Acoustic Modelling and Analysis of Vehicle Interior Noise Based on Numerical Calculation

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***Abstract*—In this paper, various numerical calculation methods are rigorously combined to model and analyze the acoustic characteristics of a heavy truck cab. Firstly a brief introduction of different numerical calculation methods is given, then the finite element structure model and cavity model of the cab are built. The statistical energy analysis models of the cab, including structural subsystems and cavity subsystems ,are also built and used in the high-frequency range. The panel acoustic contribution analysis and the energy transfer path contribution analysis are carried out to find the panels which contribute most to the sound pressure inside the cab. Based on the analysis results, structure modification of these panels has been proposed with acoustics topology optimized. The optimization result shows that the sound pressure level on the side of the driver's right ear can be reduced by a few decibels. Necessary tests to validate the models are also administered.**

Keywords-numerical calculation; system modelling ; contribution analysis

I. INTRODUCTION

Nowadays, numerical calculation is becoming more and more widely used in the vibro-acoustic analysis and design of vehicles, so as to improve the acoustics performance and ride comfort of the vehicles. Numerical calculation methods used in the vibro-acoustic analysis of vehicles mainly include finite element method (FEM), boundary element method(BEM) and statistical energy analysis(SEA), each of which is suitable for different frequency range and the type of the structure.

In the automotive industry, it is used to divide the vibro-acoustics problems in three separate domains, according to the frequency range. The low-frequency (LF) range is identified as the domain for which the dimensions of the subsystems may be considered short with respect to the wavelength. The high-frequency (HF) field is defined as the frequency range for which the components of a system are long with respect to the wavelength. This characteristic implies that the presence of small uncertainties in the properties of the subsystems can dramatically influence the response of the structure. Finally, the mid-frequency (MF) domain is defined as a transition region. In this field, the structure is constituted of two classes of subsystems, respectively, exhibiting a LF and a HF behavior[1].

FEM and BEM are deterministic element-based methods. They are successfully used to predict the dynamical response of a structure, and are able to provide local and narrow-band solutions. The current computational resources allows these numerical methods to be efficient even for complex structures as far as the LF domain is concerned.

However, as the frequency increases, the wavelengths decrease and hence the discretization mesh of the structures must be refined. On the other hand, the increasing sensitivity of the responses to small perturbations implies that performing deterministic simulations is meaningless, and it is therefore much more relevant to develop formulations able to predict a priori the statistical vibrational response in terms of expectations and statistical moments.

The statistical energy analysis is an established approach for simulating the behavior of a vibro-acoustic system at high frequencies. SEA is a substructuring analysis method which is aimed at predicting the energy levels space and frequency averaged. The SEA formulation is developed by dividing a complex system into groups of similar normal modes. The energy stored in each subsystem is derived by summarizing incoherently the energy stored in each one of the modes that comprise the subsystem. The power balance equations over each subsystem, the relationships between energy stored in adjacent subsystems and power flow between them, and the relationships between dissipated power and energy stored in a subsystem allow to formulate the SEA system of equations. SEA is generally used for structure-borne or air-borne excitations.

Analysis and optimization of acoustic behavior of vehicles with FEM and BEM has been an important research field. Amiya R.Mohanty used FEM and BEM to characterize the interior acoustic field of a truck cab[2]. Jun Dong employed the adjoint variable method to calculate the design sensitivity for structural-acoustic optimization problems[3]. C. R. Fredö researched the NVH optimization of a cab floor by means of sound transmission analysis[4]. Maria B. Dühring employed topology optimization to minimize the squared sound pressure amplitude by distribution of reflecting or absorbing materials in a chosen design domain [5]. Jianghua Deng developed a computer-aided-engineering (CAE) methodology to optimize damping treatments of cars with acoustic-structural sensitivity analysis for coupled acoustic-structural systems[6]. S. Subramanian used modal strain-energy information of bare structural panels to identify flexible regions and optimize damping treatments[7].

As to the application of SEA in vehicles, Rohit Gujarathi modelled and studied the interior noise in an off-highway truck cab using SEA. The analysis was performed using two

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different modeling techniques[8]. Terence Connelly applied SEA to the sound package design for a cnvertible and optimize the acoustic insulation in the top-stack area which were shown to be important transmission paths for tire noise[9]. Arnaud Charpentier used the hybrid FE-SEA method to create fast and efficient model of structure-borne noise in a fully trimmed vehicle from 200Hz to 1kHz[10]. Francis Poradek and Mohan D. Rao utilized SEA to simulate the effects of a variety of noise control treatments on the interior sound pressure level(SPL) of a commercial excavator cab and the effects of leaks on the SPL of the excavator cab were also investigated[11]. Xin Chen built an SEA model with all the parameters based on a domestic car and introduced the simulation method to design car interior trims based on NVH performance[12].

In this paper, according to different analysis frequency range, methods of FEM, BEM and SEA are synthetically used to model and analyze the acoustic characteristics of a heavy truck cab. The FE models and SEA models of the cab are built. Acoustic characteristics are analyzed for different frequency range. The panel acoustic contribution analysis and the energy transfer path contribution analysis are carried out to find the panels which contribute most to the sound pressure inside the cab. When calculating the nodal velocities of each panel necessary to the panel acoustic contribution analysis, the coupling modes are used instead of the structural modes in the mode superposition method. Based on the analysis results, topology optimization has been carried out to achieve the goal of low noise. The optimization result is given, which proves to be satisfactory.

II. FE MODEL OF THE CAB AND CAVITY

The cab is composed of a large number of components and parts , such as beams, plates and pillars. Modeling all the components and parts will result in a model of a large number of elements and the computation cost will be very high . On the other hand, it is not necessary to build such a model, as not all the parts play the same important role in the determination of the vibro-acoustic characteristics of the cab. It is enough to just model the ones which dominate the vibro-acoustic behavior of the cab, such as floor, side panels, roof, toolbox, door, glass and main beams. The connections between window glass and door are supposed to be rigid, the same is with the connections between the door and the cab. The elasticity of rubber sealing strips is not considered.

The FE modeling of the cab is carried out in the software HYPERWORKS. The average element size is 40mm, the number of the total elements is 96,052 and the number of the total nodes is 60,011.There are two kinds of materials in the model, being named Steel and Glass respectively. The parameters of the material Steel are as follows: density ρ=7800kg/m3,elastic modulus E=2.1×1011Pa,passion ratio u=0.3,while those of the material Glass are respectively ρ=2400kg/m3,elastic modulus E=0.72×1011Pa,passion ratio u=0.22.

The FE model is shown in the Fig.1.

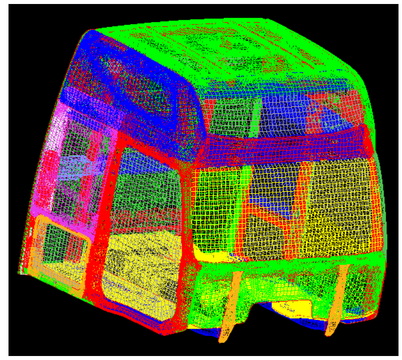

Figure 1. Structure FE model

In order to validate the FE model of the cab, tests were administered to analyze the modes of the cab.

The result of the tests is compared with that of the calculations of the FE model, the comparison is shown in Table I.

TABLE I. COMPARISON OF THE RESULTS BETWEEN TEST AND CALCULATION OF THE MODEL

No	test	calculation	mode shape
1	18.709	19.061	1st order torsion
2	20.672	21.775	local vibration
3	25.966	23.053	local vibration
4	43.088	43.532	1st order flexure

It can be seen that the two results correlate well, which assures us that the model can be used in the following research.

The cavity of the cab is the volume of the air inside the cab enclosed by the panels. According to the six-elements-per-wavelength rule, firstly the elements on the inner surfaces of the structure FE model are extracted, duplicate elements are deleted and discontinuous regions are patched. Then fill the volume enclosed by theses elements with tetrahedral elements. In this way, the one-to-one correspondence between the the nodes of the cavity model and structure model in contact areas can be guaranteed. The number of the total elements is 321,159 and that of the total nodes is 69,625. The fluid property is the same as that of air, with the sound velocity of 340m/s and the density of 1.225kg/.m3. The FE model of the cavity is shown is Fig.2.

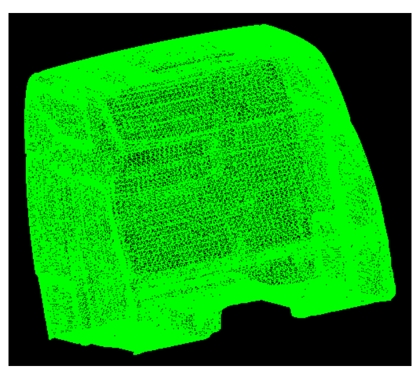

Figure 2. Cavity FE model of the cab

III. SEA MODEL OF THE CAB AND CAVITY

In SEA, the modeled structure is divided into subsystems. Based on the FE model of the cab and the principles of modal similarity, all of the structural subsystems were built. The panels, windshields and glass were built as plates or curved shells; the door pillar and the frame of the toolbox were modeled as FE subsystems, as they exhibits long wavelength behavior over a wide frequency range. The whole structure of the cab was divided into 20 SEA subsystems and 4 FE subsystems. The structural SEA model is shown in Fig.3. The subsystems in green is made of steel, while the subsystems in white is made of glass.

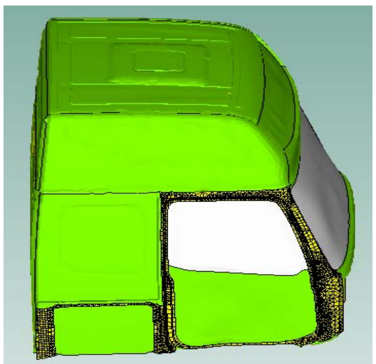

Figure 3. The structural SEA model of the cab

The occupant cavity of the cab was divided into 15 subsystems ,as shown in Fig.4.

The dynamic behaviors of SEA subsystems are described by three parameters: modal density, damping loss factor and coupling loss factor. The determination of these parameters are omitted here, due to the space limit.

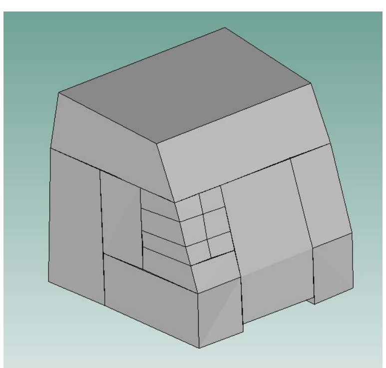

Figure 4. The cavity SEA model of the cab

With the SEA models, the responses of acoustic and vibration can be calculated. Four tested forces excitations were input at the four mounts of the cab, the SPL on the side of the driver's right ear was calculated. The result is shown in Fig.5.

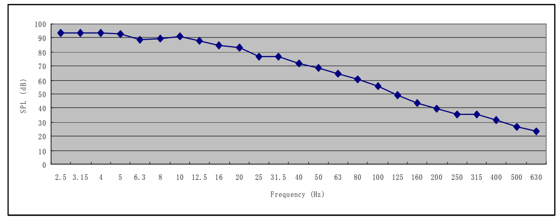

Figure 5. Calculated SPL on the driver's right ear

Energy flow between subsystems can also be determined . As to the cavity near the driver's right ear, the power input relation is shown in Fig.6.

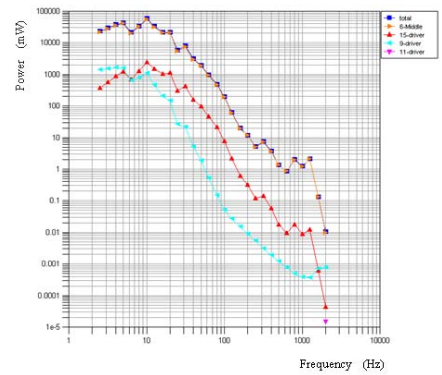

Figure 6. Power inputs to the cavity near the driver's right ear

In this way, the dominant power input sources to the cavity near the driver's right ear can be located. They are the left floor and right floor.

IV. COUPLING PANEL ACOUSTIC CONTRIBUTION ANALYSIS AND OPTIMIZATION

A. Panel acoustic contribution analysis (PACA)

PACA is used to determine the contribution of the panel to the sound pressure level(SPL) at an isolate field point. PACA allows for the classification of vibrating panels according to their contribution to the total sound pressure at a given location. This knowledge is very critical for a good sound quality design.

First, nodal vibration velocity of each panel needs to be calculated. Taking the effect of structural-acoustic coupling into account, nodal velocity is calculated with the mode superposition method.

After the velocity boundary condition has been determined, The contribution of a panel to the SPL at the point on the side of the driver's right ear can be calculated according to (2)[13].

$$c(S_p) = \int_{S_p} (p\frac{\partial G}{\partial n} + i\rho\omega v_n G)dS \qquad (2)$$

Where S_p is the panel surface, G is the Green's Function, $c(S_p)$ is the panel contribution , p is the pressure, ρ is the density, v_n is the normal velocity, ω is the frequency.

The rank of contributions of different panels of the cab is shown in Table II.

TABLE II. CONTRIBUTIONS OF THE CAB'S MAIN PANELS

No	component	31.5Hz	100Hz
1	left panel	-0.55	2.54
2	right panel	-1.27	-5.43
3	Middle floor	52.79	82.16
4	Left toolbox cover	4.91	-0.15
5	Left toolbox	-1.89	-0.88
6	Right toolbox cover	1.39	11.77
7	Right toolbox	0.04	0.75
8	Front-top panel1	-36.64	-84.84
9	Wind glass	101.17	-56.75
10	Instrument board	10.37	-42.56
11	Front-top panel2	40.35	152.91
12	Left floor	-38.96	-18.21
13	Right floor	2.46	-3.02
14	Right door glass	-10.25	2.09
15	Right door	-8.07	-3.86
16	Left door glass	-28.7	44.53
17	Left door	-17.09	28.55
18	Rear panel 1	-56.9	-86.37
19	Rear panel 2	-34.88	19.74
20	Roof panel	85.47	-9.81
21	Left-top panel	46.15	92.47
22	Left side panel	-26.66	-42.52
23	left doorframe	-10.35	3.86
24	Right-top panel	24.4	6.49
25	Right side panel	6.81	5.08
26	Right doorframe	-4.1	1.44

In Table II, The contribution is positive means that the SPL increases as the vibration level increases, while the contribution is negative means just the opposite. From the table, it can be seen that at these two frequencies, the contributions of the middle floor are both positive and large.

B. Acooustic Optimization

Based on the above analysis, panels which contribute most to the SPL on the driver's right ear can be determined. Measures can be taken to reduce the vibration level of those panels so as to improve the sound quality of the cab. One way is acoustic topology optimization. The detailed optimization process is omitted here, due to the space limit. The SPL on the side of the driver's right ear before and after optimization is shown in Fig.7.

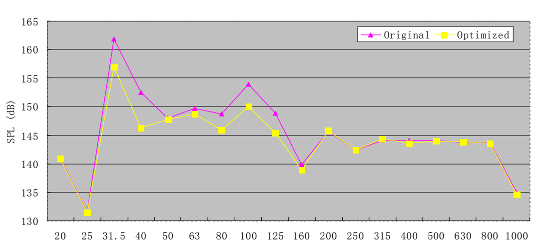

Figure 7. Comparison of SPL before and after optimization

It can be seen that at the frequency 31.5Hz, the SPL is reduced by nearly 5dB, while at 100Hz, the SPL is reduced by 3dB or so. At other frequencies, the change is not so apparent.

V. CONCLUSIONS

Some conclusions can be drawn from the paper, as listed below:

- Various numerical calculation methods, such as FEM, BEM and SEA, can be rigorously combined to model and analyze the acoustic characteristics of vehicles.
- The sound quality of the cab has been improved after acoustic topology optimization. The interior noise level has been reduced by a few decibels.

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