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Finite element model updating of low frequency acoustic structure coupling based on vehicle test

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Abstract: The finite element model of a car is established, and the acoustic-structure coupling finite element model of the car is modified considering the influence of structural damping and acoustic impedance of interior material. Through the road test of a real vehicle, the excitation signal and the corresponding interior noise signal under the condition of 50 km/h uniform speed are obtained. Taking the measured excitation signal as the input of the finite element model, the frequency response analysis of the low frequency noise in the frequency range of 20-200 Hz is carried out. The results show that the modified acoustic-structure coupling finite element model can be used to predict the low-frequency noise in the vehicle. The research in this paper can provide reference for vehicle NVH forward design, vehicle vibration and noise prediction and improvement.

Keywords: low frequency noise; acoustic structure coupling; finite element model; damping modification; vehicle test.

1. Introduction

NVH characteristic, as one of the important evaluation indices for the vehicle riding comfortability [1-2] , is increasingly emphasized by the automobile companies. The internal noise has an immediate effect on the vehicle's riding comfortability and becomes one key index for people to measure the vehicle quality[3]. When the car is running, the factors such as engine rotation, transmission system torque vibration, road surface excitation and stream excitation can cause the noises inside the car[4]. The internal noise can affect the riding comfortability of both the driver and passengers, even causing the fatigue and threatening the driving safety. Therefore, it is of great significance to do the research into the internal noise.

The vehicle body, usually composed of stamped steel plates after welding, is a non-ideal rigid body. These plates will produce vibrations under the external excitation. The confined space inside the vehicle body forms an acoustic cavity and the air, as the elastic body, will be forced to vibrate under the excitation of body panels, which will in turn cause the vibration of the body panels. The interplay between the structure and the air produces the acoustic-solid coupling phenomenon[5]. In normal circumstances, the coupling effect in the LF range is obvious and the LF coupling mode will make the sound pressure in some areas inside the car increase dramatically under the excitation of its mode frequency, producing the strong noise response[6].

At present, there are mainly three simulation research methods of noises inside the car, namely, Statistical Energy Analysis (SEA), the finite element method and boundary element method [7-8]. Among them, SEA applies to the vibration noise transmitted by the air in MF and HF range. The boundary element method is mainly used to solve the external problem of sound field or the internal sound field with complex boundary conditions. The LF acoustic-solid coupling noise(20-200 Hz) is mainly transmitted by the solid, which can be analyzed and studied with the finite element method based on the wave acoustics[9-10].

The paper, taking one certain kind of car as the subject and based on the finite element model of body-in-white, considers the effect of structural damping and internal decoration material acoustic impedance, builds the modified acoustic-solid coupling infinite element model and makes a predication about the LF noise inside the car. By comparing it with the actual vehicle test data, the paper verifies the accuracy of modified acoustic-solid coupling infinite element model. Besides, the research in the paper can provide reference for the sequential engineering of vehicle NVH(Noise, Vibration and Harshness) and the predication of vehicle vibration noise as well as the improvement of acoustic performance.

2. Finite Element Model of Body-in-white

The paper takes one home-made car as the subject and puts the previously drawn model of the body-in-white and the framework into Hypermesh to do some pre-treatment such as geometry cleanup and meshing. Do some necessary simplification during the meshing process, for example, ignoring some small holes, steps and chamfer and some unloading and unwelded parts with small quality. Since most parts of the vehicle body belong to thin-wall parts, mainly do the meshing to the body-in-white with the shell meshing and use the elastic beam element to simulate the welding spots. After meshing, the body-in-white model includes 755796 quadrilateral elements, 9978 triangle elements, 332 hexahedron elements, 20471 beam elements, in which there are 5627 elastic beam elements with simulation welding spots, and it meets the requirement that the number of triangle elements doesn't exceed 5% of the total model element number. Then confirm that the element quality meets the requirement of calculation accuracy before setting up the material properties properly. Lastly, build the model of the framework in a similar way. The finite element model of body-in-white is shown as Fig.1.

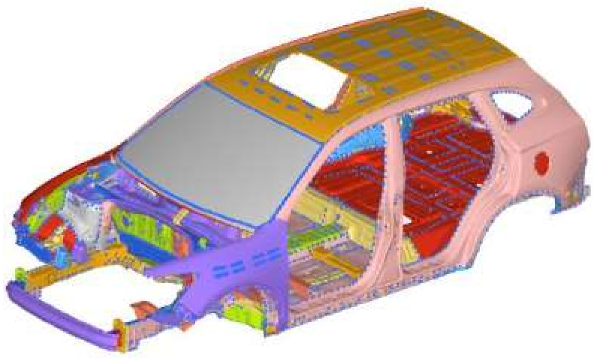


Fig.1 The Finite Element Model of Body-in-white

Add the drawn finite element model of front and rear door into the finite element model of vehicle body and use the revolute to simulate the actual connection status of the part linking the car door with the vehicle body. The connection between the lock and vehicle body is rigid to simulate the actual closing status of the car door. Further add the limit element grid of glass into the model, enclose the material property and choose the shell meshing for the glass. The final vehicle limit element model is shown as Fig.2. Build the finite element model of seats in accordance with the size of seat model and on the basis of vehicle model.

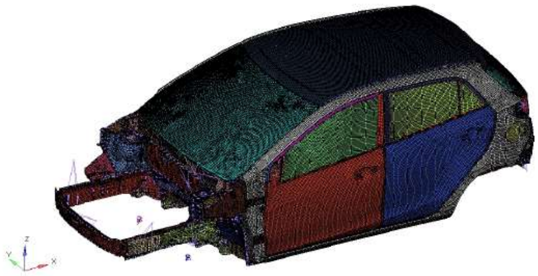


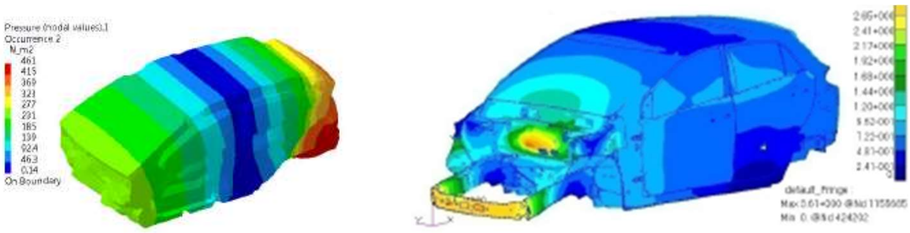
Fig.2 Vehicle Limit Element Model

3. Acoustic Cavity Finite Element Model

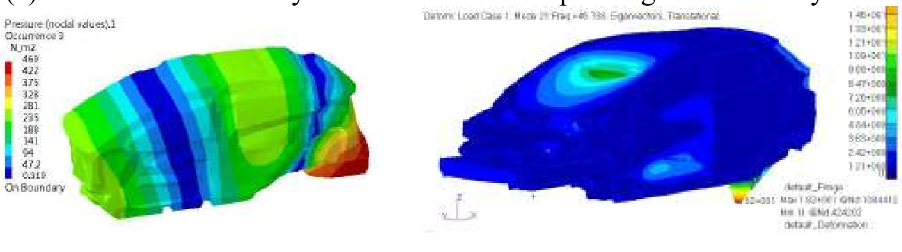
The element classification length of acoustic cavity grid is 100mm, and in this way, while each analyzed sound wave length range includes at least 6 acoustic elements, the shape of acoustic cavity matches better with the surface of vehicle body, limiting the element number of acoustic cavity and reducing the calculation burden.

To guarantee the higher quality of empty cavity and acoustic-solid coupling degree, adopt the Acoustic Cavity Mesh in the Hypermesh to do the meshing for the acoustic cavity grid. The acoustic cavity division area is between the vehicle body and seats, producing the mixed grid of tetrahedron and hexahedron as well as 20698 individual grids. Enclose the properties of air: the bulk modulus is $1.42 \times 10^5 Pa$, the density is $1.21 kg/m^3$, and the acoustic velocity is $340 m/s$.

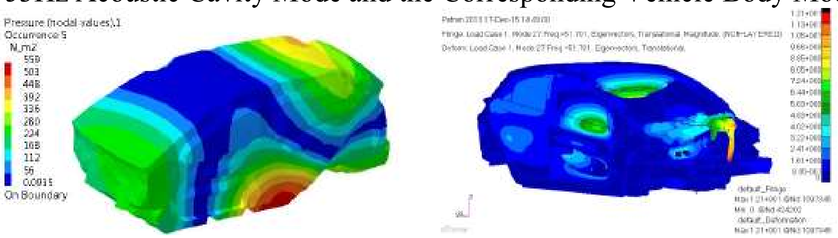
4. Acoustic-solid Coupling Modification Limit Element Model



(a) 31Hz Acoustic Cavity Mode and the Corresponding Vehicle Body Mode



(b) 53Hz Acoustic Cavity Mode and the Corresponding Vehicle Body Mode



(c) 105Hz Acoustic Cavity Mode and the Corresponding Vehicle Body Mode

Fig.3. The Acoustic-solid Coupling Mode

Load the model of vehicle, seats and acoustic cavity into the Virtual.Lab and produce the surface mesh on the surface of acoustic cavity model corresponded with the interior material to simulate the interior. Use the impedance tube to measure the acoustic resistance and acoustic holography of the interior material (including the floor, seats, ceiling and side walls' interiors) and impose the acoustic impedance of the sound absorbing material obtained from the tests on the interior material and seats to build the complete acoustic model.

Due to the model's complexity, the joints of acoustic cavity and those of vehicle body element don't coincide. Solve to acquire the acoustic-solid coupling mode and the mode frequency and vibration mode of a few coupling modes are shown as Fig.3 below.

5. Predication of LF Noise Inside the Car

0-300 Hz: Put the built acoustic-solid coupling finite element model into Virtual.Lab to do the frequency response analysis calculation. The hearing range of people's ear is 20-20000 Hz and the LF acoustic-solid coupling noise is mainly within 200 Hz, therefore, the frequency mainly analyzed with the finite element simulation is in the range of 20-200 Hz, which is the intersection of the two. To reduce the calculation time, the paper adopts the mode method for calculation: The ending frequency in the vehicle body mode calculation is 1.5 times the ending frequency in the frequency response calculation. Besides, the starting frequency should be 0Hz, that is, the frequency range of mode calculation is 0-300 Hz.

Do the vehicle test according to the test standard GB/T12534.1990 and GB/T 18697-2002 to gain the vibration noise response value and indirectly, the excitation signal. Take the even running condition of the car at the speed of 50km/h for example, the main excitation borne by the acoustic-solid coupling system established above comes from the passive end excitation point of power train mount, the passive end excitation point of driving shaft mount, the passive end excitation point of exhaust suspension and the passive end excitation point of front and rear suspension.

The paper adopts the load identification method applied in the transfer path analysis (TPA) -- generalized inverse matrix to do the load identification for the car. The mathematical expression of generalized inverse matrix is:

$$\begin{bmatrix} F_1(\omega) \\ F_2(\omega) \\ \vdots \\ F_n(\omega) \end{bmatrix} = \begin{bmatrix} H_{11,str}(\omega) & H_{12,str}(\omega) & \cdots & H_{1m,str}(\omega) \\ H_{21,str}(\omega) & H_{22,str}(\omega) & \cdots & H_{2m,str}(\omega) \\ \vdots & \vdots & \ddots & \vdots \\ H_{n1,str}(\omega) & H_{n2,str}(\omega) & \cdots & H_{nm,str}(\omega) \end{bmatrix}^{+} \begin{bmatrix} P_{1,\mathrm{str}}(\omega) \\ P_{2,\mathrm{str}}(\omega) \\ \vdots \\ P_{\mathrm{n},str}(\omega) \end{bmatrix} \tag{1}$$

$[F_n(\omega)]$ is load matrix; $[H_{str}(\omega)]$ is structure transfer function matrix obtained by the hammering method; "+" on the right upper side of transfer function matrix stands for generalized inverse matrix; $[P_{\mathrm{str}}(\omega)]$ is the response matrix.

During the process of load identification, it's required that the number of response points are no less than twice of the transfer path number to avoid the appearance of sick equation set. It means that except the target response points, it's necessary to arrange additional reference points in the car and the number of reference points should be at least twice of the studied transfer path number.

From the equation (1), when doing the load identification, there are two major steps: first, acquire the transfer function from the excitation point to target response point and reference point; second, acquire the working condition data of each target response point and reference point. Then get the load through the equation (1).

First, use the hammering method to acquire the transfer function of each transfer path. Get the load by putting the test data into the generalized inverse matrix and in the Virtual.Lab, impose the load on the engine mount, driving shaft support passive end, exhaust suspension support passive end and the corresponding positions of passive end of front and rear suspension at both sides. Mainly verify the acceleration speed of vehicle body reference point and the noise around the driver's ear, since the acoustic-solid coupling is mainly the coupling between the vehicle body and acoustic cavity. The position of noise response point should be around the driver's ear to gain the response value of noise around the driver's ear and meanwhile measure the vibration of vehicle body panel position. The obtained noise frequency spectrum is shown as Fig.4.

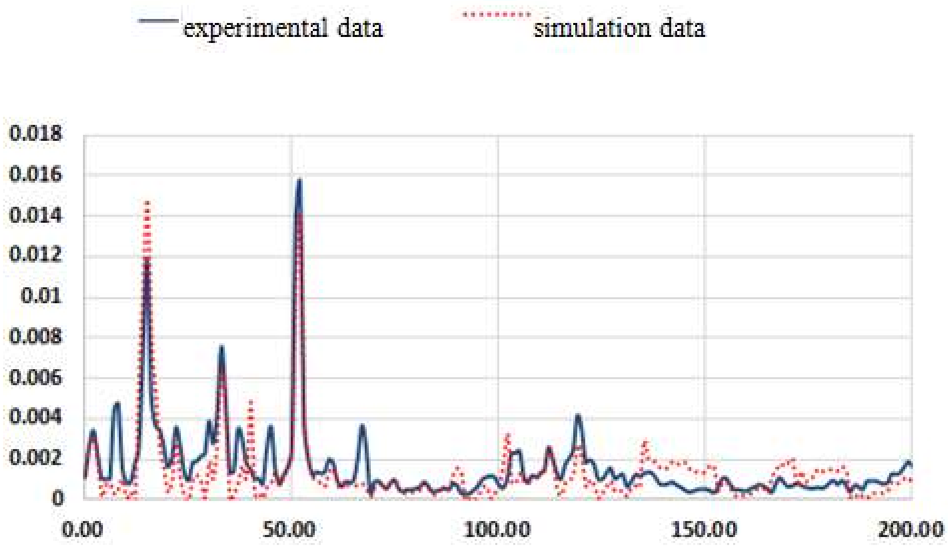


Fig.4 The Comparison Between Frequency Spectrum Simulation and Test Data of LF Noise Inside the CarTest DataSimulation Data

Through the comparison between the calculation result of frequency response and the test results, we can see that the simulation calculation value and test value of acceleration match well and there are obvious peak values in 33 Hz, 52 Hz, 103 Hz and 155 Hz in LF range.

6. Conclusion

The paper, on the basis of vehicle vibration noise test and taking the 50 km/h even running condition for example, builds the modified limit element model, makes a predication of vehicle LF noise and draws the following conclusions:

(1) Based on the finite element modeling of body-in-white and taking the effect of structural damping and interior material acoustic impedance into consideration, the paper builds the acoustic-solid coupling modified limit element model of vehicle LF noise. Compared with the test, the frequency response simulation of noise around the driver's ear under the even driving condition basically matches with the test frequency spectrum, with the main peak values coinciding, which indicates that the acoustic-solid coupling modified limit element model can effectively predicate the LF noise inside the car.

(2) Based on the established acoustic-solid coupling mode and through the panel acoustic contribution degree calculation, confirm three main peak points at 31 Hz, 53 Hz and 105 Hz and do analysis in combination with the coupling mode to lay a foundation for the follow-up improvement and optimization of vehicle body panel.

(3) The LF acoustic-solid coupling modeling method adopted in the paper can be used for the predication and improvement of LF noise inside the car and can serve as an effective simulation tool for improving the vehicle's NVH performance.

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