Optimization of acoustic performance of a vehicle dash sound package

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ABSTRACT

Acoustic performance of a vehicle dash sound package was optimized to reduce vehicle interior noise. In this paper, Statistical Energy Analysis (SEA) model of the dash was firstly built. By comparing simulation data and experimental data of Transmission Loss (TL), the accuracy of SEA model of the dash was verified. Then, acoustic parameters of several new materials were identified and sound absorption performance of materials was analyzed. The TLs of the dash with several new material combinations as the sound package were analyzed and compared with that of the dash with the traditional one. By analyzing vehicle noise reduction (power based noise reduction, PBNR) and vehicle interior sound pressure level (SPL), a new material combination was chosen as the dash sound package. With considering TL of the dash, sound absorption of the sound package and total mass of the sound package, a sound package optimization method was proposed to optimize the dash sound package. After sound package optimization, TL of the dash was increased by 2dB on average and sound absorption of the dash sound package was improved greatly. PBNR from driver's head cavity to engine cavity was increased by 1.2dB, and vehicle interior SPL of driver's head cavity was decreased by 0.6dB on average. Meanwhile, the total mass of the dash sound package was decreased by 45%.

Keywords: SEA, Sound package, TL   I-INCE Classification of Subjects Number(s): 75.2

1. INTRODUCTION

Vehicle dash system is one of the major paths for vehicle interior noise and its sound package (1, 2, 3, 4, 5, 6) is mainly used to isolate the transmission of the powertrain noise into vehicle interior. To analyze vehicle vibrations and noise problems, there are mainly three methods, which are finite element method (FEM), boundary element method (BEM) and statistical energy analysis (SEA) method (1, 2, 3, 4, 5, 7). For TL analysis of dash and the design of sound package, SEA method is usually preferred.

Based on the problem that acoustic performance of a vehicle dash was so poor that the vehicle interior noise on high frequency was high, we need to optimize acoustic performance of the dash sound package to reduce the vehicle interior noise. When traditional dash sound package, which was often composed of PUFoam and EVA, was optimized, TL of dash and weight of sound package were treated as optimization objectives. But sound absorption of sound package, which had great influence on the vehicle interior noise reduction, should also be considered. So in this paper, sound absorption coefficient of sound package was taken as one of optimization objectives. Then, a sound package optimization method based on SEA method was proposed to optimize the dash sound package. Finally, the optimized sound package was applied to the SEA model of full vehicle to verify vehicle interior noise reduction.

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2. EXPERIMENT AND SIMULATION OF TL

2.1 Experiment of TL of dash

In this paper, the experiment of the dash TL was conducted in reverberation-anechoic chamber. Structure layout of the reverberation-anechoic chamber was shown in figure 1. The TLs of trimmed and untrimmed dash were shown in figure 2.

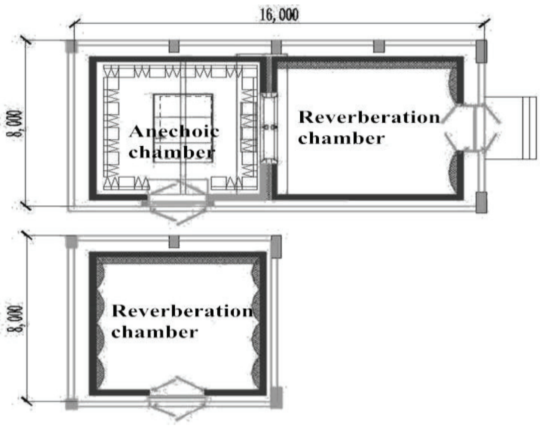


Figure 1 – Reverberation-anechoic chamber

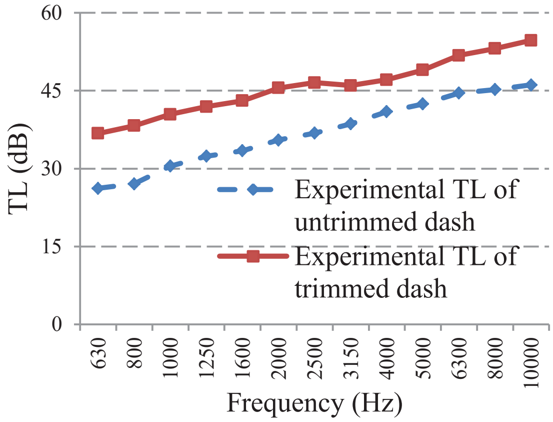


Figure 2 - Experimental TL of the dash

2.2 SEA model building and experimental validation

In the paper, SEA model of the dash was built in VAone (8) software. SEA model of the dash was shown in figure 3. The simulation result of untrimmed dash TL was obtained as shown in figure. 4.

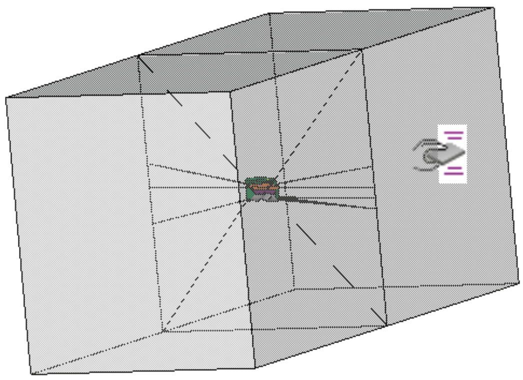


Figure 3 – SEA model of the dash

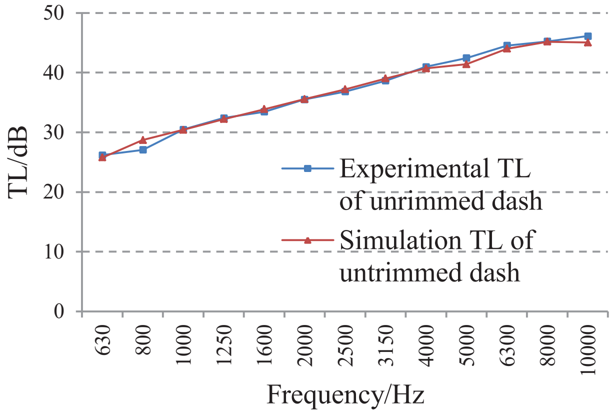


Figure 4 – Experimental TL and simulation TL of untrimmed dash

Figure 4 showed that error between experimental TL and simulation TL of untrimmed dash was less than 1dB. It indicated that the SEA model of dash satisfied calculation accuracy in 630Hz~10000Hz and could be used for sound package optimization.

3. SOUND PACKAGE MATERIAL

In this paper, the original sound package material of the dash was PUFoam and EVA. For the purpose of reducing weight and improving sound absorption performance of the dash sound package, we decided to replace the dash sound package with a new material combination. So we needed to identify acoustic parameters of new materials and analyze acoustic performance of different material combinations.

3.1 Identification of acoustic parameters

Foam-X (9) was used to identify acoustic parameters of materials. Foam-X provides two parameter identification methods: inverse method and indirect method. Inverse method can identify five parameters: open porosity (ϕ), static flow resistivity (σ), geometrical tortuosity (α_∞), viscous characteristic length (Λ) and thermal characteristic length (Λ'). Indirect identification method can identify three parameters: geometrical tortuosity, viscous characteristic length and thermal characteristic length. Based on the Biot theory (9), Foam-X took impedance tube test (9) results as input parameters.

In order to obtain sound package material with excellent acoustic performance, three kinds of sound-absorbing material were picked out, as follows: Half Solidified Felt (HSF), Thermoplastic Cotton Felt (TCF), High Molecular Polymer (HMP). Foam-X was used to calculate five acoustic parameters of the material as shown in table 1. And acoustic parameters of PUFoam were also shown in table 1.

Table 1 – Acoustic parameters of material

Material	$\rho/(\mathrm{kg/m^3})$	ϕ	$\sigma/(\mathrm{Ns/m^4})$	α_∞	$\Lambda/(\mu\mathrm{m})$	$\Lambda'/(\mu\mathrm{m})$
PUFoam	55	0.95	30000	1.31	105	1000
HSF	67	0.97	24670	1	19.1	126
TCF	435	0.95	1133000	1	150	28
HMP	80	0.95	40000	1	150	28

3.2 Acoustic performance analysis of material combination

3.2.1 Analysis of sound absorption of material combination

NOVA (10), which is based on transfer matrix, was used to analyze sound absorption performance of material combination. Transfer matrix mainly includes three parts: the internal transfer matrix of the material, the continuity condition matrix at the interface of different material and the boundary condition matrix. Acoustic impedance rate of multilayer material combination can be accurately calculated using this method, and the sound absorption coefficient of material combination can be obtained.

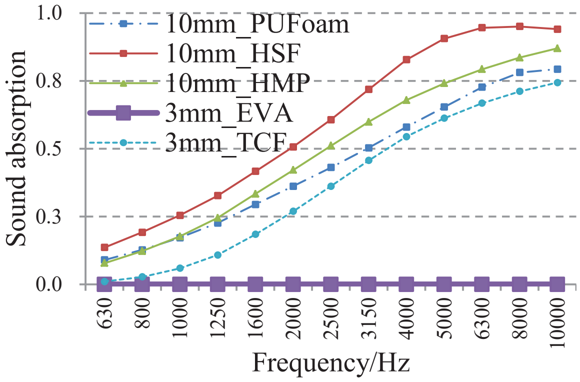


Figure 5 – Sound absorption of single material

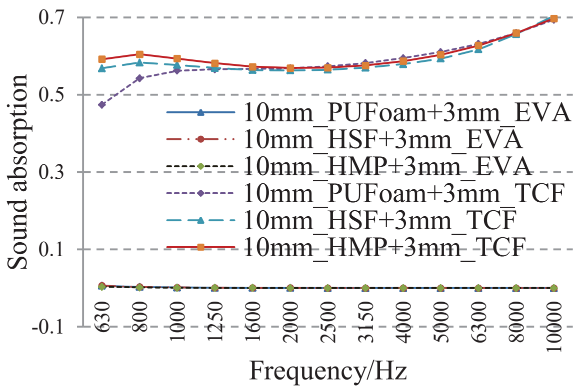


Figure 6 - Sound absorption of material combination

In this paper, the optional materials of sound package were the following five kinds: PU Foam($55\,\mathrm{kg/m^3}$), HSF($67\,\mathrm{kg/m^3}$), TCF($435\,\mathrm{kg/m^3}$), HMP($80\,\mathrm{kg/m^3}$), EVA($1600\,\mathrm{kg/m^3}$), and we would analyze sound performance of these material combinations. The sound package usually consists of hard sound insulating material which is high density and soft sound-absorbing material which is small density. So we defined denser TCF and EVA as the hard layer, the remaining three materials as soft layer. By setting hard layer thickness as 3 mm, thickness of soft layer as 10 mm, the sound absorption performance of the five kinds of single material and six kinds of material combinations were analyzed. The results were shown in figure 5 and figure 6.

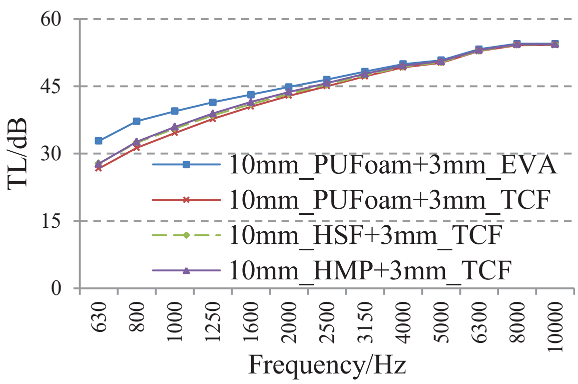


Figure 7 – TL of material combinations

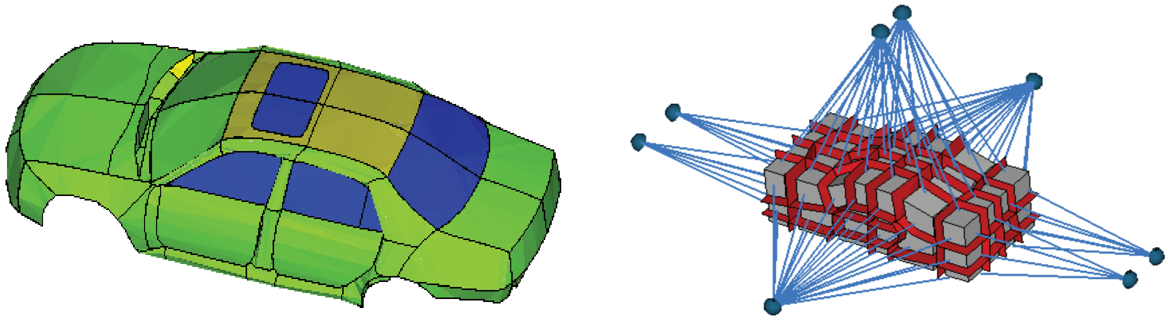
By the analysis of the sound absorption curves in figure 5 and figure 6, it concluded that: 1) PU Foam, HSF, HMP and TCF all had good sound absorption performance. 10 mm HSF had best sound absorption performance in the five kinds of single material, and sound absorption coefficient of EVA was close to zero; 2) When the hard layer material was EVA, whichever material the soft layer was, the sound absorption coefficient was close to zero. When the hard layer material was TCF, whichever the material of soft layer was, the sound absorption of material combination was significantly improved.

3.2.1 TL analysis

Three kinds of new material combinations with high sound absorption coefficients were respectively applied to SEA model of the dash, with the coverage setting to 90%. The TL curves of three combinations with comparing the original traditional sound package (PUFoam-EVE) were shown in figure 7.

Figure 7 showed that: 1) The TLs of three new material combinations were almost the same; 2) The TLs of the new material combinations were slightly lower than that of PUFoam and EVA in 630Hz~4000Hz frequency range; Above 4000 Hz, the TLs of new material combinations and PU Foam-EVA were almost the same.

Based on above analysis of sound absorption and TL, we could know that acoustic performance of three new material combinations was very close. However, by analyzing the total weight of four kinds of material combinations, it showed that the weight of PUfoam-TCF, HSF-TCF, HMP-TCF was reduced by 59.5%, 55.5% and 51.5% compared to that of PUfoam-EVA, respectively. Therefore, by taking the weight of sound package into account, PUFoam-TCF was chosen as the new material combination of the dash sound package.



(a) (b)

Figure 8 – SEA model of full vehicle

3.2.2 Analysis of vehicle interior noise reduction

In order to further compare the acoustic performances of the original traditional sound package PUFoam-EVA and the new sound package PUFoam-TCF, vehicle interior noise needed to be analyzed (11, 12). Full vehicle SEA model was built in Vaone, as shown in figure 8.

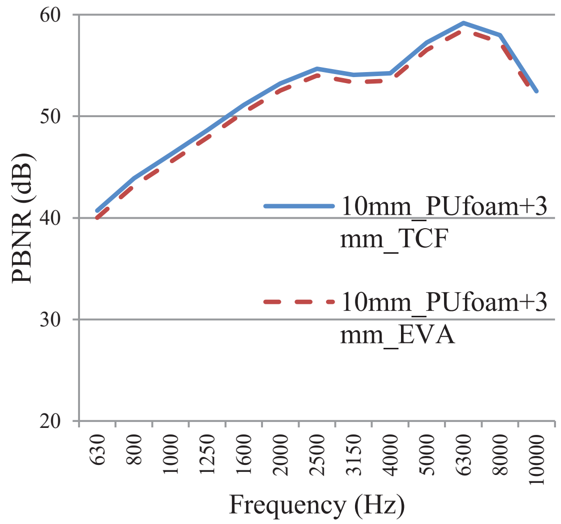


Figure 9 – Comparison of PBNR

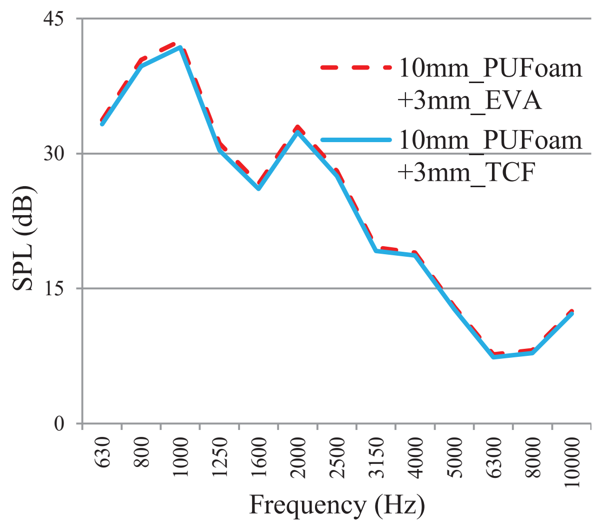


Figure 10 – Comparison of SPL

Here, power based noise reduction (PBNR) of vehicle from the driver's head cavity to engine cavity and the sound pressure level (SPL) of the driver's head cavity were calculated to evaluate the acoustic performances of two sound packages (5). The results of PBNR from the driver's head cavity to engine cavity and SPL of the driver's head cavity were shown in figure 9 and 10, respectively. For the new sound package PUFoam-TCF, PBNR was larger and SPL of the driver's head cavity was lower. S0 we could conclude that the new sound package PUFoam-TCF had better acoustic performance than the original one PUFoam-EVA. Therefore, the sound package PUFoam-TCF was taken as the dash sound package.

4. SOUND PACKAGE OPTIMIZATION

In order to optimize the acoustic parameters of the new sound package of the dash, multi-objective optimization method would be conducted. The multi-objective optimization design of the dash sound package contained five main steps: 1) Determining the design variables and objectives of optimization; 2) Selecting sample points by means of design of experiment (DOE); 3) The response values corresponding the design variables were calculated from simulation, and approximation model was built with the sample points and response values; 4) Based on the established approximation model, optimal solution was acquired using optimization algorithms; 5) Validation of the optimization results.

4.1 Design variables

The parameters that influenced acoustic performance of porous materials included: thickness (h), density (ρ), open porosity (ϕ), flow resistivity (σ), geometrical tortuosity (α_∞), viscosity characteristic length (Λ) and thermal characteristic length (Λ'). Also the parameter that affected sound insulation performance of dash was sound package coverage (f).

For PUFoam layer, the thickness (h_1) was taken as one of the design variables in optimization and the other parameters remained unchanged. For TCF layer, the thickness (h_2), density (ρ), open porosity (ϕ), flow resistivity (σ) and geometrical tortuosity (α_∞) were taken as design variables, while viscosity characteristic length and thermal characteristic length stayed unchanged. In addition, sound package coverage (f) was taken as design variable as well. The optimization goals were: TL of the trimmed dash, sound absorption coefficient (α) and total mass (M) of the sound package.

However, approximate model with high precision was hard to be established to optimize the 7 parameters simultaneously, and good optimization results could not be reached. As different parameter had different effect on the TL, sound absorption coefficient and the total mass of sound package. In order to reach good optimization results, 7 parameters were divided into 2 groups. The first group (4 parameters) included: h_1, h_2, ρ and f, the second group (3 parameters) included: ϕ, σ and α_∞.

The objectives of the first group were TL, α and M. For all of the parameters in the second group only influenced the sound absorption coefficient, the objective of the second group was α.

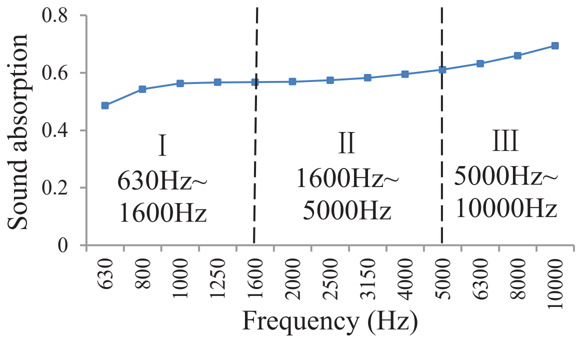


Figure 11 - Segments of sound absorption curve

Table 2 – Initial values and range of variables

Variables	Initial values	Range
ϕ	0.95	(0.9, 0.99)
$\sigma/(\mathrm{Ns/m^4})$	1133000	(1000000, 2000000)
α_∞	1	(1, 2)
h_1/(mm)	10	(10, 25)
h_2/(mm)	3	(2, 4)
ρ/(kg/m^3)	435	(300, 500)
f(%)	90	(85, 95)

For convenience of optimization, the average TL of the dash in 630Hz~10000Hz was acted as optimization objective. The sound absorption coefficient was calculated in the range of 630Hz~10000Hz. Moreover, with taking consideration of the curve characteristics of sound absorption in the frequency of 630Hz～10000Hz, the curve was divided into three sections in order to improve the accuracy of the approximate mode and the average value of each section was chosen as the optimization objectives.

The curve of sound absorption coefficient was shown in figure 11. The average sound absorption coefficient was respectively represented by α_1, α_2 and $\alpha_3$in the curve segments of Ⅰ Ⅱ, Ⅲ.

After the design variables were determined, the initial values and the ranges were given for the 3 parameters optimization design and 4 parameters optimization design. The results were shown in the table 2.

Table 3 - Sample points and responses of four parameters optimization

Numbers	*f*(%)	h_1/(mm)	h_2/(mm)	ρ/(kg/m^3)	α_1	α_2	α_3	*TL*/(dB)	*M*/(kg)
1	85	22.31	2.513	376.92	0.63429	0.590411	0.66079	45.7182	3.18959
2	85.26	16.15	3.179	371.79	0.57991	0.561429	0.66188	45.714	3.04621
3	85.51	13.08	2.41	300	0.66619	0.634095	0.65786	45.032	2.12868
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38	94.49	15.38	2	315.88	0.70502	0.673332	0.66475	47.364	2.40809
39	94.74	11.15	3.795	335.9	0.55449	0.552353	0.67208	48.733	3.08703
40	95	23.46	2.256	356.41	0.66933	0.612383	0.66547	49.142	3.43386

4.2 Four parameters optimization

4.2.1 Design of experiment

Design of Experiment (DOE) is a technical test based on probability theory and mathematical statistics, which can arrange experiments economically and scientifically. There are many methods of DOE, among which Optimal Latin hypercube design (Opt LHD) can make all test points distribute on the design space evenly (13). It has good qualities of space filling and balance. In the paper, Opt LHD was used to choose 40 groups of sample points, and the response values were calculated. The results were shown in table 3.

4.2.2 Approximation model building

In order to find the function between the design variables and the responses, the approximate model, which refers to a mathematical model with a small amount of calculation and short calculation period and calculation result of which is almost the same to experimental value, should be established. Kriging model is an unbiased estimation model with smallest estimation variance, which is one of the methods to build approximate models (14). Kriging model can include all the sample points and the quality of approximate surfaces is good. The Kriging model was built according to the relations between the design variables and the responses of four parameters optimization shown in the table 3.

In order to validate the fitting precision of the approximate model, Opt LHD was used to randomly generate other 10 groups of sample points and calculate the responses. R^2 was often used to evaluate the fitting accuracy. R^2 was defined as the following formula:

$$R^2 = \sum_{i=1}^{p} (\hat{y}_i - \bar{y}_i)^2 / \sum_{i=1}^{p} (y_i - \bar{y}_i)^2 \tag{2}$$

Where: p was number of the design point; $\hat{y}_i$, $\bar{y}_i$, y_i was predicted value, average actual value and actual value. The accuracy was higher when R^2 was closer to 1.0. Generally, R^2 was expected to be above 0.9. R^2 of the Kriging model were shown in the table 4.

Table 4 - Accuracy of Kriging model

Response	α_1	α_2	α_3	*TL*	*M*
R^2	0.9848	0.9679	0.9105	0.9554	0.9769

As we could see in the Table 4, R^2 of every objective was above 0.9, so the accuracy of the approximate model was reliable.

4.2.3 Performing multi-objective optimization

The target of multi-objective optimization of four parameters of the dash sound package was to maximize average sound absorption coefficient of the sound package and TL of the dash, meanwhile to minimize total mass of the sound package, as shown in following formula:

$$\begin{cases} \max & \alpha_1, \alpha_2, \alpha_3 \\ \max & TL \\ \min & M \end{cases} \tag{3}$$

In the process of optimization, the constraint was set as $TL \geq 47.2 + 2 = 49.2$.

Table 5 - Optimal values of design variables

Variable	h_1/(mm)	h_1/(mm)	ρ/(kg/m^3)	f(%)
Optimal value	23.4666	2.36838	395.5069	94.4949

Based on the Kriging model of four parameters optimization, we could solve the optimal solution in ISIGHT. Generally, optimal solution needed to be solved through optimization algorithm. Genetic algorithm is one of the global search algorithms. It is based on Darwin's 'survival of the fittest' ideology, which will finally converge to an optimal solution by crossover and mutation. NSGA-II genetic algorithm is currently the most widely used multi-objective genetic algorithm (15). In this paper, NSGA-II was used for solving optimal solution in ISGHT. By repeating the calculation, the weights of α_1, α_2, α_3, TL, M were set as: 0.12, 0.12, 0.12, 0.55, 0.09 respectively. The value of design variables of the optimal solution were shown in the table 5.

Table 6 – Comparison of simulation and optimization

Response	Simulation	Optimization	Error
α_1	0.645211	0.655973	1.64%
α_2	0.599978	0.604621	0.77%
α_3	0.662759	0.663973	0.18%
TL	48.99366	49.29216	0.61%
M	3.832522	3.827494	0.14%

The simulation value and the optimized value were compared, as shown in the table 6. We could see from the table that the errors were pretty small and all of them were below 2%, thus the results were accurate and reliable.

4.3 Three parameters optimization

Firstly, 30 groups of sample points were selected in three parameters optimization design space by optimal Latin hypercube method, and the responses of 30 groups of sample points were obtained by simulation. Due to limitation of the length of this paper, the sample points of three parameters optimization were not listed here.

According to the 30 groups of sample points and responses, Kriging model was adopted to establish the approximation model of three parameters optimization. In order to validate the fitting precision of the approximation model, other eight group of sample points were randomly generated using the optimal Latin hypercube method. R^2 was still used to evaluate the fitting precision. The corresponding values R^2 of α_1, α_2, and α_3 were: 0.9838, 0.98437 and 0.98129, which were all larger than 0.9 and satisfied the requirement of engineering precision.

Table 7 – Comparison of simulation and optimization

Response	Simulation	Optimization	Error
α_1	0.670111	0.677478	1.09%
α_2	0.629671	0.637346	1.20%
α_3	0.684222	0.688767	0.66%

The aim of three parameters optimization of sound package was that α_1, α_2 and α_3 reach the maximum at the same time. Optimum solution was found out in ISIGHT software using NSGA-II genetic algorithm. The weight values of α_1, α_2 and α_3 were set to 0.25. The optimum values of design variables obtained were as the follow: $\phi = 0.946$, $\sigma = 1013855\text{Ns/m}^4$, $\alpha_\infty = 1.405$.

Simulation values of responses were calculated by optimum values of design variables and the results were listed in table 7.Table 7 showed that the deviation between the simulation value of and optimal value of three parameters optimization was small, which was within 2% and met the requirements of practical engineering precision.

4.4 Validation

The TL of dash after sound package optimization and the TL of original trimmed dash (acquired by experiment) were compared, as shown in figure 12. And sound absorption coefficient before and after three parameters optimization was also compared, as shown in figure 13. We could see that the TL of the dash was increased by more than 1dB in 1600Hz~10000Hz and less than 1dB in 630Hz~1250Hz after sound package optimization. In addition, in figure 13, we could see that sound absorption coefficient of the dash sound package was improved after 3 parameters optimization.

When the optimized dash sound package was applied to SEA model of full vehicle, PBNR from driver's head cavity to engine cavity was calculated, and SPL of driver's head cavity was also calculated. The results were shown in figure 14 and figure 15.

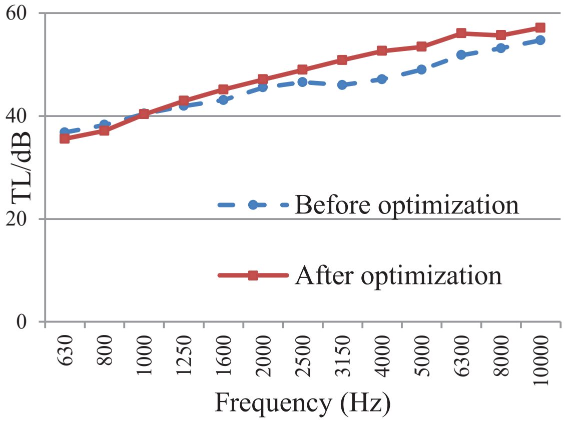

Figure 12 – The dash TL before and after sound package optimization

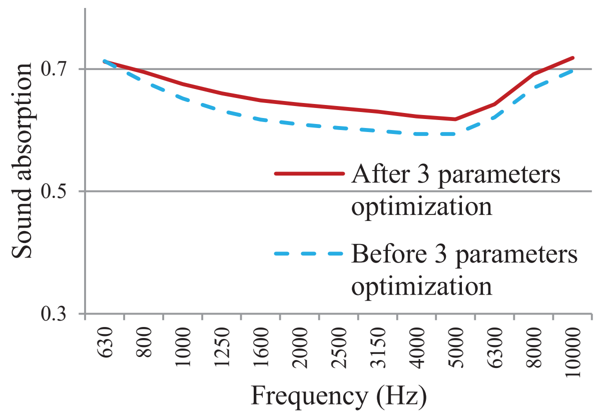

Figure 13 – Sound absorption of the dash sound package

Figure 14 showed that PBNR was increased by 1.2dB on average after optimization. Figure 15 showed that SPL of driver's head cavity was decreased by 0.6dB on average. And after optimization, the total mass of the dash sound package was decreased from 7kg to 3.8kg.

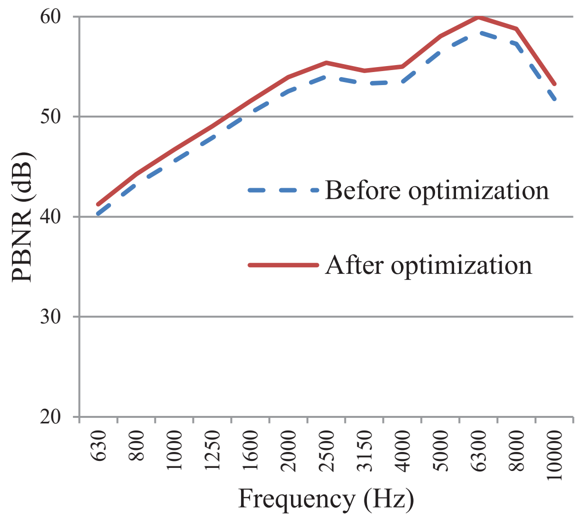

Figure 14 – PBNR before and after sound package optimization

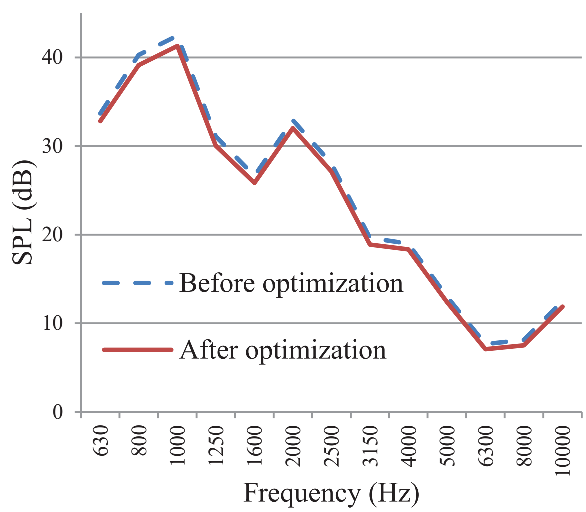

Figure 15 – SPL before and after sound package optimization

Based on optimization results, we could conclude that acoustic performance of the dash sound package was improved and the weight of the dash sound package was reduced after optimization.

5. CONCLUSIONS

In this paper, the sound package of a vehicle dash was optimized, the detailed was as follows::

(1) SEA model of a vehicle dash was built. By comparing simulation data and experiment data of TL, the accuracy of the SEA model of the dash was verified.

(2) Foam-X was used to identify acoustic parameters of material and Nova was used to analyze sound absorption of different kinds of material combinations. Based on SEA model of the dash, TLs of the dash with different kinds of material combinations as sound package were compared. Based on SEA model of full vehicle, vehicle PBNR and interior SPL were compared. As a result, PUFoam-TCF was finally chosen as the dash sound package material.

(3) The acoustic parameters of PUFoam and TCF were divided into 2 groups. Then, by considering TL of dash, sound absorption coefficient of sound package and total mass of sound package as optimization objectives, multi-objective optimization of four parameters and three parameters of sound package were conducted, respectively.

(4) After optimization, TL of the dash was increased by 2dB on average. Sound absorption coefficient was improved after 3 parameters optimization. When the optimized dash sound package was applied to SEA model of full vehicle, PBNR from driver's head cavity to engine cavity was increased by 1.2dB on average, and SPL of driver's head cavity was decreased by 0.6dB. After optimization, the mass of sound package was decreased by 45% (from 7kg to 3.8kg).

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