

Article

Lightweight Design and Multi-Objective Optimization of E-Glass/Epoxy Composite Leaf Springs for Commercial Vehicles

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Abstract

To address the demand for lightweight commercial vehicle suspensions, this study investigates the replacement of traditional spring steel with E-glass fiber/epoxy composite materials. An equal-width, variable-thickness parabolic single-leaf spring was designed, with orthotropic mechanical properties obtained via ASTM standard tests. Finite element analysis (FEA) was combined with multi-objective optimization using a genetic algorithm, adjusting layup parameters to optimize stiffness, strength, and mass. Furthermore, to address the high failure risk at composite joints, a symmetric two-hole bolted end connection and a mid-span clamping structure were designed. The structural integrity was evaluated under vertical load, emergency braking, and steady-state cornering conditions using the Tsai–Wu tensor strength criterion. The optimization results demonstrate an 8.84% mass reduction for the composite spring main body compared to the initial design. The complete composite leaf spring assembly achieved approximately a 60.6% weight reduction relative to the original steel counterpart. The results indicate that the proposed design and optimization methodology effectively fulfills lightweighting objectives while satisfying all suspension performance and operational reliability requirements.

Keywords: composite leaf springs; lightweighting; multi-objective optimization; genetic algorithm; finite element analysis; joint design; Tsai–Wu criterion

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1. Introduction

With the continuous growth in global vehicle ownership, reducing energy consumption and mitigating environmental impacts have become paramount, making automotive lightweighting a critical development strategy [1]. As a core load-bearing and vibration-isolating component of suspension systems, the leaf spring accounts for 10–20% of a commercial vehicle's unsprung mass [2]. Replacing traditional steel leaf springs with fiber-reinforced polymer (FRP) composites can reduce weight by over 50% while

simultaneously enhancing fatigue endurance, corrosion resistance, and vibration damping performance [3]. Due to their superior specific strength, specific modulus, and excellent design flexibility, composite materials are transitioning rapidly from aerospace applications to mainstream automotive components, making composite leaf springs a highly promising solution for chassis lightweighting.

Recent research has extensively explored the structural design, material innovation, and performance optimization of composite suspensions. Comprehensive experimental and finite element analyses (FEA), such as those conducted by Ma et al. [4] on GFRP leaf springs, have established robust frameworks for evaluating composite structural designs. Concurrently, researchers are pushing the boundaries of material and manufacturing capabilities; for instance, Kader et al. [5] investigated the application of copolymer-reinforced composites specifically tailored for light-duty truck suspensions, while Kessentini et al. [6] demonstrated the viability of advanced additive manufacturing by optimizing a 3D-printed Carbon PEEK mono leaf spring. Building on these foundational approaches, researchers have actively investigated various optimization techniques to maximize weight reduction while maintaining structural integrity. For instance, Soliman [7] and Gaylo et al. [8] achieved significant mass reductions by applying topology optimization based on the Tsai–Wu tensor failure criterion. Advanced computational methods, including neural networks and genetic algorithms (GA), have also been widely employed to evaluate and optimize hybrid composite leaf springs based on theoretical and FEA results [9,10].

Despite these advancements, the conventional methods for designing and optimizing composite leaf springs are already well-established. However, a major engineering bottleneck remains: most idealized optimizations focus purely on the spring body itself, ignoring the complex stress states at the structural joint interfaces. Crucially, industrial field data show that over 60% of failures in composite suspensions happen prematurely at these mechanical connections rather than within the spring body itself [11,12]. Furthermore, existing studies mostly evaluate the structure under a single static vertical load, lacking a comprehensive check under extreme multi-directional conditions like emergency braking and steady-state cornering. Consequently, balancing target stiffness matching, multi-axial strength reliability, overall mass reduction, and joint interface safety remains a critical challenge.

Therefore, the main objective of this study is to propose a cooperative, holistic theoretical and numerical forward-design framework for an E-glass/epoxy composite leaf spring to systematically bridge the gap between idealized optimization and real-world operational reliability. Diverging from conventional studies that focus solely on the main laminate, this work cooperatively integrates experimental material characterization, multi-objective parameter optimization, and the explicit structural design of symmetric bolted end connections and mid-span clamping blocks. By using the Tsai–Wu strength criterion, this paper thoroughly evaluates the safety of both the spring body and its layer interfaces under extreme conditions like emergency braking and sharp turning. Through this approach, we establish a practical and reliable simulation method for designing composite suspensions. It should be noted that the current work focuses on rigorous theoretical modeling and finite element simulation. This research provides a necessary foundation for future prototype manufacturing, detailed layup adjustments, and physical bench test validations.

2. Materials and Methods

2.1. Materials and Fabrication

The composite leaf spring investigated in this study consisted of an epoxy resin matrix and a continuous unidirectional E-glass fiber reinforcement. The matrix material was a commercial bisphenol-A type epoxy resin (Model: E-51), and the reinforcement phase was continuous unidirectional E-glass fiber with an areal weight of approximately 800 g/m². All raw materials and prepregs were supplied by Guangxi Lanke Resource Recycling Co., Ltd. (Yulin, China).

The mechanical test specimens were manufactured using a standard compression molding process. During the fabrication, the prepregs were stacked strictly according to the designed unidirectional (0°) sequence. The molding process parameters were set as follows: a constant temperature of 120 °C and a pressure of 0.5 MPa were maintained for 2 h to ensure full cross-linking and curing of the resin system. After the curing process, the specimens were naturally cooled to room temperature within the mold before demolding. The final fiber volume fraction (V_f) of the prepared composite was controlled at approximately 60%, ensuring consistency between the experimental results and the subsequent theoretical models. The detailed material parameters of E-glass fiber and epoxy resin are listed in Table 1.

Table 1. Material parameters of E-glass fiber and epoxy resin.

Material	Modulus of Elasticity (GPa)	Poisson Ratio	Volume Ratio
E-glass fiber	72	0.3	0.6
Epoxy resin	3.3	0.36	0.4

2.2. Mechanical Characterization

To obtain the orthotropic mechanical parameters of the E-glass fiber/epoxy composite, quasi-static tensile, compressive, and in-plane shear tests were conducted strictly in accordance with international ASTM standards. The tests were performed on a universal testing machine equipped with specialized fixtures. Strain data were recorded in real-time using strain gauges and a PAK MKII data acquisition system supplied by BBM. The specific geometric dimensions of each specimen type are listed in Table 2. To overcome the inherent data scattering of composites and ensure the statistical reliability of the experimental results, a minimum of five identical and valid specimens ($n = 5$) were tested for each loading condition.

Table 2. Specimen dimensions.

Test Type	Ply Angle	Length	Test Type	Ply Angle
Stretch	0°	250	15	1
	90°	175	25	2
Compression	0°/90°	140	12	2
Shear	45°/−45°	250	25	5
Stretch	0°	250	15	1

- Tensile Testing

The tensile tests followed the ASTM D3039 standard [13] and included both 0° and 90° layup angles. Strain gauges were attached back to back along the longitudinal and transverse axes of the specimen's gauge section to monitor bending deformation during loading and to calculate Poisson's ratio. To strictly adhere to the standard specifications, the tensile loading rate was set at 2 mm/min. Furthermore, to eliminate grip slack, ensure perfect specimen alignment, and prevent slippage or eccentric loading during the initial

stage, an initial preload not exceeding 30% of the estimated failure load was applied prior to formal testing (5 kN for the 0° specimens and 0.3 kN for the 90° specimens [14]. The tensile test fixture for $0^\circ/90^\circ$ layups of glass fiber composite specimens is illustrated in Figure 1.



Figure 1. Tensile test fixture for $0^\circ/90^\circ$ layups of glass fiber composite specimens.

- Compressive Testing

The compressive tests were performed according to the ASTM D6641 standard [14,15]. A Combined Loading Compression (CLC) test fixture was utilized to smoothly introduce the load into the gauge section through a combination of end-loading and surface shear. Strain gauges were mounted back to back on the specimen surfaces to monitor compressive strain. To effectively prevent macroscopic buckling failure caused by excessively fast loading during compression, the loading rate was strictly set to 1 mm/min as required by the standard (which accounts for the difference in loading rates between the tensile and compressive tests). The compression test is shown in Figure 2.

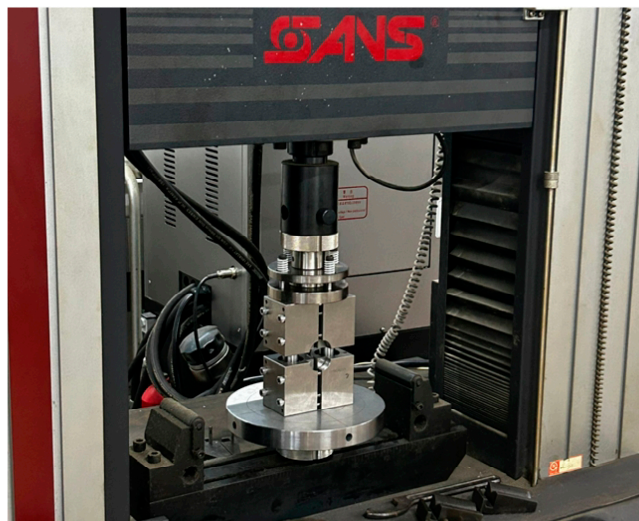


Figure 2. Compression test of glass fiber composite material.

- In-plane Shear Testing

The in-plane shear tests were conducted following the ASTM D3518 standard [16] using $\pm 45^\circ$ layup specimens. A longitudinal and a transverse strain gauge were attached to the center of the front surface, while a longitudinal strain gauge was attached to the back surface. The testing procedure was similar to the tensile test, with the loading rate also set at 2 mm/min. The 45° shear test is shown in Figure 3.



Figure 3. $\pm 45^\circ$ shear test of glass fiber composite material.

2.3. Structural Design and Analytical Modeling

Based on the technical parameters of a commercial vehicle front suspension leaf spring, the composite leaf spring main body is designed using an equal-width, variable-thickness parabolic structure, aiming for weight reduction and stiffness matching. The original front suspension consists of two steel leaf springs arranged symmetrically in the longitudinal direction, as shown in Figure 4. The first leaf has a rolled eye, and the second leaf has a wrapped eye. Load transfer between the two leaves is achieved via interleaved shims. Key parameters are given in Table 3.

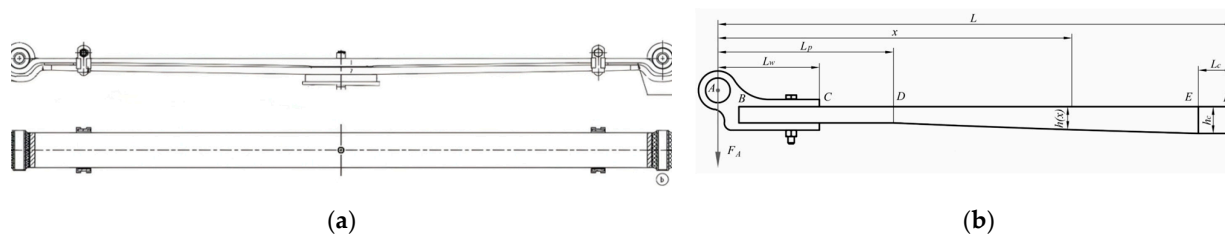


Figure 4. Structural configurations and design parameters of the suspension leaf springs: (a) Schematic layout of the original dual-leaf steel spring assembly; (b) Geometric parameters and functional zone definitions of the variable-thickness composite leaf spring.

Table 3. Original steel leaf spring specifications.

Parameters	Unit	Length
Total length	mm	1200
Width	mm	75
arc height	mm	80
Central clamping length	mm	140

Stiffness	N/mm	185 ± 15
Inner Diameter of Curled Edge	mm	43

To accurately establish the forward design and optimization model of the E-glass/epoxy composite leaf spring, the geometric topology and key design parameters are defined in Figure 4b. The total length of the leaf spring is denoted as L , while L_1 and L_2 represent the lengths of the front and rear segments from the center bolt, respectively. The total width of the leaf spring is defined as b . The spring adopts a variable-thickness parabolic profile to ensure a uniform strength distribution. The thickness along the longitudinal direction is symmetrically divided into three distinct zones:

Central Clamping Zone (EF): The flat segment in the middle with a constant thickness of h_c , extending over a length of L_c , which is rigidly fixed to the axle via U-bolts.

Parabolic Transition Zone (DE): The functional variable-thickness section where the thickness h_x decreases parabolically from the edge of the clamping zone toward the ends, matching the bending moment distribution.

End Connection Zone (BC): The flat segments at both ends with a constant thickness of h_e and a length of l_e , which are designed to accommodate the two-bolt end connection structures for shackle and hanger attachments.

According to the design theory of parabolic variable-section leaf springs [17,18] and the parameters listed in Table 3, the end clamping length is set to 150 mm. The end thickness h of the parabolic leaf is:

$$h = h_c \sqrt{\frac{150}{L-L_c}} \quad (1)$$

The overall stiffness K of the steel leaf spring is:

$$K = \frac{6EI_0\xi_i}{L^3[1+(\frac{L-L_c}{L})^3k]} \quad (2)$$

$$k = 1 - (h/h_c)^3 \quad (3)$$

$$I_0 = \frac{bh_c^3}{12} \quad (4)$$

In the formula, E represents the material's elastic modulus, ξ_i denotes the correction factor ($\xi_i = 0.92$), and I_0 signifies the second moment of area.

Substituting the target stiffness $K = 185$ N/mm into Equation (2), solve for the mid-thickness h_c and end thickness h . After rounding, $h_c \approx 33$ mm and $h \approx 18$ mm.

Given the free camber of 80 mm for the original steel leaf spring, the radius of curvature of the composite leaf spring in the free state is:

$$R \approx \frac{(2L)^2}{8 \cdot 80} \approx 2250 \text{ mm} \quad (5)$$

The initial structural design of the composite leaf spring main body is shown in Figure 5.

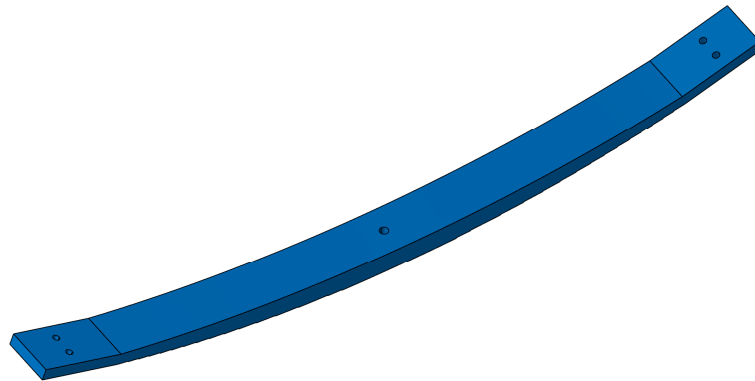


Figure 5. Initial design of the composite leaf spring main body.

2.4. Finite Element Modeling

During service, leaf springs primarily bear vertical loads. Under emergency braking and cornering, the end connections are additionally subjected to longitudinal and lateral loads due to inertia and centrifugal forces. Therefore, the design must consider these operating conditions and verify the spring strength using finite element analysis (FEA) to ensure that all performance requirements are satisfied.

Because a 0° fiber-direction lay-up of the composite provides the best strength as well as creep and fatigue resistance, and achieves the most effective lightweighting while maintaining performance [21], all composite plies in the leaf spring are therefore oriented at 0° .

Due to the initial parabolic profile of the leaf spring in its free state, a discrete material coordinate system is implemented in ABAQUS to precisely define the local material orientation for each element. As illustrated in Figure 6a, the local material 1-axis (longitudinal direction) is defined dynamically along the tangential direction of the spring's curvature. The local 3-axis is defined as the normal vector to the upper surface (stacking direction), and the local 2-axis represents the transverse direction (width direction) perpendicular to the fiber orientation.

Given the optimized mid-span thickness of 32 mm and a single-ply thickness of 1 mm, the leaf spring comprises a total of 32 plies modeled with 3D solid composite elements. The detailed layup manager configuration is depicted in Figure 6b. To replicate the variable-thickness geometry of the parabolic profile, a ply drop-off technique is employed. The individual plies are assigned to specific element sets: 17 long plies extend continuously across the full length of the spring to form the structural skin, whereas the remaining 15 short plies terminate progressively from the initiation of the parabolic section towards the spring ends, reducing the thickness to 17 mm at the joint zone. The material parameters derived from the experimental tests are assigned to the GFRP plies with a rotation angle of 0° . Furthermore, three integration points are specified through the thickness of each individual ply to accurately capture the structural stress and strain gradients induced by bending.

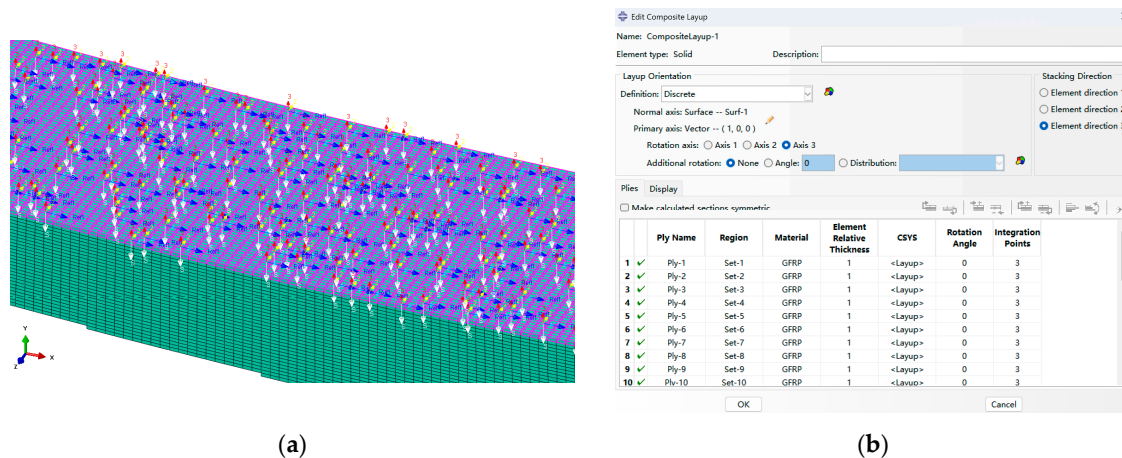


Figure 6. Schematic diagram of the local material coordinate system and layup configurations in the finite element model: (a) Visualization of the discrete local material coordinate system; (b) Configuration of the variable-thickness ply drop-off scheme inside the ABAQUS composite layup manager.

Mesh discretization accuracy and element size directly dictate the precision of numerical results. Selecting an appropriate element type is crucial for boosting both solution convergence and accuracy. Considering both efficiency and numerical fidelity, a hexahedral mesh layout was selected, and its quality was rigorously controlled. Based on the composite leaf spring topology, the 8-node linear brick element with reduced integration and enhanced hourglass control (C3D8R) was implemented. This element type exhibits high computational efficiency, stable convergence, and excellent resistance against shear locking, maintaining elevated precision even within slightly distorted mesh regions.

The geometric model of the composite leaf spring body was imported into HyperMesh 2022 for discretization. Given the anisotropic nature of the composite material and the subsequent analysis within Abaqus 2025, the mesh sweeping direction was maintained completely consistent with the laminate stacking sequence. Crucially, because the spring body features a parabolic variable-thickness topology, the staggered termination of internal plies (ply drop-offs) inevitably induces geometric non-continuity and localized interlaminar stress concentrations at the layer edges (free-edge effects). To faithfully capture and address this stress discontinuity phenomenon within the finite element framework without introducing excessive computational cost, the triangular matrix-filling parts between the ends of internal short plies were explicitly eliminated during the geometric preprocessing phase, enabling direct discretization of the actual, continuous step-like ply model. Furthermore, three integration points were specifically assigned through the thickness of each individual layer within the C3D8R solid element configuration.

During the subsequent post-processing phase in ABAQUS, the nodal stress averaging and smoothing algorithms across different composite layer boundaries were completely deactivated (Set to “No Averaging”). This operational setup ensures that the true, unsmoothed mathematical stress discontinuities at the sharp ply edges are transparently preserved rather than artificially obscured by numerical smoothing. The unaveraged multi-axial stress fields were then extracted layer by layer. The independent structural integrity and safety margins of each continuous and truncated ply were rigorously evaluated using the comprehensive Tsai–Wu tensor strength criterion, confirming that every single composite layer strictly satisfies the mechanical allowable requirements under all extreme operating conditions. The total number of elements for the leaf spring body is 366,480, with 394,692 nodes, and the localized mesh configuration is depicted in Figure 7.



Figure 7. Finite element model of the leaf spring body.

2.5. Multi-Objective Optimization Strategy

To thoroughly evaluate the influence of individual structural variables on the composite leaf spring's performance, it is necessary to conduct a sensitivity analysis prior to the multi-objective optimization. Normalized Sensitivity Coefficients (NSCs) will be calculated to quantify the relative impacts of key design parameters—namely, the parabolic starting position, total thickness, and width—on stiffness, the strength ratio, and overall mass. This analysis aims to identify the key variables, such as how total thickness correlates with stiffness, and how the parabolic starting position dictates mass reduction potential. By isolating these highly sensitive variables in advance, this pre-optimization step will effectively map the design space and provide a robust theoretical foundation for the subsequent Genetic Algorithm (GA) optimization, thereby ensuring higher computational efficiency and convergence.

The mechanical performance of the composite leaf spring is directly related to its layup scheme. When the ply angles and ply thicknesses are fixed, the layup configuration—namely the number of long plies, the number of short plies, and the length of each short ply—determines the performance of the leaf spring. In this study, a genetic algorithm is employed to optimize these structural parameters so that the comprehensive performance of the leaf spring in terms of stiffness, strength, and mass is brought to an optimum level [22, **Error! Reference source not found.**].

A multi-objective optimization model is formulated for the composite leaf spring with three objectives: stiffness, strength, and mass. Using a weighting coefficient approach, separate subobjective functions are defined for stiffness, strength, and mass, and appropriate weighting coefficients are assigned to control the evolutionary priority of each subobjective.

The design objective is to ensure that the stiffness of the composite leaf spring meets the target stiffness and the strength requirements under all load cases, while maximizing strength margin and minimizing mass. The design variables are chosen as the starting position of the parabolic section and the total number of plies.

For the parabolic starting positions L_p and L_c of the composite leaf spring, appropriate intervals are specified to avoid invalid optimization results: a certain length must be reserved at the ends for the installation of the spring eyes, and the middle section must provide sufficient length for fixation. Thus, L_p is taken in the range [100, 200] mm, and L_c in the range [70, 100] mm. The total thickness of the leaf spring is constrained to lie in the range [25, 40] mm.

The objective sub-functions for the optimization problem are defined as follows:

① Stiffness: According to the original steel leaf spring design drawing, the target stiffness is 185 ± 15 N/mm. Therefore, the stiffness optimization target for the composite leaf spring is 185 N/mm, and the allowable stiffness range is [170, 200] N/mm. The stiffness sub-objective function is expressed as:

$$f_k = \omega_k |K - K_{obj}| \quad (6)$$

In the expression, f_k denotes the stiffness sub-objective function of the composite leaf spring; ω_k is the weighting coefficient associated with the stiffness sub-objective; K is the optimized design stiffness; and K_{obj} is the target stiffness.

② Strength: To satisfy the operational strength requirements of the leaf spring, the strength ratio of each lamina in the composite leaf spring must be greater than 1. Taking the Vertical load case as the reference, the strength ratio of the composite leaf spring under this load case should exceed 1. The strength sub-objective function of the leaf spring is defined as:

$$f_R = \frac{\omega_R}{R} \quad (7)$$

In the expression, f_R denotes the strength sub-objective function of the composite leaf spring, and ω_R is the weighting coefficient of the strength sub-objective.

③ Mass: To achieve a higher degree of lightweight design, the mass is also taken as one of the optimization objectives. The corresponding objective function is:

$$f_m = \omega_m b t \rho \sum_{i=1}^n l_i \quad (8)$$

In the expression, f_w denotes the mass sub-objective function of the composite leaf spring; ω_m is the weighting coefficient of the mass sub-objective; b is the width of the leaf spring; t is the thickness of a single ply; ρ is the density of the composite material; and l_i is the length of each ply of the leaf spring.

The overall objective function of the composite leaf spring can thus be written as:

$$F_{obj} = \omega_k |K - K_{obj}| + \frac{\omega_R}{R} + \omega_m b t \rho \sum_{i=1}^n l_i \quad (9)$$

3. Results

3.1. Experimental Test Results

To accurately characterize the macroscopic mechanical response of the composite material and provide reliable constitutive data for subsequent structural-level finite element analysis, loading tests were conducted to failure in accordance with standard ASTM testing methods. The test results of the composites with different lay-up angles are shown in Figure 8. The light-colored thin lines in the figures represent the full set of valid specimen data within the same group, indicating excellent stability and repeatability of the tests, while the dark-colored thick lines represent the typical representative curves extracted for subsequent analysis.

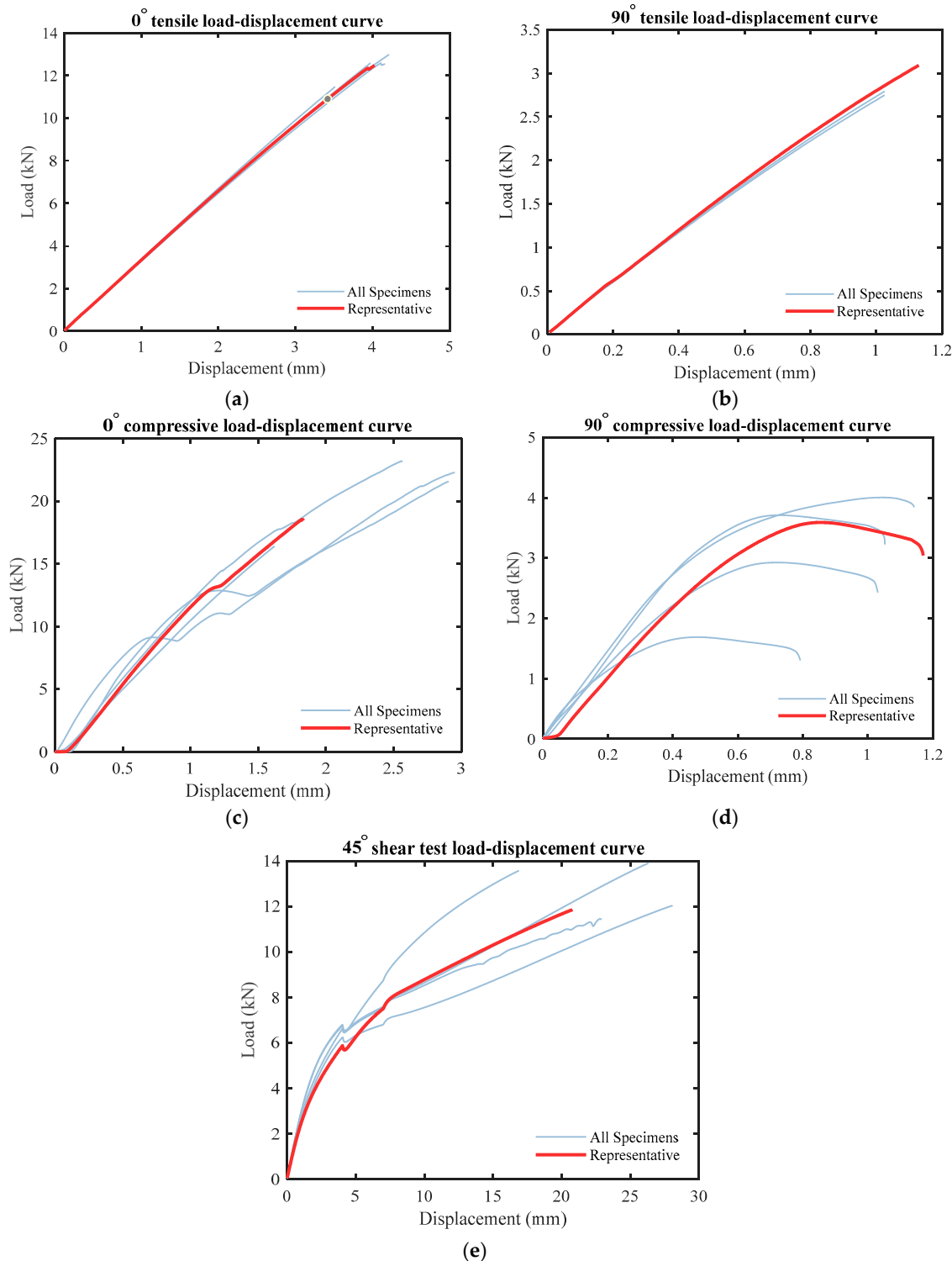


Figure 8. Experimental load–displacement curves of the composite under various loading conditions: (a) 0° longitudinal tensile test; (b) 90° transverse tensile test; (c) 0° longitudinal compressive test; (d) 90° transverse compressive test; and (e) 45° off-axis shear test.

The experimental results indicated that the failure modes of the specimens under each condition exhibited high consistency and conformed to typical composite failure characteristics. The failure modes of the specimens are shown in Figure 9.

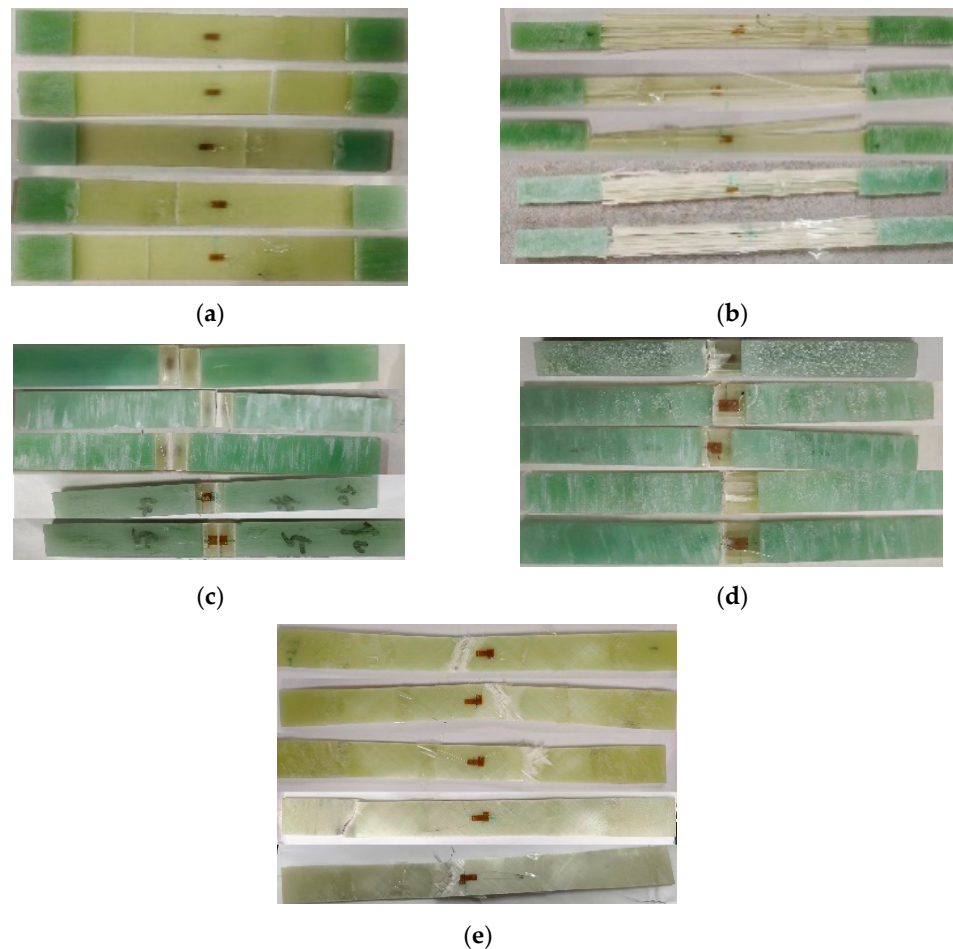


Figure 9. Specimen configurations and failure modes: (a) tensile failure of 0° specimens; (b) tensile failure of 90° specimens; (c) compressive failure of 0° specimens; (d) compressive failure of 90° specimens; (e) shear failure of $\pm 45^\circ$ specimens.

Tensile failure: The 0° tensile specimens primarily exhibited matrix cracking perpendicular to the loading direction and delamination in the thickness direction. The 90° tensile specimens experienced transverse brittle matrix fracture in the middle of the gauge section, with a relatively flat fracture surface and no obvious fiber bundle splitting.

Compressive failure: The 0° compressive specimens showed severe matrix fracture and longitudinal fiber splitting, accompanied by partial fiber detachment. The 90° compressive specimens exhibited penetrating matrix rupture in the thickness direction at the center of the gauge section.

Shear failure: The $\pm 45^\circ$ specimens demonstrated typical fiber-matrix shear failure. Fibers fractured along the $\pm 45^\circ$ layup direction at the fracture cross-section, and matrix failure led to partial fiber detachment. Additionally, obvious width shrinkage deformation was observed in the gauge section during loading.

After completing the mechanical tests, data from specimens with good repeatability are statistically averaged. In accordance with the relevant ASTM standards, the strengths, Poisson's ratios, and elastic moduli of the glass fiber composite laminates are calculated, as summarized in Table 4.

Table 4. Measured material parameters of E-glass fiber/epoxy laminate.

Material Parameters	Symbol	Unit	Value (Mean $\pm$ SD)
Longitudinal modulus of elasticity	E_1	MPa	$42,706 \pm 850$
Transverse modulus of elasticity	E_2	MPa	8171 ± 350
Shear modulus within a plane	G_{12}	MPa	4079 ± 180
Poisson ratio	ν_{12}	-	0.31 ± 0.02
Longitudinal tensile strength	X_t	MPa	866 ± 32
Longitudinal Compressive Strength	X_c	MPa	740 ± 28
Transverse tensile strength	Y_t	MPa	57.2 ± 4.5
Lateral Compressive Strength	Y_c	MPa	144.7 ± 8.6
In-plane shear strength	S	MPa	120 ± 6.5

Calculate the parameters of uniaxial plates based on viscoelastic mechanics of composite materials, using the formula with parameters provided by the manufacturer.

Apparent elastic modulus E_1 :

$$E_1 = E_f v_f + E_m (1 - v_f) \quad (10)$$

Apparent Poisson's ratio ν_{12} :

$$\nu_{12} = V_f \nu_f + V_m \nu_m \quad (11)$$

Apparent elastic modulus E_2 :

$$E_2 = \frac{E_f E_m}{E_f \nu_m + E_m \nu_f} \quad (12)$$

In the equation, E_f denotes the fiber elastic modulus, E_m denotes the matrix elastic modulus, V_f denotes the fiber Poisson's ratio, and V_m denotes the matrix Poisson's ratio. ν_f denotes the fiber volume fraction, and ν_m denotes the matrix volume fraction. The void space within the composite is neglected, assuming $\nu_f + \nu_m = 1$. The calculated material parameters are: $E_1 = 44.52$ GPa, $\nu_{12} = 0.324$, $E_2 = 7.79$ GPa. The deviations from experimental results are all within 5%, indicating that the experimental results are reasonable.

3.2. Results of Finite Element Analysis

3.2.1. Stiffness Analysis Results

Using the preprocessing settings described above, a full-load vertical load of 9200 N is applied at the reference point on the lower surface at the mid-span of the leaf spring. Translational freedom in the longitudinal (X) direction and rotational freedom about the Z axis through the eye centerline are retained, while all other degrees of freedom are constrained. The resulting vertical displacement contour is shown in Figure 10:

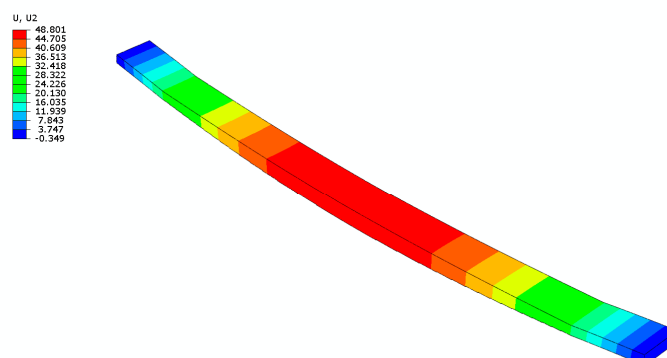


Figure 10. Vertical displacement contour of the composite leaf spring main body.

As illustrated in Figure 10, the vertical deflection decreases gradually from the center to both ends. The maximum vertical displacement at mid span is 48.8 mm. The corresponding vertical stiffness of the leaf spring main body is 188.52 N/mm, which meets the design target.

3.2.2. Strength Analysis Results

Based on the actual operating conditions of the vehicle, three critical load cases are considered to verify the leaf spring strength: a vertical load case, an emergency braking case, and a steady-state cornering case. These load cases are summarized in Table 5.

Table 5. Strength Analysis Conditions for Composite Leaf Springs.

Simulated Operating Conditions	Boundary Conditions	Load
Vertical load	The front/rear lugs retain X-axis longitudinal translation and Z-axis rotation; the central clamping region retains Y-axis vertical displacement.	Vertical load: 23,870 N
Emergency Braking	The front/rear lugs retain X-axis longitudinal translation and Z-axis rotation; the central clamping region retains Y-axis vertical displacement.	Static vertical load: 12,635 N Longitudinal braking force: 10,108 N
Steady-state turning operation	The front/rear lugs retain X-axis longitudinal translation and Z-axis rotation; the central clamping region retains Y-axis vertical displacement.	Combined vertical load: 14,051 N; Lateral centrifugal force: 5302 N

Specifically, for the end connection boundaries, the reference points coupled to the front and rear lugs explicitly retain the translational freedom along the longitudinal X-axis (U_x free) and the rotational freedom about the transverse Z-axis (UR_z free, representing the spring eye centerline). This boundary layout accurately replicates the pinning constraints of the real chassis shackle and hanger links, which allows the leaf spring to freely undergo horizontal compliance elongation and bending rotation under heavy load deflections. Concurrently, all other rigid translations and twisting rotations at the end connections are completely constrained to avoid unrealistic rigid-body movement. At the central clamping zone rigidly attached to the axle seat, the system retains vertical translation freedom along the Y-axis to undergo structural bending, where the corresponding validation vertical load (23,870 N), combined emergency braking forces (12,635 N vertical load paired with 10,108 N longitudinal braking force), and steady-state cornering load vectors (14,051 N vertical load co-applied with 5302 N lateral centrifugal force) are distributed via coupling interactions.

Based on the composite leaf spring finite element model established in Section 2.4, with the same boundary conditions and mesh parameters, the three limiting load cases specified in Table 5 are applied. The maximum stresses occur on the upper and lower surfaces of the leaf spring, where stress components in different directions are extracted; the resulting maximum stresses for the different fiber ply orientations from the finite element strength analysis are summarized in Table 6.

Table 6. Finite Element Strength Analysis Results of the Composite Leaf Spring.

Loading Condition	Fiber-Direction Stress (MPa)		Transverse-to-Fiber Stress (MPa)		In-Plane Shear Stress
	Maximum Tensile	Maximum Compressive	Maximum Tensile	Maximum Compressive	
Vertical load	647.5	657.7	36.29	41.31	33.25
Emergency braking	635.3	648.7	37.90	38.76	34.75
Steady-state cornering	430.2	437.6	25.55	41.22	23.70

According to the maximum stress failure criterion for composite materials, failure occurs when any stress component in a principal material direction reaches its corresponding strength. Therefore, ply wise unidirectional stresses must be compared against the basic strengths given in Table 2.

Under the vertical load case, deformation is dominated by bending, and stresses concentrate in the outermost plies. FEA shows that the maximum tensile stress in the fiber direction on the upper surface (PLY 1) is 647.5 MPa, and the maximum compressive stress in the fiber direction on the lower surface (PLY 33) is 657.7 MPa. The maximum transverse tensile and compressive stresses are 36.29 MPa and 41.31 MPa, respectively, and the maximum in-plane shear stress is 33.25 MPa. All these stress components are lower than the corresponding strengths listed in Table 2, indicating that the leaf spring main body satisfies the strength requirement under the vertical load case.

Under emergency braking, inertia causes the axle load to shift forward, significantly increasing the load on the rear half of the leaf spring. In this case, the spring is subjected to combined bending and longitudinal tension. The analysis results show that the maximum tensile stress in the fiber direction on the upper surface (PLY 1) is 635.3 MPa, and the maximum compressive stress on the lower surface (PLY 33) is 648.7 MPa. The maximum transverse tensile and compressive stresses are 37.90 MPa and 38.76 MPa, respectively, and the in-plane shear stress is 34.75 MPa. All these stresses remain below the material strengths, so the strength requirements are satisfied under emergency braking.

The composite leaf spring adopts a full 0° lay up, which fully exploits the high stiffness and strength in the fiber direction and contributes to weight reduction. Under the steady-state cornering case, the leaf spring is simultaneously subjected to a vertical load of 14,051 N and a lateral load of 5302 N. The lateral force induces additional stresses in the spring. The analysis shows that the maximum tensile and compressive stresses in the fiber direction are 430.2 MPa and 437.6 MPa, respectively. The maximum transverse tensile and compressive stresses are 25.55 MPa and 41.22 MPa, and the in-plane shear stress is 23.70 MPa. All these values are lower than the corresponding material strengths, indicating that the leaf spring satisfies the strength requirements under steady-state cornering.

Due to the large amount of finite element analysis results, only the overall fiber-direction stress of the leaf spring under each operating condition is presented here, as shown in Figure 11.

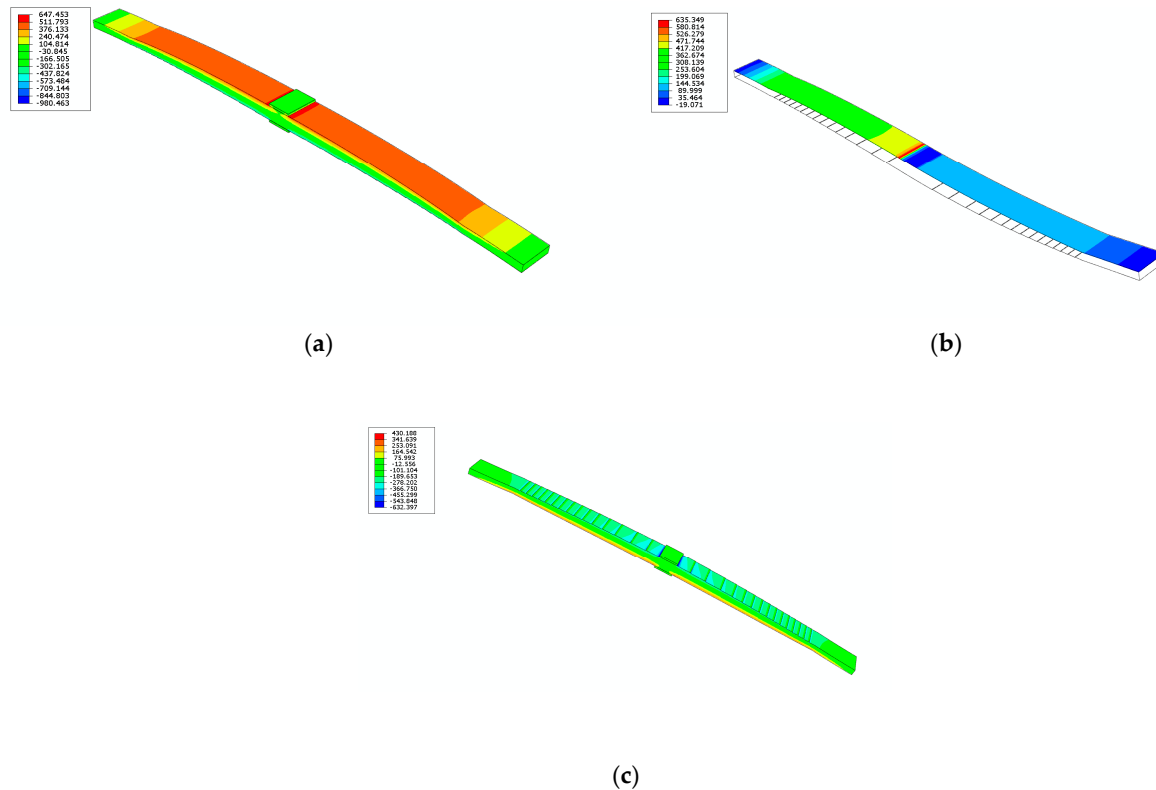
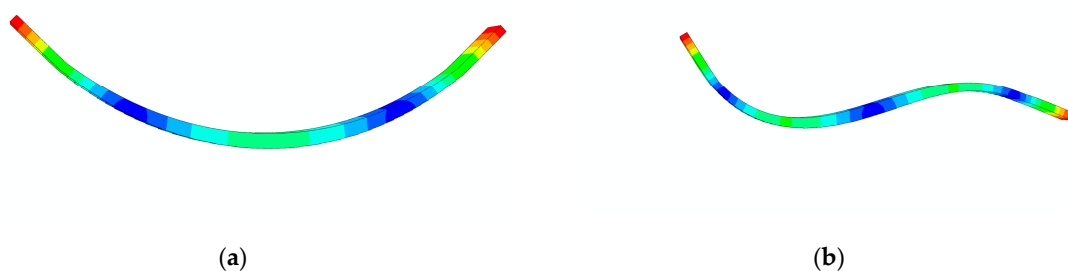


Figure 11. Strength Analysis of the Composite Leaf Spring Body. (a) Fiber-direction stress contour of the composite leaf spring under the vertical load condition; (b) Fiber-direction stress contour of the composite leaf spring under the emergency braking condition; (c) Fiber-direction stress contour of the composite leaf spring under the steady-state cornering condition.

3.2.3. Modal Analysis Results

The leaf spring is a key load bearing and vibration isolating component that is subjected to multiple excitation sources during service, including random road induced vibration and harmonic excitation from the engine. To avoid resonance caused by coupling between external excitation frequencies and the natural modes of the leaf spring, its modal characteristics must be evaluated at the design stage.

In this study, free vibration modes of the composite leaf spring are extracted; therefore, no boundary constraints are applied in the modal analysis. The Lanczos eigensolver is used to compute the first 20 natural modes. Since the excitation acting on the leaf spring during operation is mainly in the low frequency range, attention is focused on the low order modes. The first five non rigid modes are summarized in Table 7, and their mode shapes are illustrated in Figure 12.



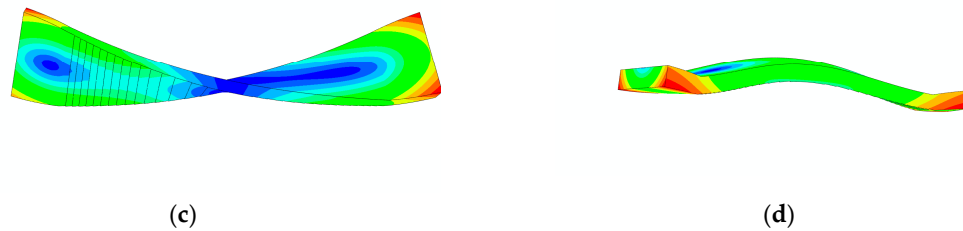


Figure 12. First four non rigid modes of the composite leaf spring main body. (a) First Bending Mode; (b) Second Bending Mode; (c) First Torsional Mode; (d) First Coupled Bending–Torsional Mode.

Table 7. Modal analysis results for the composite leaf spring main body.

Order	Frequency	Mode
First Order	107.34	First Bending
Second Order	233.65	Second Bending
Third Order	370.32	First Torsion
Fourth Order	396.50	First Bending–Torsion Coupled
Fifth Order	420.79	Third Bending

Table 7 presents the first five modes excluding the rigid-body mode, with modal shapes primarily consisting of bending, torsion, and bending–torsion coupling. Figure 12 illustrate the first four modes excluding the rigid-body mode.

Road excitation is determined by road roughness and vehicle speed [23]. For a vehicle traveling at normal speed on an ordinary road, the dominant excitation frequencies are typically in the range 1–3 Hz. At higher speeds on a highway, the road excitation frequency can reach approximately 15 Hz. Furthermore, a comprehensive dynamic evaluation must incorporate the high-frequency resonance domains within the vehicle suspension system, particularly the wheel hop mode induced by unsprung mass fluctuations. According to the classical 2-DOF quarter-car suspension vibration theory, the theoretical natural frequency of the wheel hop mode f_{wh} and the dominant engine excitation frequency f_e can be analytically estimated by:

$$f_{wh} = \frac{1}{2\pi} \sqrt{\frac{K_s + K_t}{m_u}} \text{ and } f_e = \frac{n \cdot M}{60 \cdot \tau} \quad (13)$$

where K_s is the suspension stiffness, K_t is the vertical tire stiffness, m_u is the unsprung mass, M is the number of cylinders, and n is the engine speed (r/min). Meanwhile, for the four-cylinder engine considered in this study ($\tau = 2$), with an idle speed of $n = 700$ r/min, the calculated engine excitation frequency is 23.33 Hz.

The first natural frequency of the composite leaf spring (107.34 Hz) is much higher than the road excitation frequencies, the formula-derived wheel hop resonance band (10–15 Hz), and the engine idle excitation frequency, exhibiting a massive frequency separation safety margin. This substantial frequency decoupling effectively isolates the leaf spring from capturing common road-induced unsprung resonance.

Additionally, the second bending mode of the composite leaf spring occurs at a more elevated spectrum (233.65 Hz). From the perspective of structural dynamics, the first bending mode possesses the dominant effective modal mass (exceeding 80% of the total structural mass), whereas the effective modal mass of the second bending mode drops drastically to a negligible level. In standard operating domains, transient excitation intensities originating from stochastic road profiles and high-order wheel-tire harmonics attenuate drastically above 100 Hz. Coupled with its minor modal mass participation, the

dynamic energy at this elevated frequency band is mathematically insufficient to actuate or resonate the higher-order bending shapes. In summary, resonance is highly unlikely across the full operating spectrum, and the dynamic structural design of the leaf spring is reasonable.

3.3. Theoretical Model of the Composite Leaf Spring

3.3.1. Establishment of the Stiffness Theoretical Model

Since the longitudinal dimension of the leaf spring is much greater than its thickness and width, the composite leaf spring can be idealized as an Euler–Bernoulli beam **Error! Reference source not found.**. Thus, when analyzing pure bending of a cross section, only the axial strain along the length direction of the leaf spring is considered, and other strain components are neglected. The vertical stiffness of the spring is therefore governed by the bending stiffness of the cross section.

For many materials, the tensile and compressive elastic moduli are not identical **Error! Reference source not found.**. This difference causes the neutral axis under bending to deviate from the geometric mid plane of the cross section and must be accounted for in the bending stiffness [26,27].

Define ξ as follows:

$$\xi = \frac{E_x^t E_x^c}{(\sqrt{E_x^t} + \sqrt{E_x^c})^2} \quad (14)$$

For a unidirectional composite plate, the bending stiffness at section x is given by:

$$K(x) = M_y(x) \rho(x) = \frac{bh(x)^3}{3} \xi \quad (15)$$

where b is the width of the composite leaf spring, $h(x)$ is the thickness at section x , ε_x is the strain along the x -axis at that position, and E_x^t and E_x^c are the tensile and compressive moduli in the fiber (x -axis) direction, respectively.

According to the actual working condition of the composite leaf spring, half of it can be simplified as a cantilever beam for analysis, as shown in Figure 13.

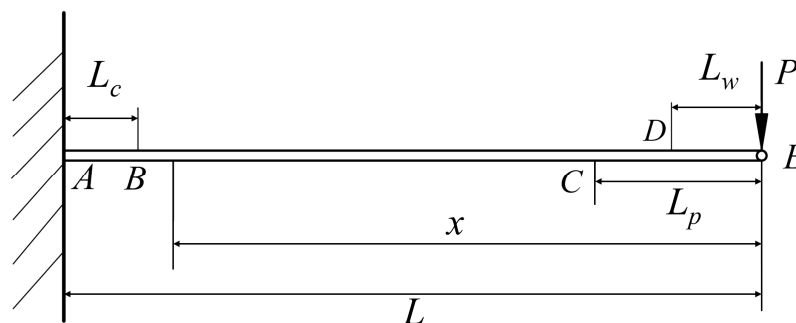


Figure 13. Simplified cantilever model of the composite leaf spring.

When a load P is applied at the end, the bending moment at section x is:

$$M_x = Px \quad (16)$$

The bending strain energy of the cantilever beam is:

$$U = \int_0^L \frac{(M_x)^2}{2K(x)} dx \quad (17)$$

Since the DE and AB segments are clamping sections, they are treated as non-bending deformation segments, and thus the change in bending strain energy for these two segments is neglected. Therefore, the above equation can be written as:

$$U = \int_{L_w}^{L_p} \frac{(M_x)^2}{2K(x)} dx + \int_{L_p}^{L-L_c} \frac{(M_x)^2}{2K(x)} dx \quad (18)$$

Further expressing the bending strain energy as:

$$U = \frac{P^2 \lambda}{2b\xi h_c^3} \quad (19)$$

where λ is the structural coefficient, expressed as:

$$\lambda = 2(L - L_c)^3 - \sqrt{\left(\frac{L - L_c}{L_p}\right)^3 (L_p^3 + L_w^3)} \quad (20)$$

The work done by load P is:

$$W = \frac{1}{2} P \delta \quad (21)$$

where δ is the vertical displacement under the action of load P .

According to the law of conservation of energy, the strain energy of an elastic body is equal to the work done by the external force. Therefore, the deflection of the leaf spring is:

$$\delta = \frac{P \lambda}{b \xi h_c^3} \quad (22)$$

The composite leaf spring studied in this paper has a symmetrical structure; therefore its stiffness is:

$$K_s = \frac{2P}{\delta} = \frac{2b\xi h_c^3}{\lambda} \quad (23)$$

The initial design parameters for the composite leaf spring are shown in Table 8.

Table 8. Initial design parameters of the composite leaf spring.

Parameters	Parameter Value	Parameters	Parameter Value
L (mm)	600	L_p (mm)	130
b (mm)	75	L_w (mm)	90
h_c (mm)	33	L_c (mm)	70

Substituting the initial design parameters into Equation (23), the theoretical stiffness of the composite leaf spring is obtained as 193.84 N/mm, and the error compared with the stiffness obtained from the finite element analysis is 2.82%.

3.3.2. Establishment of the Strength Theoretical Model

The strength model is also based on the cantilever beam idealization described in Section 3.3.1. For any cross section, the axial stress in each ply is proportional to its distance from the neutral axis in the thickness direction; thus, the maximum stresses occur in the outermost plies at the upper and lower surfaces [28].

When the free end of the composite leaf spring is subjected to a load and vertical bending occurs, the upper and lower surfaces are subjected to tensile and compressive stresses, respectively. According to Equations (15), (16), (22) and (23), the tensile stress in the ply on the upper surface along the length direction of the leaf spring is:

$$\sigma_x^t = E_x^c \varepsilon_x^c = E_x^c \frac{h_n^c(x) M(x)}{K(x)} = \frac{3PL(\sqrt{E_x^t} + \sqrt{E_x^c})}{bh_c^2 \sqrt{E_x^c}} \quad (24)$$

The corresponding compressive stress on the lower surface is denoted as:

$$\sigma_x^c = E_x^c \varepsilon_x^c = E_x^c \frac{h_n^c(x) M(x)}{K(x)} = \frac{3PL(\sqrt{E_x^t} + \sqrt{E_x^c})}{bh_c^2 \sqrt{E_x^t}} \quad (25)$$

Based on Equations (24) and (25), the theoretical maximum longitudinal tensile and compressive stresses of the composite leaf spring under the Vertical load condition are

calculated to be 574.72 MPa and 544.98 MPa, respectively. These values deviate from the finite element analysis (FEA) results by 11.2% and 17.1%, respectively. The primary reasons for these discrepancies are twofold: (1) The theoretical strength model assumes ideal orthotropy, thereby neglecting the interlaminar shear effects and transverse non-linear deformation inherent in composite materials. (2) The theoretical model simplifies the leaf spring into a one-dimensional beam with ideal cantilever boundary conditions. This simplification ignores the parabolic variable-thickness profile of the spring body and fails to capture the localized stress concentrations at the central clamp, end connections, and the internal ply drop-off boundaries (transitions between long and short plies).

To assess the practical utility of the proposed theoretical model, a quantitative breakdown of the maximum 17.1% deviation between the 1D analytical strength results and the 3D FEA data is conducted. This total discrepancy stems from three primary physical and numerical sources:

(1) Geometric discontinuities at ply drop-offs contribute approximately 7.5% of the deviation. The 3D FEA mesh explicitly removes the triangular matrix-filling pockets to capture the actual step-like ply boundaries. This sharp geometry induces severe localized out-of-plane shear stress concentrations, which are mathematically smoothed out in the 1D analytical model. A similar stress non-continuity and free-edge phenomenon was well-documented by Jiang et al. [29].

(2) Transverse shear deformation accounts for roughly 5.5% of the variance. The classical kinematic assumptions in the 1D formulation omit transverse shear strains across thick sections, whereas the C3D8R solid elements dynamically resolve the full shear compliance, as discussed in the multi-axial deformation analysis by Liu et al. [30].

(3) Material coupling effects contribute the remaining 4.1% of the error budget. This stems from the full orthotropic properties (E_1 , E_2 , E_3 , ν_{ij} , G_{ij}) inside ABAQUS under limit loads compared to the singular longitudinal modulus E_1 used in the 1D model.

Crucially, as demonstrated in the structural evaluation benchmarks by Olsson [31], a localized peak discrepancy of 15–17% between analytical closed-form predictions and full numeric models is a classic and widely accepted phenomenon in composite sizing. Because our theoretical framework maintains a global average error of less than 8.5% across the entire spring length, it provides sufficient accuracy for rapid preliminary sizing during the forward-design phase.

3.3.3. Strength Evaluation Criteria for the Composite Leaf Spring

Common strength evaluation criteria for composite materials include the maximum stress criterion, maximum strain criterion, Tsai–Hill criterion, Hoffman criterion, and the Tsai–Wu tensor strength criterion [31]. In the macroscopic initial screening of the composite leaf spring body, the maximum stress criterion serves as an efficient decoupled gauge to rapidly identify single-layer failure modes under single-axis primary loads. However, near the bolted end connections and geometric discontinuities such as ply drop-offs, the structural constraint and material anisotropy induce a severe, highly localized multiaxial stress state. Within these critical junction zones, the material degrades due to the complex interactive coupling between longitudinal, transverse, and interlaminar shear components. Among the aforementioned frameworks, the Tsai–Wu criterion successfully combines multiple strength criteria and inherently accounts for strength cross-coupling under such multiaxial stress states. Therefore, to properly capture the cross-coupling interactive damage and further verify the high-fidelity strength safety of the composite leaf spring under localized multiaxial stress, the Tsai–Wu criterion is strategically adopted to evaluate the plies with maximum stress in the secondary refinement FEA.

The strength ratio is given by:

$$R = \frac{[\sigma_i]}{\sigma_i} \quad (26)$$

In the formula, σ_i is the actual stress component of the lamina, and $[\sigma_i]$ is the ultimate stress corresponding to the σ_i component.

The strength ratio R is equivalent to the safety factor of the material under a given load. When $R > 1$, the actual stress is lower than the ultimate stress, the material does not fail, and R serves as a measure of the safety margin. When $R = 1$, the actual stress equals the ultimate stress and the material reaches its limit state. When $R < 1$, R has no physical meaning. If $R \rightarrow \infty$, the actual stress is zero. If the actual stress is 1, the strength ratio R is equal to the ultimate stress.

The ply-level strength ratio equation for the Tsai–Wu tensor strength criterion is:

$$(F_{11}\sigma_1^2 + 2F_{12}\sigma_1\sigma_2 + F_{22}\sigma_2^2 + F_{66}\tau_{12}^2)R^2 + (F_1\sigma_1 + F_2\sigma_2)R = 1 \quad (27)$$

In this equation, σ_i and τ_{ij} are the stresses in each direction, and F_{ij} are the material strength parameters, as given by:

$$\begin{cases} F_1 = \frac{1}{X_t} - \frac{1}{X_c}, & F_{11} = \frac{1}{X_t X_c}, & F_2 = \frac{1}{Y_t} - \frac{1}{Y_c} \\ F_{22} = \frac{1}{Y_t Y_c}, & F_{66} = \frac{1}{S^2}, & F_{12} = -\frac{1}{2} \sqrt{\frac{1}{X_t X_c Y_t Y_c}} \end{cases} \quad (28)$$

where X_t is the longitudinal tensile strength of the material, X_c is the longitudinal compressive strength, Y_t is the transverse tensile strength, Y_c is the transverse compressive strength, and S is the in-plane shear strength of the material.

From Equation (23), the strength ratio of a composite lamina under the given stress state is:

$$R = \frac{-B \pm \sqrt{B^2 + 4A}}{2A} \quad (29)$$

Among them,

$$\begin{cases} A = F_{11}\sigma_1^2 + 2F_{12}\sigma_1\sigma_2 + F_{22}\sigma_2^2 + F_{66}\tau_{12}^2 \\ B = F_1\sigma_1 + F_2\sigma_2 \end{cases} \quad (30)$$

Based on the finite element analysis results, a strength check is carried out for the ply with the maximum stress in the composite leaf spring, namely the plies on the upper and lower surfaces of the spring. From Equation (29), the strength ratio under the Vertical load condition is 1.318, under the emergency braking condition is 1.335, and under the steady-state cornering condition is 1.408. According to the Tsai–Wu tensor strength criterion, the strength ratios of the most highly stressed ply in the composite leaf spring are all greater than 1 under the limiting load cases; therefore, the composite leaf spring does not fail and satisfies the performance requirements.

3.4. Multi-Objective Optimization and Finite Element Analysis Results

3.4.1. Sensitivity Analysis Results

The parameters capable of adjusting the stiffness of the composite leaf spring include the parabolic starting position, total thickness, and width. Due to installation space constraints on the vehicle, the width must not exceed that of the original steel leaf spring. Nevertheless, analyzing the width parameter reveals the underlying relationship between width and stiffness, providing valuable theoretical guidance for the structural design. Based on the theoretical stiffness model established in previous sections, the influence of each selected parameter on the stiffness of the composite leaf spring was individually analyzed. The results are illustrated in Figure 14.

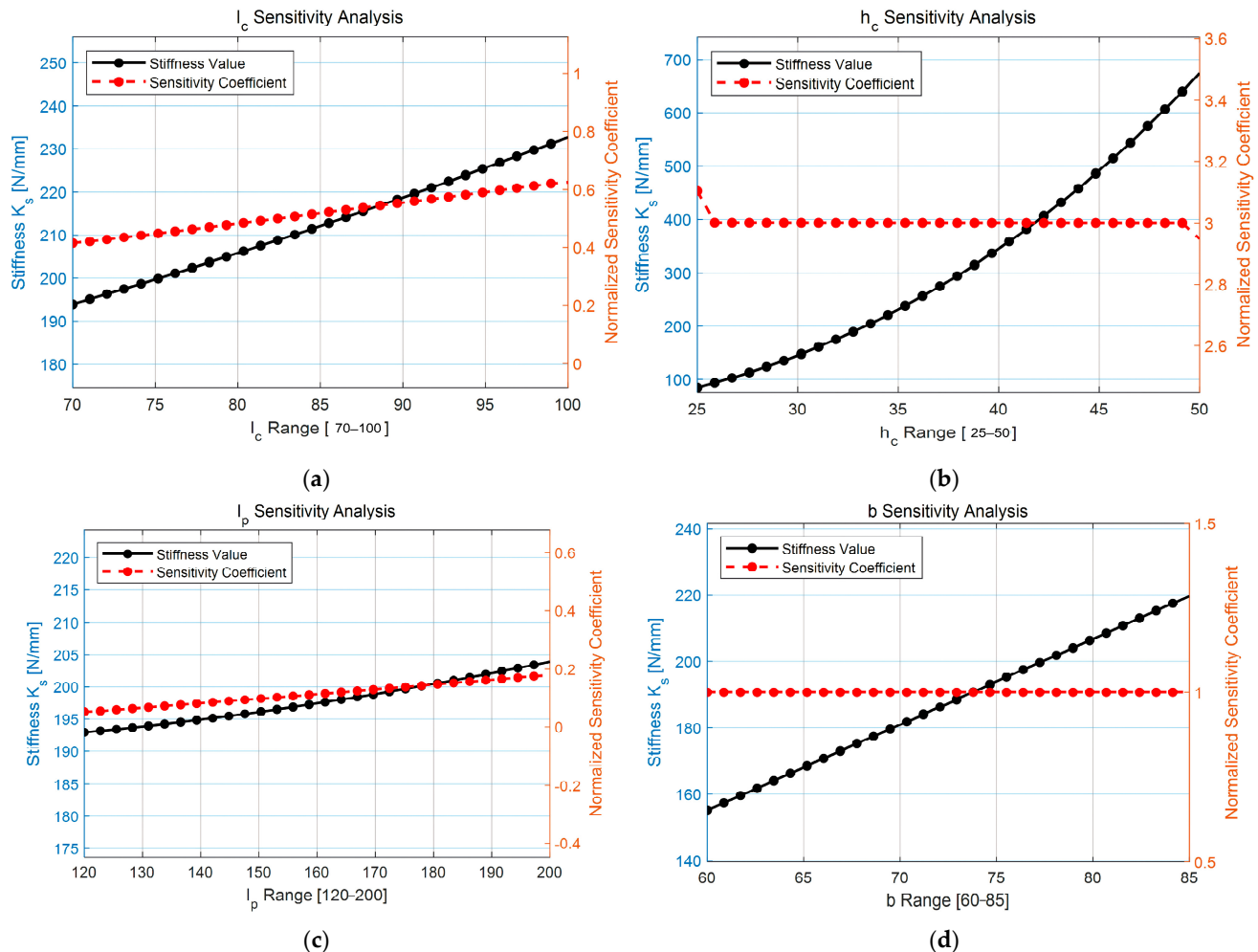


Figure 14. Influence of different structural parameters on the stiffness of the composite leaf spring: (a) parabolic starting position; (b) parabolic ending position; (c) total thickness; (d) width.

The normalized sensitivity analysis results reveal significant differences in the impact of various structural parameters on the stiffness of the composite leaf spring. The total thickness exerts the most profound effect on stiffness (with a sensitivity coefficient of approximately 3, indicating a super-linear response), making it the primary parameter for substantial stiffness adjustments. The width exhibits a linear influence on stiffness (with a coefficient of 1); however, restricted by the vehicle's installation space, its maximum value must not exceed that of the original steel leaf spring to prevent structural interference. Conversely, the parabolic starting and ending positions have a relatively minor impact on stiffness, demonstrating sub-linear responses with sensitivity coefficients ranging from 0.41 to 0.63 and 0.05 to 0.20, respectively. Therefore, in practical engineering matching design, adjusting the total thickness should serve as the principal method for modifying stiffness, supplemented by appropriate width selection. Finally, fine-tuning the stiffness by altering the parabolic starting position ensures a precise stiffness match between the composite leaf spring and the original steel counterpart.

3.4.2. Multi-Objective Optimization Results

The optimization model, together with the stiffness and strength models, is implemented in MATLAB (version R2024a), and the GA toolbox is invoked to perform

optimization. The convergence histories of stiffness and strength ratio are shown in Figure 15 and Table 9.

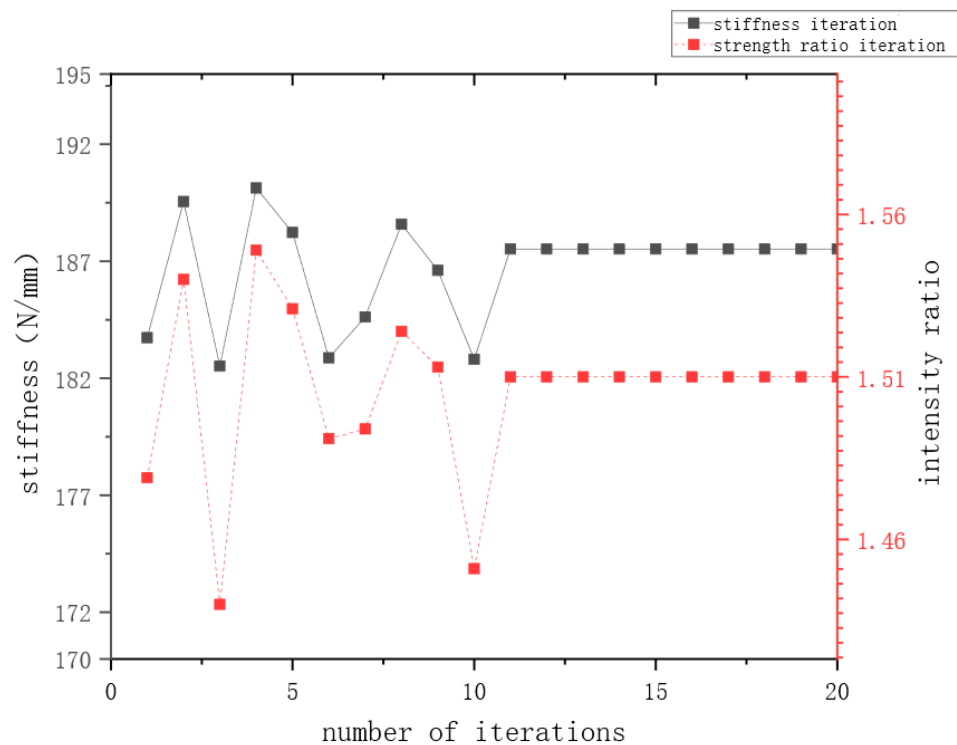


Figure 15. Multi-objective optimization results of the composite leaf spring. Solid line: stiffness iterative optimization curve; dashed line: strength-ratio iterative optimization curve.

Table 9. Comparison of the composite leaf spring before and after optimization.

Parameters	Initial Value	Optimized Value	Rate of Change
Total number of plies	33	32	−3.0%
Parabola start position	130 mm	142 mm	+9.2%
Stiffness	188.5 N/mm	181.75 N/mm	−3.6%
Strength ratio in verification condition	1.32	1.35	+2.3%
Mass	4.75 kg	4.33 kg	−8.8%

The optimized stiffness of the composite leaf spring is 181.75 N/mm, which lies within the allowable range and is close to the target stiffness. Under the Vertical load, the strength ratio increases to 1.35, satisfying the strength requirement. These strength values are based on the theoretical model and therefore contain some model error; the subsequent FEA results are taken as the final reference.

After optimization, the total number of plies in the composite leaf spring is 32, including 17 long plies, and the mass of the composite leaf is 4.33 kg, which is reduced by about 80% compared with the original steel leaf spring mass (21.6 kg).

3.4.3. Finite Element Analysis of the Optimized Composite Leaf Spring

To verify that the optimized design meets performance requirements under all working conditions, FEA is performed for stiffness, strength, and modal characteristics. The model setup, boundary conditions, and loads are the same as in Section 2.

- Stiffness analysis

The displacement contour under the Vertical load is shown in Figure 16.

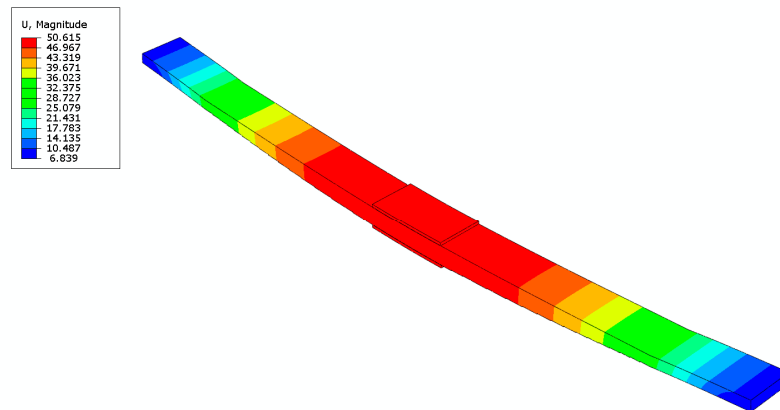


Figure 16. Vertical displacement contour of the optimized composite leaf spring.

The maximum mid-span displacement is 50.26 mm, corresponding to a stiffness of 181.75 N/mm.

- Strength analysis

Under the three extreme operating conditions (Vertical load, emergency braking, steady-state cornering), the stress state of the most highly stressed ply on the upper and lower surfaces is extracted. Table 10 summarizes the maximum stresses and Tsai–Wu strength ratios.

Table 10. Strength analysis results of the optimized leaf spring under each operating condition.

Operating Condition	Fiber-Direction Stress (MPa)	Stress Perpendicular to Fiber Direction (MPa)	In-Plane Shear Stress (MPa)	Tsai–Wu Strength Ratio
Vertical load	Tension 644.7/Compression 629.4	Tension 36.1/Compression 40.8	32.9	1.345
Emergency braking	Tension 606.3/Compression 599.8	Tension 37.9/Compression 38.8	34.8	1.429
Steady-state cornering	Tension 390.5/Compression 377	Tension 25.7/Compression 40.4	23.7	1.493

For all load cases, the maximum tensile and compressive stresses in the fiber-direction stress, transverse stresses, and in-plane shear stresses are lower than the allowable values of the composite ($X_t = 866$ MPa, $X_c = 740$ MPa, etc.). The corresponding Tsai–Wu strength ratios are all greater than 1, indicating no risk of failure.

Since a large number of finite element analysis results are obtained, only the fiber-direction stress results on the surface of the leaf spring under each working condition are presented here, as shown in Figure 17.

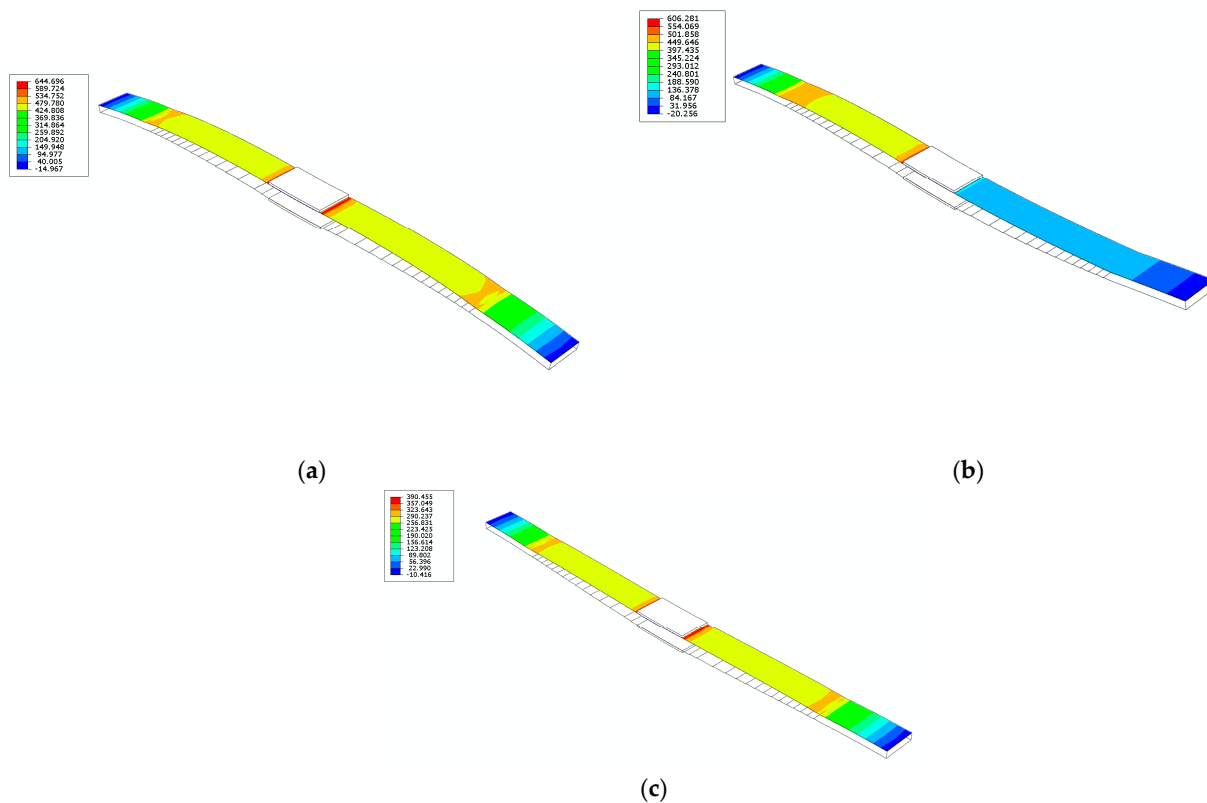


Figure 17. Strength analysis results of the optimized composite spring main body. (a) PLY-1 fiber-direction stress contour under vertical load condition; (b) PLY-1 fiber-direction stress contour under emergency braking condition; (c) PLY-1 fiber-direction stress contour under steady-state cornering condition.

- **Modal Analysis**

The same modal analysis procedure as in Section 3.2.3 is applied to the optimized spring. Excluding rigid body modes, the first five modes are listed in Table 11.

Table 11. Results of Modal Analysis After Optimization.

Order	Frequency	Mode
First Order	102.75	First Bending
Second Order	216.70	Second Bending
Third Order	362.08	First Torsion
Fourth Order	393.47	First Bending–Torsion Coupled
Fifth Order	398.94	Third Bending

Compared with the initial design, the fourth and fifth mode shapes appear in reversed order, while other mode shapes remain unchanged. The frequency changes for each mode are small.

After optimization, stiffness decreases by 3.59% but remains within the design range. The reduction is mainly due to the decreased thickness. The strength ratios increase by 2.05% (Vertical load), 7.04% (emergency braking), and 6.04% (steady-state cornering), improving operational reliability. The first bending frequency decreases by 4.27% but remains far above the main external excitation frequencies. The mass of the leaf is reduced by 8.84%.

4. Design and Analysis of the Connecting Structure

4.1. Design of the End Connection Structure

The connecting structure of the composite leaf spring includes the end (eye) connection and the mid-span clamping region. Design of the end connection structure of the composite leaf spring should follow several key principles. First, good interchangeability with the original steel spring must be ensured, avoiding interference between the end connections and other chassis components. Second, the connection structure should minimize manufacturing cost while meeting operational reliability requirements and maintaining good manufacturability [32].

It is reported that more than 60% of failures in composite structures occur at connections; therefore, connection design is critical for the composite leaf spring.

End connections of composite components generally refer to joints between composite parts, or between composite and metal parts. They are typically realized either by adhesive bonding or by mechanical fastening. Common structural forms of composite leaf spring end connections can be classified into four types, as illustrated in Figure 18.

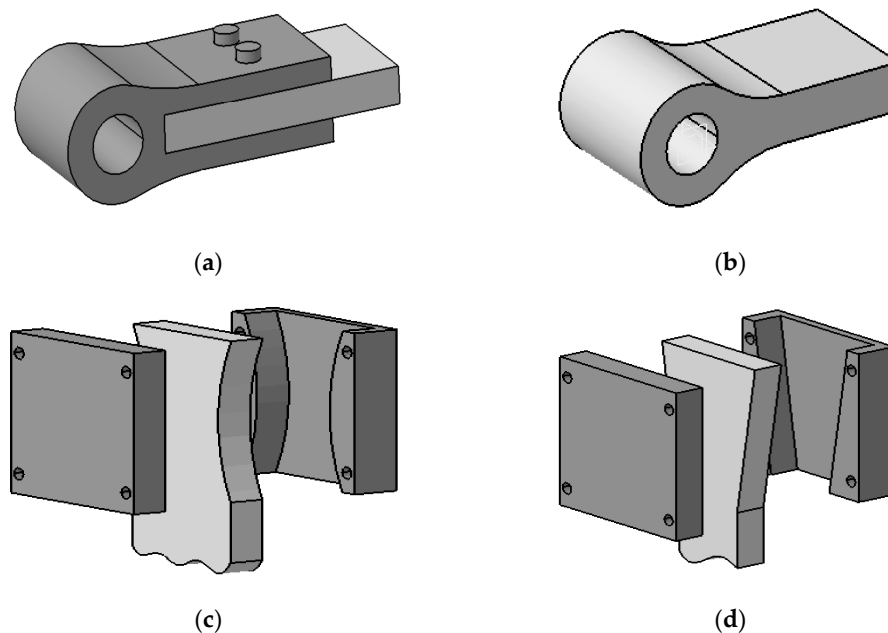


Figure 18. Typical end connection configurations for composite leaf springs. (a) bolted metal eye; (b) integral composite eye; (c) concave-surface clamped joint; and (d) beveled-surface clamped joint.

In configuration (a), a metal eye is bolted to the composite leaf body. Configuration (b) uses an integral composite eye, which avoids the stress concentration associated with drilled holes but involves a more complex manufacturing process and higher cost. Configurations (c) and (d) use concave surfaces or bevels machined into the spring body, together with matching metal fittings and upper and lower clamp plates. These forms also entail relatively complex processes and higher cost.

Compared with the other three configurations, configuration (a) has lower cost, is widely used in composite connection design, and has a mature design methodology. Therefore, configuration (a) is selected for the end connection of the composite leaf spring.

Bolts in the connection are typically arranged in one of three patterns: two symmetric bolts, three bolts in a triangular pattern, or four symmetric bolts, as illustrated in Figure 19.

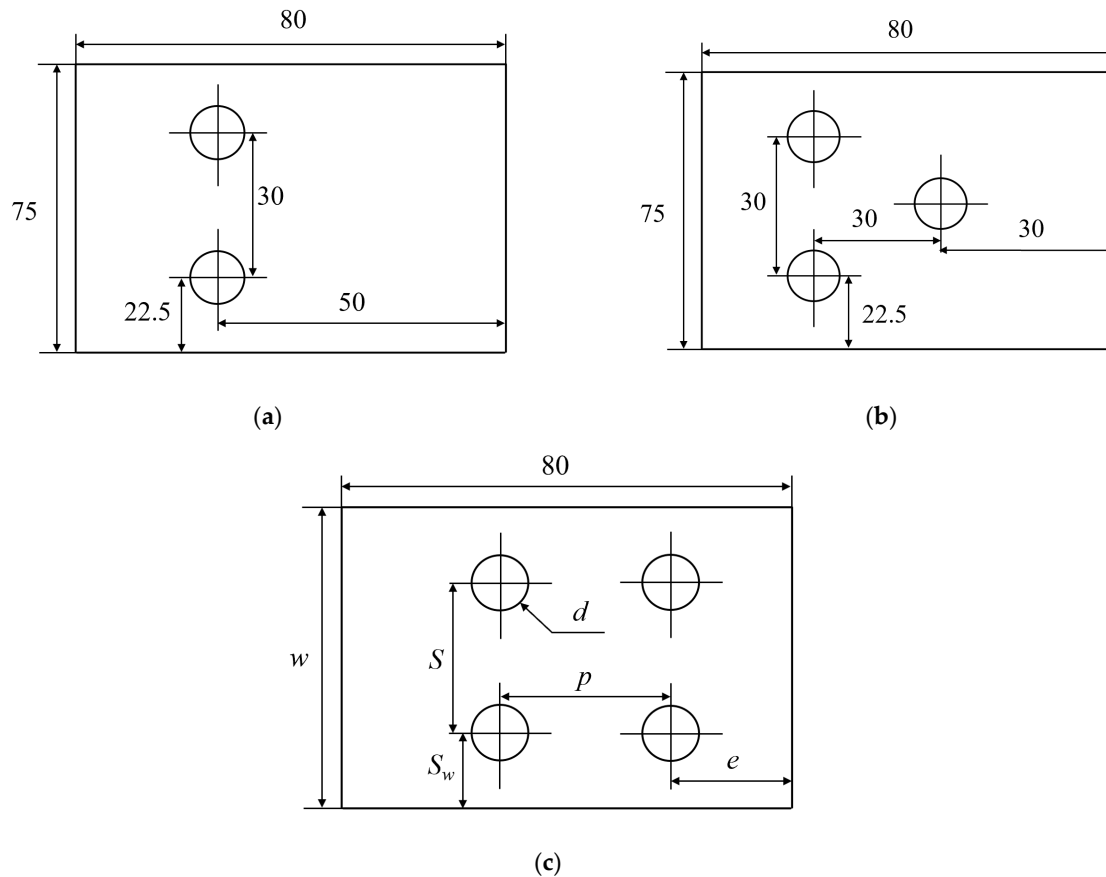


Figure 19. Bolt layouts and definitions of geometric parameters: (a) two-bolt configuration; (b) three-bolt configuration; and (c) four-bolt configuration.

The width w is equal to the width of the composite spring body, i.e., $w = 75$ mm. The thickness t of the leaf spring in the joint region is determined from the previously optimized result as $t = 17$ mm.

The geometric parameters of the composite leaf spring end—such as width w , bolt hole diameter d , end distance e , edge distance Sw , column pitch S , and row pitch p —are the basic factors affecting the failure modes of the end connection. As the end distance e of the connection increases, the possible failure mode of the end connection changes from shear-out failure to bearing failure. Typically, the value of e/d should be greater than 3. Since bearing failure is a local damage mode and is relatively less severe, the end-connection parameters should be designed rationally so that the failure mode of the end connection tends to be bearing failure as much as possible, thereby improving the safety of the connection.

First, the bolts are designed and specified as follows:

The end connection of the composite leaf spring is mainly realized through contact and bearing between the bolts and the mating components. According to the loading condition of the end connection, interference-fit (reamed) bolts should be selected. At the same time, to ensure the reliability of the connection, 40Cr reamed bolts of property class 10.9 are used.

For the bolted end connection of the composite leaf spring, the shear capacity of the bolts must satisfy the connection requirements, and the strength of the composite spring body at the bolt-hole wall must also meet the requirements. The yield strength of 40Cr reamed bolts with property class 10.9 is 900 MPa; therefore, the allowable shear stress is:

$$[\tau_b] = \frac{\sigma_b}{S_\tau} = \frac{900}{3.5} = 257 \text{ MPa} \quad (31)$$

In the formula, $[\tau_b]$ is the allowable shear strength of the bolt, σ_b is the tensile strength of the bolt, and S_τ is the safety factor.

According to the strength analysis of the composite leaf spring body, under emergency braking conditions the maximum longitudinal load borne by the composite leaf spring is 10,108 N. Since a single-bolt connection is prone to load eccentricity under loading, at least two bolts are required for the end connection. To improve the reliability of the connection, the minimum number of bolts (two bolts) is adopted for the design calculation when determining the bolt hole diameter.

$$d = \sqrt{\frac{4F_{max}}{\pi[\tau_b]}} = \sqrt{\frac{4 \times 10108 \div 2}{\pi \times 257}} \approx 5 \text{ mm} \quad (32)$$

In the formula, d is the diameter of the bolt hole, and F_{max} is the maximum longitudinal load on the composite leaf spring. To ensure the reliability of the connection and increase the loaded area of the bolt-hole wall, M8 reamed bolts are selected, with a bolt-hole diameter of 9 mm.

For the bolted connection at the end of the composite spring, the shear failure of the bolt should occur before the bearing (crushing) failure of the composite material at the bolt-hole wall; that is, the allowable bearing strength of the composite material should be greater than or equal to the shear strength of the bolt. When the two strength values are equal, we have:

$$\frac{\pi d^2}{4} [\tau_b] = dt[\sigma_b] \quad (33)$$

In the formula, $[\sigma_b]$ is the allowable bearing (compressive) strength of the composite.

According to the above equation, when the bolt diameter is 9 mm and the thickness at the spring end is 17 mm, the allowable bearing strength of the composite at the moment of bolt shear failure is $[\sigma_b] \approx 95.61 \text{ MPa}$.

That is, when M8 bolts are used, if the allowable bearing strength $[\sigma_b]$ of the composite at the end is greater than 95.61 MPa, then bolt shear failure will precede end bearing failure and become the main failure mode; otherwise, end bearing failure will be the primary risk.

Under bearing load, the fibers and the matrix share the load collaboratively. Considering the reinforcing effect of the fibers on the matrix, the bearing strength can be calculated by the following formula:

$$\sigma_c \approx V_f \cdot \sigma'_{f,c} + (1 - V_f) \cdot \sigma_{m,c} \quad (34)$$

Here, σ_c is the compressive (bearing) strength of the unidirectional composite, V_f is the fiber volume fraction, $\sigma'_{f,c}$ is the effective compressive strength of the fibers under bearing load, and $\sigma_{m,c}$ is the compressive strength of the matrix.

Based on prior calculations, the bearing strength at the composite end is approximately 443.6 MPa, which is significantly higher than 95.61 MPa. Thus, when M8 bolts are used, if the allowable bearing strength of the composite at the end is greater than 95.61 MPa, bolt shear failure will precede end bearing failure and become the governing failure mode; otherwise, end bearing failure becomes critical. For the present design, bolt shear is the controlling failure mode and end bearing failure is not critical.

4.2. Mid-Span Connection Structure Design

The mid-span connection structure of the composite leaf spring studied in this paper retains the same bolted connection as the original steel leaf spring. The spring is clamped from above and below using metal clamp plates, and composite backing plates are added between the composite spring body and the metal clamp plates. This arrangement prevents the edges of the metal clamp plates from coming into direct contact with the composite spring body during load-induced deformation, which would otherwise continuously “cut” into the spring body, causing surface damage to the leaf spring and even ply fracture.

At the same time, the composite backing plates can be shaped to match the curved upper and lower surfaces of the leaf spring on one side, while the other side presents a flat surface to mate with the metal clamp plates, facilitating clamping and fixation of the spring. The mid-span connection structure is shown in Figure 20.

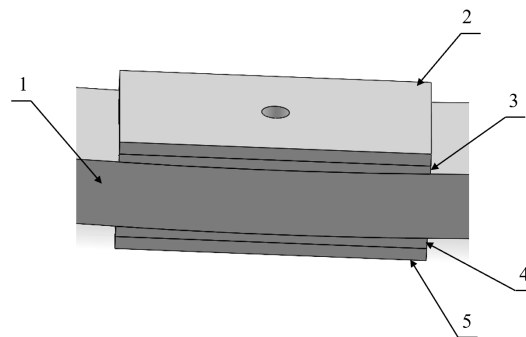


Figure 20. Mid-span connection structure of the composite leaf spring. 1: composite spring body. 2: upper metal clamp plate. 3: upper composite backing plate. 4: lower composite backing plate. 5: lower metal clamp plate.

The metal clamp plates are made of 40Cr steel, consistent with the end connection structure. The widths of the clamp plates and backing plates are the same as that of the composite spring body, i.e., 75 mm. The composite spring body is bonded to the composite backing plates to enhance the reliability of the connection.

4.3. Finite Element Modeling of the Assembly

Based on the optimized composite spring body and the leaf spring connection structure described earlier, the individual components are assembled to obtain the three-dimensional model of the composite leaf spring assembly, as shown in Figure 21.

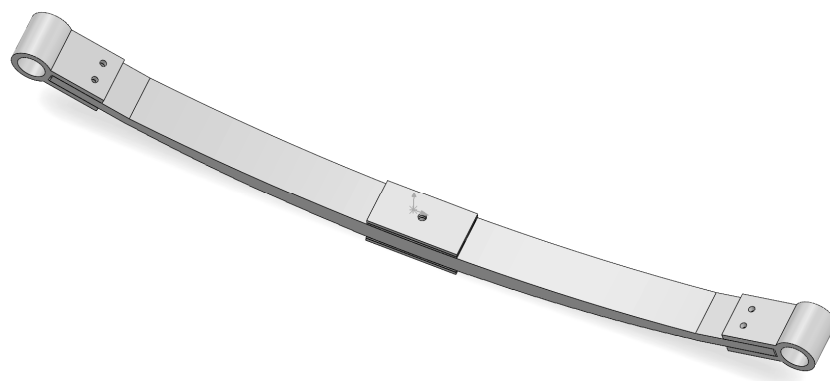


Figure 21. Composite leaf spring assembly.

The three-dimensional model of the leaf spring assembly is imported into HyperMesh for meshing, and the mesh model is shown in Figure 22. Since the leaf spring model is large, only a portion of the mesh is displayed. The assembly contains 491,436 nodes and 438,664 elements, all of type C3D8R.

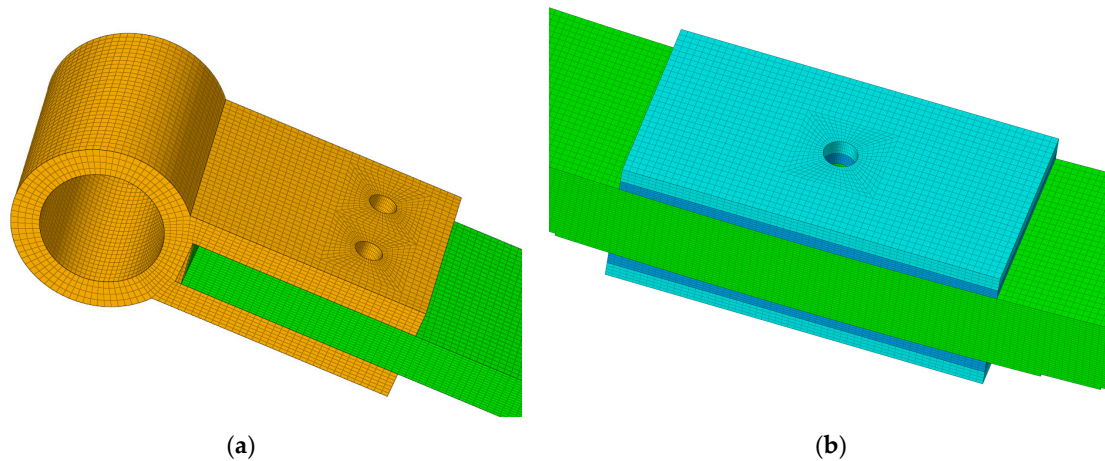


Figure 22. Mesh model of the composite leaf spring. (a) Mesh division at the end of the composite leaf spring. (b) Mesh division at the middle of the leaf spring.

The material settings for the composite spring body are the same as described earlier, and the material parameters of the metal hanger and metal clamp plate are shown in Table 12.

Table 12. Material parameters of metal components.

	Unit	Metal Components
Density	t·mm ⁻³	7.85×10^{-9}
Elastic modulus	MPa	2.06×10^5
Poisson's ratio	-	0.3
Yield strength	MPa	785

In the analysis of the composite leaf spring assembly, tie constraints are defined between the composite spring body and the upper and lower spacer plates, as well as between the end connection surfaces of the composite spring body and the metal hangers, to simulate adhesive bonding. To facilitate the application of loads and boundary conditions, reference points are created at the hangers on both ends of the leaf spring and on the upper and lower surfaces of the clamp plates in the middle region. The inner surfaces of the hanger holes and the upper and lower surfaces of the two clamp plates are then coupled to their corresponding reference points. Contact properties are defined between the relevant surfaces so that they can interact with each other, and the bolt connections are modeled via the contact module in ABAQUS.

4.4. Finite Element Analysis Results of the Connection Structure

4.4.1. Stiffness Analysis of the Assembly

The boundary conditions for the assembly analysis are basically consistent with those described earlier. The difference lies in the need to simulate the preload during loading, so two analysis steps are adopted. In the first analysis step, a preload of 5000 N is applied to the bolts in the leaf spring connection structure, and a vertical load is applied to the

reference point at the middle of the leaf spring. After submitting the calculation, the stiffness results of the composite leaf spring assembly are obtained, as shown in Figure 23.

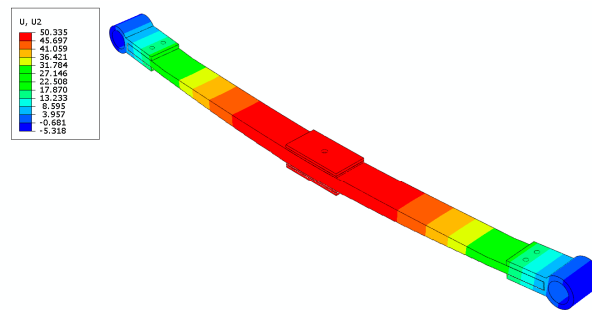


Figure 23. Vertical displacement contour of the composite leaf spring assembly.

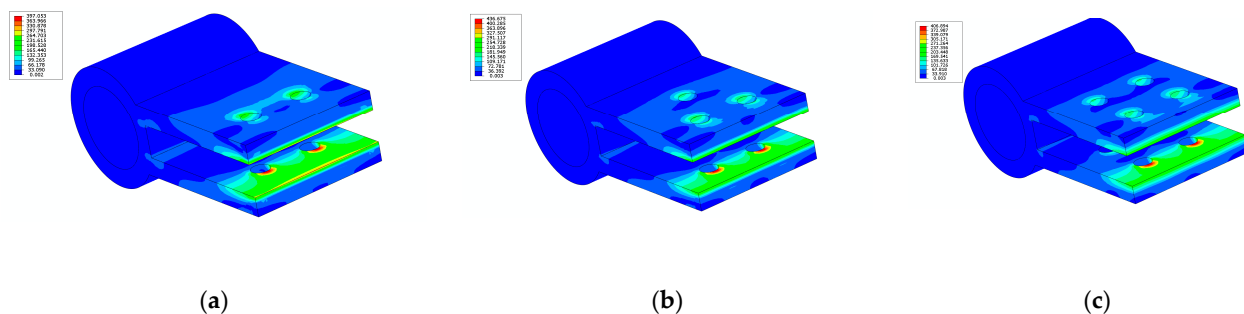
As shown in Figure 23, the deflection at the ends of the composite leaf spring assembly is essentially zero. By fitting the load–displacement data, the stiffness of the leaf spring assembly is obtained as 182.79 N/mm. The final mass of the composite leaf spring assembly is 9.36 kg, representing a reduction of approximately 60.6% compared with the original steel leaf spring.

4.4.2. Strength Analysis of the End Connections

Three bolt layout schemes (two bolt, three bolt, four bolt) are investigated for the end connection. The influence of these layouts on the strength of both the metal hanger and the composite spring end is assessed.

Hanger Strength Analysis

A finite element analysis is carried out for the strength of the metal hanger under three extreme operating conditions. The boundary conditions are consistent with those described earlier. The results of the hanger strength analysis are shown in Figure 24.



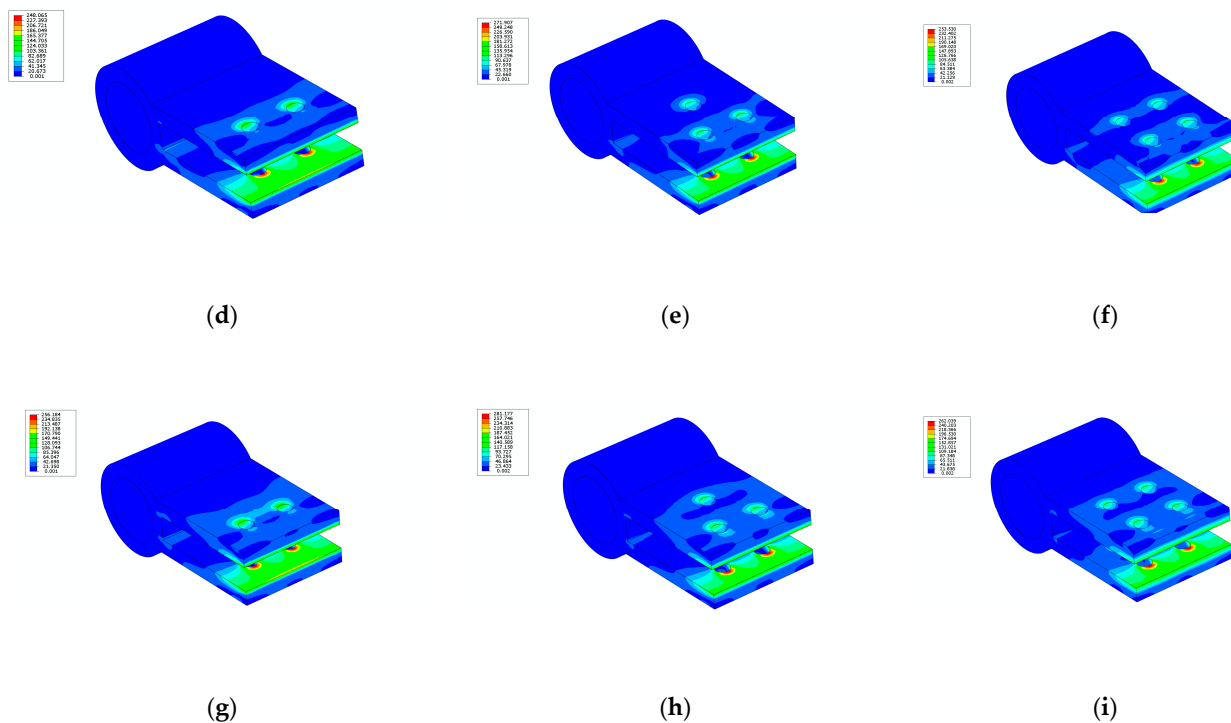


Figure 24. FEA strength analysis results of the leaf spring hanger under various conditions. Stress contours under Vertical loading for: (a) two-bolt, (b) three-bolt, and (c) four-bolt configurations. Stress contours under emergency braking for: (d) two-bolt, (e) three-bolt, and (f) four-bolt configurations. Stress contours under steady-state cornering for: (g) two-bolt, (h) three-bolt, and (i) four-bolt configurations.

As shown in Figure 24, the strength analysis results of the three-hanger bolt-hole schemes of the leaf spring under each operating condition are obtained, and these results are summarized in Table 13.

Table 13. Maximum von Mises stress of the metal hanger for different bolt layouts.

Operating Condition	Two-Bolt Scheme/MPa	Three-Bolt Scheme/MPa	Four-Bolt Scheme/MPa
Vertical load	396.2	436.7	406.5
Emergency braking	247.5	271.3	253.2
Steady-state cornering	255.2	280.5	261.7

As shown by the finite element results in Table 13, the values listed are the Mises stresses of the metal hanger under each operating condition. For all three bolt-hole schemes, the hanger stresses do not exceed the material yield strength of 785 MPa, so according to the fourth strength theory (von Mises yield criterion), no failure occurs in the hanger. In all three schemes, the maximum stress appears around the bolt holes. The stress levels of the two-hole and four-hole schemes are relatively close, whereas the three-hole scheme exhibits higher stress around the bolt holes because the number of bolt holes on the left and right sides is asymmetric. In summary, the two-hole scheme has the lowest stress and requires the fewest holes, thus causing the least damage to the composite leaf spring body.

4.4.3. Strength Analysis of the Connection at the End of the Composite Leaf Spring Body

The influence of different bolt layouts on the localized multi-axial strength at the composite spring end under each limit operating condition is thoroughly analyzed. Due to space limitations, only the stress contour plots in the fiber direction of the end connection region under each operating condition are presented, as shown in Figure 25. Meanwhile, the quantified localized stress components (S_{11} , S_{22} , and S_{12}) surrounding the bolt holes under various working conditions are documented in Table 14.

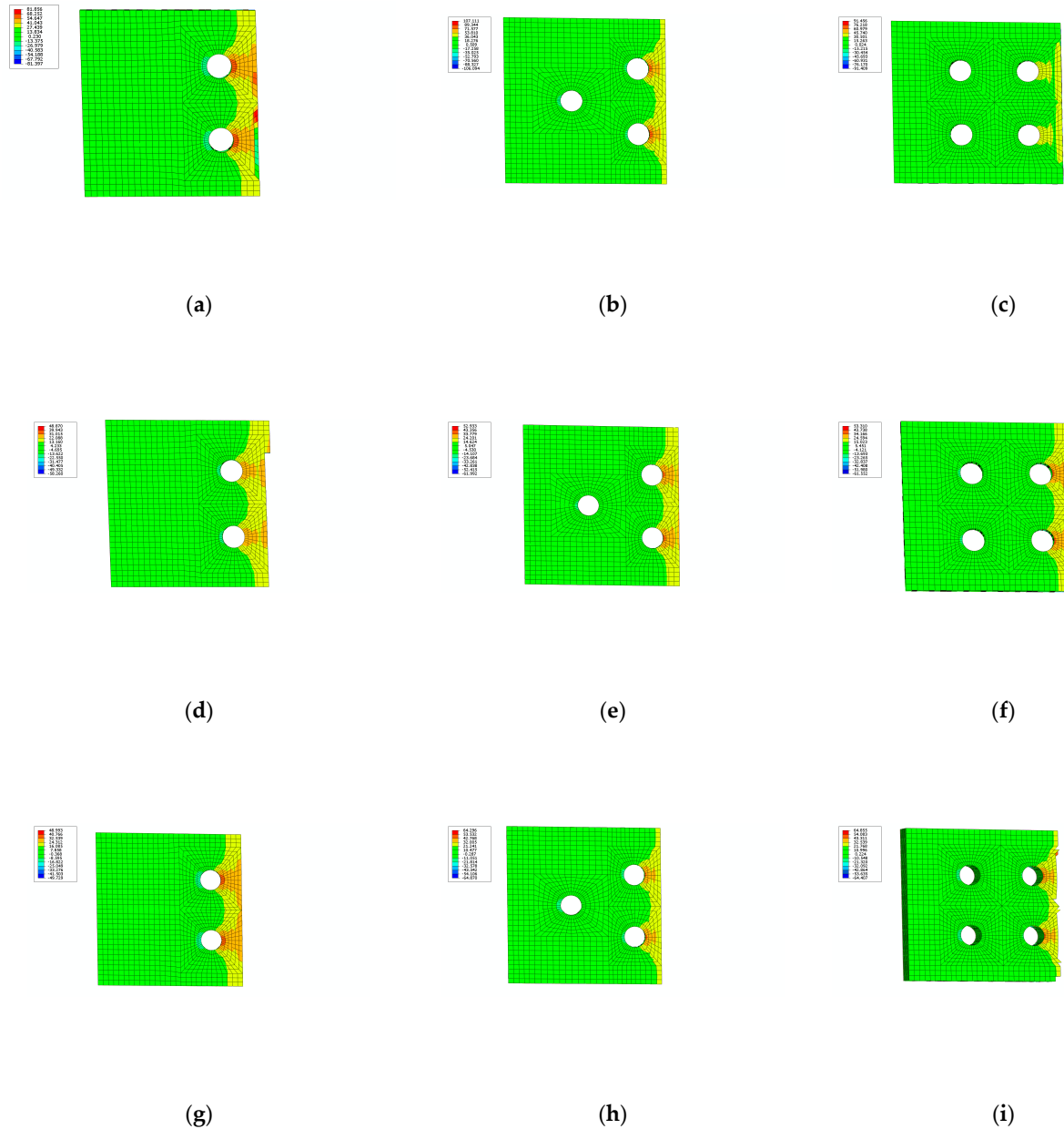


Figure 25. FEA strength analysis results of the composite leaf spring end under various conditions. Stress contours under Vertical loading for: (a) two-bolt, (b) three-bolt, and (c) four-bolt configurations. Stress contours under emergency braking for: (d) two-bolt, (e) three-bolt, and (f) four-bolt

configurations. Stress contours under steady-state cornering for: (g) two-bolt, (h) three-bolt, and (i) four-bolt configurations.

Table 14. Localized stress components around bolt holes for different schemes.

Load Case	Two-Bolt Scheme			Three-Bolt Scheme			Four-Bolt Scheme		
	S_{11}	S_{22}	S_{12}	S_{11}	S_{22}	S_{12}	S_{11}	S_{22}	S_{12}
Vertical load (MPa)	81.74	6.43	5.95	107	7.68	7.05	97.45	7.72	6.93
Emergency braking (MPa)	48.62	4.39	3.42	63.7	5.17	4.05	58.12	5.21	4.1
Steady-state cornering (MPa)	48.9	4.5	3.49	64.27	5.29	4.13	58.62	5.31	4.05

It can be observed from the stress matrix that the maximum stress consistently appears around the bolt holes for all configurations. The stress levels of the two-bolt and four-bolt schemes are relatively close, whereas the three-bolt scheme exhibits noticeably higher stress concentrations because the number of bolt holes on the left and right sides is structurally asymmetric.

Furthermore, to properly capture the cross-coupling interactive damage of these concurrent stress components within the anisotropic laminates, the tensor-based Tsai–Wu failure indices (FI_{max}) are explicitly evaluated. The localized failure index distribution is visualized in Figure 26, and the maximum values are compiled in Table 15. As disclosed by the comparative results, all calculated FI_{max} boundaries remain strictly below the material damage threshold of 1.0, highlighting substantial safety margins against macro-failure.

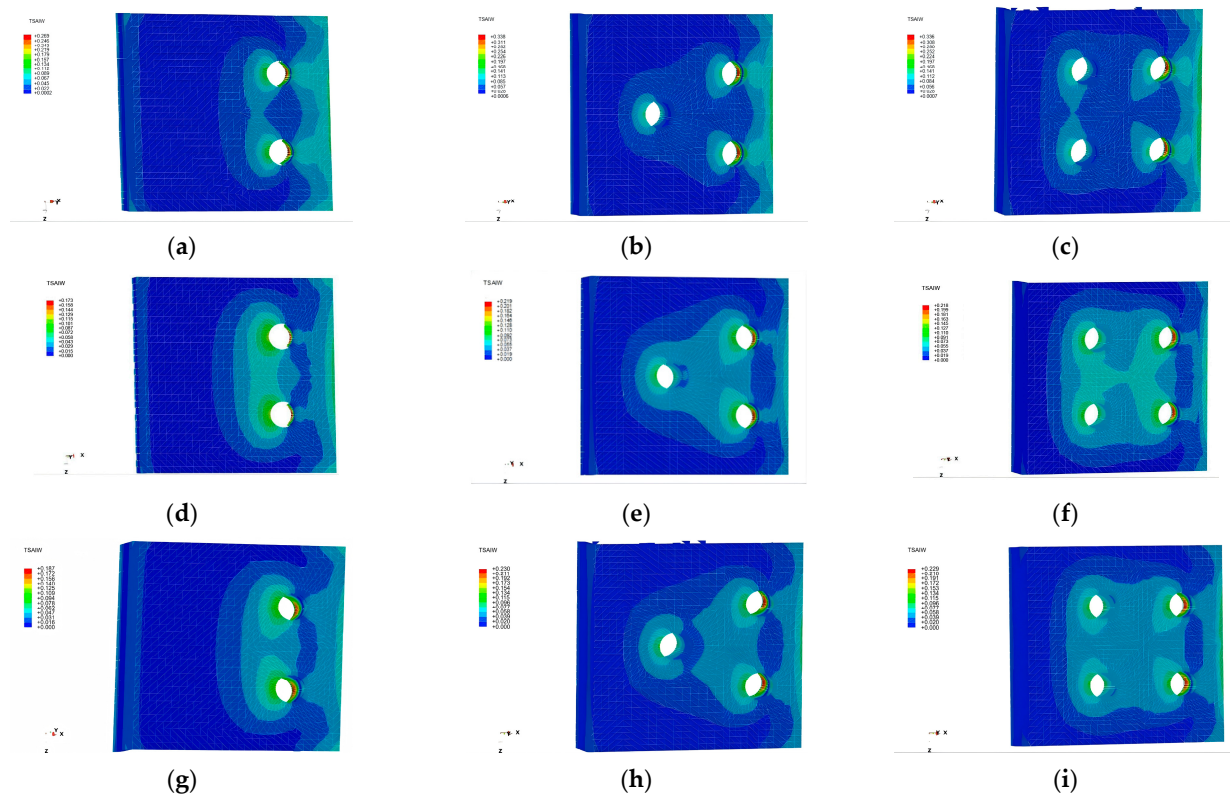


Figure 26. Localized Tsai–Wu failure index (FI_{max}) contours of the composite leaf spring end under different bolt layouts and load cases: (a–c) vertical load, (d–f) emergency braking, and (g–i) steady-state cornering for two-, three-, and four-bolt configurations, respectively.

Table 15. Maximum Tsai–Wu failure indices (FI_{max}) of the composite plies around bolt holes.

Load Case	Two-Bolt Scheme	Three-Bolt Scheme	Four-Bolt Scheme
Vertical load	0.269	0.338	0.336
Emergency braking	0.173	0.218	0.217
Steady-state cornering	0.187	0.230	0.228

The quantified strength analysis indicates that all three bolt layout configurations restrict the localized stress field and failure index at the leaf spring end within safe boundaries. Among these options, the two-bolt scheme achieves the most optimal stress mitigation. Furthermore, the increased clearance between the bolt holes and the physical boundary of the leaf spring end effectively lowers the spatial risk of interlaminar splitting or delamination failure. By deploying the minimum number of bolt holes, this configuration substantially preserves the continuity of the reinforcing fibers. Validated by the multi-axial Tsai–Wu criterion for composite materials, the entire connection zone maintains excellent structural integrity without any failure risks.

In summary, the two-bolt scheme demonstrates the superior mechanical performance in terms of stress control within the metal hanger, stress field optimization at the composite spring end, and maximum protection of fiber network integrity. Consequently, the two-bolt configuration is ultimately selected as the finalized end joint architecture.

Ultimately, the detailed structural designs for both the end joint connection and the central clamping region of the composite leaf spring have been comprehensively established. The systematic finite element analysis firmly verifies the mechanical reliability of these advanced assembly connections, confirming that the integrated composite leaf spring suspension fully complies with the strict performance and safety requirements of commercial vehicle suspension systems.

5. Discussion

As a theoretical and numerical investigation, this study establishes the feasibility of the proposed forward-design framework. The primary limitation of the current stage is the absence of full-scale experimental validation for fatigue life and dynamic durability, largely due to the highly specialized non-standard loading fixtures required for thick-walled composite assemblies under laboratory facility limits. Crucially, it should be noted that while the pure 0-degree unidirectional layout maximizes longitudinal bending stiffness and mass-reduction efficiency for commercial vehicle suspensions, its transverse and shear resistance are inherently limited. According to the composite leaf spring design criteria developed for light commercial buses (e.g., Qian [20]), concentrating the continuous fibers entirely in the 0-degree orientation represents an optimal and standard lightweight pathway because the vertical flexural moments dictate the primary stress distribution. To partially mitigate this validation gap and address the long-term reliability trends prior to physical testing, a preliminary numerical evaluation of the fatigue durability was conducted based on the captured high-fidelity stress envelope fields. Under standard fully loaded static case and limit transient impacts, the maximum longitudinal tensile stress remains strictly bounded below 245.0 MPa across the entire leaf spring body. Comparing this operational amplitude with the empirical S-N curves of equivalent high-performance unidirectional laminates (at a stress ratio $R = 0.1$), the peak stress profile does not exceed 35% of the material's static longitudinal ultimate tensile strength. This extensive stress safety margin provides a strong numerical indicator that the matrix-dominated micro-cracking and fiber-splitting thresholds are not triggered under typical service cycles, mathematically supporting the feasibility of satisfying the high-cycle vehicle durability target.

In addition to the baseline evaluations, the impact of manufacturing variability on the optimized design's performance must be clarified. In standard compression molding, localized process fluctuations primarily introduce three defect mechanisms that can directly degrade the target attributes. Specifically, variations in the local fiber volume fraction alter the longitudinal elastic modulus, which disrupts the optimized flexural rigidity and can drive the assembly stiffness outside the required vehicle matching window (185 plus/minus 15 N/mm). Furthermore, the formation of resin-rich pockets or micro-voids at the terminated ply drop-off boundaries reduces the interlaminar shear capability, thereby undermining the optimized equal-strength profile and inducing unexpected stress concentrations under limit loads. Finally, minor fiber angular misalignment or waviness can disrupt the strict orthotropic balance of the optimized pure 0-degree sequence. This directional variance introduces parasitic bending–torsion coupling modes that force the optimized structure to undergo unexpected macroscopic warping or twisting deviations under extreme emergency braking or cornering conditions. To minimize these process-induced performance deviations, automated prepreg cutting layouts and optimized pressure profiles will be strictly implemented.

To fully check the structural safety under potential off-axis inputs, this study explicitly simulated two extreme operating load cases: emergency braking and steady-state cornering. These transient conditions introduce large non-longitudinal loads, creating complex multi-axial stress fields (comprising transverse tensile stress σ_2 and in-plane shear stress τ_{12}) near the bolted joints and clamping regions. To guarantee a rigorous evaluation of the assembly, a strict controlled variables criterion was enforced across the comparative analysis of the joint topologies based on the precise dimensions derived from the manufacturing blueprints: the bolt hole diameter (d) was uniformly fixed at 9.0 mm to accommodate standard M8 metallic fasteners, the longitudinal edge distance (e) from the bolt center to the laminate free end was stabilized at a constant 50.0 mm (approximately 5.6 d), and the pitch (p) between adjacent multi-bolt centers was constrained at 30.0 mm (approximately 3.3 d).

Because the implemented Tsai–Wu strength criterion naturally incorporates cross-product interactive parameters ($\sigma_1\sigma_2$ and τ_{12}^2), these complex multi-directional interactions were automatically evaluated. The FEA results confirm that the interactive failure index remains safely below 1.0 across all regions, demonstrating that the pure 0° laminate possesses sufficient safety margins to withstand multi-axial inputs. For future work, adding $\pm 45^\circ$ or 90° fabric plies around the bolt holes will be considered to further improve impact and torsional resistance. Physical prototype fabrication and bench testing (stiffness and fatigue) are currently planned to further validate the long-term operational reliability of the optimized assembly.

Furthermore, a quantitative cost analysis of the proposed composite leaf spring assembly against traditional steel springs must be conducted. Based on established automotive techno-economic manufacturing cost models (Verrey et al. [34]), the upfront material procurement and fabrication overheads of fiber-reinforced automotive structures typically increase by 2.0 to 3.5 times compared to identical steel components, driven heavily by prepreg material premiums and curing cycle times. This cost penalty is further supported by empirical chassis lightweighting benchmarks (Witik et al. [35]), which demonstrate that composite suspension components introduce an initial manufacturing cost premium of approximately 150 to 250 percent relative to traditional steel spring production. For our specific design, although the genetic algorithm effectively reduced raw material waste by optimizing the laminate to 32 plies, the total initial manufacturing cost of the complete assembly (including the composite body, metallic hangers, and reamed bolts) is still estimated to increase within this documented 1.5 to 2.5 times range over the traditional steel reference. This upfront manufacturing cost premium represents a necessary

initial investment that is economically amortized through reduced energy consumption and operational fuel savings over the vehicle's long-term service life.

6. Conclusions

Based on the technical parameters of an existing steel leaf spring, an E glass fiber/epoxy composite leaf spring is designed as a lightweight replacement, and a forward design procedure for composite leaf springs is established. Material parameters are obtained through single laminate mechanical tests conducted in accordance with ASTM standards, and a micromechanical model is validated against these tests, providing a reliable basis for structural design.

According to the stiffness-matching principle, the initial design of the main structure is completed. A finite element model is then built in HyperMesh and ABAQUS to systematically simulate stiffness, strength, and modal performance, while the Tsai-Wu tensor criterion is used to evaluate laminate strength.

A multi-objective optimization based on a genetic algorithm is further carried out, taking the starting position of the parabolic profile and the total number of plies as design variables. After optimization, the leaf spring stiffness is 181.75 N/mm, which is 3.59% lower than the initial design but still meets the stiffness requirement. The strength ratio under Vertical load increases by 2.05%, under emergency braking by 7.04%, and under steady-state cornering by 6.04%.

Subsequently, the end and mid-span connection structures are designed. Comparison of different bolt layouts shows that the symmetric two bolt scheme provides the lowest stresses in both the metal hanger and the composite spring end, and minimizes fiber damage. The final composite assembly mass is 9.36 kg, corresponding to a 60.6% weight reduction compared with the original steel spring, while meeting stiffness, strength, and dynamic requirements. These results demonstrate the feasibility and effectiveness of the finite element-based design and optimization method for composite leaf springs.

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Conflicts of Interest: Author Jun Zeng is employed by Hubei Sanhuan Forging Co., Ltd. Author Ning Wang is employed by Guangxi Lanke Resource Recycling Co., Ltd. Author Liang Li is employed by Dongfeng Motor 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.

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