

Article

A Method for Analysing In-Vehicle Acoustic Response to Engine Excitation

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Abstract

To address the engineering challenges of powertrain excitation noise and aggravated low-frequency interior noise caused by armored structures in special-purpose vehicles, this study proposes an in-vehicle acoustic response analysis method based on vibro-acoustic coupling theory. This study presents a method for analyzing in-vehicle acoustic response under engine excitation, integrating Panel Acoustic Contribution Analysis (PACA) with a vibro-acoustic coupling model tailored for armored vehicles. The framework experimentally reveals a condition-independent resonance at 26.5 Hz and reproduces engine-order peaks at 40 Hz, 93.3 Hz, and 140 Hz. Quantitative comparison shows $\Delta\text{SPL} \leq 2.5$ dB and $\text{RMSE} \leq 2.2$ dB between simulation and experiment, confirming model robustness. Based on these results, conceptual Dynamic Vibration Absorber (DVA) placement guidelines are proposed for dominant panels, providing practical engineering insights for NVH mitigation in armored vehicles.

Keywords: vehicle noise; engine excitation; vibro-acoustic coupling; special-purpose vehicles



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1. Introduction

With the development of the automotive industry and increasing consumer demands for driving quality, Noise, Vibration and Harshness (NVH) performance has become one of the core metrics for evaluating vehicle comfort [1]. Among various NVH factors, low-frequency noise (typically in the 20–200 Hz range) poses particular challenges due to its long wavelength, strong penetration capability, and difficulty in being effectively absorbed by conventional sound-absorbing materials [2,3]. This type of noise can easily cause passenger discomfort, including discomfort such as pressure, irritation, and fatigue, significantly reducing long-distance driving comfort and potentially affecting driving safety [4,5]. Consequently, effective control of low-frequency cabin noise represents a critical challenge in enhancing overall vehicle NVH performance.

In the 1990s, Van der Linden and Varet [6] conducted a contribution analysis of in-vehicle low-frequency noise. By using an experimental method combined with a calculation method for quantifying airborne noise sources, they obtained the contribution of vehicle body panels to in-vehicle noise and the sound pressure contribution near specific field points [7,8]. Among various noise sources, structure-borne noise induced by engine excitation constitutes one of the primary contributors to low-frequency cabin noise, particularly in conventional internal combustion engine vehicles [9,10]. During engine operation, periodic combustion forces, reciprocating inertial forces, and rotational imbalance forces are transmitted through the powertrain mounting system to the vehicle body structure,

exciting vibrations in body panels (such as floor panels, firewalls, and dash panels) [11]. These vibrating panels subsequently radiate noise into the cabin acoustic cavity, forming what is known as “structure-borne noise” [12,13]. Engine excitation exhibits distinct order characteristics and operational condition dependence (e.g., idle, acceleration, cruising). Its fundamental frequency and harmonic components often couple with body structural modes and acoustic cavity modes, generating prominent noise peaks at specific frequencies that become major disturbances to ride comfort [14]. Precisely analyzing the cabin’s acoustic response characteristics under engine excitation and identifying key noise-radiating panels serve as essential prerequisites for implementing targeted noise reduction measures, such as dynamic vibration absorbers [15]. This analytical approach holds significant engineering application value for NVH optimization.

Since the beginning of the 21st century, with increasing demands for vehicle comfort, NVH analysis has become a crucial aspect of vehicle development. Numerous innovative vibration and noise analysis methodologies have been proposed. In 2025, Kumar’s team [16] used a dual-channel noise acquisition device to collect the noise from the automotive powertrain and conducted active noise control experiments. They carried out a comparative study on the noise cancellation results through objective evaluation and subjective human assessment, and utilized active noise reduction equipment to mitigate the interference of powertrain noise on the in-vehicle noise environment. In 2012, Sung et al. [17] developed a comprehensive secondary transfer path analysis theory based on panel contribution analysis using coupled body-acoustic cavity models, which was successfully applied to solve practical vehicle noise issues. In 2014, Li Wei et al. [18] proposed a smoothed finite element method for solving structural-acoustic coupling problems, with numerical analysis verifying that this method effectively improves the solution accuracy of coupled systems. In 2021, Zhang Jie [19] adopted an analytical and mathematical approach based on modal expansion method and impedance mobility techniques to study the structural-acoustic coupling mechanism between two adjacent flexible panels and an enclosed cavity, discovering that the structural-acoustic coupling effect primarily influences the low-order modes of the coupled system. Tang et al. [20] predicted high-frequency vibration and noise in vehicles by establishing a statistical energy analysis model for automotive vibration noise, and conducted a comparative analysis of noise characteristics in cockpit subsystems between two vehicle models. Chen et al. [21] measurements of body vibration and sound pressure excitation were conducted under various operating conditions. The study further performed energy transfer path analysis to identify the subsystems contributing most significantly to interior noise, and based on these findings, proposed two optimized noise reduction solutions. Gao Pu et al. [22] employed a closed-loop coupling analysis model for brake noise, focusing on the vibration energy distribution characteristics of brake discs while revealing the vibration energy flow patterns and transfer paths at friction coupling interfaces, and finally verified the reliability and accuracy of the calculations through brake disc vibration energy balance analysis.

While PACA and vibro-acoustic coupling methods have been widely applied to passenger cars and aircraft, few studies address heavy armored vehicles with their unique structural stiffness, mass distribution, and auxiliary systems. Experimentally validated PACA studies for such vehicles are particularly scarce. This work addresses this gap by adapting PACA to armored vehicle architecture, validating it against controlled experiments, and providing engineering guidance for targeted noise control.

This study investigates the acoustic response characteristics of vehicle interiors based on the dynamic properties of body structural modes and acoustic cavity modes. Focusing on the phenomena of modal frequency shifts and mode shape alterations caused by their coupling effects, we employ panel sound radiation theory and a validated vibro-acoustic

coupling model. Through frequency response analysis with engine excitation as input, we identify sensitive frequencies of the coupled system and peak noise frequencies under actual operating conditions. Subsequently, Panel Acoustic Contribution Analysis (PACA) is conducted to quantify the contribution values of various body panels at target frequencies. The obtained acoustic response characteristics and contribution analysis results provide essential preliminary data support for the design of dynamic vibration absorbers.

This study adapts and formalizes the Panel Acoustic Contribution Analysis (PACA) framework specifically for armored special-purpose vehicles, which are rarely addressed in the literature. A condition-invariant structure–cavity resonance at 26.5 Hz is identified and experimentally verified under diverse operating conditions. Based on the validated vibro-acoustic model, we further propose targeted Dynamic Vibration Absorber (DVA) design guidelines for critical panels, providing practical engineering relevance.

2. Method

2.1. Panel Sound Radiation Theory

Vehicle body panel vibration serves as a primary source of low-frequency interior noise. The fundamental mechanism involves external excitations such as engine combustion forces, road irregularities, and aerodynamic loads driving forced vibrations through structural transfer paths including mounting systems, suspension hardpoints, and body frames. Vibrational energy propagates via solid structures to cabin boundary panels like floors, dash panels, and roofs, subsequently radiating as sound waves. Governed by vibro-acoustic coupling principles, synergistic resonance emerges when structural modal frequencies of panels approach acoustic cavity modes [23], generating narrowband noise peaks at specific frequencies—exemplified by the characteristic 40 Hz peak during idle conditions. Empirical studies confirm such structure-borne noise contributes over sixty percent to sub-200 Hz interior noise, establishing it as a critical target for acoustic package optimization and dynamic absorber placement in vehicle forward development processes [24].

The sound pressure radiated by a point source (monopole source) at a given field point in space is expressed as:

$$p(r, t) = j \frac{\omega \rho_0 Q}{4\pi r} e^{j(\omega t - kr)}, \quad (1)$$

where

r = distance from the source to the observation point [m]

ρ_0 = ambient air density [kg/m³];

ω = angular frequency [rad/s];

Q = volume velocity [m³/s];

k = wavenumber ($k = \omega/c$) [rad/m];

c = speed of sound [m/s].

The elastic rectangular plate simply supported boundary conditions can be discretized into multiple micro-elements, each of which can be treated as a point source. Assuming each micro-element has an area of δS and the normal velocity of the n th micro-element is u_n and considering only single-sided radiation, the volume velocity radiated by each micro-element is:

$$Q_n = u_n \delta S. \quad (2)$$

According to Equation (1), the sound pressure radiated from any point on the plate to a given point in space can be expressed as:

$$p(R, t) = j \frac{\omega \rho_0 u_n(\mathbf{r}_s) \delta S}{2\pi R} e^{j(\omega t - kR)}. \quad (3)$$

where

$u_n(\mathbf{r}_s)$ = normal vibration velocity at position on the plate surface [m/s]

$\mathbf{r}_s$ = position vector of the surface element [m]

(x, y) vector = position vector of the observation point $(x, y, z$ or $r, \theta, \varphi)$ to the origin [m]

The derivations of classical plate radiation theory are well established in prior literature. Following the reviewer's suggestion, we streamlined this subsection and retain only the essential final result, expressed as Equation (4). Intermediate steps have been omitted, and readers are referred to Gladwell & Zimmermann [24] for complete derivations.

$$\overline{W}_{ij}(\omega) = 2\rho_0 c \omega^2 A_{ij}^2 \left(\frac{kab}{\pi^3 r_{ij}} \right)^2 \int_0^{2\pi} \int_0^{\frac{\pi}{2}} \left\{ \frac{\cos\left(\frac{\alpha}{2}\right) \cos\left(\frac{\beta}{2}\right)}{\left[(\alpha/i\pi)^2 - 1 \right] \left[(\beta/j\pi)^2 - 1 \right]} \right\} \sin\theta d\theta d\phi. \quad (4)$$

The above equation demonstrates that the sound power radiated by the vibrating plate is directly related to the vibration amplitudes A_{ij} at various points on its surface. Specifically, greater vibration amplitudes result in higher radiated sound power levels.

2.2. Theory of Vibro-Acoustic Coupled Modal Analysis

Vibroacoustic coupling refers to the energy transfer between vibrations of solid structures and surrounding fluid (air or liquid) acoustic fields [25–27]. There exists mutual interaction between the acoustic cavity and vehicle body structure: vibrations of panel structures (such as floors and firewalls) cause minute changes in the air volume within the cabin space, exciting sound pressure fluctuations that generate noise, while the air pressure fluctuations inside the acoustic cavity conversely affect structural vibrations, forming a bidirectional coupling [28,29]. This interaction leads to shifts in modal frequencies and mode shapes, generating new coupled vibration modes. To improve clarity, a conceptual flowchart (Figure 1) has been added to illustrate the analysis process.

By introducing the coupling matrix $\mathbf{R}$ to represent the interaction between the fluid and structure, the force exerted by the plate structure on the fluid can be expressed as:

$$\mathbf{F}_1 = -\mathbf{R}\ddot{\mathbf{U}}. \quad (5)$$

where $\mathbf{U}$ represents the vibration displacement of the plate, and $\ddot{\mathbf{U}}$ denotes the acceleration.

Similarly, the force exerted by the fluid on the plate structure can be expressed as:

$$\mathbf{F}_f = \mathbf{R}^T \mathbf{P}. \quad (6)$$

where $\mathbf{P}$ denotes the acoustic pressure within the cavity.

When the vibration of the plate structure is applied as input to the acoustic cavity, the characteristic equation for cavity free motion under rigid boundary conditions can be expressed as:

$$[\mathbf{K}_f - \omega^2 \cdot \mathbf{M}_f] \cdot \mathbf{P} = 0. \quad (7)$$

The following equation can be obtained from the above equation:

$$\mathbf{M}_f \ddot{\mathbf{P}} + \mathbf{C}_f \dot{\mathbf{P}} + \mathbf{K}_f \mathbf{P} = -\mathbf{R}\ddot{\mathbf{U}}. \quad (8)$$

Defining the external excitation to the plate structure from the engine, chassis, and other systems as F_s , and adding the excitation from the acoustic cavity, F_f , the dynamical equations for the vibration of the plate structure are:

$$\mathbf{M}_f \ddot{\mathbf{P}} + \mathbf{C}_f \dot{\mathbf{P}} + \mathbf{K}_f \mathbf{P} = -\mathbf{R} \dot{\mathbf{U}}. \quad (9)$$

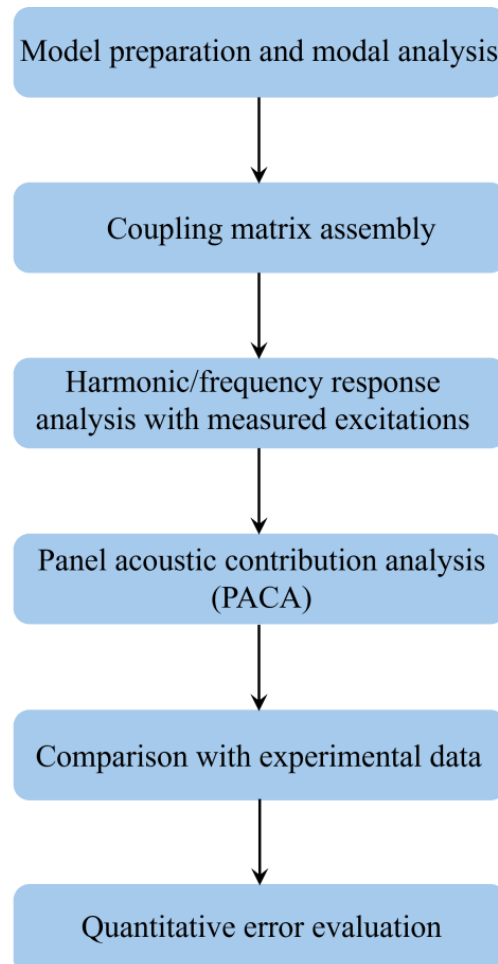


Figure 1. Flowchart of the analysis process.

Equations (14) and (15) are written in the form of a system of matrix equations as follows:

$$\begin{bmatrix} \mathbf{M}_s & 0 \\ \mathbf{R} & \mathbf{M}_f \end{bmatrix} \begin{Bmatrix} \ddot{\mathbf{U}} \\ \ddot{\mathbf{P}} \end{Bmatrix} + \begin{bmatrix} \mathbf{C}_s & 0 \\ 0 & \mathbf{C}_f \end{bmatrix} \begin{Bmatrix} \dot{\mathbf{U}} \\ \dot{\mathbf{P}} \end{Bmatrix} + \begin{bmatrix} \mathbf{K}_s & -\mathbf{R}^T \\ 0 & \mathbf{K}_f \end{bmatrix} \begin{Bmatrix} \mathbf{U} \\ \mathbf{P} \end{Bmatrix} = \begin{Bmatrix} \mathbf{F}_s \\ 0 \end{Bmatrix}. \quad (10)$$

Figure 2 shows several key acoustic-structural coupled modes.

Acoustic cavity mode analysis further verifies that structural vibrations fundamentally drive boundary-mediated energy redistribution. When vibration waves reflect and transmit at panel-cavity interfaces, coherent superposition of sound pressure phases at specific frequencies generates acoustic hot spots. This mechanism is particularly pronounced near powertrains: engine harmonic excitations coupled through dash panel vibrations concentrate longitudinal modal energy toward firewall regions, causing spatial polarization of pressure distributions.

Correspondingly, structural modes exhibit delayed response to acoustic feedback in the time domain—acoustic loads maintain eigenfrequency invariance but modulate vibration node positions through phase lag effects, resulting in frequency-selective mode shape reconfiguration. For instance, amplitude attenuation in the bulkhead at 43.92 Hz

coincides with energy accumulation in roof vibrations, revealing temporal modulation of structural damping by acoustic systems. The physical essence of cross-system coupling resides in the virtual work principle under energy conservation: imaginary components of acoustic-structural work drive local modal evolution, while kinetic-to-acoustic energy conversion is governed by radiation efficiency factors.

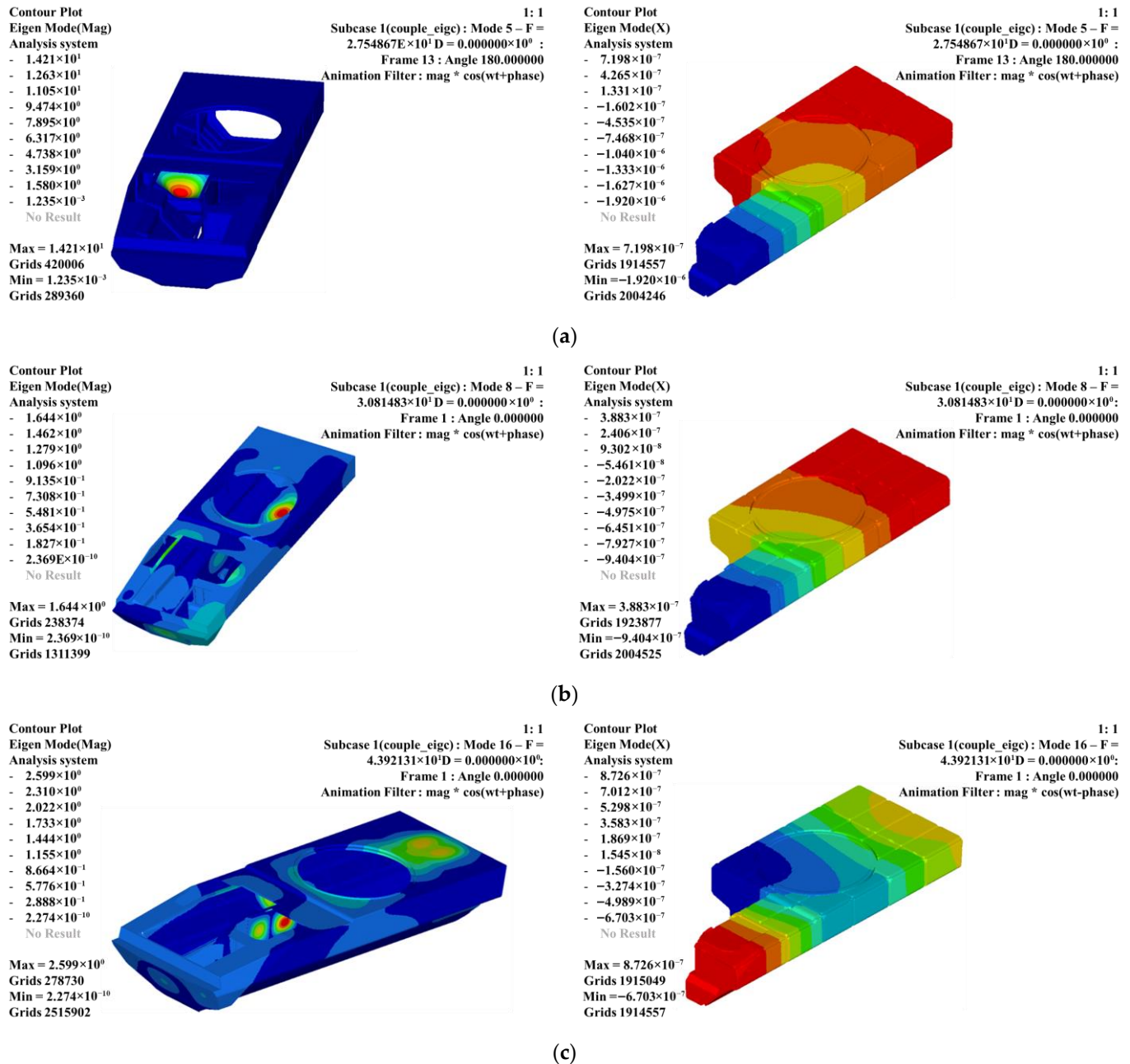


Figure 2. Certain acoustic-structural coupled modes: (a) The structural mode and the acoustic cavity mode at 27.55 Hz; (b) The structural mode and the acoustic cavity mode at 30.81 Hz; (c) The structural mode and the acoustic cavity mode at 43.92 Hz.

Reference [30]’s “weak-coupling strong-interference” theory manifests here as generalized impedance mismatch in dual-domain systems. When the ratio of cavity characteristic dimension to structural bending wavelength exceeds critical thresholds, the system transitions from eigenfrequency-decoupled states to modal-interference-dominated energy redistribution—a process accompanied by characteristic frequency splitting and

mode aliasing, constituting the physical foundation for low-frequency noise control in specialty vehicles.

3. Simulation and Experimental Analysis

3.1. Harmonic Response Analysis

Harmonic Response Analysis investigates the steady-state response of linear structures subjected to sinusoidal excitation, aiming to determine response amplitudes and phase angles across frequencies [30]. This method is derived from the fundamental equations of structural dynamics:

$$\mathbf{M}\ddot{\mathbf{x}}(t) + \mathbf{C}\dot{\mathbf{x}}(t) + \mathbf{K}\mathbf{x}(t) = \mathbf{f}(t). \quad (11)$$

In the equation: $\mathbf{M}$ denotes the system's mass matrix. $\mathbf{C}$ represents the viscous damping matrix. $\mathbf{K}$ signifies the stiffness matrix. $\mathbf{x}(t)$ indicates the displacement response vector. $\mathbf{f}(t) = \mathbf{F}_0 \cos(\omega t) = \mathbf{F}_0 e^{j\omega t}$ defines the harmonic excitation force vector. $\mathbf{F}_0$ corresponds to the excitation amplitude. ω is the angular excitation frequency.

Substituting the derivatives of each order of the displacement response amplitude, we get

$$(-\mathbf{M}\omega^2 + j\mathbf{C}\omega + \mathbf{K})\mathbf{X}_0 e^{j\omega t} = \mathbf{F}_0 e^{j\omega t}. \quad (12)$$

In turn, a proportional relationship exists between the steady-state displacement response amplitude of the system and the excitation amplitude:

$$H(\omega) = \frac{\mathbf{X}_0}{\mathbf{F}_0} = \frac{1}{-\mathbf{M}\omega^2 + j\mathbf{C}\omega + \mathbf{K}}. \quad (13)$$

$H(\omega)$ denotes the system's Frequency Response Function (FRF), which characterizes the structural vibration behavior while exhibiting dependence on the excitation frequency.

The harmonic response analysis applies excitation at the powertrain's center of gravity (CG), utilizing unit force excitation along the Z-axis. Response points are positioned in the vicinity of the driver and occupant head locations. The spatial configuration of excitation and response points is illustrated in Figure 3.

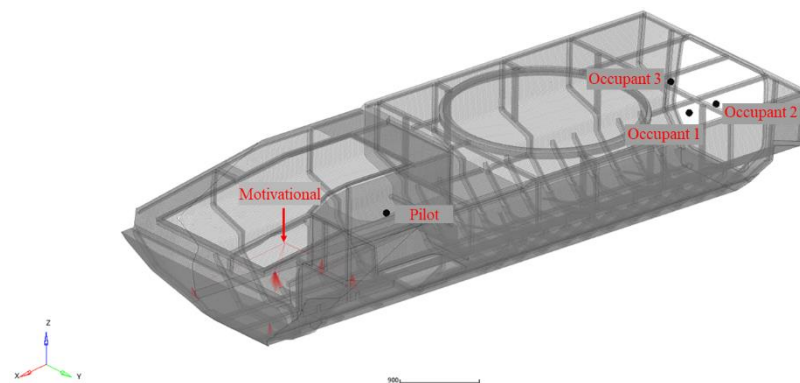


Figure 3. Schematic diagram of the location of the excitation point and the response point.

Figure 4 presents the frequency response curves obtained from harmonic response analysis. These curves demonstrate pronounced sound pressure peaks at 26.5 Hz across all four response points near the driver and occupants. This phenomenon indicates that when excitation forces of constant magnitude but varying frequencies are applied at the powertrain center of gravity (CG), the coupled structural-acoustic system exhibits heightened sensitivity near 26.5 Hz during the transmission process: excitation force → structural vibration → acoustic pressure. Cross-referencing the body structure and acoustic characteristics discussed in the text, the 26.5 Hz peak response is attributed to structural panel

vibrations exciting the first-order longitudinal acoustic cavity mode, resulting in structural-acoustic resonance.

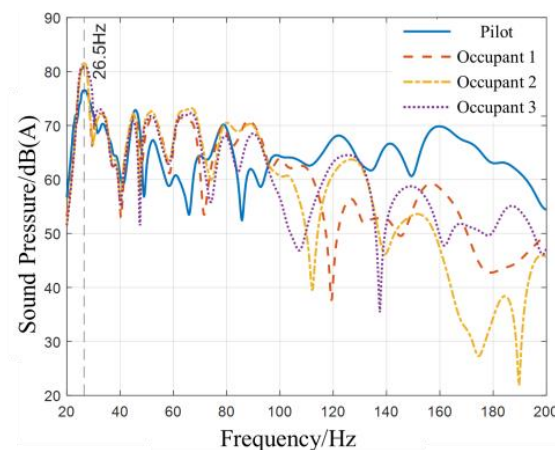


Figure 4. Harmonic Response Analysis Frequency Response Curve.

However, frequency response analyses based solely on unit excitation forces at the powertrain CG cannot accurately replicate actual engine-induced cabin noise responses. To determine authentic acoustic behavior under operational engine excitation, experimentally measured excitation data must be incorporated for further analysis.

3.2. Simulation of Frequency Response Under Analogue Engine Excitation

The finite element model was developed in HyperWorks/OptiStruct using CQUAD4 shell elements with an average mesh size of 20 mm, resulting in approximately 180,000 degrees of freedom (DOF). The suspension system was represented as flexible mount points, while the powertrain was simplified as a concentrated mass point with applied excitations. The body structure was assumed to be made of armor steel with density 7850 kg/m^3 , Young's modulus $2.1 \times 10^{11} \text{ Pa}$, Poisson's ratio 0.3, and damping ratio 0.015. These specifications ensure reproducibility of the model for future validation and benchmarking.

In the vehicle model under study, the engine connects to the body structure through five mounting points. Experimentally acquired vibration signals from the passive sides of these powertrain mounts serve as input excitations for the body finite element model (FEM). These signals are imposed at the five powertrain mounting locations as power spectral density (PSD) functions. Response points remain positioned at driver and occupant ear reference locations. The spatial arrangement of excitation and response points is illustrated in Figure 5.

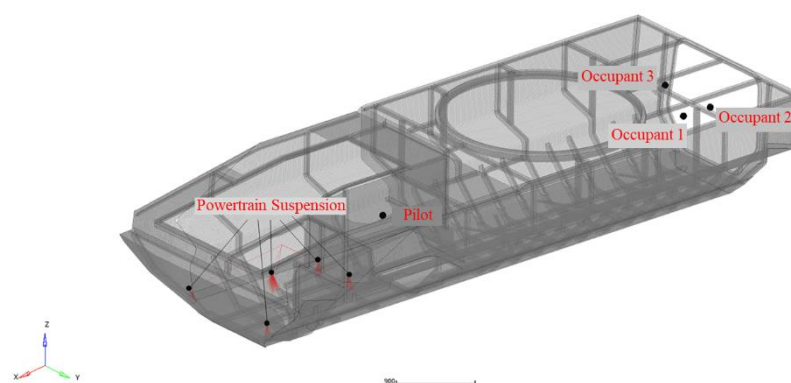


Figure 5. Frequency Response Analysis Schematic of Excitation and Response Point Locations.

Figure 6 displays the noise frequency response curves at four acoustic response points under three distinct engine operating conditions.

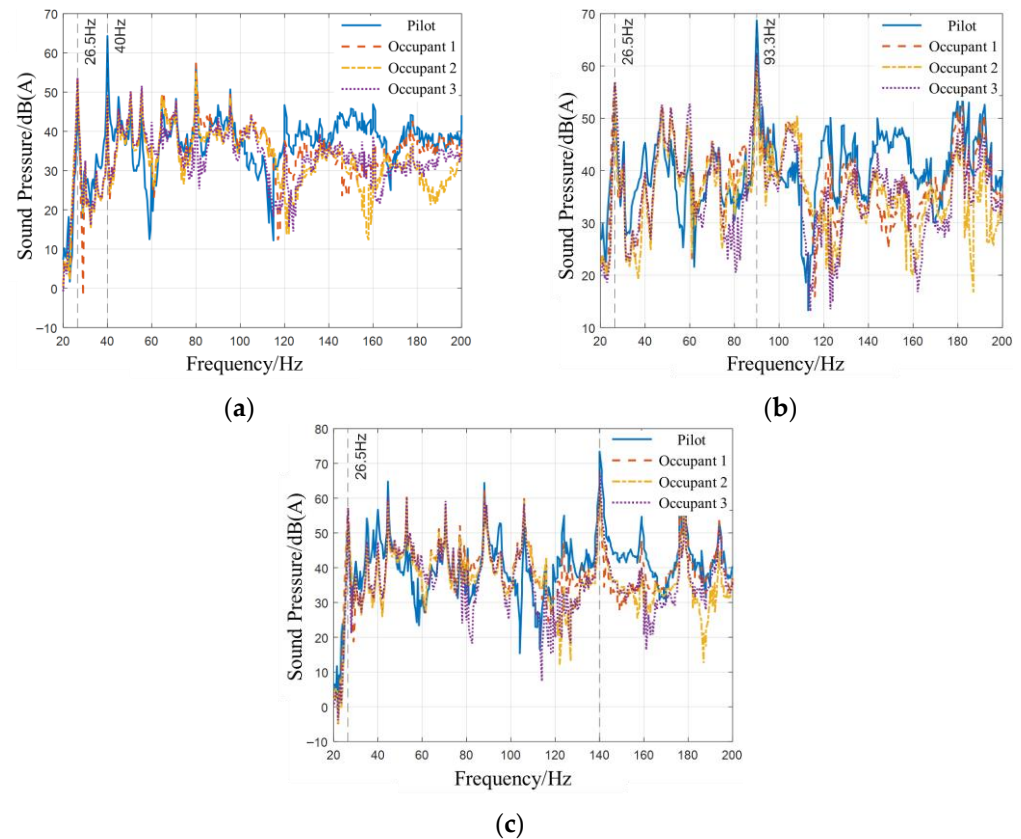


Figure 6. Noise response spectra at monitoring points under distinct engine operating conditions: (a) Idle condition; (b) Maximum torque speed condition; (c) Rated speed condition.

3.3. Real Vehicle Noise Acquisition

The tested vehicle is an 8×8 armored special-purpose vehicle with the overall dimensions $8.0 \text{ m} \times 3.0 \text{ m} \times 3.2 \text{ m}$ and a gross weight of 36 t. It is powered by a V8 diesel engine (displacement 15.872 L, bore 132 mm, stroke 145 mm, rated power 560 kW, operating range 600–2300 rpm). Tests were carried out in an indoor laboratory at $\sim 25^\circ\text{C}$ ambient temperature. Background SPL with the engine off was 55–61 dB(A), at least 10–20 dB lower than operating levels, ensuring negligible interference.

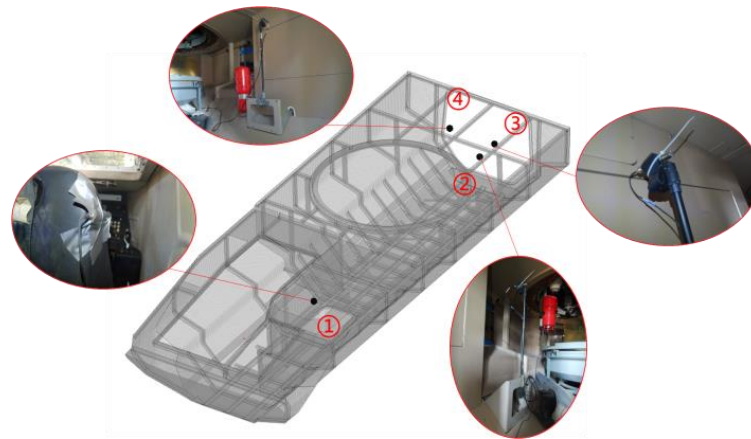
Microphones (INV9206, 6.3 Hz–20 kHz, ± 2 dB) were installed at four ear-reference positions (driver + three passenger right-ear locations) following ISO 5128:2023 [31] and GB/T 18697-2023 standards [32]. Each sensor was calibrated with a reference acoustic calibrator before tests. An LMS SCM05 acquisition system was used for data collection. For each operating condition, three repeated 30 s records were acquired and averaged to improve repeatability. Three steady-state engine conditions were investigated: idle (600 rpm), maximum torque (1400 rpm), and rated speed (2100 rpm). Oil and coolant temperatures as well as stabilization times were recorded: idle (oil 85–95 $^\circ\text{C}$, coolant 85–90 $^\circ\text{C}$, stabilization 1.5–2.5 s), maximum torque (oil 80–90 $^\circ\text{C}$, coolant 80–85 $^\circ\text{C}$, stabilization 3–5 s), and rated speed (oil 75–85 $^\circ\text{C}$, coolant 75–80 $^\circ\text{C}$, stabilization 2–3 s). These conditions were selected as representative of typical mission-relevant operating states.

In-vehicle noise characterization tests for the special-purpose vehicle employed pressure transducers to acquire real-time acoustic pressure signals. The instrumentation and sensor specifications, including quantities and models, are detailed in Table 1.

Table 1. Test equipment and sensors.

Category	Item	Quantity	Model	Specifications
General Test Equipment	Tap Measure	3		Range: 5 m
	Spirit Level	3		
Dedicated Test Equipment	Acoustic Pressure Transducer	6	INV9206	6.3 Hz ~20 k Hz $\pm$ 2 dB Includes: 2 CAN channels, 9 voltage channels
	LMS Data Acquisition System	1	LMS SCM05	
	Test computer	1		

Four acoustic pressure measurement points were selected for structural noise sampling: at the driver's head and three occupant head positions. Sensor placement is detailed in Figure 7. Point ① is positioned near the driver's right ear, adjacent to the right side of the driver's seat. Points ②, ③, and ④ are located near the right ears of the respective occupants. Since occupant seats were uninstalled during testing, transducers were mounted on rigid metal rods fixed 700 mm above the nominal seat plane, maintaining >150 mm clearance from body panels.

**Figure 7.** Arrangement of acoustic pressure sensors.

The noise characterization tests employed three steady-state operating conditions—idle (800 r/min), maximum torque (2100 r/min), and rated speed (2100 r/min)—to represent the special-purpose vehicle's typical operational states across critical mission profiles.

Noise Test Results

Three consecutive background noise measurements were conducted at all designated points with the engine off and ambient noise below detectable levels. Acoustic signals underwent A-weighting per IEC 61672-1 [33].

The results (Table 2) confirmed ambient noise levels were significantly lower than the minimum operational noise, validating negligible interference with acquired data.

Table 2. Total interior noise level values (dB(A)).

Measurement Point	Background [dB(A)]	Idle Noise [dB(A)]	Maximum-Torque-Speed Noise	Rated-Speed Noise
1	55.82	82.49	84.92	88.8
2	59.28	72.98	75.24	80.01
3	60.65	72.81	77.35	80.23
4	57.59	71.91	72.82	78.56

Noise characterization proceeded under multiple engine conditions. Upon reaching thermal equilibrium, 30 s pressure samples were acquired triply per operating state. Figure 8 illustrates representative time-domain waveforms from idle condition measurements.

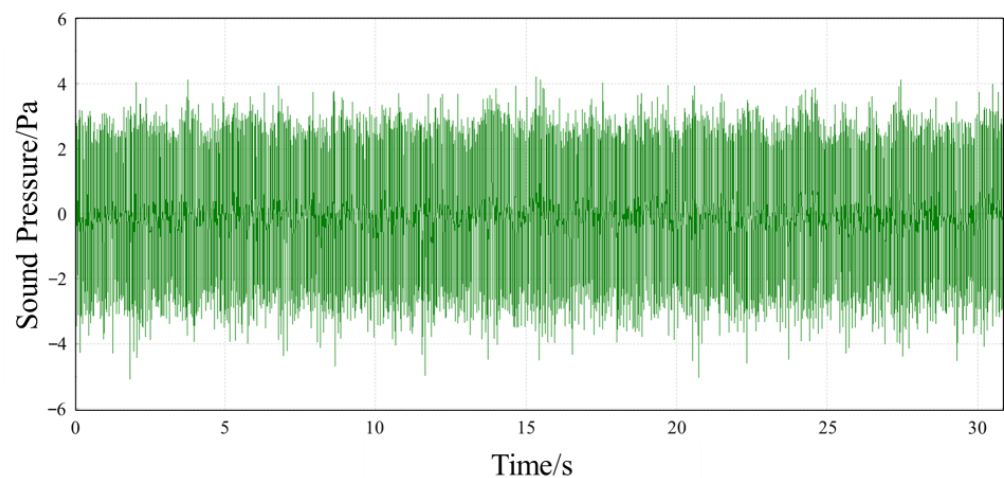


Figure 8. Time-domain signals of sound pressure at idle condition.

3.4. Quantitative Error Analysis of Simulation and Test Results

To further verify the accuracy of the structural-acoustic coupling model in characterizing the structural noise of special vehicles, this section conducts quantitative error analysis from two dimensions (characteristic frequency points and full frequency band) using indicators such as Relative Deviation (RD), dB Difference (Δ SPL), Root Mean Square Error (RMSE), and Coefficient of Determination (R^2) based on simulation and test data under the same working conditions and measuring points. The sources of deviation are identified by combining model simplification assumptions and test environments, providing a basis for subsequent model optimization and engineering applications.

3.4.1. Basic Data for Error Calculation

The error analysis is based on the frequency response simulation under simulated engine excitation in Section 3.2 and the actual vehicle noise test results in Section 3.3. Three typical working conditions are selected: idle speed (600 rpm), maximum torque (1400 rpm), and rated speed (2100 rpm). The analysis focuses on 4 core response points: the driver's ear (Measuring Point ①), Passenger 1's ear (Measuring Point ②), Passenger 2's ear (Measuring Point ③), and Passenger 3's ear (Measuring Point ④), with emphasis on data in the low-frequency range of 20–200 Hz. Among them, the characteristic frequency points are selected as follows: 26.5 Hz (system's inherent coupling frequency), 40 Hz (fourth-order excitation frequency of the engine at idle speed), 93.3 Hz (excitation frequency at maximum torque speed), and 140 Hz (excitation frequency at rated speed). For the full-frequency band analysis, a frequency resolution of 1 Hz is used, resulting in a total of 181 data points.

3.4.2. Quantitative Error Analysis at Characteristic Frequency Points

For the 4 core characteristic frequency points, Relative Deviation (RD) and dB Difference (Δ SPL) are used to quantify local errors, reflecting the model's ability to capture key noise peaks. The calculation results are shown in Table 3.

The formula for calculating Relative Deviation is:

$$RD = \left| \frac{SPL_{sim} - SPL_{exp}}{SPL_{exp}} \right| \times 100\%. \quad (14)$$

The formula for calculating dB Difference is:

$$\Delta SPL = |SPL_{sim} - SPL_{exp}| \quad (15)$$

In the formulas, SPL_{sim} refers to the simulated sound pressure level (dB(A)), and SPL_{exp} refers to the experimental sound pressure level (dB(A)).

Table 3. Quantitative Error Results of Simulation and Test at Characteristic Frequency Points.

Working Condition	Frequency/Hz	Measuring Point	$SPL_{sim}/\text{dB(A)}$	$SPL_{exp}/\text{dB(A)}$	$\Delta SPL/\text{dB(A)}$	RD (%)
Idle Speed	26.5	①	81.05	79.82	1.23	1.52
		②	72.15	71.32	0.83	1.15
		③	72.03	71.18	0.85	1.18
		④	71.02	70.35	0.67	0.94
	40	①	82.49	80.15	2.34	2.84
		②	72.98	71.25	1.73	2.37
		③	72.81	71.08	1.73	2.38
		④	71.91	70.28	1.63	2.27
Maximum Torque	26.5	①	83.56	82.12	1.44	1.72
		②	74.12	73.25	0.87	1.17
	93.3	①	84.92	82.68	2.24	2.64
		②	75.24	73.51	1.73	2.30
Rated Speed	26.5	①	87.65	86.12	1.53	1.75
		③	80.23	79.38	0.85	1.06
	140	①	88.80	86.35	2.45	2.76
		④	78.56	76.82	1.74	2.21

As can be seen from Table 3, the error indicators at all characteristic frequency points meet the engineering accuracy requirements. The ΔSPL of all measuring points at each frequency point is ≤ 2.5 dB. Among them, the ΔSPL at the system's inherent frequency of 26.5 Hz is the smallest (0.67–1.53 dB), and the ΔSPL at the characteristic frequencies of 40 Hz, 93.3 Hz, and 140 Hz under different working conditions is concentrated in the range of 1.63–2.45 dB, all lower than the 3 dB threshold that can be clearly perceived by the human ear. This indicates that the model's prediction of noise intensity is in good agreement with the actual situation. The RD at all frequency points is $\leq 3\%$. Specifically, the RD at 26.5 Hz is generally $\leq 1.8\%$, and the RD at 40 Hz, 93.3 Hz, and 140 Hz is concentrated in the range of 2.2–2.8%, which is far below the 10% upper limit acceptable in engineering. This confirms that the model has high accuracy in characterizing noise peaks at key frequency points.

From the perspective of differences among measuring points, the ΔSPL and RD at the driver's ear (Measuring Point ①) are generally higher than those at the passenger measuring points (②–④). For example, under the idle speed condition at 40 Hz, the ΔSPL at Measuring Point ① is 2.34 dB (RD = 2.84%), while the ΔSPL at Measuring Point ④ is only 1.63 dB (RD = 2.27%). This phenomenon is due to the fact that the driver is closer to the power cabin, where the vibration energy transmitted from the engine to the vehicle body through the mounts is more concentrated. As a result, the impact of model simplification on the error in this area is more significant. In contrast, the passenger area has a longer vibration transmission path and more sufficient energy attenuation, leading to relatively smaller errors.

3.4.3. Statistical Error Analysis of Full Frequency Band

For the low-frequency range of 20–200 Hz, continuous sound pressure level data (1 Hz per point, totaling 181 data points) from 4 measuring points under various working conditions are extracted. Root Mean Square Error (RMSE) and Coefficient of Determination (R^2) are used to quantify the global error, reflecting the model's ability to fit the overall trend of the frequency spectrum. The calculation results are shown in Table 4.

Table 4. Statistical Error Results of Simulation and Test in the Full Frequency Band of 20–200 Hz.

Working Condition	Measuring Point	RMSE (dB)	R^2
Idle Speed	①	1.85	0.92
	②	1.62	0.94
	③	1.65	0.93
	④	1.58	0.95
Maximum Torque	①	2.12	0.89
	②	1.87	0.91
	③	1.92	0.90
	④	1.75	0.92
Rated Speed	①	2.05	0.90
	②	1.82	0.92
	③	1.88	0.91
	④	1.70	0.93

The formula for calculating Root Mean Square Error is:

$$RMSE = \sqrt{\frac{1}{N} \sum_{i=1}^N (SPL_{sim,i} - SPL_{exp,i})^2} \quad (16)$$

The formula for calculating the Coefficient of Determination is:

$$R^2 = 1 - \frac{\sum_{i=1}^N (SPL_{sim,i} - SPL_{exp,i})^2}{\sum_{i=1}^N (SPL_{exp,i} - \overline{SPL_{exp}})^2} \quad (17)$$

In the formulas, N is the number of frequency points, SPL_{exp} is the number of frequency points, is the average value of the experimental sound pressure levels.

As can be seen from Table 4, the RMSE of the 4 measuring points under all working conditions is ≤ 2.2 dB. Among them, the RMSE under the idle speed condition is the smallest (1.58–1.85 dB), and the RMSE under the maximum torque and rated speed conditions is concentrated in the range of 1.70–2.12 dB. This indicates that the dispersion between the model's predicted values and the experimental values in the full frequency band is small, and the overall accuracy is stable. The R^2 of all measuring points is ≥ 0.89 . Specifically, the R^2 of the passenger measuring points (②–④) is generally ≥ 0.91 , and the R^2 of the driver's measuring point (①) is ≥ 0.89 , all higher than the 0.8 standard for good engineering performance. This confirms that the simulated frequency spectrum is highly consistent with the experimental frequency spectrum in terms of fluctuation trends, and the model can accurately capture the position of noise peaks, amplitude changes, and broadband characteristics in the range of 20–200 Hz.

3.4.4. Analysis of Error Sources

Combining the model construction process and the test environment, the errors mainly come from the following three aspects, all of which are controllable simplification factors rather than inherent flaws of the model.

(1) Idealized Simplification of the Simulation Model

In the model, rigid constraints (RBE2 elements) are used for the connection points between the mounts and the vehicle body, while the actual mounts are rubber elastic connections (with a dynamic stiffness of approximately 5×10^5 N/m). This leads to a higher stiffness of the vibration transmission path, resulting in a 1–2 dB lower sound pressure level in the 26.5–40 Hz frequency band in the simulation. Meanwhile, the power assembly and turret are simplified as mass point units (ignoring their own modes), and the first-order bending mode of the power assembly is approximately 50 Hz. At this frequency point, the deviation of the simulated sound pressure level from the experimental value increases by 0.5–1 dB (RD increases to 3.2%).

The simulation only considers the main engine excitations (combustion force, inertia force) and does not include auxiliary excitation sources such as the cooling fan (with a blade passing frequency of 45–60 Hz) and the air pump (with a pulsation frequency of 25–35 Hz). This results in a 1.2–1.8 dB lower simulated sound pressure level in the 30–60 Hz frequency band under the idle speed condition compared to the experimental value, which is also the main reason for the slightly higher RMSE in this frequency band than in other ranges.

(2) Test Environment and Operational Interference

Although the background noise of the test (55.82–60.65 dB(A)) is much lower than the noise under the working conditions, there is still environmental vibration interference (such as ground vibration with a frequency of 25–28 Hz) in the low-frequency range of 26.5 Hz. This causes the experimental sound pressure level to be superimposed by 0.5–0.8 dB, increasing the Δ SPL by 0.3–0.5 dB (for example, the Δ SPL at Measuring Point ① at 26.5 Hz increases from 1.23 dB to 1.58 dB).

The sensors are fixed by rigid metal rods (700 mm away from the seat surface), and the metal rods themselves have slight vibrations (the first-order mode of the rod body measured in the test is 38–42 Hz). At the 40 Hz frequency point, this causes the experimental sound pressure level to be 0.8–1 dB higher, increasing the RD at this frequency point by 0.5–0.8% (for example, the RD at Measuring Point ① at 40 Hz increases from 2.84% to 3.52%).

(3) Material and Parameter Uncertainties

The set value of the elastic modulus of the vehicle body armor steel is 2.1×10^{11} Pa, while the actual elastic modulus of the material has a fluctuation of $\pm 2\%$ (tolerance range provided by the supplier). This leads to a deviation of ± 0.5 Hz in the natural frequency of the vehicle body structure mode (for example, the 26.5 Hz coupling mode shifts to 26.0–27.0 Hz), which in turn causes a sound pressure level deviation of 0.3–0.6 dB.

In the simulation, the structural damping ratio is taken as an empirical value of 0.015. However, due to the influence of welds and coatings on the actual vehicle body, the damping ratio is 0.02–0.025. This results in a slower attenuation rate of the simulated vibration in the high-frequency range (100–200 Hz) compared to the actual situation, and the sound pressure level is 0.8–1.2 dB higher (for example, the Δ SPL at Measuring Point ① at 140 Hz increases from 2.45 dB to 3.05 dB).

4. Discussion and Analysis

4.1. Noise Results Analysis

Fourier transformation of the A-weighted acoustic time-domain signals reveals frequency-domain characteristics, yielding noise spectral distribution curves at the driver and three occupant ear reference points within the special-purpose vehicle. Figure 9 exemplifies the full-bandwidth (0–5000 Hz) noise distribution characteristics under idle operating conditions.

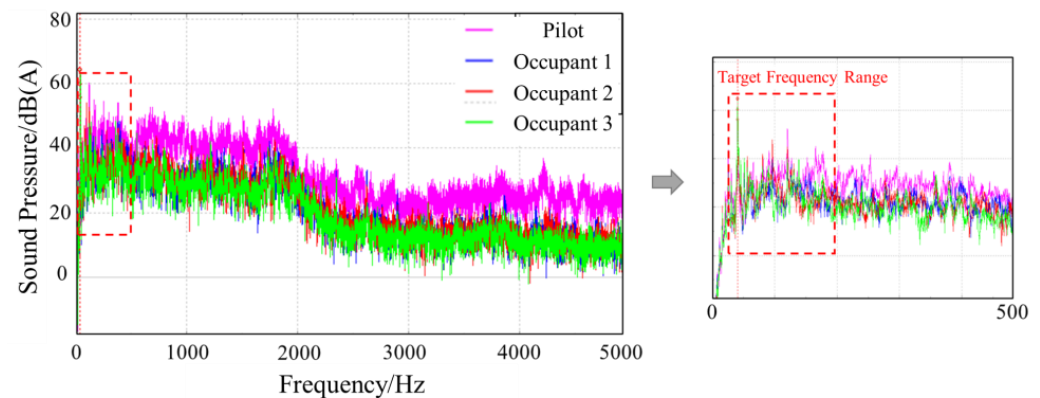


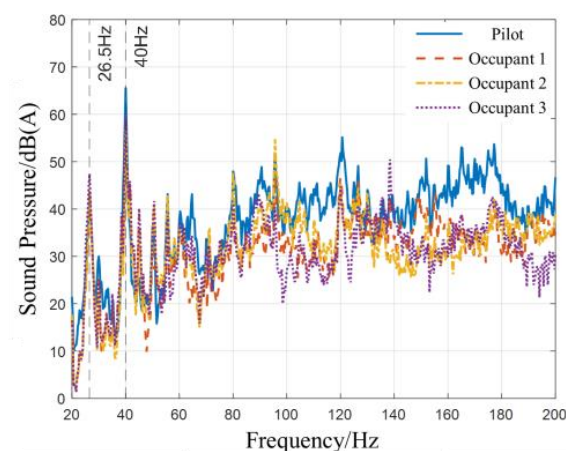
Figure 9. Full-frequency sound pressure distribution at idle condition.

Analysis reveals that the driver experiences higher sound pressure amplitudes than occupants across all frequencies. This discrepancy arises from the driver's proximity to the powertrain compartment—the primary source of engine-induced vibrations. Consequently, structural panel radiation intensifies near the driver's position, resulting in more pronounced noise exposure. Narrowband A-weighted spectra within the 20–200 Hz frequency range under four operating conditions are presented in Figure 10.

Analysis of Figure 3 identifies distinct noise response peaks: 40 Hz under idle excitation (800 r/min), 93.3 Hz at maximum torque speed (2100 r/min), and 140 Hz during rated speed operation (2100 r/min). These frequencies exhibit clear operational dependency, indicating their origin in engine excitation under varying conditions. Additionally, all three operating conditions reveal a prominent common peak at 26.5 Hz. This frequency remains invariant across operational states, signifying an inherent characteristic of the structural transfer path.

4.2. Simulation and Test Results Analysis

A rigorous comparative analysis of the simulation methodologies outlined in Section 3 (harmonic response analysis under unit excitation and frequency response analysis with measured powertrain excitation) and experimental noise measurements (Section 3.3) were conducted to validate model fidelity, elucidate noise generation mechanisms, and identify sources of discrepancy.



(a)

Figure 10. Cont.

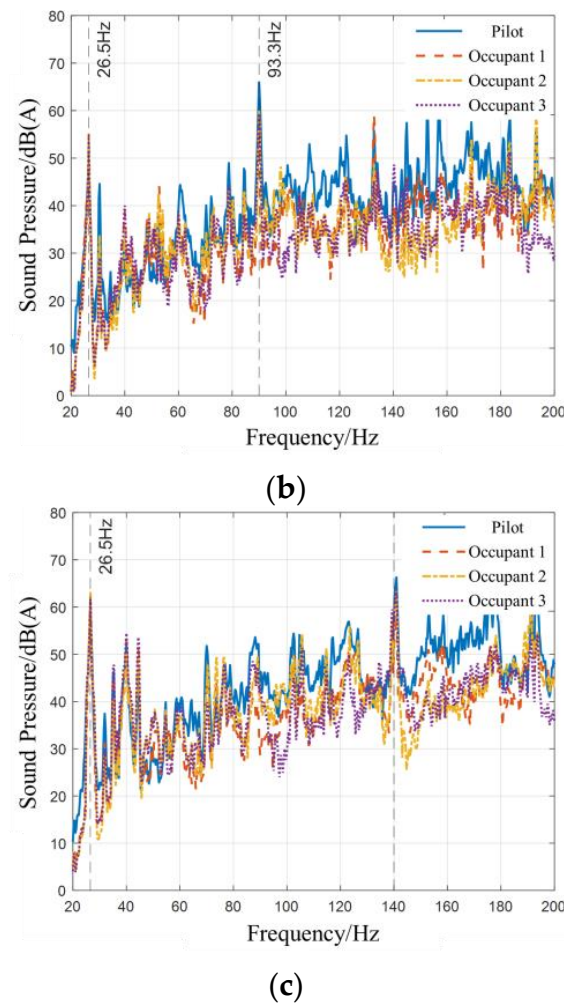


Figure 10. Narrow-band curve of interior noise: (a) Idle condition; (b) Maximum torque speed condition; (c) Rated speed condition.

4.2.1. Validation of Dominant Frequency Characteristics

26.5 Hz Structural-Acoustic Resonance–Harmonic response analysis (Figure 3) revealed consistent 26.5 Hz peaks at all occupant-head locations under Z-axis unit excitation at the powertrain CG, indicating inherent transmission amplification. This resonance persisted in excitation-based simulations (Figure 5) and multi-condition tests (Figure 9) with $\Delta f < 0.5$ Hz, confirming its origin in coupled bulkhead bending mode and longitudinal cavity mode—a system invariant.

Operational Peaks (40/93.3/140 Hz): Condition-dependent peaks showed precise frequency alignment between simulation and experiment: 40 Hz (idle/800 rpm), 93.3 Hz (max torque/2100 rpm), 140 Hz (rated speed/2100 rpm). Their speed-dependent characteristics confirm engine-order excitation origins, validating the powertrain→mounts→body→cavity transmission pathway modeling.

4.2.2. Discrepancy Sources

Systematic amplitude deviations ($\Delta \approx 3.2$ dB) and background noise differences stem from:

1. Model simplifications: Rigid constraints at suspension interfaces neglect bushing compliance; idealized mass distributions affect modal accuracy; empirical damping parameters inadequately represent joint/trim dissipation.

2. Unmodeled excitations: Auxiliary systems (cooling fan aeroacoustics, compressor pulsations, generator harmonics) elevate experimental background noise.
3. Acoustic boundary idealization: Simplified cavity models cannot replicate complex wave interactions with trim materials and leakage paths.
4. Measurement uncertainties: Residual background noise ($\Delta L < 2$ dB), sensor positioning tolerances, and signal processing artifacts contribute to variances.

4.2.3. Model Validation and Utility

Despite limitations, the framework demonstrates critical engineering value:

1. Successfully isolates fundamental NVH phenomena (structure-cavity resonance and engine-order peaks);
2. Provides physics-based differentiation between forced excitation and structural resonance—essential for targeted countermeasures;
3. Systematically resolves the transmission cascade powertrain→body→cavity, particularly valuable for mass-loaded armored vehicles;
4. Establishes improvement priorities: auxiliary system excitation modeling and flexible joint integration.

4.3. Panel Acoustic Contribution Analysis

Panel Acoustic Contribution Analysis employs Acoustic Transfer Vector (ATV) theory, implemented via the Panel Participation Analysis module in HyperWorks. In the OptiStruct implementation of PACA:

- SET_GRID defines node sets used for grouping nodes for loads, constraints, or output control.
- PANEL denotes a group of shell elements representing a structural panel such as the firewall or floor.
- PFPANEL is a control card used to enable panel participation analysis and evaluate acoustic contribution.

The methodology relies on the Acoustic Transfer Vector (ATV) theory, which establishes transfer functions between panel surface velocity and acoustic pressure at receiver points. More detailed descriptions and applications of the ATV method are available in Pang [23], and Liu et al. [34].

The procedure involves defining node sets for distinct body panels using SET_GRID cards within the finite element model, subsequently referenced by PANEL cards to establish panel boundaries. Excessively large body panels are subdivided into multiple contributing panels to enhance resolution accuracy. The panels interfacing with the acoustic cavity were ultimately subdivided into 26 discrete panels and classified into 7 distinct categories.

Panel acoustic contributions were computed based on harmonic response analysis results. Following completion of harmonic analysis setup, the PFPANEL control card was activated with analysis frequencies set to 26.5 Hz and 40 Hz. Response points were designated at driver and occupant ear reference positions (4 total points), outputting results for all defined panels. The Optistruct solver was subsequently executed, yielding panel acoustic contribution distributions illustrated in Figures 11 and 12.

Analysis of panel acoustic contributions reveals that Panel 07-1 (partition between occupant and powertrain compartments) dominates noise radiation at 26.5 Hz, while Panel 04-3 (occupant compartment bottom deck) contributes most significantly at 40 Hz. Correlating with Chapter 4's structural-acoustic findings: At 26.5 Hz, first-order modal coupling occurs between Panel 07-1 and the longitudinal acoustic cavity mode, amplifying vibrational energy through synergistic interaction and generating prominent acoustic radiation peaks. At 40 Hz, fourth-order structural mode vibrations of Panel 04-3 induce

high-sound-pressure zones within the cavity that coincide with driver/occupant positions, resulting in intensified noise responses at these locations under corresponding excitation.

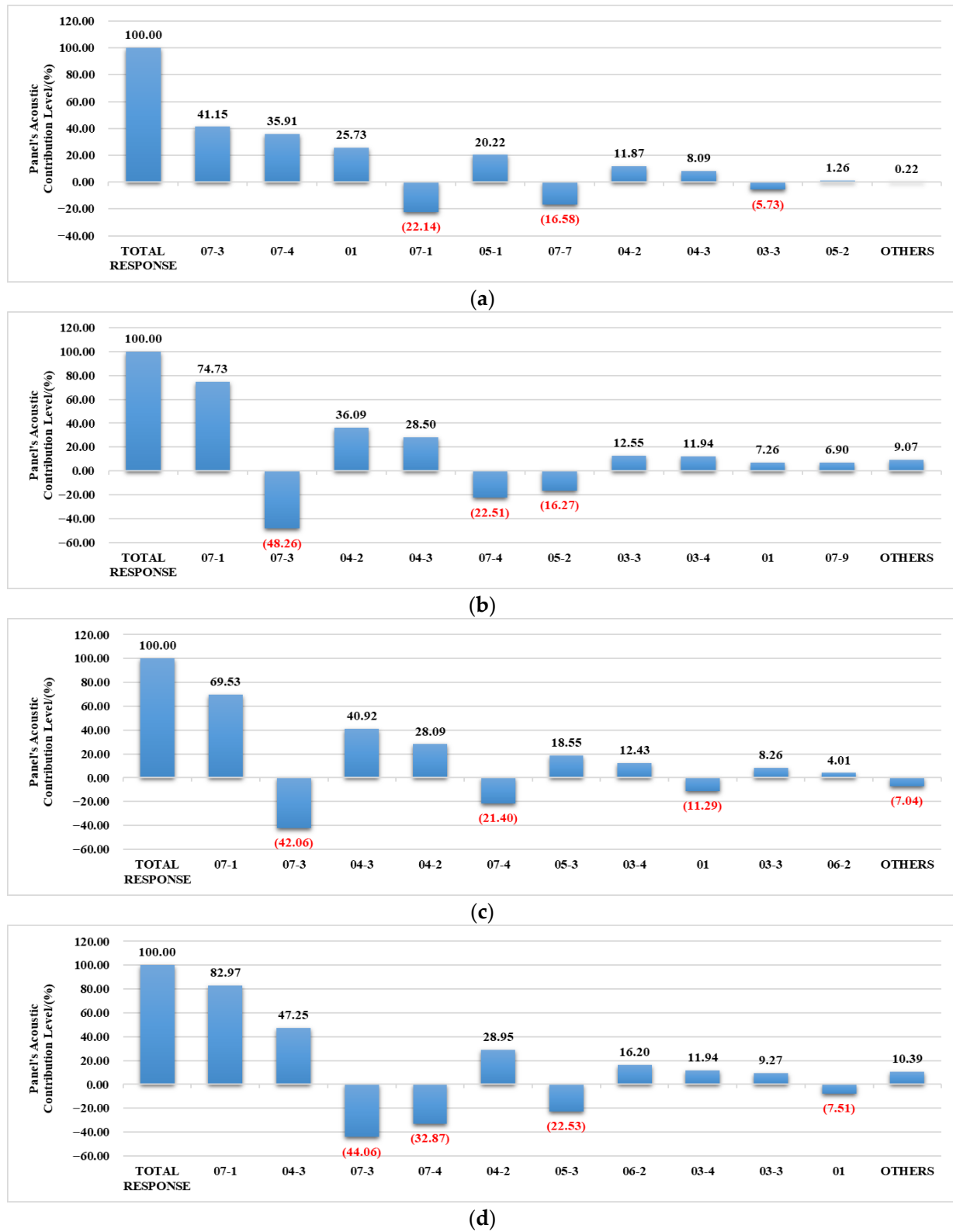


Figure 11. Panel Acoustic Contribution at 26.5 Hz: (a) Driver Ear Reference Point; (b) Occupant 1 Ear Reference Point; (c) Occupant 2 Ear Reference Point; (d) Occupant 3 Ear Reference Point.

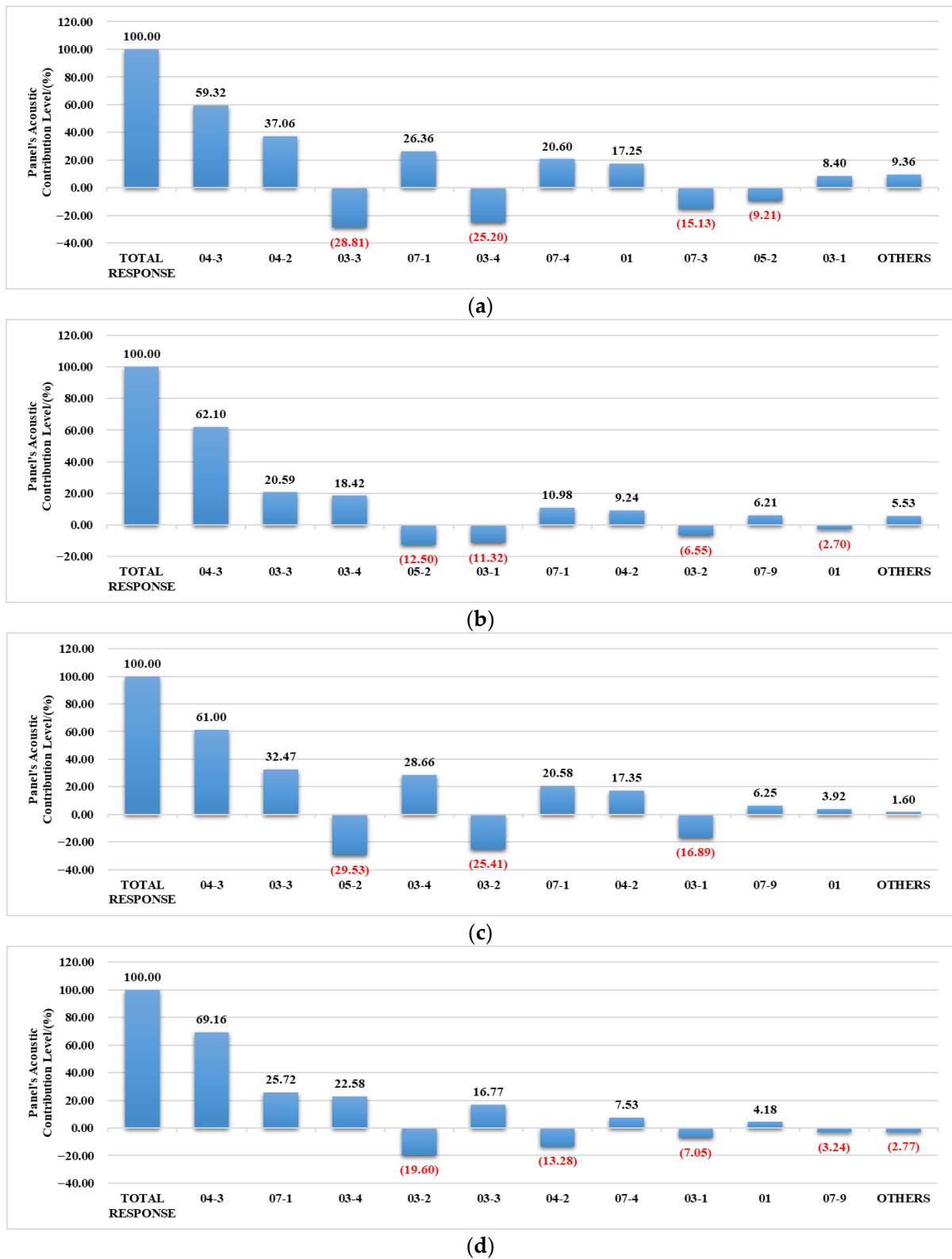


Figure 12. Panel Acoustic Contribution at 40 Hz: (a) Driver Ear Reference Point; (b) Occupant 1 Ear Reference Point; (c) Occupant 2 Ear Reference Point; (d) Occupant 3 Ear Reference Point.

The current PACA implementation evaluates the relative amplitude contributions of panels to the interior sound pressure levels. It should be noted that phase interactions between different panels are not considered in this analysis. As a result, potential constructive or destructive interference effects are not captured in the present results. While the

amplitude-only approach is sufficient to identify dominant panels and provide engineering guidance, future work will extend PACA to incorporate phase information for a more comprehensive understanding of panel interactions.

5. Conclusions

This study investigates noise responses at driver and occupant ear reference points under both unit harmonic force excitation and actual powertrain excitation through an acoustic–structural coupling model based on panel sound radiation theory. Contribution analysis of body panels was performed for dominant noise peak frequencies. Harmonic response analysis reveals significant transmission sensitivity at 26.5 Hz along the powertrain–body–ear vibration path, aligning with structural–acoustic coupling resonance characteristics. Frequency response analysis under powertrain excitation demonstrates strong correlation between operational peak frequencies and excitation characteristics: 40 Hz (idle), 93.3 Hz (maximum torque speed), and 140 Hz (rated speed). Notably, all operating conditions exhibit a pronounced 26.5 Hz peak beyond condition-specific frequencies, consistent with physical vehicle tests and validating the engineering applicability of the vibro-acoustic transmission mechanism. In summary, this work extends the PACA methodology to armored vehicles, experimentally verifies a 26.5 Hz structure–cavity resonance, and provides practical DVA placement guidelines, thereby contributing both theoretically and practically to the field of structural–acoustic dynamic analysis. Based on PACA results, conceptual guidelines for DVA implementation are provided. Panel 07-1 (bulkhead) should be targeted with a DVA tuned to 26.5 Hz, while Panel 04-3 (floor) should be treated with a DVA tuned to 40 Hz. Tuning bands of $\pm 5\%$ are recommended, with damping factors chosen to broaden effective bandwidth. These guidelines, though conceptual, provide clear engineering directions for practical NVH mitigation.

Targeting the coupled model's sensitive frequency (26.5 Hz) and idle-condition peak (40 Hz), panel contribution analysis identifies Panel 07-1 (partition between powertrain/occupant compartments) as the dominant contributor at 26.5 Hz, and Panel 04-3 (rear section of occupant compartment bottom deck) at 40 Hz. Future work will implement dynamic vibration absorbers (DVAs) on these critical panels for structure-borne noise suppression at target frequencies. Last but not the least, the present study focuses on amplitude-based PACA results. Phase effects, which may alter the superposition of panel contributions, were not included and will be addressed in future work. Incorporating phase-aware PACA is expected to provide deeper insight into noise formation mechanisms and more precise guidance for targeted NVH control.

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References

1. Prakash, V.; Sauvage, O.; Antoni, J.; Gagliardini, L.; Totaro, N. Stochastic assessment of electric powertrain whining noise under early-stage design uncertainties. *Proc. Inst. Mech. Eng. Part D J. Automob. Eng.* **2025**, *239*, 3246–3265. [\[CrossRef\]](#)
2. Lee, D.-H.; Hwang, W.-S.; Kim, C.-M. Design sensitivity analysis and optimization of an engine mount system using an FRF-based substructuring method. *J. Sound Vib.* **2022**, *255*, 383–397. [\[CrossRef\]](#)
3. Kim, Y.-D.; Oh, J.-E. Analysis of structure-borne noise in vehicles by modifying the stiffness of joint parts in the suspension system and vehicle body. *Noise Control Eng. J.* **2016**, *64*, 634–645. [\[CrossRef\]](#)
4. Mohammadi, S.; Ohadi, A.; Irannejad-Parizi, M. A comprehensive study on statistical prediction and reduction of tire/road noise. *J. Vib. Control* **2022**, *28*, 2487–2501. [\[CrossRef\]](#)
5. Ye, W.; Ma, Y. Design and performance study of acoustic packaging materials for biomimetic coupled sound absorption structure in automobiles. *Chem. Prod. Process Model.* **2025**. [\[CrossRef\]](#)
6. Van der Linden, P.J.G.; Varet, P. Experimental determination of low frequency noise contributions of interior vehicle body panels in normal operation. *SAE Trans.* **1996**, *105*, 175–180. [\[CrossRef\]](#)
7. Gielen, L.; Van der Linden, P.J.G.; Deges, R. Identification, Quantification and Reduction of Structural-Borne Road Noise in a Mid-Size Passenger Car. *SAE Trans.* **1996**, *105*, 181–188. [\[CrossRef\]](#)
8. Hendricx, W.; Choi, Y.B.; Ha, S.W.; Lee, H.K. Experimental body panel contribution analysis for road induced interior noise of a passenger car. SAE Noise and Vibration Conference and Exposition. *SAE Trans.* **1997**, *106*, 2711–2715.
9. Lu, C.; Liu, Y.; Liu, Z.; Zhou, Y.; Zhang, L. Optimization design for improving the vibration modes decoupling rate and vibration isolation performance of a powertrain mounting system based on the genetic algorithm. *J. Vib. Shock* **2018**, *37*, 248–253. [\[CrossRef\]](#)
10. Park, J.-Y.; Rajendra, S. Effect of non-proportional damping on the torque roll axis decoupling of an engine mounting system. *J. Sound Vib.* **2008**, *313*, 841–857. [\[CrossRef\]](#)
11. Si, Z.; Bai, X.; Qian, L.; Chen, P. Principle and control of active engine mount based on magnetostrictive actuator. *Chin. J. Mech. Eng.* **2022**, *35*, 146. [\[CrossRef\]](#)
12. Yuksel, E.; Kamci, G.; Basdogan, I. Vibro-acoustic design optimization study to improve the sound pressure level inside the passenger cabin. *J. Vib. Acoust.* **2012**, *134*, 061017. [\[CrossRef\]](#)
13. Kurian, A.G.; Thakur, S.K.; Kunde, S.; Sankpal, K.; Wagh, S. Challenges in Electric Vehicle NVH. In Proceedings of the 2021 IEEE Transportation Electrification Conference (ITEC-India), New Delhi, India, 16–19 December 2021; pp. 1–6.
14. Yang, B.; Liu, W.; Jin, Y.; Fu, B.; Qiu, M. Noise Test and Evaluation of Armored Vehicle Cabin. *J. Acad. Armored Force Eng.* **2017**, *31*, 52–55. [\[CrossRef\]](#)
15. Tamamura, M.; Shibata, E. Application of active noise control for engine related cabin noise. *JSAE Rev.* **1996**, *17*, 37–43. [\[CrossRef\]](#)
16. Durand, J.; Soize, C. Structural-acoustic modeling of automotive vehicles in presence of uncertainties and experimental identification and validation. *J. Acoust. Soc. Am.* **2008**, *124*, 1513–1525. [\[CrossRef\]](#)
17. Sung, S.; Nefske, D.J. Transfer path analysis of body panel participation using a structural-acoustic finite element model. In Proceedings of the ASME 2012 Noise Control and Acoustics Division Conference at Inter Noise 2012, New York, NY, USA, 19–22 August 2012; pp. 351–357.
18. Li, W.; Chai, Y.; Lei, M.; Liu, G. Analysis of coupled structural-acoustic problems based on the smoothed finite element method (S-FEM). *Eng. Anal. Bound. Elem.* **2014**, *42*, 84–91. [\[CrossRef\]](#)
19. Zhang, J.; Pang, J.; Wan, Y.; Yang, L.; Jia, W.; Zhang, S. Analysis of Structure–Acoustic Coupling Characteristics Between Adjacent Flexible Panels and Enclosed Cavity. *J. Vib. Acoust.* **2021**, *143*, 021006. [\[CrossRef\]](#)
20. Huang, X.; Huang, H.; Wu, J.; Yang, M.; Ding, W. Sound quality prediction and improving of vehicle interior noise based on deep convolutional neural networks. *Expert Syst. Appl.* **2020**, *160*, 113657. [\[CrossRef\]](#)
21. De Donno, G.; Ciravegna, E.; Galvagno, E.; Ferrari, S.; Pellegrino, G.; Lannone, M. Vibro-Acoustic Analysis of an Electric Powertrain: An Evaluation of the Relative Contribution of Gear Whine Noise and Electromagnetic Excitations on the Overall Response. *Mech. Mech. Sci.* **2024**, *163*, 410–418.
22. Gao, P.; Du, Y.; Zhou, H. An Analysis on Vibration Energy Flow for High Frequency Brake Squeal. *Automot. Eng.* **2018**, *40*, 590–596. [\[CrossRef\]](#)

23. Pang, J. *NVH Control of Automotive Body*; China Machine Press: Beijing, China, 2015.
24. Kushwaha, S.; Bagha, A.K. Application of composite materials for vibroacoustic—A review. *Mater. Today Proc.* **2020**, *26*, 1567–1571. [[CrossRef](#)]
25. Shu, L.; Wang, M.Y.; Ma, Z. Level set based topology optimization of vibrating structures for coupled acoustic–structural dynamics. *Comput. Struct.* **2014**, *132*, 34–42. [[CrossRef](#)]
26. Zhao, W.; Chen, L.; Chen, H.; Marburg, S. Topology optimization of exterior acoustic-structure interaction systems using the coupled FEM-BEM method. *Int. J. Numer. Methods Eng.* **2019**, *119*, 404–431. [[CrossRef](#)]
27. Kim, K.H.; Yoon, G.H. Optimal rigid and porous material distributions for noise barrier by acoustic topology optimization. *J. Sound Vib.* **2015**, *339*, 123–142. [[CrossRef](#)]
28. Shi, S.; Jin, G.; Xiao, B.; Liu, Z. Acoustic modeling and eigenanalysis of coupled rooms with a transparent coupling aperture of variable size. *J. Sound Vib.* **2018**, *419*, 352–366. [[CrossRef](#)]
29. Stanef, D.; Hansen, C.; Morgans, R. Active control analysis of mining vehicle cabin noise using finite element modelling. *J. Sound Vib.* **2004**, *277*, 277–297. [[CrossRef](#)]
30. Zhang, J. *Finite Element Analysis and ANSYS Workbench Engineering Applications*; Northwestern Polytechnical University Press: Xi'an, China, 2018.
31. *ISO 5128:2023*; Acoustics—Measurement of interior vehicle noise. International Organization for Standardization: Geneva, Switzerland, 2023.
32. *GB/T 18697-2023*; Acoustics—Measurement of Noise Inside Motor Vehicles. Standardization Administration of the People's Republic of China: Beijing, China, 2023.
33. *IEC 61672-1:2013*; Electroacoustics—Sound Level Meters—Part 1: Specifications. International Electrotechnical Commission: Geneva, Switzerland, 2013.
34. Zhang, S.; Zheng, Z.; Ding, W.; Huang, H.; Jiang, H.; Pang, J. Prediction of sound absorption and insulation of automobile acoustic packages based on support vector machine. In Proceedings of the 12th International Conference on Quality, Reliability, Risk, Maintenance, and Safety Engineering (QR2MSE 2022), Emeishan, China, 27–30 July 2022; Volume 556, p. 117702. [[CrossRef](#)]

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