

Article

Vibration Performance and Vibration Reduction Optimization of Diesel Generator Sets for Diesel Locomotives

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Abstract

The power package of high-speed internal combustion units generates complex excitation forces during operation. These forces cause excessive vibration in the driver's cab, negatively affecting the driver's working conditions. Therefore, optimizing the isolation system design is essential. This study established a rigid body dynamic model of a single-layer isolation system to determine initial stiffness parameters. A refined finite element (FE) model of the diesel locomotive body was also developed to analyze vibration characteristics. Using the global response surface method, multi-objective optimization was performed. The three-dimensional stiffness of the isolator served as the design variable, while the maximum force transmissibility at the vehicle reference point was the optimization objective. The optimization accuracy was verified through FE modeling, experiments, and simulations considering coupled wheel–rail excitations. Results showed that the force transmissibility at the cab's center floor decreased from 44.05% to 32.65%. Furthermore, the comfort index met the requirements under all working conditions. These findings indicate that the proposed design method effectively improves the efficiency of the power pack isolation system and provides a valuable reference for future designs.

Keywords: diesel locomotive; vibration reduction; rubber isolator; multi-objective optimization; finite element analysis

1. Introduction

Power packs are one of the most common sources of excitation for diesel power units. Due to constraints in the structural design of some rolling stock, the diesel generator set is mounted above the chassis of the locomotive head car [1]. As speed increases, the operation of the power pack generates significant excitation with complex frequencies, leading to severe vibrations in the driver's cab. These intense vibrations not only cause fatigue and structural damage to the vehicle [2] but also result in adverse effects [3–5] on the operating crew such as fatigue, dizziness, and nausea, posing serious safety risks. Therefore, it is necessary to optimize the design of the power pack isolation system.

In the field of rail transit, vibration control and energy management are critical. Jing et al. [6] conducted a comprehensive study using field data and finite element modeling to attribute frame fatigue failure to modal resonance, subsequently proposing an optimized design that meets reliability standards through experimental verification. Furthermore, Dumitriu [7] evaluated how suspended equipment affects ride comfort and car body flexible



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vibrations, utilizing a comparative numerical analysis across models with varying equipment configurations. Minasyan and Minasyan [8] investigated the vibration characteristics of rotating machinery and proposed advanced methods for harvesting and sensing vibration energy, which provides a new perspective for understanding the vibration spectrum of diesel generator sets.

In the field of vibration control for internal combustion power pack power systems, the current research is primarily concentrated on three aspects: multi-level system modeling, active–passive hybrid vibration isolation control, and shaft torsional vibration optimization.

Given the complex coupling characteristics of power pack systems, establishing precise simulation models is a prerequisite for vibration analysis. Boychuk et al. [9] proposed a multi-level methodology to simulate ship electric power systems. By introducing control models to interconnect various subsystems, this approach allows for a more accurate investigation of the overall dynamic response and interdependencies of the power system, providing a theoretical framework for subsequent vibration management. To overcome the low-frequency performance limitations of traditional passive isolation, active and semi-active control technologies have become a research hotspot. Li et al. [10] designed an intelligent vibration reduction system for marine mechanical equipment that integrates active–passive hybrid isolators. Experiments demonstrated that the system achieves an active attenuation of up to 20 dB in the low-frequency range (below 200 Hz). The comprehensive vibration reduction effect is significant, verifying the effectiveness of the hybrid control strategy for diesel generator sets. Furthermore, Yang et al. [11] conducted a real-ship active vibration isolation retrofitting experiment on a diesel generator set in a small vessel. They adopted an inertial actuator system based on a decentralized adaptive feedforward control strategy, utilizing engine speed as the reference signal. In addition to suppressing transmitted vibration via isolators, controlling torsional vibration at the source through structural parameter optimization is another effective approach. Chen et al. [12] proposed a Partial Eigenvalue Assignment technique based on the gradient flow method for complex marine propulsion shafting systems. This method shifts the system's natural frequencies out of the excitation frequency range through structural modifications, thereby avoiding resonance. Numerical simulations indicate that this method can achieve effective control of the torsional vibration of diesel generator sets and propulsion shafting systems while satisfying complex physical constraints.

However, research specifically targeting vibration reduction designs for power packs in rolling stock remains limited compared to marine applications. Unlike ships, railway vehicles are subject to the coupling effects of wheel–rail excitation and equipment excitation [13]. Zhai et al. [14] established the classic vehicle–track coupled dynamics theory, which serves as the foundation for analyzing such coupled vibration problems. Additionally, the nonlinearity of rubber isolators cannot be ignored. Liu et al. [15], Huang et al. [16] and Deng et al. [17] analyzed the nonlinear stiffness and damping characteristics of rubber mounts used in rail vehicles, proving that optimization depends on accurate parameter identification.

Regarding the optimization method, since isolation systems involve multiple design variables and conflicting objectives (e.g., vibration at different locations), traditional trial-and-error methods are inadequate. Thus, multi-objective optimization based on surrogate models has gained popularity. Cho et al. [18] applied the response surface method (RSM) to optimize the suspension parameters of a high-speed train. Xie et al. [19] proposed a novel prefabricated strengthening structure that can significantly enhance the stiffness, load-bearing capacity, and dynamic performance of short-span, simply supported bridges for heavy-haul railways without interrupting railway operations. By combining theoretical modeling with experimental validation, the RSM and multi-objective optimization were

employed to systematically establish functional relationships between strengthening parameters and key mechanical performance indicators of the bridge, thereby determining the optimal strengthening scheme.

Based on the practical operational issues and the findings of the relevant studies mentioned above, this study established a rigid body dynamics model for a power pack single-layer isolation system and determined the optimal initial parameters for this system. A refined finite element model of rolling stock including the power pack was used to investigate the vehicle vibration characteristics under power pack excitation. Based on a single transmission path from the power pack to the driver cab, the study utilized the global response surface methodology to perform multi-objective optimization. The initial experiments identified the design variables and modal matching principles to constrain the optimization process. The optimization objectives included minimizing the maximum force transmission rate at critical points on the vehicle body. Response surface fitting using LSR (Least Squares Regression) and the Moving Least Squares Method was employed, followed by optimization calculations using the GRSM (Gradient-Based Response Surface Method). The accuracy of the optimization predictions was verified using finite element model validation. Finally, the study considered the coupled effects of wheel–rail excitation and other stimuli, combining experimental and simulation results to validate the optimized stiffness parameters for the isolation system.

2. Initial Design of Power Pack Parameters

2.1. Power Pack Dynamics Mode

A multi-substructure power pack comprises multiple components, such as diesel generator sets and cooling systems, in a single framework (Figure 1). This power pack is elastically connected to the chassis of the vehicle at six points. Assuming the vehicle body to be a rigid foundation and the diesel generator set M to be a rigid body, the dynamics model of the single-layer isolation system for the power pack can be simplified, as shown in Figure 2.

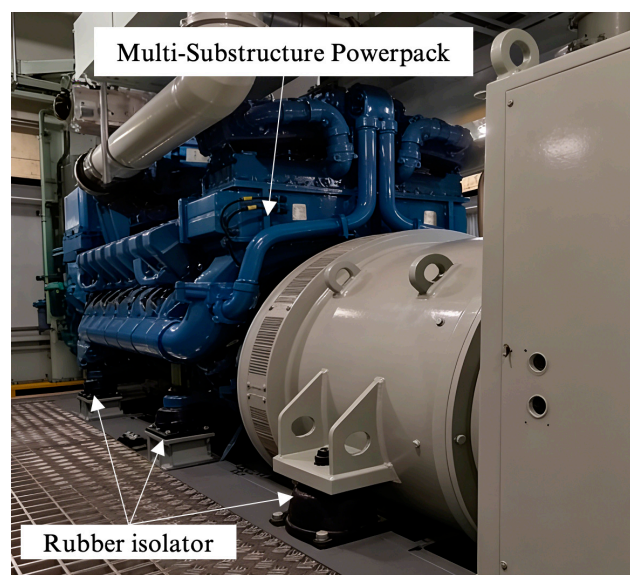


Figure 1. Structure of multi-substructure power pack.

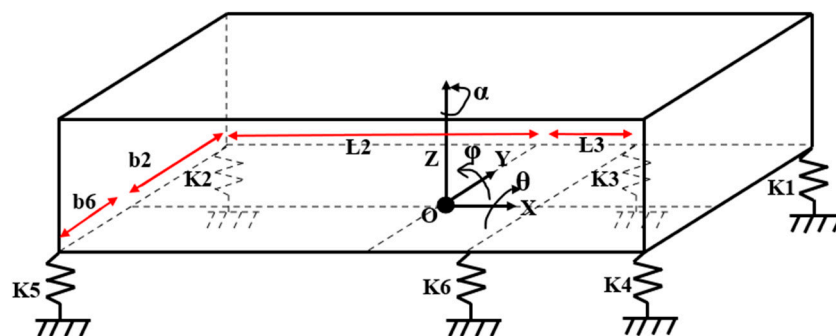


Figure 2. Simplified mechanical model of single-layer isolation system for a power pack.

The power pack has 6 degrees of freedom, and its motion differential equations are as follows:

$$M_{6 \times 6} \ddot{X}_{6 \times 1} + C_{6 \times 6} \dot{X}_{6 \times 1} + K_{6 \times 6} X_{6 \times 1} = F_{6 \times 1} \quad (1)$$

$$X = (x, y, z, \theta, \varphi, \alpha)^T \quad (2)$$

$$F = (F_x, F_y, F_z, M_x, M_y, M_z)^T \quad (3)$$

In these equations, M , K , and C represent the mass matrix, stiffness matrix, and damping matrix of the isolation system, respectively. x , y , z , θ , φ , and α denote the displacements of the rigid body along the reference coordinate system $(x, y, z, \theta, \varphi, \alpha)$; and F_x , F_y , F_z , M_x , M_y , and M_z represent the forces (or torques) exerted on the rigid body along the reference coordinate system $(x, y, z, \theta, \varphi, \alpha)$.

The damping matrix is related to the stiffness matrix and the excitation frequency, and can be expressed according to the following formula:

$$C = \eta \cdot K / \omega \quad (4)$$

where η is the damping factor and ω is the excitation frequency. In this study, the damping factor η was set to 0.1, which was determined based on the inherent material properties of the natural rubber isolators employed in the power pack system.

2.2. Stiffness Design of the Power Pack Vibration Isolator

The power pack of the vehicle is connected to the chassis via six wedge-shaped vibration isolators with known positions, coordinates, and initial static stiffness, as illustrated in Figure 1. A mechanical analysis was conducted for each isolator to establish its static equilibrium equation based on Hooke's Law:

$$F_i = (z + b_i \cdot \theta + l_i \cdot \varphi) \cdot k_i \quad (5)$$

where F_i is the reaction force of the i -th vibration isolator under the power pack; z is the static deflection of the power pack center of mass; b_i is the lateral coordinate of the i -th isolator in the local coordinate system; l_i is the longitudinal coordinate of the i -th isolator in the local coordinate system; θ and φ are the tilt angles of the power pack around its center of mass in the x -axis and y -axis directions, respectively; and k_i is the initial prescribed static stiffness of the i -th isolator.

For the power pack, conducting a force equilibrium analysis based on static principles yields the following set of equations for the static forces of the equipment mounted underneath the vehicle:

$$\begin{cases} mg = (F_1 + F_2 + F_3 + F_4 + F_5 + F_6) \\ (F_1 + F_2 + F_3) \cdot b_1 + (F_4 + F_5 + F_6) \cdot b_4 = 0 \\ (F_1 + F_4) \cdot l_1 + (F_2 + F_5) \cdot l_2 + (F_3 + F_6) \cdot l_3 = 0 \end{cases} \quad (6)$$

where m is the mass of the power pack and $g = 9.81 \text{ m/s}^2$ is the acceleration due to gravity.

Substituting each vibration isolator reaction force F_i into (6), according to the initial prescribed static stiffness of each isolator, the equation can be solved for θ , φ , and z . Subsequently, corrections are made to isolators with inconsistent static deflections to ensure that the power pack remains horizontal. The initial state variables θ , φ , and z obtained from solving Equation (6) are substituted back into Equation (5) to determine each isolator's reaction force F_i . Based on the natural frequencies of the vehicle-mounted equipment and the permissible range of isolator static deflections, the corrected static deflection z' of the isolators can be determined:

$$z' = \frac{gd}{4\pi^2\omega_n^2} \quad (7)$$

where d is the dynamic-to-static stiffness ratio of the isolator, and ω_n is the natural frequency of the power pack.

Revisiting the mechanical analysis for each isolator, we established the static equilibrium equation for each isolator based on Hooke's Law:

$$F'_i = (z' + b_i \cdot \theta' + l_i \cdot \varphi') \cdot k'_i \quad (8)$$

where $F'_i = F_i$, θ' and φ' represent the optimized roll and pitch of the vehicle-mounted equipment after adjustment, and k'_i denotes the designed static stiffness of the i -th isolator. The initial natural frequency was set at 8 Hz based on the longitudinal symmetry characteristics of the power pack, with its triaxial stiffness design as follows: 5390 N/mm for K_{x1}/K_{z1} , 1617 N/mm for K_{y1} , 6440 N/mm for K_{x2}/K_{z2} , 1932 N/mm for K_{y2} , 5880 N/mm for K_{x3}/K_{z3} , and 1764 N/mm for K_{y3} .

3. Establishment and Analysis of Finite Element Models

3.1. Model Establishment and Modal Analysis

First, the three-dimensional structural modeling of the vehicle body was conducted. The model of the vehicle body primarily included the roof, body, underframe, driver cab section, and other undercarriage equipment such as the power pack. Subsequently, these components were discretized and meshed using a multi-zone meshing approach. The model comprised a total of 1,406,933 mesh elements and 1,429,494 mesh nodes (Figure 3a). Due to the complexity of the power pack structure and its various modes of motion, it was simplified to a lumped mass model connected to the vehicle body via six isolators (Figure 3b). The mass of the power pack was 14,050 kg.

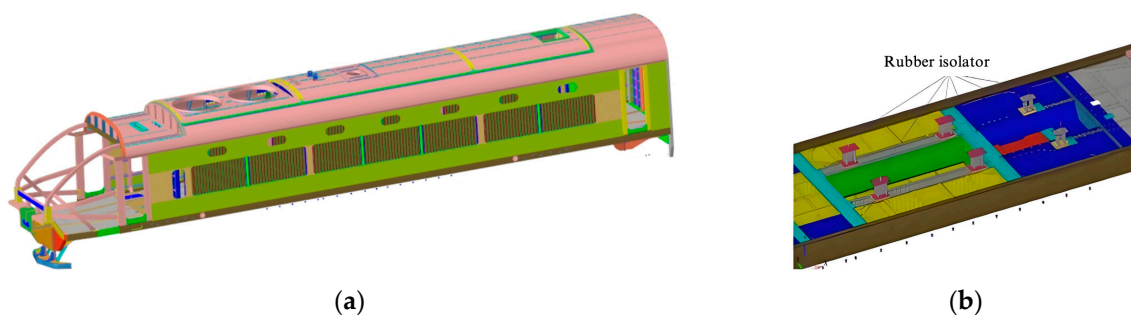


Figure 3. (a) The refined finite element model of the vehicle body including the power pack; (b) rubber isolator.

Modal analysis was performed to ascertain the inherent vibration characteristics of the vehicle body [8], including mode frequencies and shapes. Simultaneously, the analysis validated the accuracy of the established model. The first six mode shapes of the vehicle body are illustrated in Figure 4a–f, along with their modal frequencies. The modal frequencies obtained from the analysis closely match the experimental modal frequencies (Table 1), highlighting the accuracy of the established model.

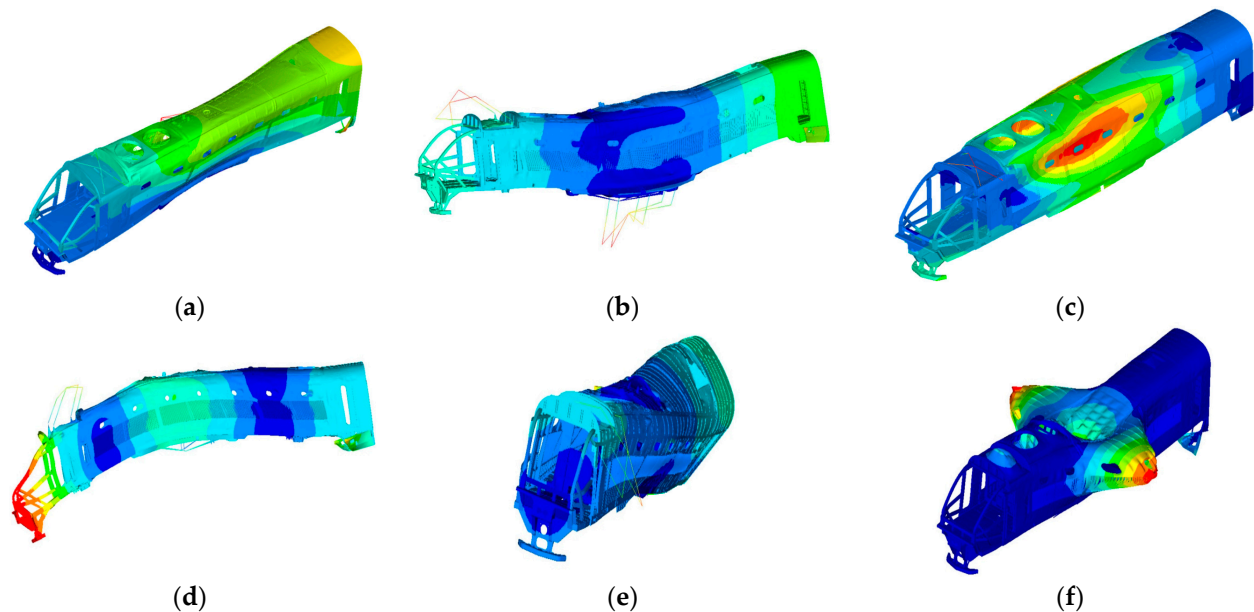


Figure 4. Diagrams of modal shapes: (a) diamond mode; (b) 1st-order vertical bending in-phase with power pack bouncing; (c) 1st-order lateral bending; (d) 1st-order vertical bending out-of-phase with power pack bouncing; (e) torsional mode; (f) breathing mode.

Table 1. Variable design.

Mode Shape	Simulation Results (Hz)	Experimental Results (Hz)	Relative Error
1st-order vertical bending in-phase with power pack bouncing	8.05	8.14	−1.11%
Diamond mode	11.64	11.19	4.02%
1st-order lateral bending	14.88	14.58	2.06%
1st-order vertical bending out-of-phase with power pack bouncing	14.89	14.76	0.88%
Torsional mode	16.28	16.80	−3.10%
Breathing mode	24.13	24.90	−3.09%

3.2. Analysis of Vehicle Vibration Characteristics Under Power Pack Excitation

The diesel engine in this study operates at an idle speed of 600 rpm and normal operational speeds of 900–1800 rpm. The excitation forces during idle and normal operation are detailed in Appendix A. When analyzing the vehicle vibration characteristics under power pack excitation, specific points of interest on the vehicle were selected for computational study, such as the center of the driver cab (S1), the center of the car body (S2), and the floor directly beneath the power pack (S3) (Figure 5a). Modal analysis was used to calculate the response accelerations at these reference points after applying power pack excitation. The frequency response curves (Figure 5b) indicate that all three reference points exhibited a primary peak frequency around 7.8 Hz. Given that the vehicle's second mode natural

frequency is 7.8 Hz, resonance occurs when the excitation frequency matches this value, resulting in enhanced vibration amplitudes. Notably, point S1 showed higher vibration amplitudes compared to the other points, which was attributed to the vehicle's second mode shape being a first vertical bending mode, where the vibration amplitudes at the sides of the vehicle are greater than those at the center.

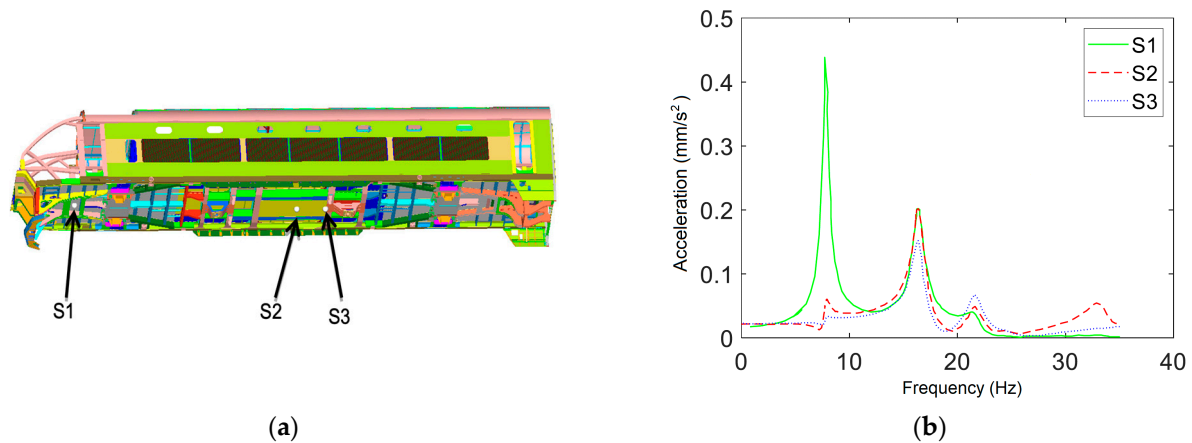


Figure 5. (a) Schematic diagram of reference point locations; (b) frequency response curves of reference points.

4. Multi-Objective Optimization of Power Pack Vibration Isolation Systems

4.1. Optimization Targets

During actual operation, the excessive vibrations in the driver cabin stem from the power pack being excessively excited. To address this practical issue, this section focuses on optimizing the parameters of the power pack vibration isolation system starting from a single vibration transmission path from the power pack to the vehicle body. Transmission efficiency is a critical metric for evaluating isolation performance, where force transmission efficiency is defined as the ratio of transmitted force to excitation force:

$$T_i = \frac{a_{ti}}{a_{ei}} (i = 1, 2, 3) \quad (9)$$

where a_{ti} is the amplitude of the response reactive force, and a_{ei} is the amplitude of the unit excitation force.

Three reference points of force transmission efficiency were used as the optimization criteria, with a higher transmission efficiency indicating poorer vibration isolation. To prevent scenarios where reducing vibrations in the driver cabin increases vibrations at other points on the vehicle body, a multi-objective optimization approach was designed:

$$\begin{cases} \text{goal} \Rightarrow \min(g(x)), g(x) = T_{s1}(X) \\ T_{s2}(X) \leq T_{s2}^0 \\ T_{s3}(X) \leq T_{s3}^0 \\ \text{s.t. } A_0 \leq X \leq A_n \end{cases} \quad (10)$$

where X is the variable, (T_{s2}^0, T_{s3}^0) denotes the force transmission rate between the original stiffness points S2 and S3, and (A_0, A_n) indicates the minimum and maximum values of the constrained variable.

4.2. Design Variables

The stiffness parameters of each shock absorber in a single-layer vibration isolation system exert a critical influence on the damping effect. Hence, it is imperative to choose

appropriate design variables to optimize the results. In this study, we used the three-directional stiffness of the shock absorber as the design variables (Table 2). By seeking both the minimum and maximum values of the objective function, we strove to achieve superior vibration isolation performance for the system.

Table 2. Variable design.

Isolator	$K_{xi} (K_{zi})$ (N/mm)			K_{yi} (N/mm)		
	A_1	A_n	d	A_1	A_n	d
D1	4990	5790	100	1417	1817	100
D2	6040	6840	100	1732	1932	100
D3	5480	6280	100	1564	1964	100

A total of $9 \times 9 \times 9 \times 5 \times 5 \times 5 = 91,125$ combinations.

When designing a power pack isolation system, consideration must be given to 18 design variables encompassing the three-directional stiffness of the six isolators, resulting in a significant computational workload. Therefore, it is essential to perform sensitivity analysis on these design variables to eliminate the ones that do not contribute to the optimization objectives and enhance computational efficiency. Utilizing a fractional factorial design, variables with substantial impacts on the optimization metrics were identified (Figure 6a). Irregular fluctuations in transmission rates among three reference points within the sampled points were observed (Figure 6b). Pareto charts were employed to visually analyze the degree of influence of each factor on the force transmission rates at various points (Figure 6c). Notably, variable K_{x1} exhibited the most significant impact on transmission rates at three points, followed by K_{x3} and K_{y1} , while the influence of the other variables on transmission rates were largely negligible.

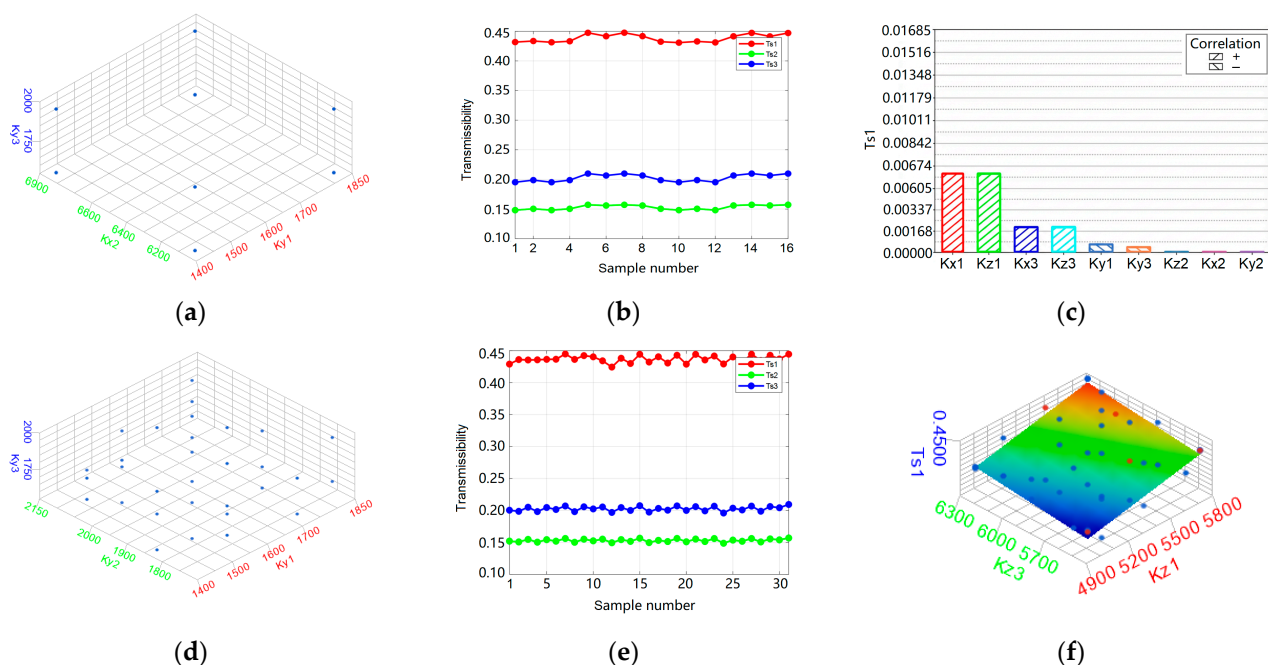


Figure 6. Cont.

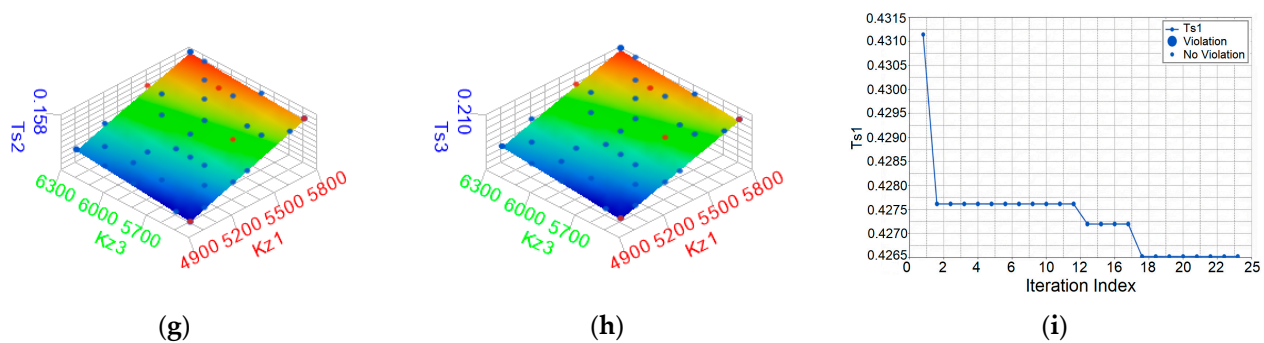


Figure 6. Multi-objective optimization of power pack vibration isolation system: (a) distribution projections of K_{x2} , K_{y1} , and K_{y3} during initial selection; (b) sampling results during initial selection; (c) Ts1 Pareto chart; (d) distribution projections of K_{y1} , K_{y2} , and K_{y3} during final selection; (e) sampling results during final selection; (f) Ts1 response surface; (g) Ts2 response surface; (h) Ts3 response surface; (i) variation in force transmissibility.

4.3. Multi-Objective Optimization Based on Global Response Surface Method

Using the Hammersley method for sampling ensures an even distribution of sampling points in the data space (Figure 6d); the simulation results were computed based on these samples (Figure 6e). Using the collected dataset, the Maximum Likelihood Estimation (MLSM) and LSR methods were employed to fit response surfaces for the maximum force transmission rates at three points (Figure 6f–h).

$$\min E = \sum_{i=1}^n (f_i^p - f_i)^2 \quad (11)$$

Here, f_i^p denotes the predicted values and f_i denotes the actual values. The fitting results are shown in Table 3, which demonstrates the high reliability of the fit.

Table 3. Fitting results.

Dependent Variable	Fitting Method	Residual Matrix of Training Set (Percentage)	Residual Matrix of Test Set (Percentage)	Cross-Validation Matrix Residual (Percentage)
Ts1	LSR	0.7929	0.7306	0.7189
Ts2	MLSM	0.9994	0.9984	0.9888
Ts3	MLSM	0.9995	0.9950	0.9929

Using the GRSM algorithm for optimization, the objective was achieved after 17 iterations (Figure 6i), with a central floor force transmission rate of 32.65% in the driver cabin. After iterative calculations, the optimized stiffness values for each rubber isolator at the minimum objective function were as follows: 5090 N/mm (K_{x1}/K_{z1}), 1517 N/mm (K_{y1}), 6440 N/mm (K_{x2}/K_{z2}), 1932 N/mm (K_{y2}), 5480 N/mm (K_{x3}/K_{z3}), and 1764 N/mm (K_{y3}).

4.4. Simulation Verification

Using the finite element model to simulate and verify the results obtained from fitting and iterative optimization, the maximum force transmission rates at three points before and after optimization were determined (Table 4). It is evident that after optimization, the predicted force transmission rates closely match the simulated ones, demonstrating the validity of the optimization predictions. Additionally, the force transmission rate in the driver cabin was enhanced by 11.6%, with no increase in transmission rates at other points, meeting the optimization criteria.

Table 4. Change in maximum force transmission rates at three points.

Reference Point	Design Parameter Force Transmission Rate	Optimized Parameter Force Transmission Rate	Simulated Calculation Force Transmission Rate
S1	0.4405	0.3265	0.3245
S2	0.1532	0.1494	0.1502
S3	0.2032	0.1969	0.1869

5. Test Verification

In the previous experiments, optimization design and simulation verification of the isolation system under single excitation (power pack excitation) were conducted. However, in actual operation, wheel–rail excitations also significantly affect cabin vibrations. Therefore, it is necessary to verify whether the optimized isolation system meets isolation requirements under coupled excitations. Operational line tests were conducted on a high-speed ballast track at speeds ranging from 120 to 200 km/h (Figure 7a). PCB ICP-type accelerometers were installed on the power pack, the interior floor, and the carbody underframe above the air springs to measure triaxial vibration accelerations (Figure 7b,c) and vibration intensity on the cabin floor (Figure 7d). Data were acquired at a sampling frequency of 1024 Hz and processed using a 0.5–80 Hz band-pass filter, in accordance with ISO 2631 standards [5]. Finally, the vibration comfort [20] in the cabin at different speeds was evaluated, and the corresponding comfort indices were measured to be 2.086, 2.304, and 2.487. The results indicate that all parameters meet the relevant standards, demonstrating the effectiveness of this isolation design method in enhancing the isolation efficiency of the power pack isolation system, indicating that this method possesses significant practical value.

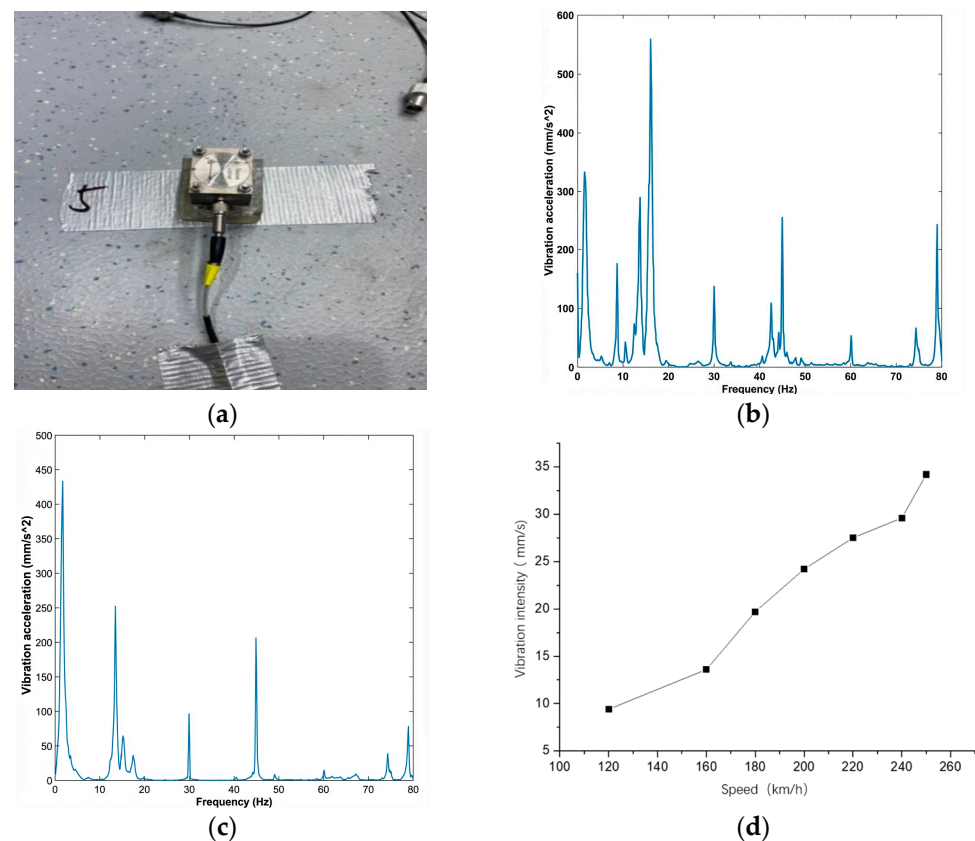


Figure 7. Test verification: (a) accelerometer; test results at speeds of (b) 200 km/h and (c) 240 km/h; (d) vibration intensity at the cabin floor.

6. Conclusions

A dynamic model of a single-layer isolation system for power packs was established. Based on vibration isolation theory and static load-bearing requirements, the optimal stiffness parameters of the isolation system were determined.

A refined finite element model of the vehicle body integrated with the power pack was developed. Modal analysis was conducted to extract the first six elastic modal frequencies and corresponding mode shapes of the vehicle body. Harmonic response analysis was then performed to characterize the vibration behavior of the vehicle under power pack excitation. The results show that resonant peaks appeared at frequencies of 7.8, 16.4, and 21.6 Hz in the driver's cabin, with the 7.8 Hz peak being the most prominent. Therefore, targeted measures to suppress excitation frequencies around 7.8 Hz are recommended.

The triaxial stiffness of the isolation device was optimized, focusing on minimizing the maximum force transmission rate at the vehicle reference point. Multi-objective optimization was implemented using the global response surface methodology to determine the optimal stiffness parameters. The accuracy of the optimization predictions was verified via finite element simulations.

Field tests were carried out on the operational line under different working conditions. The vibration intensities of the power pack and the comfort indices of the driver's cabin floor were calculated for each condition. The test results indicate that all parameters satisfy the requirements of the relevant standards, demonstrating that the proposed isolation design method can effectively improve the vibration isolation performance of power pack systems and thus possesses considerable practical engineering value.

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Conflicts of Interest: Author W.S. was employed by CRRC Qingdao Sifang Co., Ltd. The remaining authors declare that the research was conducted in the absence of any commercial or financial relationships that could be construed as a potential conflict of interest.

Appendix A. The Excitation Forces During Idle and Normal Operation

Speed	Centrifugal Inertial Force /N	Speed	Centrifugal Inertial Force /N
Idling	82	1400	115
1000	82	1500	123
1100	90	1600	131
1200	98	1700	139
1300	107	1800	148

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