A Study on Idle Vibration Analysis Technique Using Total Vehicle Model

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SUMMARY

It was required to improve the low frequency vibration characteristics of a current production car. For the sake of design study, a well correlated FE model was necessary. This paper describes the process of how to build FE model and to correlate it with test result from BIW to total vehicle.

Key Words : Low Frequency Vibration, BIW, Joint stiffness, MAC, FRF Curve, Trimmed Body, Total Vehicle, Operating test

INTRODUCTION

Low frequency vibration characteristics is one of the important factors influencing on overall quality. In particular, drivers are sensitive to steering wheel vibration in the range of engine idling frequencies. For the most automotive engineers, controlling this kind of vibration remain to be a difficult problem, although some car makers seem to already have their know-how. Well correlated finite element model is useful, if not almighty, in that it helps understanding the current phenomenon in more detail and it can predict what it would be like after design change. Although deciding what to change in finite element model and how to apply to a real structure is an addition problem. Of course to make the over all development cycle, these kinds of simulation should be done as early as possible. Even if so, tuned finite element model for previous similar platform becomes enevitable to minimize errors which might occur in analyzing the complex structure such as cars. So building reliable finite element model is a very important step to effectively control vibration problem.

To obtain a reliable vehicle model, each stage of model such as body in white, trimmed body, total vehicle model, and every main components should be validated with test results. In this paper, the process of building finite element model which can simulate the steering vibration characteristics within 40Hz is described. Fig. 1 illustrates overall process for obtaining well correlated total vehicle FE model.

FE MODELING

Body In White with glasses (Fig. 2) About 3,000 elements were used to model body in white with glasses. Shell elements were used to model panels such floor, roof, dash, etc.

On the other hand beam elements were used to model main members such as pillars, rocker, front side members, roof rail, etc. But, joints where these members are connected tend to deform locally. If beam elements were to be used in these areas, flexibility had to be taken into account. Here, joint flexibility was considered by freeing rotational degree of freedom and adding springs at joints. And spring stiffness was calculated so that the stiffness of beam model be that of shell model. At the same time new coordinate system was defined for spring. Ribs in panel which enhance bending stiffness play a negative effect on membrane stiffness. Young's modulus for membrane stiffness was modified. For sealant springs were used. Concentrated mass and non-structural mass were added so that weight of model be equal to that of real structure.

Trimmed body (Fig. 3) Promptly, trimmed body was defined as a sprung mass except power train and radiator. Door, battery, and bumbers were modeled by concentrated mass, rigid body, and spring at mounting points. But steering system was modeled as exactly as possible because vibration characteristics of steering system heavily depends on stiffness of itself and its mounting area. And MPC's were used for universal joints and rack & pinion. Shell elements were used to model cross member and center member. It goes without saying that weight should be tuned to that of corresponding structure.

Total Vehicle (Fig. 4) Power train, radiator, exhaust and suspension are added to trimmed body. Such data as inertia and bush stiffness of power plant were obtained from experiment. Also exhaust system had to be well correlated in order to be added to total vehicle model. Suspension were modeled with beams and springs.

CORRELATION

Correlation process Fig.5 illustrates process of correlation of FE model with test results. In each stage, frequency responses as well as modal parameters should be examined.

Body In White with glasses Natural frequencies and mode shapes were examined. Information of strain energy distribution was useful in controlling modal parameters. MAC(Modal Assurance Criteria) was used as the criteria for correlation of mode shapes. It was obtained from SDRC Test. Table 1 shows that MAC are above 0.7 up to 4th mode. Fig. 6 compares frequency responses when excited vertically at the front end of front side member. Uniform modal damping was assumed. Body in white model was well correlated, if not satisfactory, enough to analyze the low frequency vibration characteristics.

Table 1. MAC(Modal Assurance Criteria) of BIW model

Test \ Anal.	1	2	3	4	5	6
1	0.80					
2		0.87				
3			0.76			
4				0.69		
5					0.58	
6						0.60

Trimmed body Correlation of Trimmed body model was more difficult than that of body in white model. The number of design variables was quite many. They were stiffness of door latch, weather strip, battery mounting, and so on. Fig. 7 shows frequency response curves at steering wheel and excitation point. Correlation was more successfully done relative to body in white. But MAC was not used because complex modes rather than real ones became dominant due to increased damping.

Total Vehicle Correlation of total vehicle was relatively easy. Fig. 8 shows frequency response curves at excitation points and steering wheel.

Operating Test In order to be able to simulate vibration characteristics in engine idling range, engine excitation force had to be calculated, if roughly. It can be divides into two components. One is unbalance force due to vertical movement of piston and excites vertical force. The other is torque fluctuation due to pressure within cylinder 2and exerts rotational moment. They were shown below.

- Unbalance Force

  · $F = 4(Mp + Mcrp)\dfrac{R^2\ \omega^2}{L^2} \mathrm{COS}2\omega t$

  Mp : piston mass
  Mcrp : conrod mass
  R : crank radius
  L : conrod length
  ω : angular velocity

- Torque Fluctuation

  · Torque due to unbalance :
  $Mz = -2(Mp + Mcrp)\ \omega^2\ R^2\ \sin 2\omega t$

  · Torque due to gas force :
  $Mz = F_{Gi}\tan\gamma_i\ (L\cos\gamma_i + R\cos(\omega t + \theta_i))$

  F_{Gi} : gas force
  θ_i : phase difference between each piston

Analysis under excitation forces calculated above was carried out. Operating tests were done more than 20 times at D-range. Fig. 9 shows the comparison of two curves from tests and analysis. It was well correlated.

CONCLUSIONS

So far modeling techniques, analysis process, analysis results, and so on have been discussed. Frequency responses were well correlated with operating tests. This model will be used in design studies for improvement of steering wheel vibration characteristics. It can also be used for next car at an early stage of its development.

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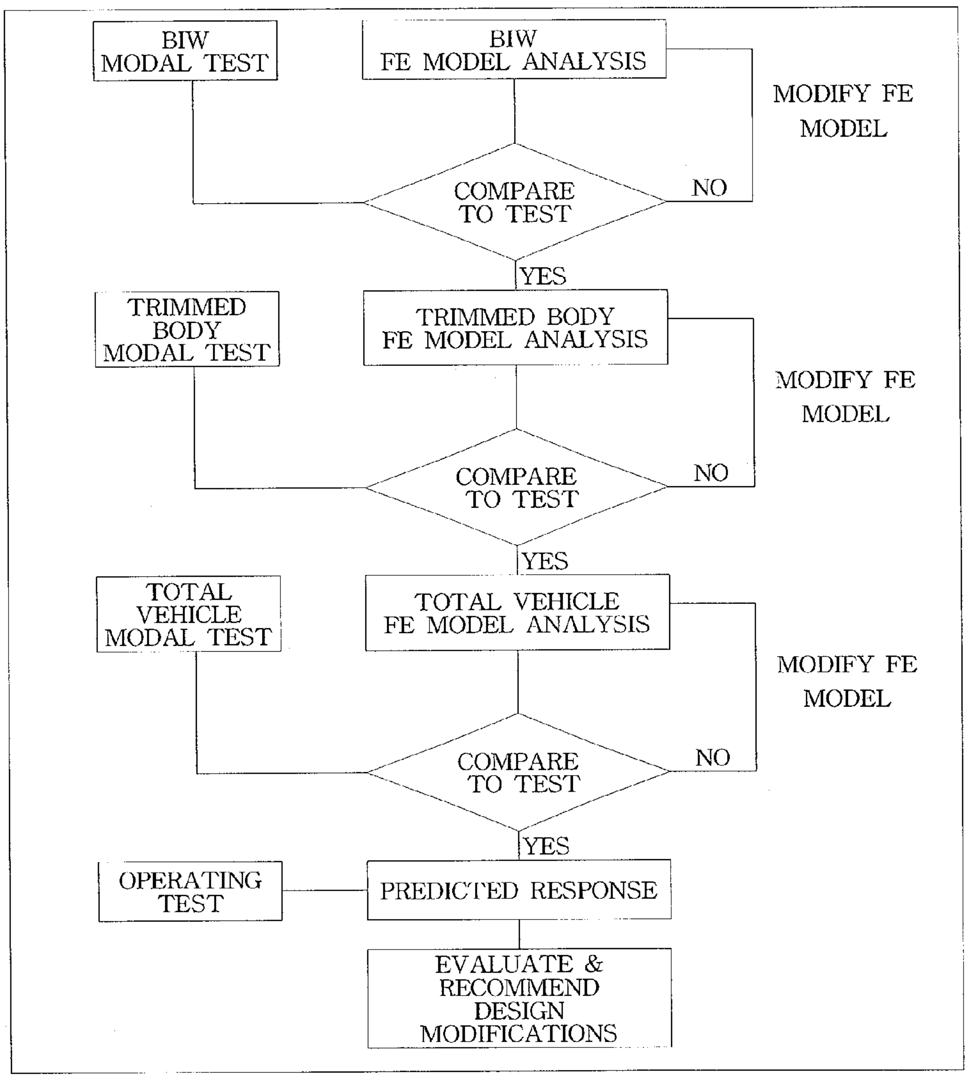

Fig. 1 Over all process of total vehicle model

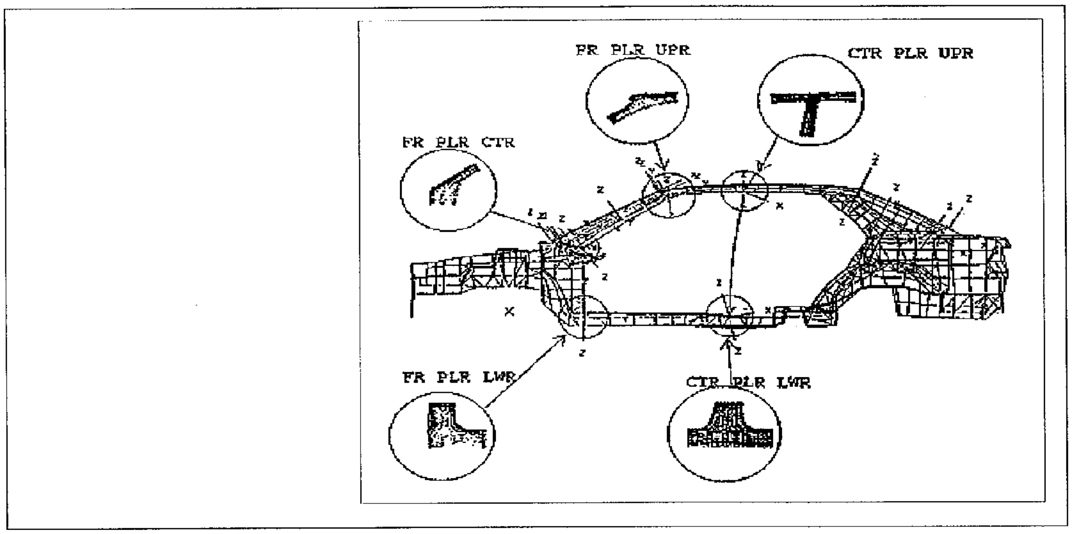

Fig. 2 BIW FE Model.

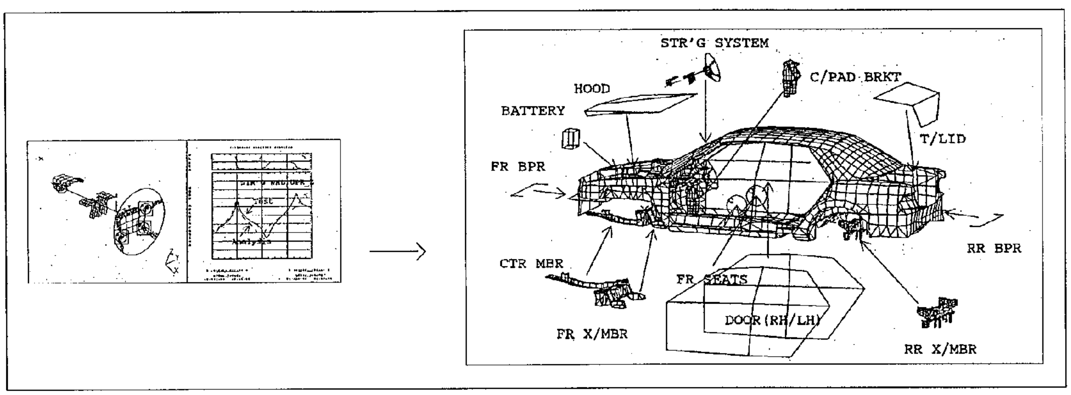

Fig. 3 Trimmed Body FE Model

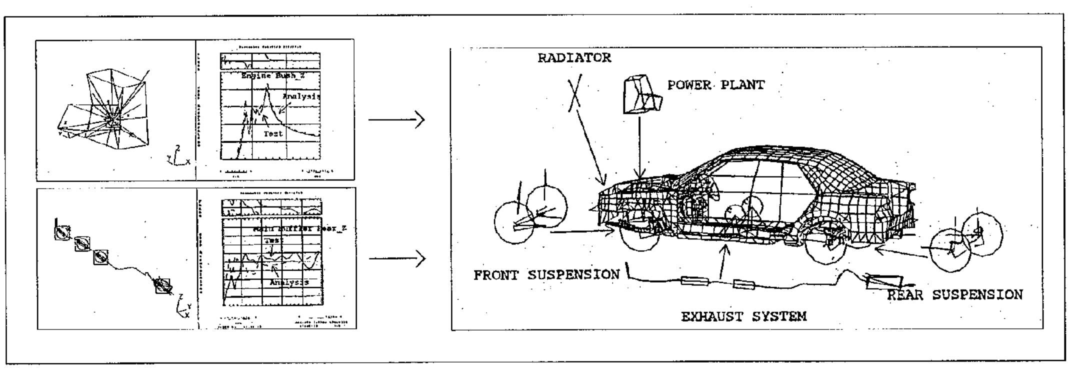

Fig. 4 Total Vehicle FE Model

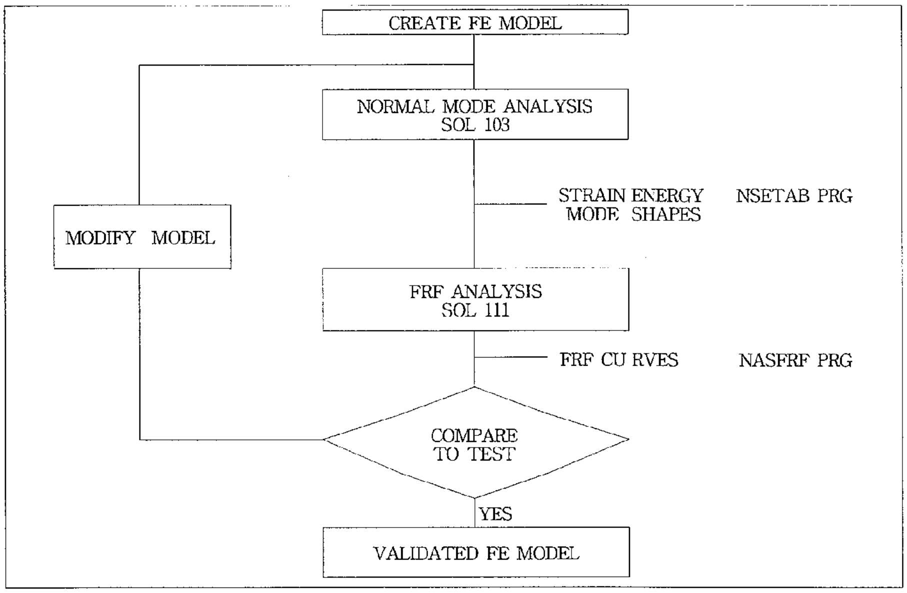

Fig. 5 Process of FE Model Correlation.

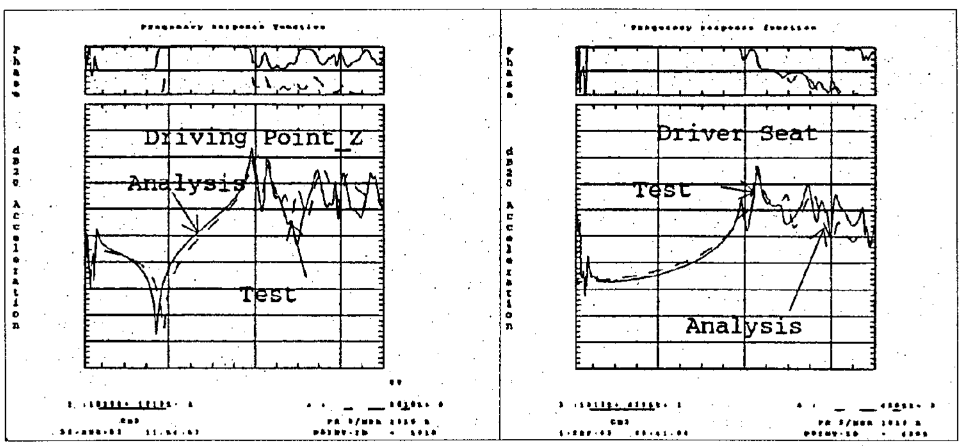

Fig. 6 Comparison of BIW Test & Analysis FRF Curve.

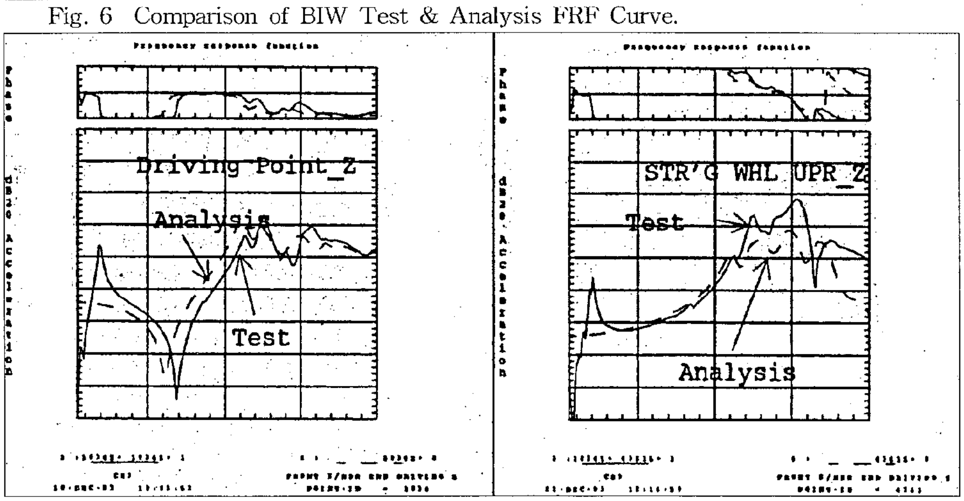

Fig. 7 Comparison of Trimmed Body Test & Analysis FRF Curve.

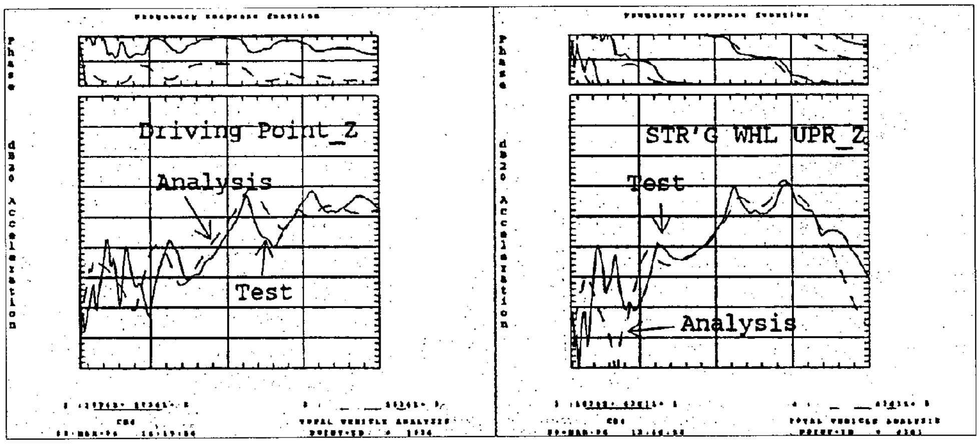

Fig. 8 Comparison of Total Vehicle Test & Analysis FRF Curve.

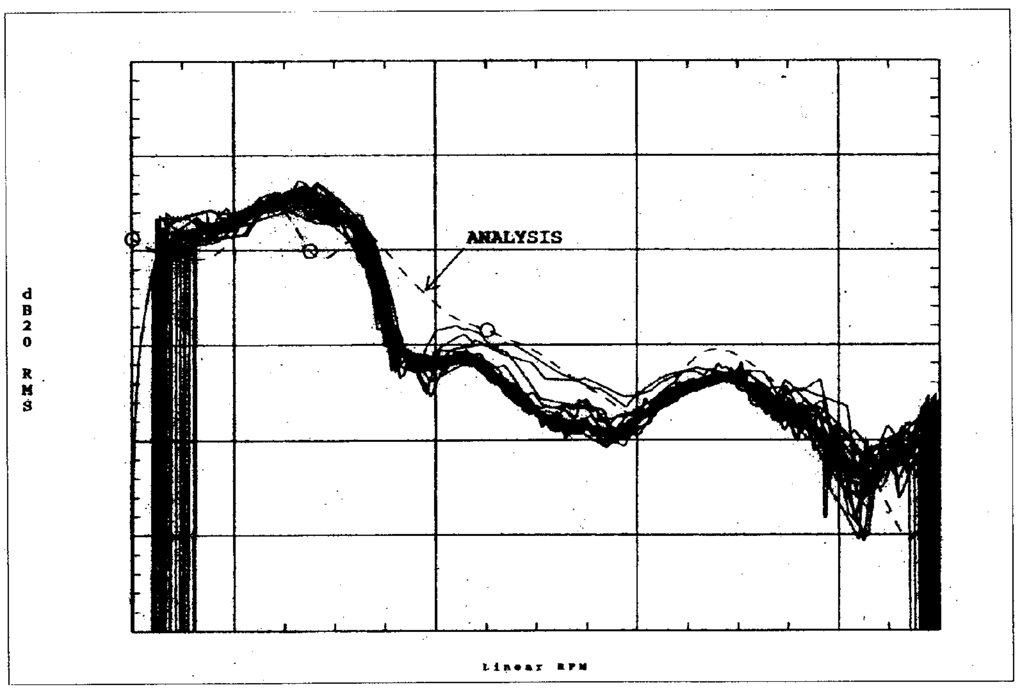

Fig. 9 Comparison of Idle Test & Analysis FRF Curve.

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