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# Prediction and Optimization of Full-Vehicle Road Noise Based on Random Response Analysis

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**Abstract.** Based on the theory of random response analysis, a road noise prediction method is proposed. A finite element model of acoustic-structure coupling for the full-vehicle was established, and the road surface texture measured in the actual vehicle test was converted into a power spectrum, which is the excitation of FE model. Under the conditions of coast down from 80km/h to 20km/h on Belgian pavement, the prediction of road noise in the 20-200Hz frequency band was achieved. The driver's ear sound pressure obtained from the simulation is in good agreement with the test results, which verifies the effectiveness of the simulation model and prediction method. The sensitivity of the driver's ear-side sound pressure level to the stiffness of the suspension mounts was further analysed. The stiffness of the bushings that had a large impact was determined. The Response Surface Methodology was used to optimize road noise. The results show that the average sound pressure level near the driver's ear is reduced by 1.81 dBA. The simulation analysis and optimization methods introduced in this paper can provide a reference for the virtual development process of vehicle road noise.

## 1. Introduction

With the development of automobiles towards electrification and light weight, the problem of road noise in vehicles has become more prominent, which seriously affects the comfort of passengers. In-vehicle road noise is mainly caused by road surface excitation, which is transmitted to the vehicle body through tires and suspension, causing body panel vibration and interior noise response. In the early stages of vehicle development, computer aided engineering (CAE) simulation analysis methods are usually used to find road noise problems in vehicles, and corresponding control measures are taken to avoid road noise risks in vehicles.

For the simulation of road noise in vehicles, scholars and engineers at home and abroad have conducted a lot of research. Zhao Chun [1] input the collected wheel center load into the vehicle finite element model and analysed the main peaks of road noise in the vehicle. Chen Changming [2] used a combination of multi-body system dynamics theory, finite element theory, and boundary element theory to simulate analysed the low-frequency noise in the car due to road surface unevenness excitation. Haste F [3] loaded the rough road excitation in the frequency domain onto the modal tire provided by the supplier, and calcu-



lated the sound pressure near the driver's ear. Bartolozzi [4] established a road model and a tire structure finite element model for vehicle road noise simulation, and calculated the vehicle noise under different road excitations. For interior road noise control, the structure is usually optimized based on a simulation model. Yu Xiongying [5] analysed the contribution of car body noise to the noise of a car. By modifying the installation structure of the front windshield, increasing the installation rigidity, and reducing the transfer function of the noise to the interior noise, the problem of drum sound in the vehicle was solved. Liu Wei [6] optimized the road noise performance of an electric vehicle based on the 3D simulation analysis model of the entire vehicle. The two schemes of changing the stiffness of the rear trailing arm bushing and adding a battery bracket were used together to verify the actual vehicle effect, which reduced the road noise of the electric vehicle. Guellec [7] established a complete vehicle finite element model, and carried out simulation of road noise inside the vehicle under actual road excitation. The road noise was optimized from the two aspects of the floor indentation and the body interior parts such as asphalt and porous sound insulation layer, which found that improving the body interior parts was more effective to improve the road noise in the car.

This paper takes a hatchback as a research object, establishes a sound-structure coupling finite element model of the whole vehicle, and applies the actual road excitation to the tire ground point in the form of a power spectrum to predict the road noise inside the vehicle. The sensitivity of driver's ear pressure to the stiffness of suspension and body connection point bushings was analyzed. With the sensitivity of the bushing with greater sensitivity as the design variable and the sound pressure level near the driver's ear as the objective function, the road surface noise in the vehicle was optimized using response surface methodology.

## 2. Simulation Model of Road Noise Inside the Vehicle

A finite element model of acoustic-structure coupling for the full-vehicle was established in this section. The vehicle road noise simulation model mainly includes six parts including the decorative body, suspension system, tire, exhaust system, power system, and interior acoustic cavity.

The panels on the decorative body are simulated using quadrilateral and triangular elements, the solids are simulated using hexahedral and tetrahedral elements, the welding points are simulated using CWELD elements, the elastic elements are simulated using CBUSH elements, and the damping elements are simulated using CDAMP1 elements. The vehicle body was divided into units with a unit size of 8 mm, and a total of 776,720 shell units and 107,559 solid units were obtained. The interior parts on the decorative body mainly include ceilings, carpets, damping layers, paint, etc., which are simulated using non-structural mass units and evenly distributed on the body panels. The non-structural mass attached to the body is 104.3 kg. The finite element model of the decorative body is shown in Figure 1.



**Figure 1.** Finite element model of decorative body.

Suspension systems include MacPherson front suspension and torsion beam rear suspension. Suspension springs, axles and steering rods are simulated using CBEAM units. For solid and thin-walled structures, a unit size of 8mm is used for unit division. The suspension system includes 26,909 shell units and 111,338 solid units.



a) Finite element model of front suspension.

b) Finite element model of rear suspension.

**Figure 2.** Finite element model of suspension system.

The tire finite element model uses the SAIC-Volkswagen internal tire model [8]. The digital tyre modeling has the following steps. First, the material parameters, modal and mechanical properties of the tire were measured. Based on this, a finite element model of the tire structure is established and analyzed, and the analysis results are compared with the test data and the finite element model is modified. Finally, the DMIG parameters used in NASTRAN calculations are output. The established tire finite element model is shown in Figure 3.

The power system includes an engine and a transmission. For the simulation of road noise in the vehicle, the road surface unevenness excitation is mainly considered, so the power system is simplified. The position of the center of gravity of the power system is measured, and a centralized mass unit CONM2 is simulated at the position to simulate the power system.

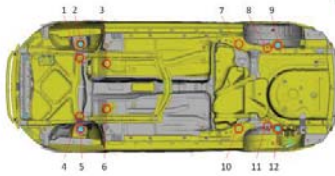
Using the finite element model of the decorative body, a finite element model of the interior acoustic cavity considering the seat and instrument panel is established. In order to ensure the calculation accuracy, at least 6 units are distributed for each wavelength. Since the maximum frequency of the modal analysis of the acoustic cavity is 900 Hz, a 60 mm tetrahedral element is used for division. The finite element model of the interior acoustic cavity is composed of 841,910 units. The finite element model of the interior acoustic cavity is shown in Figure 4.

**Figure 3.** Tire finite element model.**Figure 4.** Finite element model of the acoustic cavity in the car.

The coupling relationship between the finite element model of the decorative body and the finite element model of the interior acoustic cavity was established by the ACMOL card in MSC.NASTRAN, and the acoustic-solid coupling finite element model of the decorative body was obtained.

After the finite element model of each part is established, the connection relationship is established between the suspension and tire, the suspension and the body, the exhaust system and the body, the power system and the body to form a vehicle finite element model of road noise simulation. Regarding the connection between the suspension and the tire, ignoring parts such as hub bearings, a REB2 unit is established between the rim and the drive shaft. There are 12 connection points between the suspension and the body, whose position on the body is shown in Figure 5. The CBUSH unit is used to simulate the connection of the bushings on the real car. The rubber connection between the exhaust system and the vehicle body, the bushing between the power system mount point and the vehicle body are all simulated using

CBUSH units. The parameters of each CBUSH unit are set according to the bush stiffness and damping measured in the test. So far, the vehicle-structure-coupled finite element model for vehicle interior road noise simulation is obtained, as shown in Figure 6.



**Figure 5.** Connection points between body and suspension.



**Figure 6.** Finite element model of sound-structure coupling of the vehicle.

### 3. Prediction and Verification of Road Noise in Vehicles

#### 3.1. Random Response Analysis Theory

When a random system is linear and excited by a stationary process, the frequency response analysis method can be used for the analysis of the random process.

According to transfer function theory, if  $H_{ja}(\omega)$  is the frequency response function of  $u_j$  under excitation  $Q_a$ , then the Fourier transform  $Q_a(\omega)$ ,  $u_j(\omega)$  of the excitation and response can be expressed as

$$u_j(\omega) = H_{ja}(\omega) \cdot Q_a(\omega) \quad (1)$$

Therefore, the power spectral density of the response can be expressed as

$$S_j(\omega) = |H_{ja}(\omega)|^2 \cdot S_a(\omega) \quad (2)$$

In the formula,  $S_a(\omega)$  is the power spectral density of the excitation source.

According to the formula (1), the statistically significant response of the system under random excitation can be estimated by the frequency response analysis method. If the excitation source  $Q_1, Q_2, Q_3 \dots$  is statistically independent, that is, the cross-correlation function between two pairs

$$R_{ab}(\tau) = \lim_{T \rightarrow \infty} \frac{1}{T} \int_0^T q_a(\tau) q_b(t - \tau) dt = 0 \quad (3)$$

Then the power spectral density of the total response is equal to the sum of the power spectral density of the responses under each excitation, that is,

$$S_j(\omega) = \sum_a S_{ja}(\omega) = \sum_a |H_{ja}(\omega)|^2 \cdot S_a(\omega) \quad (4)$$

If the excitation sources are related, and the mutual power spectral density  $S_{ab}$  is used to indicate the degree of correlation between the excitation sources, the power spectral density of the response can be expressed as

$$S_j = \sum_a \sum_b H_{ja} H_{jb}^* S_{ab} \quad (5)$$

$H_{jb}^*$  is the complex conjugate of  $H_{jb}$ .

#### 3.2. Simulation Analysis Process of Road Noise in the Car

In-vehicle road noise is mainly structural noise caused by road surface excitation. Its mechanism is shown in Figure 7. The unevenness of the road surface is excited on the wheels, and the vibrations on the wheels are transmitted to the vehicle body through the suspension and sub-frames, causing the vibration of the vehicle body structure and the response of the sound field in the vehicle. Based on this, the simulation calculation process of the road noise in the vehicle can be obtained, as shown in Figure 8. Based on the vehicle acoustic-solid coupling finite element model, the MSC.NASTRAN modal frequency response analysis module is used to calculate the transfer function of the wheel ground point displacement to the response point sound pressure or acceleration. The road roughness power spectrum is used as the input of random response analysis to calculate the sound pressure response near the ears of the driver and occupants.

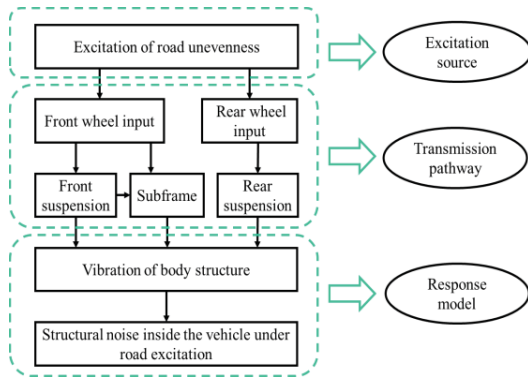


Figure 7. Mechanism of road noise.

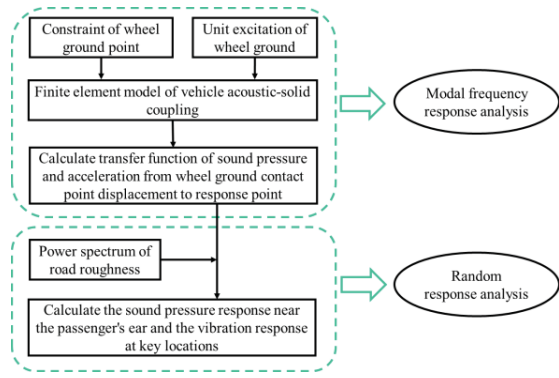


Figure 8. Simulation calculation process of road noise.

The road noise inside the car is mainly low frequency noise below 300Hz, but considering the limitation of the tire finite element model, the frequency range of the modal frequency response analysis is set to 20-150Hz with a step size of 1Hz. In order to ensure the calculation accuracy of the model, the termination frequency of the modal calculation should usually be greater than 1.5 times the frequency response analysis frequency. Therefore, the modal analysis frequency of the vehicle finite element model is set to 0-300Hz, and the modal analysis frequency range of the vehicle interior acoustic cavity finite element model is set to 0-900Hz.

The pavement used in the simulation is derived from a Belgian pavement, with a length of 500m and a width of 4m, as shown in Figure 9. Scan the actual road surface to obtain its unevenness data, and use the Fourier transform to obtain the power spectrum of the road surface unevenness [9], as shown in Figure 10.



Figure 9. Belgian pavement.

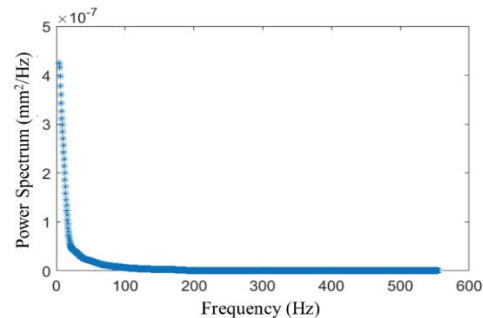
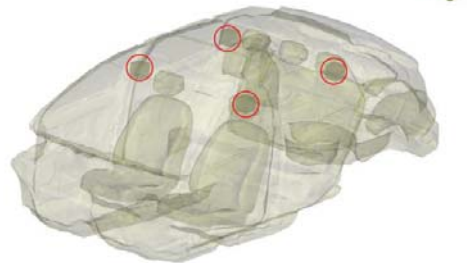


Figure 10. Power spectrum of unevenness of Belgian pavement.

The simulation conditions are consistent with the test. During the test, the vehicle accelerated to 80km/h and then coasted to 20km/h in neutral. In the simulation, the vehicle finite element model is stationary, and the relative motion with the road surface is mainly reflected in the interaction between the power spectrum of the road surface roughness and the tire ground point. Therefore, the simulation conditions are mainly realized through the power spectrum of road surface roughness. The road noise simulation in the car calculates the sound pressure near the ears of the driver and occupants. The position of the measurement point is consistent with the actual vehicle test. It is determined according to the ISO 5128-80

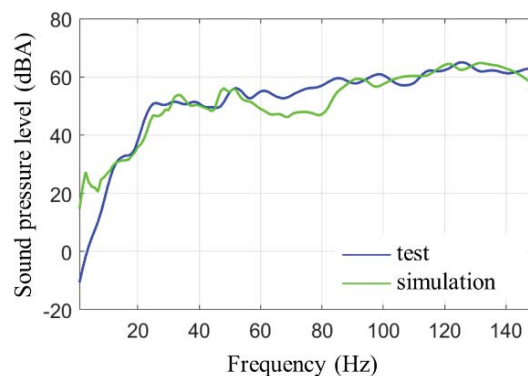
interior noise measurement standard, and it is placed near the ears of the driver and occupants. The location of the sound pressure measurement point in the simulation model is shown in Figure 11.



**Figure 11.** Sound pressure measurement points near the ears of the driver and occupants.

### 3.3. Simulation Analysis and Verification of Road Noise in Vehicles

The comparison between the simulation results of the driver's ear pressure and the results measured in the real vehicle test is shown in Figure 12. The driver's ear pressure trend obtained from the simulation and the test is basically the same, and the corresponding road noise peaks appear in the simulation results, indicating that the simulation model can better predict the road noise in the vehicle.



**Figure 12.** Simulation and test comparison of road noise in the car.

## 4. Bushing Stiffness Sensitivity Analysis and Optimization

### 4.1. Suspension Bushing Stiffness Sensitivity Analysis

Based on the prediction model of road noise in the vehicle, the sensitivity analysis of the bushing was carried out. Because the road surface excitation is transmitted to the vehicle body through the suspension, the stiffness of the bushing at the connection point between the suspension and the body plays a vital role in the transmission of vibration, the noise sensitivity analysis of which should be performed. The stiffness in the six directions of each bushing was changed from 50% to 150% of the initial value, and the driver's ear pressure level after adjustment was calculated. The comparison results are shown in Tables 1 and 2. The unit of sound pressure level in the table is dBA, and greater than 0 means that the road noise performance in the car is worse when the stiffness becomes larger.

**Table 1.** Sensitivity analysis of front suspension bushing stiffness to driver's ear pressure level.

Position	Changes in average sound pressure level	Changes in peak sound pressure level
X direction of the upper end of the shock absorber	-0.0990	0.0042
Y direction of the upper end of the shock absorber	0.5672	0.1981
Z direction of the upper end of the shock absorber	0.0659	-0.0190
X direction of sub frame connection point	0.2149	0.0122
Y direction of sub frame connection point	0.2184	0.1386
Z direction of sub frame connection point	0.0149	0.0081

**Table 2.** Sensitivity analysis of the stiffness of the rear suspension bushing to the driver's ear pressure level.

Position	Changes in average sound pressure level	Changes in peak sound pressure level
X direction of torsion beam	0.4875	0.1201
Y direction of torsion beam	0.0400	-0.1657
Z direction of torsion beam	-0.2660	-0.1999
X direction of spring upper end	0.0003	-0.0003
Y direction of spring upper end	-0.0001	0.0000
Z direction of spring upper end	-0.0002	0.0000
X direction of the upper end of the shock absorber	0.0038	0.0011
Y direction of the upper end of the shock absorber	0.0002	-0.0094
Z direction of the upper end of the shock absorber	0.0020	-0.0122

As can be seen from Table 1, the stiffness of the bushing in the upper end of the shock absorber in the front suspension in the Y direction, and the stiffness in the X and Y directions of the bushing at the junction point of the sub-frame and the vehicle body have more influence on the average sound pressure level near the driver's ear. Increasing the stiffness at these locations will increase the sound pressure level near the driver's ear. As can be seen from Table 2, the stiffness of the torsion beams in the X and Z directions of the rear suspension has a greater effect on the average sound pressure level near the driver's ear. Increasing the stiffness in the X direction will increase the sound pressure level near the ear of the driver, while decreasing the stiffness in the Z direction will increase the sound pressure level near the ear of the driver.

#### 4.2. Optimization of Bushing Stiffness

Based on the analysis results of the sensitivity of the bushing stiffness to the road noise inside the vehicle, the optimization of the stiffness of the bushing was performed to reduce the road noise inside the vehicle. In order to obtain the optimal bushing stiffness, the response surface method is used for experimental design, model establishment, model verification and model optimization. The application of response surface method can not only get the change relationship between the response target and the design variables, but also the optimization plan, that is, the optimal combination of design variables, so that the objective function can be optimized. The design variables are the highly sensitive bushing stiffness, that is, the Y-direction stiffness of the bushing at the upper end of the shock absorber in the front suspension, the X- and Y-direction stiffness of the bushing at the junction point of the subframe and the body, and the X and Z-



direction stiffness of torsion beam in rear suspension. The objective function is the average sound pressure level near the driver's ear.

**Table 3.** Optimized bushing stiffness (N / mm).

Position	X direction	Y direction	Z direction
Upper end of shock absorber	2586.21	2500	424.96
Connection point of subframe and body	6804.37	5340.89	6480
Torsion beam	2159.07	1000	2776.27
Upper end of spring	100000	100000	100000
Upper end of shock absorber	1710	1710	900

The optimized bushing stiffness is shown in Table 3. Under optimal bushing stiffness, the sound pressure level near the ears of the front-line driver was reduced by 0.81 dBA, and the sound pressure level near the ears of the right-hand rear occupant was reduced by 0.61 dBA.

## 5. Conclusion

In this paper, we did the following work and got the following conclusions.

(1) A finite element model of sound-structure coupling for the entire vehicle is established, and a road noise prediction method is established based on the theory of random response analysis. The simulation results and experimental results have good consistency, which verifies the validity of the model.

(2) Through the analysis of the sensitivity of the stiffness of the bushing, the position of the bushing which has a great influence on the road noise in the vehicle is obtained, the design variables for the optimization of the road noise in the vehicle are reduced, and the optimization direction is determined.

(3) Based on the response surface method, the optimal bushing stiffness was obtained, which reduced the average sound pressure level near the driver's ear by 0.81dBA.

## References

- [1] Zhao C, Xing Y, Wang H 2018 Method to improve vehicle noise excited with some road *Automotive Electrical Appliances* **2018**(07) 30-3
- [2] Chen C, Sun W 2008 Numerical simulation and analysis of cab interior noise with the consideration of road surface roughness *Journal of Vibration and Shock* **2008**(08) 102-105+180
- [3] Haste F et al 2004 A Comprehensive Vehicle NVH Performance Correlation Study *Proceedings of IMAC-2004*
- [4] Bartolozzi G, Danti M, Camia A, and Vige D 2016 Enhancement of full-vehicle road noise simulation including detailed road surface and innovative tire modeling *SAE Int J Passeng Cars - Mech Syst* **9**(3) 1091-9
- [5] Yu X, Liu B, Zhang J 2015 Application and research on control of auto booming noise due to road exciting *2015 China Automotive Engineering Society Annual Meeting*
- [6] Liu W, Han P, Yang S 2016 Road Noise Optimization of an Electric Vehicle Model *Auto Engineer* **2016**(11) 46-49+51
- [7] Guellec A, Cabrol M, Jacqmot J, and Van den Nieuwenhof B 2018 Optimization of trim component and reduction of the road noise transmission based on finite element methods *SAE Technical Paper* 2018-01-1547
- [8] Chen J, Yang Y, Liu X 2018 Road noise and TPA analysis by simulation *2018 China Automotive Engineering Society Annual Meeting*
- [9] Kido I, Ueyama S, Hashioka M, Yamamoto S et al 2011 Tire and Road Input Modeling for Low-Frequency Road Noise Prediction *SAE Int J Passeng Cars – Mech Syst* **4**(2) 1277-82