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Prediction and Control Cabin Noise of High Speed Train

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Abstract:

The noise problem generated by the rail transportation is attracting more and more attention. To address the noise issue in the VIP cabin of a high-speed train, the statistical energy analysis method is employed for predicting and diagnosing noise levels. The prediction model indicates that the vibration of the inner floor is the primary source of interior noise. The transfer path analysis reveals that structure-borne noise through connections between the inner floor and outer floor needs to be addressed. Experimental results confirm that dominant frequencies of noise range from 100 to 300Hz. Distributed vibration absorbers with targeted frequencies are proposed to reduce interior noise. The distributed vibration absorbers is applied to the inner floor of the train, and the cabin noise level is tested at a speed of 350km/hr. The noise in the VIP cabin is reduced from 75.1dBA to 71.3dBA, achieving the target of noise reduction.

Key words: noise, vibration, high speed train, statistical energy analysis, sound transmission loss, distributed vibration absorber.

1. Introduction

Rail transportation has become the most popular way of traveling in China and is gaining increasing attention regarding passenger comfort. With the rise in operational speeds, the issue of noise within the train cabin becomes more prominent. Due to the complexity of high-speed train structures and the diversity of transmission paths, such as track noise being transmitted to the interior through a double-layer floor, there are four energy transfer paths within the floor structure alone [1]: Double-wall transmission path, Non-resonant zone transmission path, Resonance transmission path, Structural transmission path. Determining all these paths through experiments poses a significant challenge. Utilizing statistical energy analysis (SEA) models enables the rapid identification of energy contributions from different transmission paths. Based on the results, optimizing the vehicle structure significantly enhances the efficiency of overall noise optimization efforts.

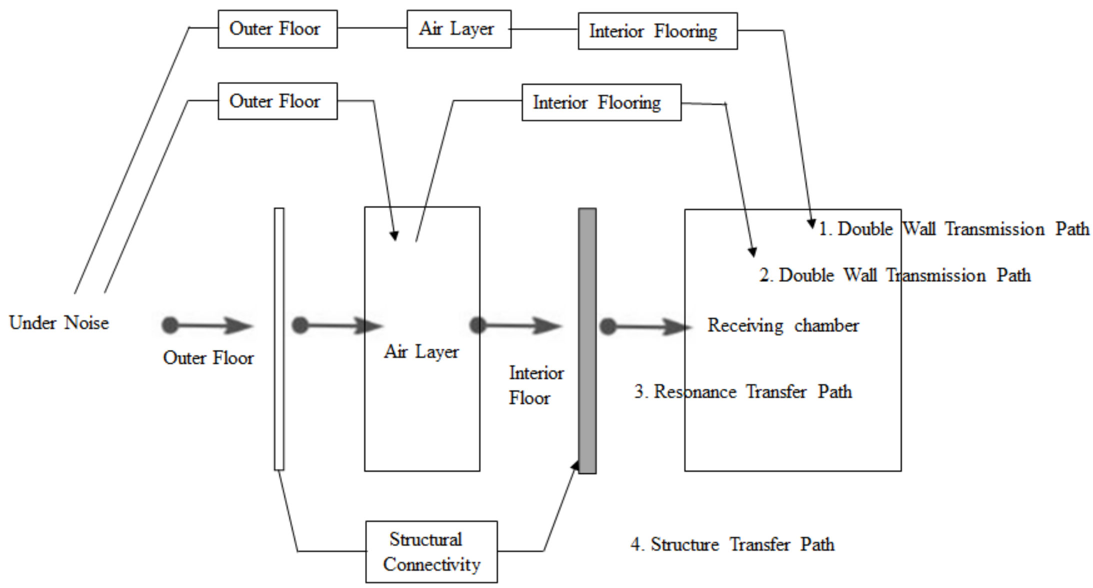

Figure 1 The Energy Transfer Path of Double-Layer Flooring

The operational noise level in the VIP cabin area of the high-speed train, as designed by the customer, exceeded the target at 75 dBA. To address this issue, a comprehensive analysis of noise sources and acoustic design became imperative for interior noise reduction. This study employed the statistical energy method to create an acoustic model of the VIP cabin area, validating it through experiments. Based on this model, noise transmission paths were analyzed, leading to the design of vibration reduction and noise reduction schemes targeting the primary noise sources. Ultimately, the implemented measures successfully lowered the cabin noise to 71.3 dBA, achieving the specified noise reduction goal for the compartment.

1.1 Modelling Cabin Noise

The statistical energy analysis method serves as a potent tool for tackling the high-frequency dynamics of intricate systems by characterizing the system's state through the dynamic energy of its subsystems. External inputs to the model manifest as energy flows within the system, and the power flow between subsystems adheres to specific principles: energy moves from high modal energy substructures to low modal energy substructures [2].

Figure 2 shown the energy transfer between two subsystems. By constructing energy balance equations and analyzing each subsystem separately, referred to as subsystem 1 and subsystem 2, respectively:

$$P_1 = \omega\eta_1 E_1 + \omega\eta_{12} n_1 \left[\frac{E_1}{n_1} - \frac{E_2}{n_2}\right] \tag{1}$$

$$P_2 = \omega\eta_2 E_2 + \omega\eta_{21} n_2 \left[\frac{E_2}{n_2} - \frac{E_1}{n_1}\right] \tag{2}$$

In the equation 1 and 2, P_i represents the input energy, ω is the center frequency of the analyzed frequency band, η_i is the damping loss factor, η_{ij} is the coupling loss factor, n_i is the modal density, and E_i represents the energy of the subsystem.

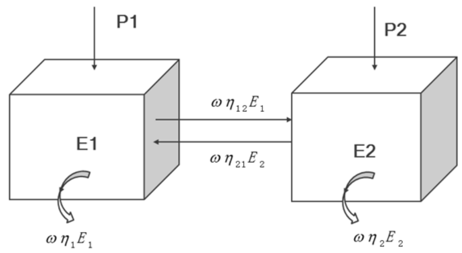

Figure 2 Energy Flow Transfer between Two Subsystems

For multiple subsystems, an energy transfer matrix can be established:

$$\omega[A]\begin{bmatrix} E_1/n \\ \cdots \\ \cdots \\ E_k/n_k \end{bmatrix} = \begin{bmatrix} P_1 \\ \cdots \\ \cdots \\ P_k \end{bmatrix} \tag{3}$$

The [A] here is the damping matrix, which can be expressed as:

$$[A] = \begin{bmatrix} (\eta_1 + \sum_{i=1} \eta_{1i})n_1 & -\eta_{12}n_1 & \cdots & -\eta_{1k}n_1 \\ -\eta_{21}n_2 & (\eta_2 + \sum_{i=2} \eta_{2i})n_2 & \cdots & \cdots \\ \cdots & \cdots & \cdots & \cdots \\ -\eta_{k_1}n_k & \cdots & \cdots & (\eta_k + \sum_{i=k} \eta_{ki})n_k \end{bmatrix} \tag{4}$$

In the analysis of high-speed train noise, the focus is on the acoustic cavity subsystem within the cabin. Solving the energy balance equation enables the identification of energy sources and losses within the studied subsystem.

The application of the statistical energy method to address high-speed train noise issues is shown in Figure 3. Initially, key subsystems are created and analyzed, followed by experimental validation. Then, a comprehensive vehicle model is established, validated through experiments, and employed for problem diagnosis and the design of optimization solutions. Finally, real-world validation of the proposed solutions is conducted to achieve the design goals.

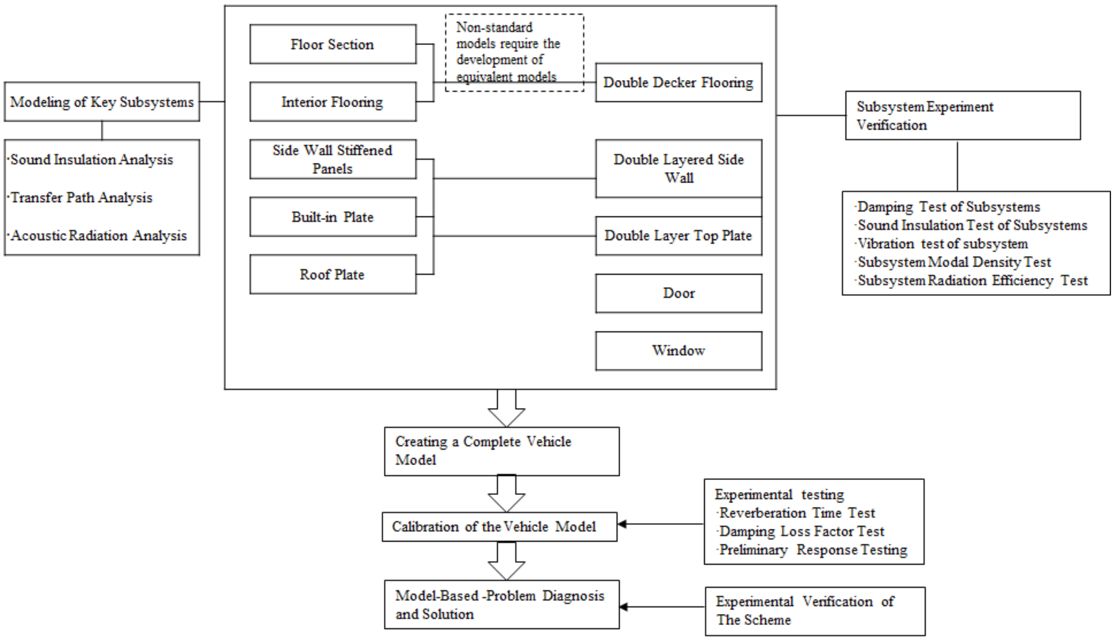

Figure 3 The Statistical Energy Method is Applied to Solve the Noise Analysis Process of High-Speed Railway

2. Model-based Transfer Path Analysis

When a high-speed train is operating at elevated speeds, the interaction between the wheels and rails gives rise to structural noise. This noise, on one hand, is transmitted to the floor through the bogie, causing floor vibrations and radiating noise into the cabin. On the other hand, air-borne noise generated by the wheels and rails is transmitted to the interior through components like the double-layered floor, sidewalls, and windows. Additionally, the high-speed interaction between the vehicle and the air produces turbulent boundary layer noise, and this energy is transmitted to the interior through structures such as sidewalls, roof panels, windows, doors, etc.

Before constructing the complete vehicle model, it is essential to perform acoustic analysis on key subsystems, including the floor, sidewalls, roof, windows, doors, windshields, etc. The subsystem models are precisely defined through experiments before the creation of the comprehensive vehicle model. Due to space constraints, the modeling and experimental verification of subsystems are not elaborated further here.

To model the VIP area, the statistical energy model in Figure 4 is established. The comprehensive model comprises 68 structural subsystems and 29 acoustic cavity subsystems. The floor, sidewalls, and roof are modeled as double-layer structures, closely resembling actual structures. Manual connections are employed between the inner and outer structures to simulate the structural transmission paths.

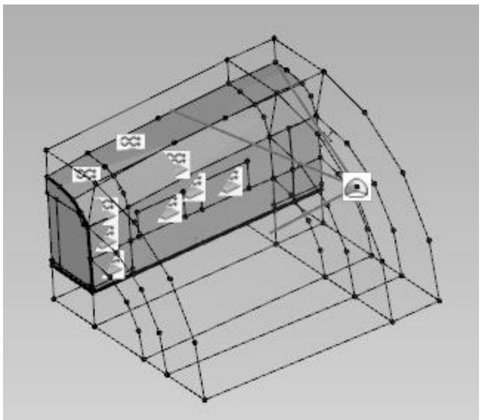

Figure 4 Vehicle Statistical Energy Model

The sound pressure level within the VIP cabin is determined collaboratively by input energy and loss energy. When energy balance is achieved, the input energy equals the loss energy [3]. At this point,

$$P_{in} = \eta\omega \frac{p^2}{\rho_0 c^2} V \quad (5)$$

In equation 5, η is the total loss factor, p=sound pressure, ρ_0=air density, c=velocity of sound, V= the volume of the VIP class sound chamber.

Therefore, once the input energy and loss energy are determined, the sound pressure response p can be calculated. Typically, the average absorption coefficient is utilized to simulate sound absorption in the cavity. The average absorption coefficient can be obtained through the reverberation time test method [4]. The experimental setup for the average absorption coefficient test is shown in Figure 5.

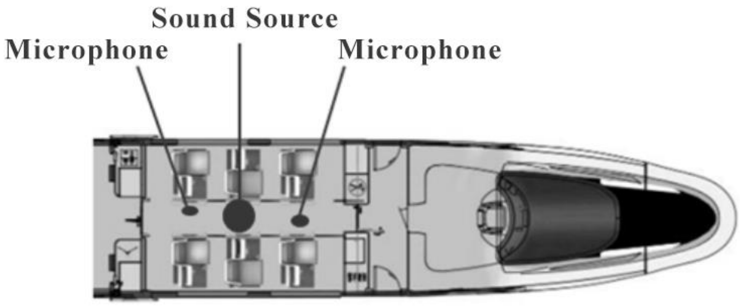

Figure 5 VIP Cass Reverberation Time Test

To validate the accuracy of the model, a spherical sound source was employed for excitation under the train. The vibration velocities of the inner and outer floors were compared under the excitation, and the responses from the model aligned with the experimental results. This validation process confirms the accuracy of the model.

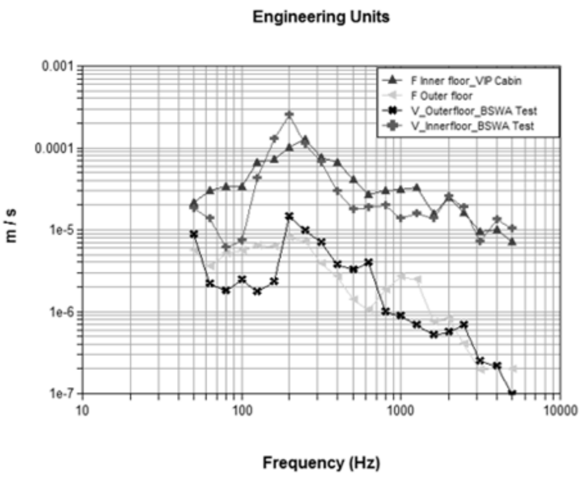

Figure 6 Comparison of Vibration Response between Nnner and Outer Floor Tests and Models under Ball Sound Source Excitation

The noise sources in the actual model encompass wheel-rail structure excitation, wheel-rail noise, and turbulent airflow noise on the surface of the vehicle. The excitation transmitted from the bogie to the vehicle structure can be determined by impedance and the vibration velocity at the attachment points [5]. External acoustic cavities are connected to a semi-infinite free field to simulate the propagation of noise to the far field. Analyzing the model provides the noise response inside the vehicle and contribution paths, as shown in Figure 7.

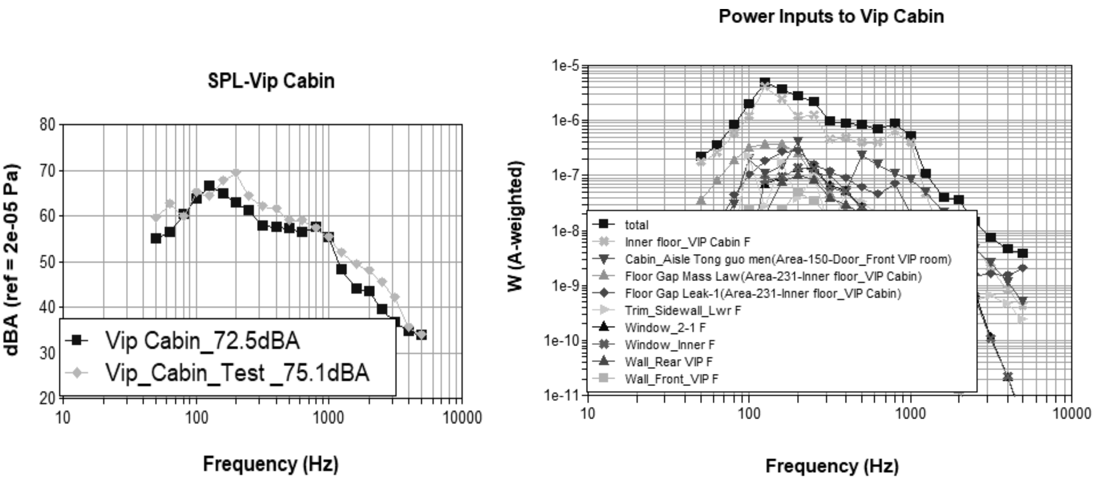

Figure 7 Cabin noise Response (Left) and Energy Contribution (Right) of Statistical Energy Model

The noise trends between the model and experimental tests are generally consistent, with errors within 3 dB. The model indicates that the main source of noise inside the vehicle is the vibration radiation from the interior floor (Inner floor_VIP Cabin F). Further analysis of the interior floor reveals that the noise primarily originates from the connection between the exterior floor and the interior floor.

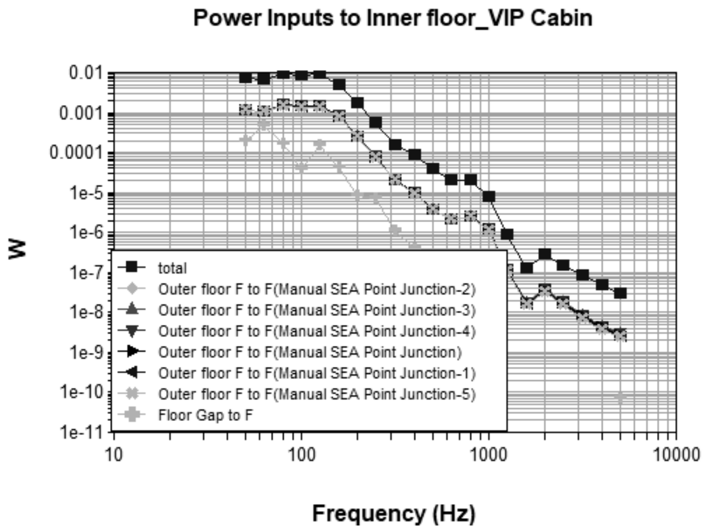

Figure 8 Energy Contribution of Interior Flooring

3. Noise Reduction Solution and Validation

Based on the analysis results from the model, coupled with the examination of the noise spectrum in Figure 7, it is observed that the noise is mainly concentrated in the frequency range of 100-300 Hz, with measured noise peak points at 100 Hz, 160 Hz, and 200 Hz. As the noise is primarily attributed to the vibration radiation of the interior floor, it is imperative to decrease its vibration velocity. This Chapter introduce the specific measures encompass reducing the noise source, modifying the transmission path, and directly applying damping and absorption treatments to mitigate structural vibration.

In the development of high-speed trains, making alterations to existing structures can be challenging. Three noise reduction solutions are proposed based on the difficulty of implementation, with detailed information and model verification results provided in Table 1.

Table 1 Noise Reduction Scheme and its Effect

Solution	Noise Reduction
Add distributed absorbers at frequencies of 100, 160, and 200Hz, with a mass of 15% - 20% of the total mass of the interior floor.	Interior noise is reduced by 1.2dB
Add Low-Frequency Sound-Absorbing Panels to the Bogie Area	Interior noise is reduced by 1.5dB
Change the Connection Structure to Reduce the Floor Connection Stiffness	Interior noise is reduced by 2.7dB

To validate the effectiveness of the proposed solutions, experiments were conducted both in a laboratory setting and on an actual train. Initially, the effectiveness of the distributed absorber was verified in a Acoustics laboratory. Considering the double-layer structure of the floor, distributed absorbers were developed with characteristics such as small volume and easy installation. Absorbers corresponding to peak frequencies of 100 Hz, 150 Hz, and 200 Hz were designed. During actual installation, the total mass of the absorbers was set at 15%-20% of the wooden floor, with the masses of the absorbers at the three frequencies distributed in a 1:1:1 ratio.

Figure 9 shows the results of the laboratory experiment validating the effectiveness of the distributed absorbers. The experiments indicated that distributed absorbers could increase the sound insulation of a single-layer interior floor by approximately 10 dB in the 100-200 Hz frequency range and enhance the sound insulation of the double-layer floor by 3 dB. The effective frequencies corresponded to the peak frequencies of noise inside the vehicle, demonstrating the significant effectiveness of the distributed absorbers.

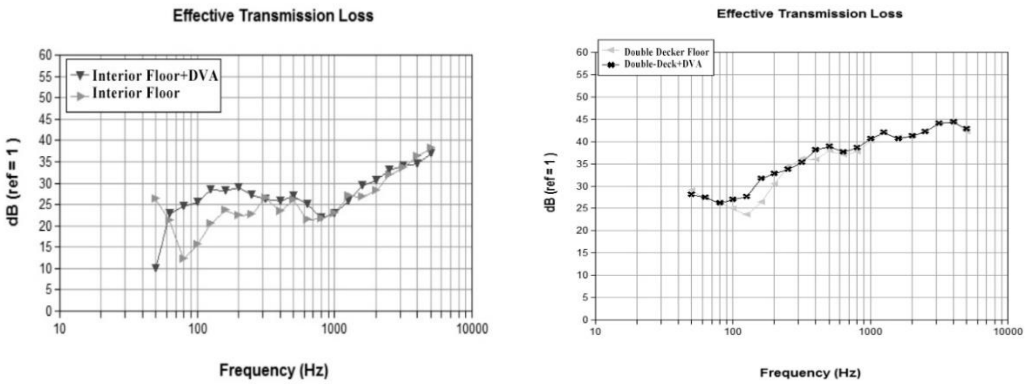

Figure 9 Experiment and Effect of Distributed Vibration Absorber

Before optimization, there was no sound-absorbing material beneath the bogie, and the sound absorption coefficient under the train was close to 1% [4]. Adding acoustic panels in the bogie area can effectively increase the sound absorption coefficient beneath the floor, thereby reducing the energy of the sound source on the floor structure. The designed acoustic panel consists of 40mm sound-absorbing foam and aluminum fiber structure. The left side of Figure 10 shows the absorption coefficient of the acoustic panel, while the right side shows the change in the average absorption coefficient under the train after adding the acoustic panel.

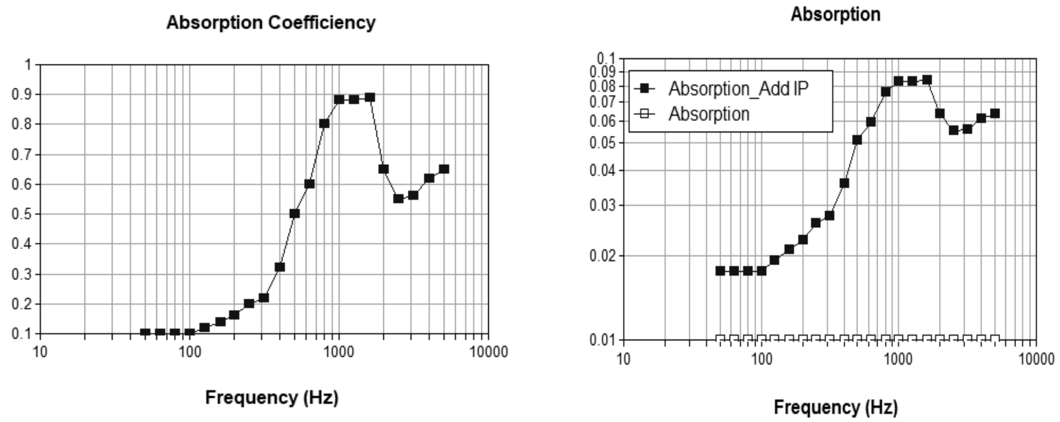

Figure 10 Sound Absorption Coefficient of the Sound-Absorbing Board (Left Figure) after Increasing the Sound-Absorbing Board, the Average Sound Absorption Coefficient under the Train (Right Figure)

Before optimization, there was a rigid connection between the inner and outer floors, allowing energy to easily transfer from the outer floor to the inner floor through the rigid connection points. To address this, modifying the connection between the inner and outer floors can reduce the energy transmitted from the outer floor to the inner floor, thereby decreasing the vibration velocity of the inner floor. The rigid connection structure was modified to the anti-pull structure shown on the right side of the figure. This structure remains inactive when subjected to downward displacement but only engages when there is a significant separation between the inner and outer floors, limiting their movement. This design allows the vibration damping pad between the floors to effectively reduce vibrations.

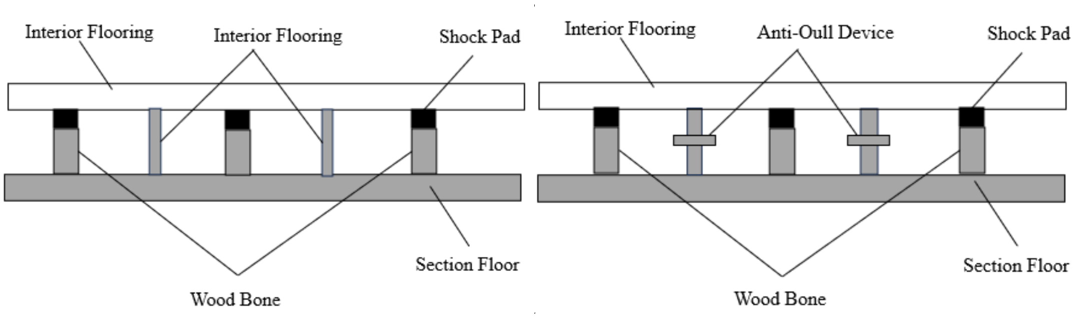

Figure 11 Connections between Floors (Left for the Original Structure, Right for the Optimized Structure)

The three aforementioned solutions above were implemented on the actual train, and noise tests inside the high-speed train were conducted under constant speed conditions at 350 km/h. The test results are shown in the figure below.

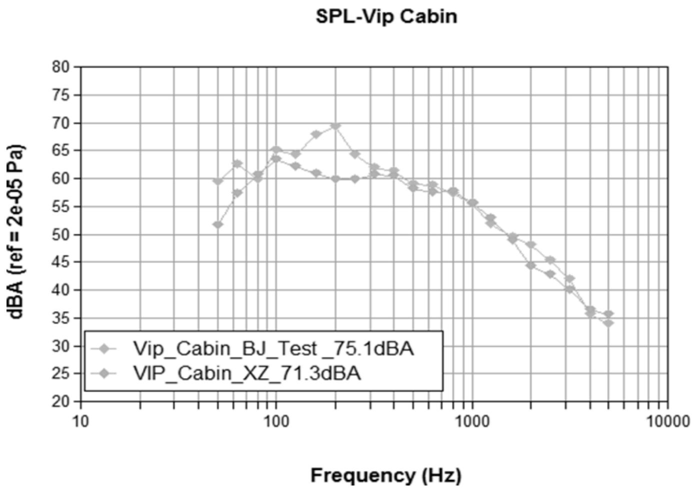

Figure 12 The Comparison of Internal Noise before and after Optimization

A comparison shows a substantial reduction in the overall sound pressure level of interior noise, decreasing from the previous 75.1 dBA to 71.3 dBA. Simultaneously, there is a noticeable decrease in noise within the primary contributing frequency range of 100-300 Hz. This indicates that the primary transfer path analysis and optimization design based on the statistical energy model are highly practical and effective means of noise design.

4. Conclusion

This paper addresses the issue of excessive noise in the high-speed train's VIP cabin using a statistical energy model. Through the analysis of energy paths of noise within the VIP cabin of high-speed trains, it was determined that the primary source of noise is the vibration of the interior floor. Subsequently, three optimization solutions were proposed for the main noise paths, targeting noise sources, transmission paths, and receiving points. As a result, the interior noise level decreased from 75.1 dBA to 71.3 dBA. From the diagnosis and analysis of the problem to the proposal and validation of the solutions, this project provides a comprehensive approach based on the statistical energy model for addressing real vibration noise issues. It serves as a guide and process for engineers to solve practical problems through simulation methods in future projects.

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