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NVH analysis and improvement of a vehicle body structure using DOE method †

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Abstract

The improvement of a vehicle body structure under the constraint of noise, vibration and harshness (NVH) behavior is investigated by using design of experiments (DOE) method. First, car body geometry is modeled in CATIA and meshed in HYPERMESH software. Then, a modal analysis between 0-50 Hz is done by MSC NASTRAN. A frequency map of the body is extracted and compared with a reference map to identify shortcomings. By using factorial and response surface Methods (RSM), optimization of the NVH performance is accomplished. An algorithm is proposed to improve the car NVH behavior. At last, in order to verify the present algorithm, forced vibration is analyzed under real road inputs for the model before and after improvement.

***Keywords*:** Vehicle body; FEM; DOE method; Modal analysis; Frequency map; NVH; RSM

1. Introduction

The automotive industry is more and more characterized by serious pressures to shorten product development cycles, to broaden the spectrum of vehicles, and to develop and produce the product more efficiently. Automotive systems have become more complex, integrating a higher number of functions and features. Thus, not only increasing material costs, but also rising effort due to a higher number of exigencies on the vehicles, enforces the use of numerical simulation in vehicle development. A number of disciplines to be studied have risen enormously; starting from classical disciplines like driving simulation, structural static analysis and crashworthiness analysis, additional features have emerged such as interior and exterior acoustics, electromagnetic compatibility become important.

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In the two last decades, the numerical simulation via finite element methods (FEM) has been well integrated into the product development process (PDP) of the automotive industries. Non-linear (such as crashworthiness) and linear cases (such as NVH) have been accomplished successfully, so that the PDP is currently more and more driven by numerical simulation. Noise, vibration, and harshness (NVH) is one of the most important attributes for car product development. A vehicle with a good NVH behavior often results in much higher customer satisfaction. In vehicle development, different NVH models are used for different purposes and systems that lead together the quality of the NVH behavior.

First studies were realized by simple Monte Carlo search approaches, e.g., Kodiyalam [1], Kodiyalam and Sobieski [2], but it became soon evident that the computational effort remains cumbersome, although software and hardware engineering were improved remarkably. The complexity of numerical models often grows faster than the computational power. A detailed study on the most appropriate algorithms is still missing. NVH behavior is normally optimized via a gradient-based approach often integrated directly into the FE codes (e.g., SOL200 in NASTRAN software). As long as the problem at hand is convex, these methods are successful. For multimodal problems, they are not appropriate. The optimization is then often terminating at local instead of global extreme, e.g., Duddeck et al. [3], Kropp [4]. These studies have shown that the low frequency problems and linear statics can be optimized with local search algorithms, whereas global optimization strategies have to be used for higher frequency ranges encountered, for example, in interior car body acoustics. For all dynamic cases, attention has to be paid to mode tracking algorithms. Relevant eigen modes of the structure identified at the initial design are changing during the optimization process, but should be followed by the optimization procedure. Normally, a strategy based on the modal assurance criterion (MAC) is adapted, which is an indicator of the co-linearity of the original mode and the mode of the modified model. Unfortunately, this strategy is not always successful, and in some cases a restart of the optimization is necessary with an updated definition of the reference dynamic mode [5]. In some literatures, optimization of NVH problems is often solved via surrogate models, i.e., via the construction of response surface models like Kriging, polynomial regression, neural networks, radial basis functions, or similar approaches. In Myers and Montgomery [6], the general aspects of RSM are given, and in Schramm [7], Sobieszczanski-Sobieski et al. [8], Stander and Craig [9], Wauquiez et al. [10], the particularities are discussed when RSM is applied to targets such as weight, NVH behavior and crashworthiness problems.

In this article the vehicle body structure is represented by a finite element model of about 1.6 million degrees of freedom. The modal analysis in a frequency interval between 0-50 Hz is performed by the NASTRAN software, while surface modeling is carried out by CATIA and the meshed model has been constructed in HYPERMESH environment. In the optimization procedure, a DOE method including factorial and response surface methods is applied on natural modes compared with reference modes. Based on these data, an analytical approximation to the NVH behavior as a function of the design variables was constructed and used in the optimization procedure. Finally, an algorithm is presented for optimization of the vehicle NVH behavior and forced vibration analysis is employed to verify this algorithm.

2. The vehicle FE modeling

The vehicle model used for this study is fully surface modeled in CATIA consisting of suspension and powertrain sub-systems. All the closures (door, hood, etc.) and other sub-systems (steering column, pedal bracket, seats structure, etc.) are also modeled. The full scale NVH finite element model is shown in Fig. 1. Current FE models consist of up to 320,000 elements including shell, beam, mass and spring elements and the rigid links to model the spot welds.

3. Verification of the vehicle FE model

The normal modes are calculated under the free-free boundary condition. The natural frequencies are shown in Table 1. The first six modes are approximately zero due to rigid-body modes.

Table 1. Natural frequencies in free-free conditions.

Modes	Frequency (Hz)	Mode Shape
1	0.000133670	Rigid roll motion
2	0.000100390	Roll and for/aft motions of the car body
3	0.000085279	Roll and lateral motions
4	0.000059742	Bounce motion
5	0.004785400	Pitch motion
6	0.005587100	Yaw motion
7	6.6707	Torsion mode of the car body
8	7.5409	Flexible bending mode of the car
9	9.7455	Opposite torsion mode of the chassis and back-cabin
10	11.341	First bending mode of the car body

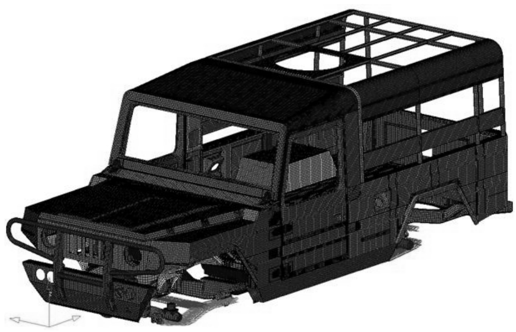

Fig. 1. Full vehicle finite element model.

4. Optimization of vehicle body structure

Optimization of the vehicle body structure under the constraints imposed by the regulations, pertains to the passenger isolation against the noise, vibration and harshness (NVH). Modal analysis has been done in a frequency interval between 0-50 Hz is investigated by using MSC NASTRAN software. Then, a frequency map (shown in Fig. 14 at the end of this paper) is drawn and compared with a referenced map to identify defects of the model. Consequently, on optimization procedure in which the above analysis would be accomplished is imposed by a DOE method such as factorial and response surface methods by using MINITAB software.

The vehicle body modal modes which have been modified are as follow.

To validate the results of the modal test and the finite element model, the results of the experimental modal test of a vehicle in the same class, which is extracted by Ricardo Company, is used (shown in Fig. 15). According to that figure, the range of the primary rigid body modes of the experiment is between 1.5 Hz to 4.5 Hz. That range for FE model and the modal analysis by MSC.NASTRAN software is between 2 Hz to 7 Hz. Therefore, the validation can be seen to some extent.

4.1 The torsion mode of the engine hood

The frequency of first torsion mode of the engine hood is 12.25 Hz, although the reference shows a limit of frequency between 25-30 Hz. To correct the mode of engine hood, the lock place is changed from the middle of the hood in to the two side corner locks. By analyzing the new model, the first mode of the engine hood becomes a bending mode with a frequency of 30.986 Hz (Fig. 2). It should be mentioned that this mode is a combinative mode with the vehicle ceiling bending mode as shown in this figure.

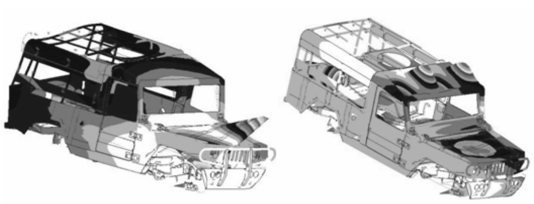

Fig. 2. First modes of the engine hood before (right) and after (left) modification.

4.2 The global bending and torsion modes of the body

First bending and torsion modes of the body are approximately started with a frequency of 7 and 12 Hz. Because of the limits from the reference frequency for this mode (15-20 Hz), the vehicle structure should be closed form. To correct these modes, the driver cabin and back-cabin are connected together with rigid-link elements (Fig. 3 and Fig. 4). By repeated modal analysis, those modes are approximately increased to 13 Hz (13.193 Hz for torsion and 13.833 Hz for bending modes).

4.3 The torsion mode of the engine hood

The first bending mode of the body ceiling begins at a frequency of 12 Hz (as shown in Fig. 3). Considering a frequency interval between 25-30 Hz for the ceiling in the reference, the factorial method is used to correct this mode. The total number of design variables is three, including adding the longitudinal and transversal beams with the section as illustrated in Fig. 5, and the thickness of the ceiling plate. The design variables are considered in two levels shown in Table 2. Experiments of the present method are listed in Table 3. The sensivity results are drawn in Fig. 6 which shows that the longitudinal beam is more effective among the other parameters.

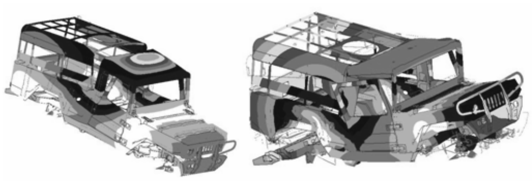

Fig. 3. The first body bending modes before (right) and after (left) correction.

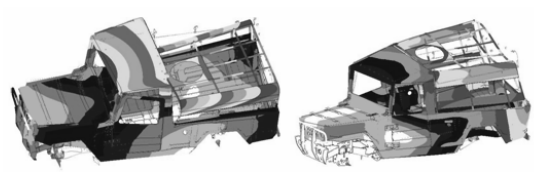

Fig. 4. The first body torsion modes before (right) and after (left) correction.

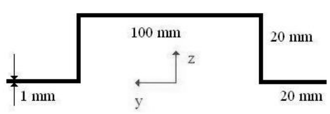

Fig. 5. The initial beam cross section used in the optimization process.

Table 2. Design variables and levels in the factorial method.

Level (1)	Level (-1)	Variables	Factors
2 beams	1 beam	Transversal beam	A
2 beams	1 beam	Longitudinal beam	B
2.0 (mm)	1.7 (mm)	Thickness of ceiling plate	C

Table 3. Experiments of Factorial method.

Run Order	A	B	C	Frequency (Hz)
1	1	1	1	19.590
2	1	1	-1	32.566
3	1	-1	1	12.559
4	1	-1	-1	12.478
5	-1	1	1	19.586
6	-1	1	-1	20.087
7	-1	-1	1	12.559
8	-1	-1	-1	12.478

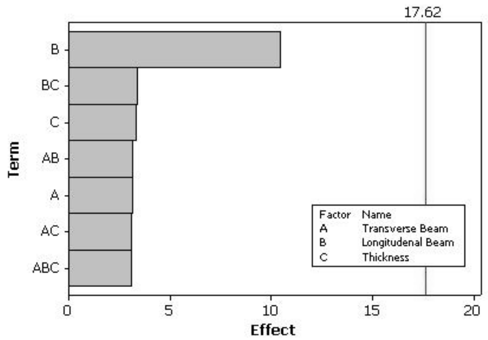

Fig. 6. Evaluation of the effectiveness of parameters used in the factorial method.

According to Fig. 6 and also choosing A, B, C and BC parameters, the regression equation is extracted as the following equation.

$$f = 17.738 + 1.56 \times A + 5.219 \times B \\ - 1.664 \times C - 3.41 \times B \times C \tag{1}$$

Optimized amounts are level (-1) for thickness, (0.7521) for the transversal beam and (1) for the longitudinal beam. Thus, the ceiling with two transversal beams (with approximating 0.7521 to 1), two longitudinal beams and ceiling thickness about 1.7 mm is optimum.

4.4 The bending mode of the back-bonnet

The frequency of first bending mode of the back-

Table 4. Design variable and levels in RSM.

Level (1)	Level (0)	Level (-1)	Variables	Factors
3 beams	2 beams	1 beam	Longitudinal beam	A
4 beams	2 beams	0 beam	Transversal beam	B
2.0 (mm)	1.8 (mm)	1.6 (mm)	Thickness of bonnet plate	C
3.0 (mm)	2.0 (mm)	1.0 (mm)	Thickness of beams	D

Table 5. Experiments of RSM.

A	B	C	D	Frequency (Hz)
0	0	0	1	23.004
1	-1	1	1	21.471
0	0	0	0	22.628
0	-1	0	0	23.032
-1	-1	1	1	22.620
0	0	1	0	20.126
-1	1	-1	1	21.496
0	1	0	0	21.526
-1	-1	1	-1	21.563
1	-1	-1	1	22.601
1	1	1	-1	22.627
-1	-1	-1	1	20.366
-1	1	-1	-1	22.637
1	1	-1	-1	22.641
1	-1	-1	-1	21.563
1	1	-1	1	22.583
-1	1	1	1	21.487
-1	0	0	0	22.631
0	0	-1	0	22.630
1	1	1	1	22.578
0	0	0	-1	20.055
1	-1	1	-1	22.642
-1	1	1	-1	23.044
1	0	0	0	21.527
-1	-1	-1	-1	22.653

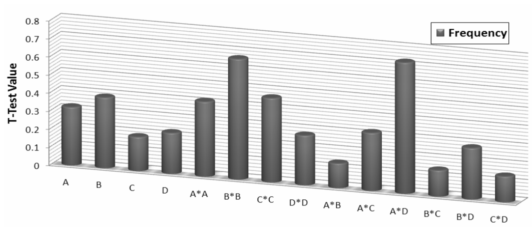

Fig. 7. The effectiveness of each parameter in RSM.

bonnet is 20.605 Hz. The total number of design variables for this component is considered as four parameters, including the additional longitudinal and transversal beam with the section which is shown in Fig. 5 and the thickness for bonnet plate and the beams. The design variables are considered in three levels as listed in Table 4. To modify this mode, RSM method is employed. Results of the design of experiments method including statistic data are presented in Tables 5-6. The sensivity analysis (Fig. 7) shows that the B parameter (the transversal beam) and the A parameter (the longitudinal beam) are more effective.

Considering variations of the A and B parameters versus D parameter, response surface and contour are drawn in Fig. 8 and Fig. 9. By choosing a frequency interval between 25-30 Hz, level (0) is selected for each parameter. Thus, the back-bonnet with two longitudinal beams, two transversal beams, with 1.8 millimeter thickness for bonnet plate and 2.0 millimeter thickness for the beams is improved.

The modified vehicle FE model is illustrated in Fig. 10. By modal analysis of this model, the final frequency map is prepared to identify changes in the modes as shown in Fig. 14 at the end of this paper.

5. Forced vibration

By imposing the real road inputs (vertical displacements) in two types, B & D according to ISO-2631 standard (as shown in Fig. 11) to the wheels of

Table 6. Statistical Results of RSM.

P-Value	T-Test	Coefficient	Term
0.000	39.899	21.8743	Const.
0.754	0.322	0.0964	A
0.704	0.391	0.1171	B
0.855	-0.188	-0.0562	C
0.826	-0.226	-0.0677	D
0.687	0.415	0.3303	A*A
0.520	0.667	0.5303	B*B
0.651	-0.466	-0.3707	C*C
0.789	-0.275	-0.2192	D*D
0.895	0.136	0.0431	A*B
0.755	-0.321	-0.1020	A*C
0.484	0.726	0.2305	A*D
0.888	-0.145	-0.0459	B*C
0.782	-0.284	-0.0901	B*D
0.889	0.143	0.0455	C*D

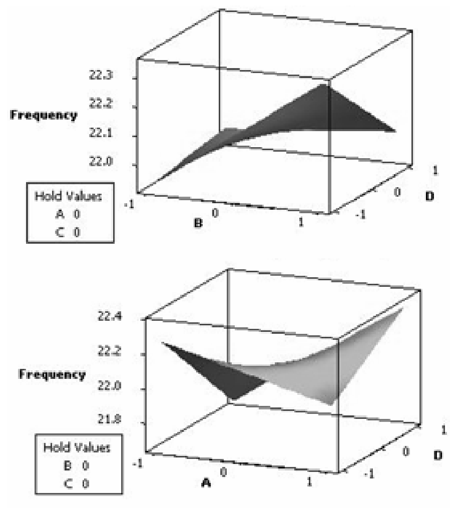

Fig. 8. Response surface considering variations of A and B parameters vs. D parameter.

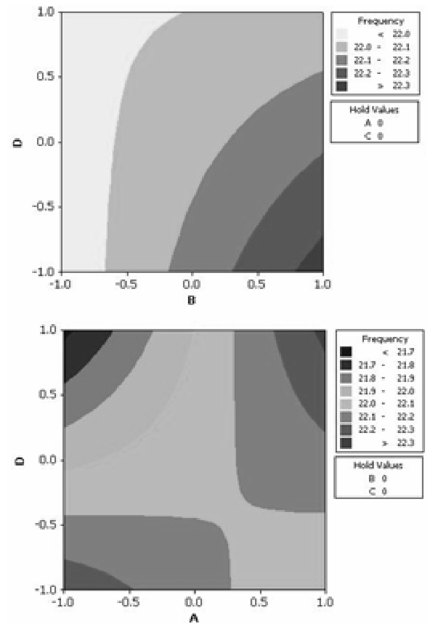

Fig. 9. Response contour considering variations of A and B parameters vs. D parameter.

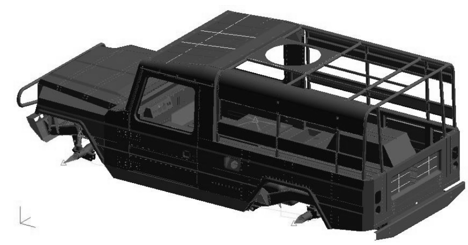
Fig. 10. The modified vehicle FE model.

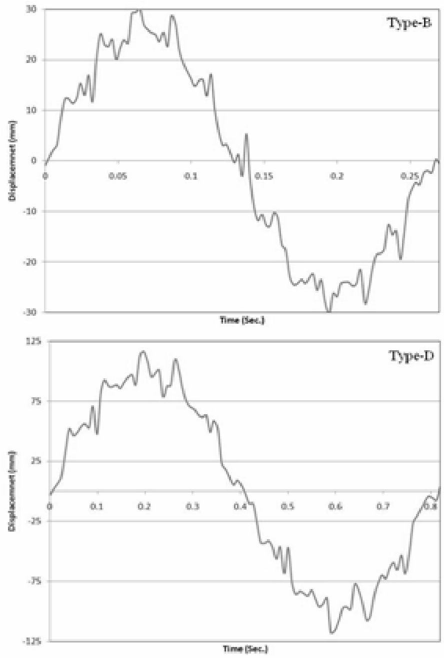

Fig. 11. Real road inputs type-B (up) and type-D (down).

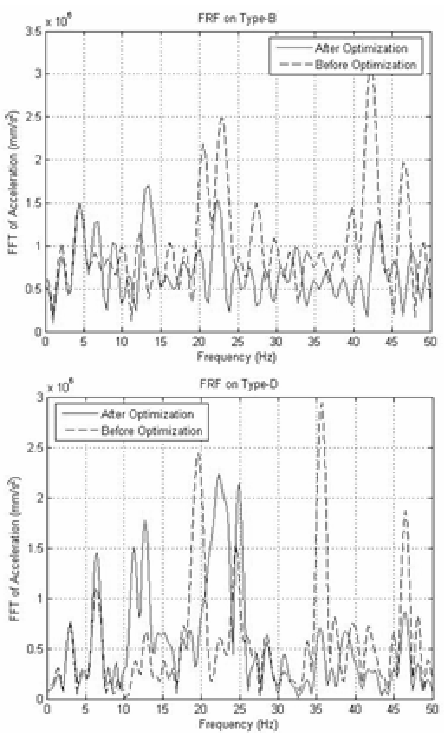

Fig. 12. FFT of driver seat acceleration for type-B (up) and type-D (down) road inputs.

the vehicle, the transient response can be investigated. The outputs in terms of acceleration in the driver seat location are transformed from time to frequency domain (Fig. 12). According to the vehicle responses, after and before the optimization process, it can be seen the sensible improvement in the modified model, so the validation of the present algorithm can be observed.

6. Proposed algorithm for vehicle body structure optimization based on the NVH behavior

Here, an algorithm is presented to optimize the car NVH behavior (Fig. 13). According to this procedure, for the vehicle structure optimization it should modify the body structure based on the modal characteristics and after that to compare the result of the frequency map with the reference frequency map (which is depend on the type of the vehicle) and then doing the forced vibration for verifying the real vehicle NVH performance.

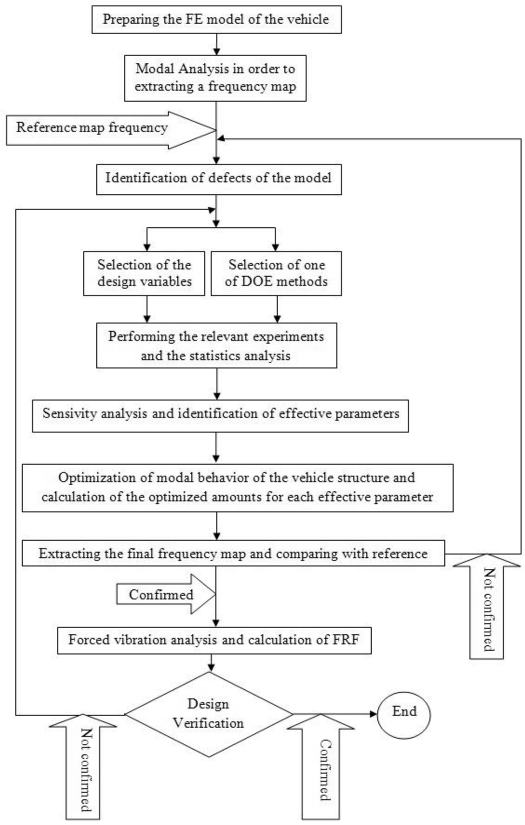

Fig. 13. An algorithm to optimize the car NVH behavior.

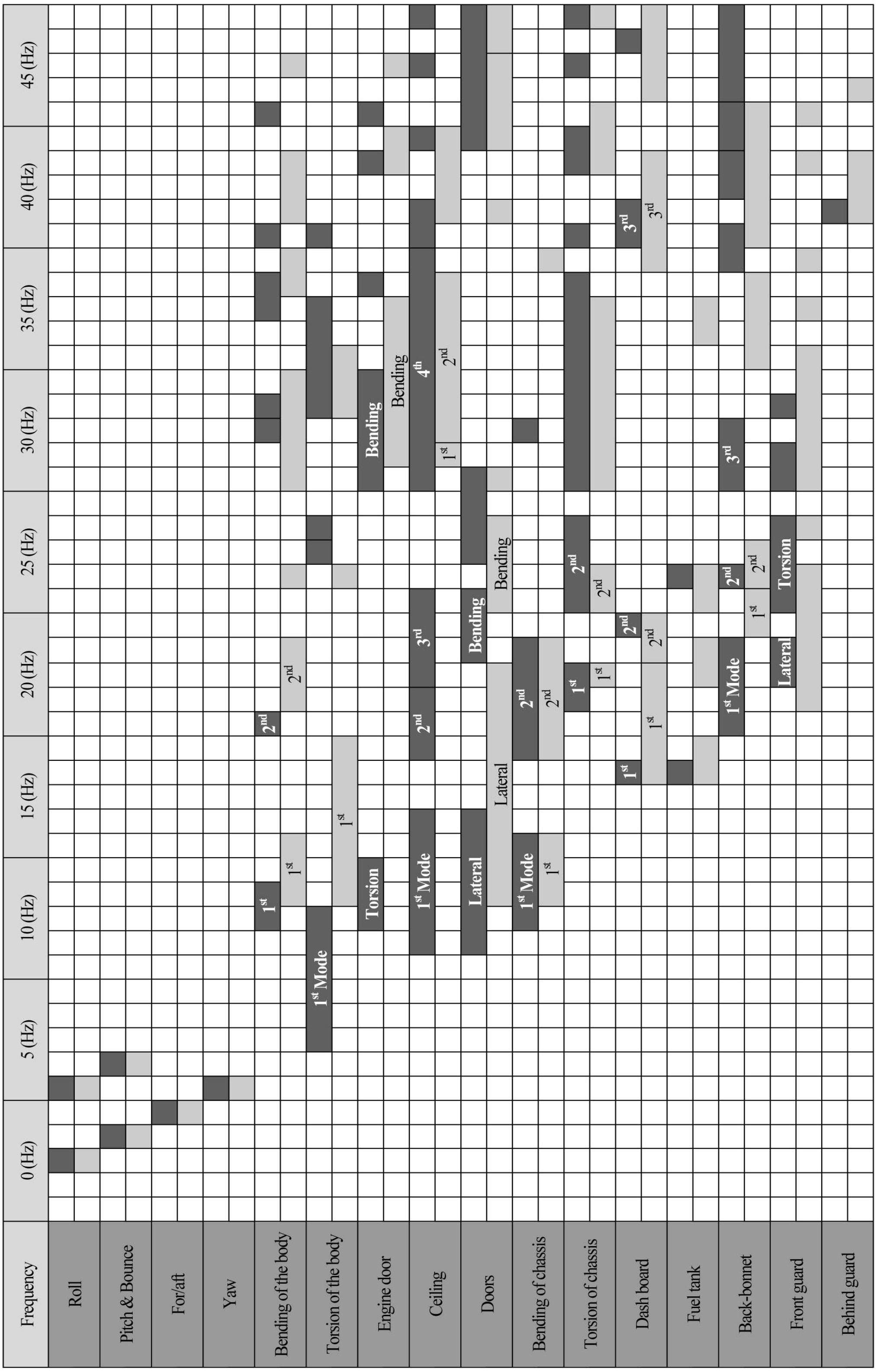

Fig. 14. The frequency map of the FE model, clearer color for before optimization and darker color for after optimization.

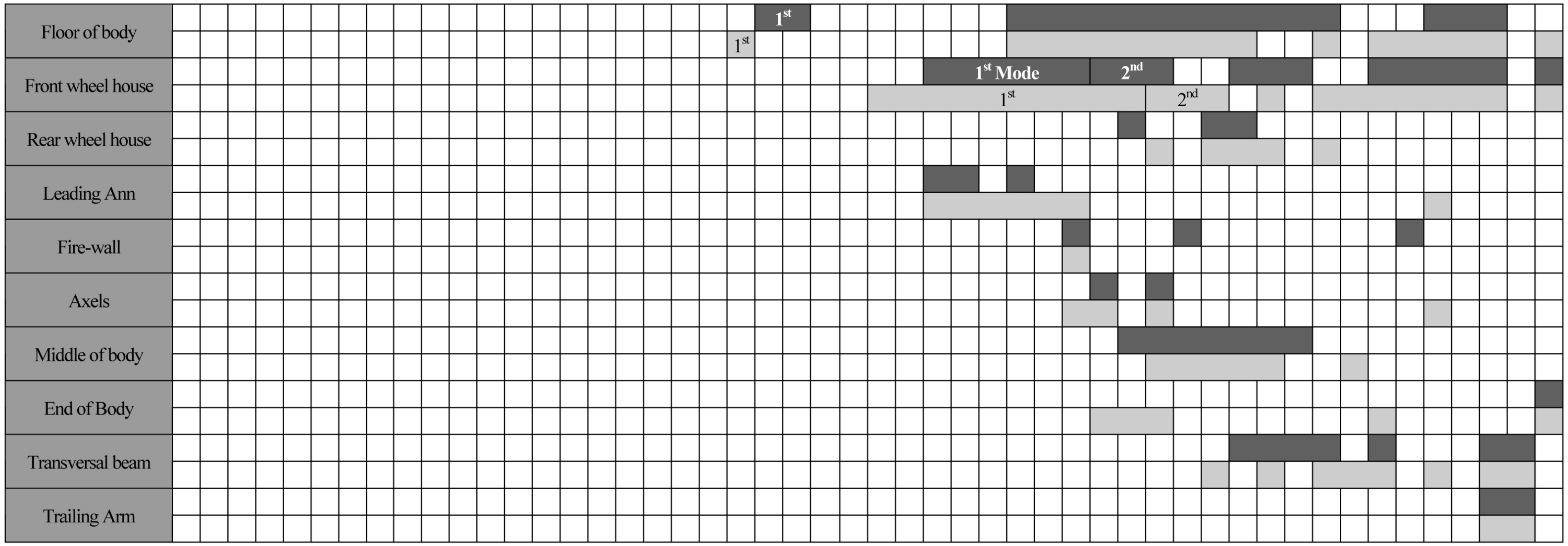

Fig. 14. The frequency map of the FE model, clearer color for before optimization and darker color for after optimization. (Continued)

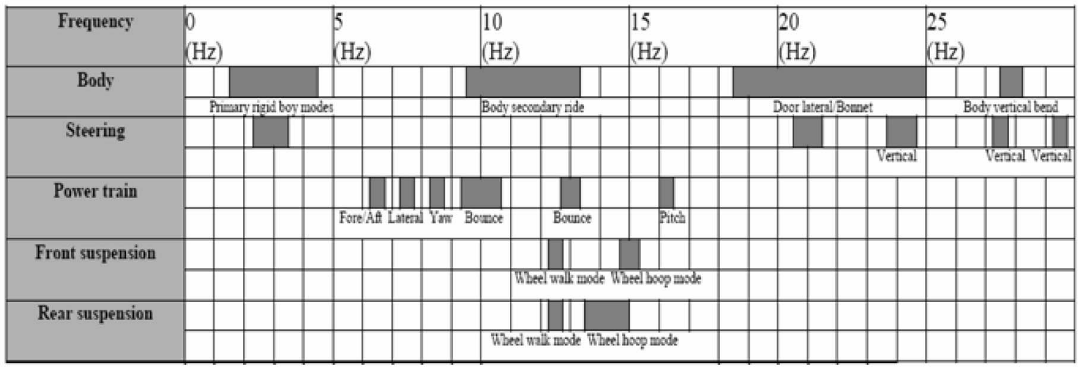

Fig. 15. The frequency map of the experimental modal test.

7. Conclusion

The analysis and improvement of a vehicle body structure based on NVH behavior is investigated by using design of experiments (DOE) method. Firstly, the surface modeling is accomplished for the vehicle body in CATIA and meshed in HYPERMESH software. Then, modal analysis in a frequency range between 0-50 Hz is done by NASTRAN. A frequency map of the vehicle body is extracted and compared with a reference map to identify the defects.

By changing the bonnet lock place, the torsion mode of the engine hood is eliminated and the bending mode frequency is increased. By closing the car structure, the frequency of the torsion and bending modes of the car body are increased. By using factorial method for the ceiling design parameters, the results show that the additional longitudinal beam is more effective among other parameters. Therefore, the optimization is accomplished for this part with adding two transversal beams, two longitudinal beams with thickness of 1.7 mm. By using the response surface method (RSM) for the back-bonnet parameters, the sensitivity analysis shows that the transversal and longitudinal beams are more effective. Thus, the back-bonnet with two longitudinal beams, two transversal beams, with 1.8 mm thickness for bonnet plate and 2.0 mm thickness for the beams is improved.

Finally, an algorithm is proposed to improve the overall car NVH behavior. To verify the present algorithm, forced vibration is performed under real road inputs for the model before and after optimization.

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