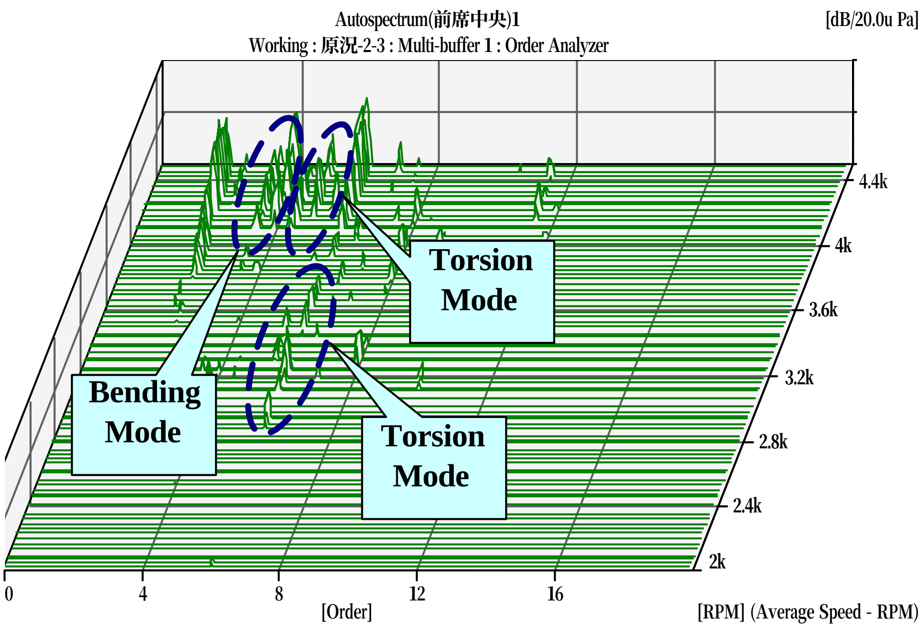
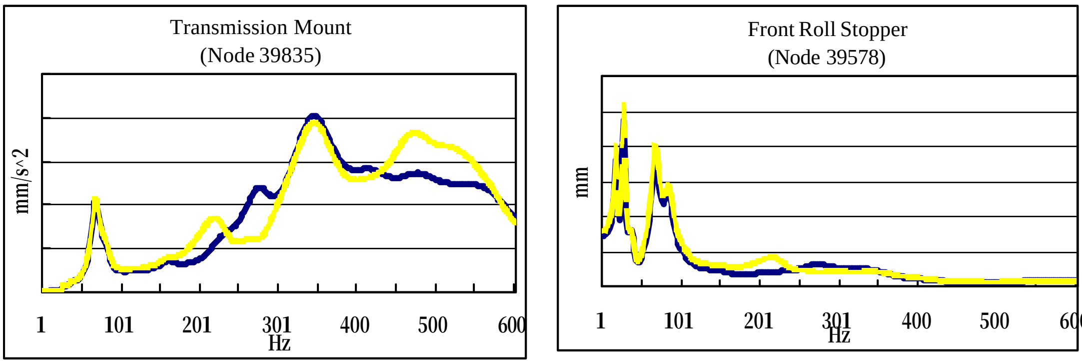


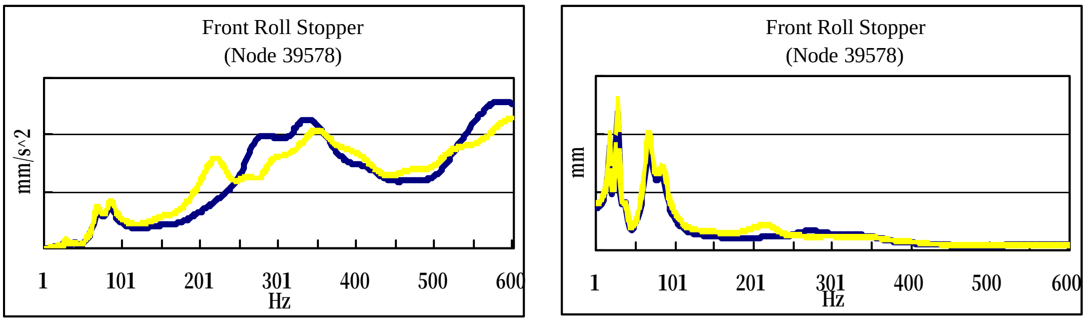
Figure 2 Comparison of mode shape and waterfall results



Acceleration *Displacement*

With Reinforcement ***Without Reinforcement***

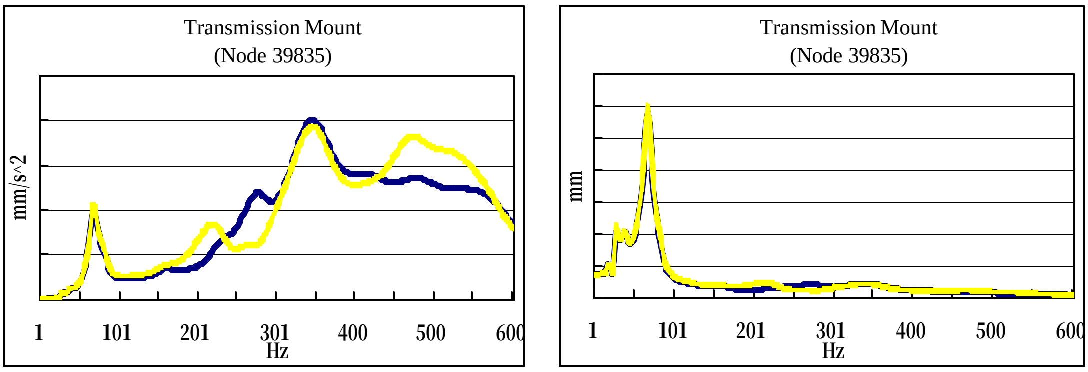
Figure 3(a) FRF of engine mount side with and without reinforcement



Acceleration *Displacement*

With Reinforcement ***Without Reinforcement***

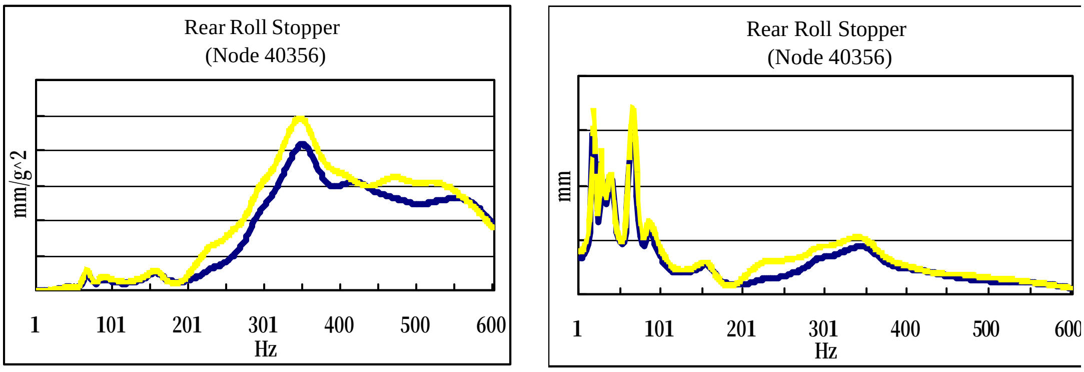
Figure 3(b) FRF of front roll stopper side with and without reinforcement



Acceleration *Displacement*

With Reinforcement ***Without Reinforcement***

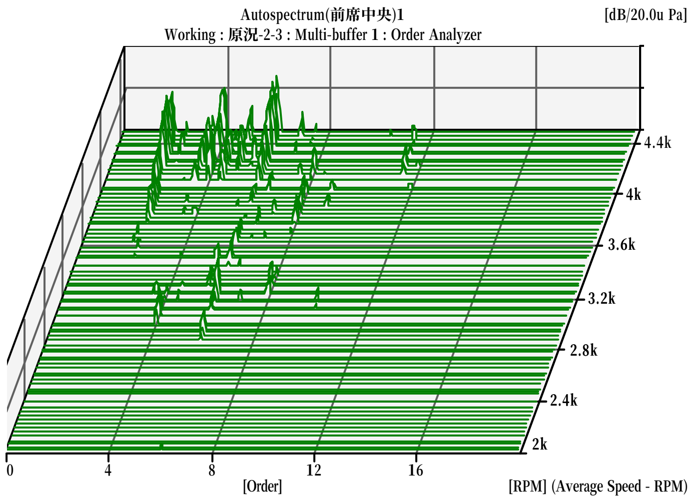
Figure 3(c) FRF of transmission mount side with and without reinforcement



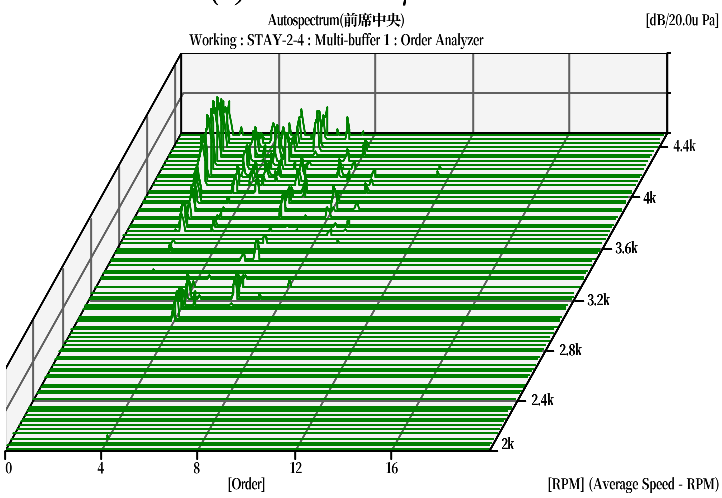
Acceleration *Displacement*

With Reinforcement ***Without Reinforcement***

Figure 3(d) FRF of rear roll stopper side with and without reinforcement



(a)Without Reinforcement



(b) With Reinforcement

Figure 4 waterfall of front seat with and without reinforcement