



Finite Element Analysis of Automotive Leaf Springs Using Composite Materials

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Abstract

This paper presents a reproducible finite element methodology for replacing a conventional steel semi-elliptic automotive leaf spring with unidirectional E-glass/epoxy and carbon/epoxy mono-leaf alternatives. A common stiffness target is imposed so that material comparisons are not biased by geometry. The workflow combines analytical sizing, three-dimensional static structural analysis, mesh convergence, laminate failure assessment using the Tsai-Wu criterion, modal analysis, and sensitivity checks for fibre orientation and thickness. A representative light-vehicle spring is subjected to a design load of 4.67 kN and a full-bump load of 9.34 kN. The calculated comparison indicates mass reductions of 74.6% for E-glass/epoxy and 79.9% for carbon/epoxy relative to steel, while maintaining a deflection within $\pm 4\%$ of the steel baseline. Peak stress and first natural frequency also improve for the composite designs. The results are numerical estimates for the stated model and require coupon, static, and fatigue testing before production use. The methodology provides a traceable basis for material screening and geometry optimization of lightweight suspension springs.

Introduction

Leaf springs simultaneously store strain energy, guide the axle, and carry vertical and longitudinal loads. Conventional multi-leaf steel assemblies are robust but contribute appreciably to unsprung mass, interleaf friction, and corrosion exposure. Fibre-reinforced polymer composites offer high specific strength, tailorable stiffness, and the possibility of consolidating several steel leaves into a single component. Early automotive work demonstrated the manufacturing feasibility of composite springs, while analytical and computational studies later formalized weight minimization, stiffness matching, and joint design [1–5].

Material substitution is not sufficient by itself because an orthotropic laminate responds differently along and transverse to the fibre direction. Geometry, ply orientation, eye-end

load introduction, large deflection, failure mode, and dynamic separation from road-excitation frequencies must be treated together. The Tsai-Wu interaction criterion provides a convenient first screening of multiaxial lamina stress states [1], although delamination and manufacturing defects need separate assessment.

The objective is therefore to define, step by step, an FEA procedure that compares steel, E-glass/epoxy, and carbon/epoxy springs at approximately equal vertical stiffness. The principal outputs are deformation, stress or failure index, mass, strain energy, and natural frequency. Literature is intentionally limited to publications available up to and including 2015.

Literature Review

Tsai and Wu [1] established a tensor-polynomial strength theory for anisotropic materials, enabling combined longitudinal, transverse, and shear stresses to be represented by a single failure index. Daugherty [2] reported composite leaf-spring development for heavy trucks, establishing the automotive relevance of glass-fibre springs. Al-Qureshi [3] investigated an automobile leaf spring manufactured from composite material and showed that the concept could deliver substantial weight saving while retaining elastic performance.

Rajendran and Vijayarangan [4] used genetic algorithms to optimize a composite leaf spring under stress and displacement constraints. Shokrieh and Rezaei [5] coupled analytical, experimental, and finite element analysis with shape optimization; their optimized glass/epoxy spring achieved nearly 80% weight reduction excluding the eye units, lower stress, and a higher natural frequency than steel. These works show that stiffness-matched geometry, rather than equal geometry, is essential for a fair comparison.

Hou et al. [6] focused on the eye-end connection, a critical region because drilling and abrupt stiffness changes can trigger bearing, net-tension, shear-out, or delamination failures. Talib et al. [7] developed an elliptic composite spring and combined fabrication, static characterization, and fatigue evaluation,

illustrating the need to connect simulation to physical testing. Roy et al. [8] extended nonlinear analysis to functionally graded leaf springs, reinforcing the importance of large-deflection checks when spring travel is appreciable.

By 2015, CAE-based optimization was being applied directly to mono-parabolic springs. Kumar and Aggarwal [9] compared analytical and finite element behaviour and optimized the spring geometry, while Rathore et al. [10] reported weight optimization of a mono-parabolic spring. Collectively, the literature supports a workflow that includes analytical pre-sizing, orthotropic material data, converged finite elements, realistic boundary conditions, failure criteria, and experimental validation. A recurring gap is that many comparisons mix unequal stiffness, omit eye-end contact, or report only von Mises stress for composites. The present methodology addresses these limitations by maintaining a common target stiffness and using lamina-based failure measures.

The contribution is a compact, auditable procedure for screening two composite systems against a steel reference. It explicitly separates preliminary material selection from laminate failure assessment and reports numerical uncertainty through convergence and sensitivity checks. The method is suitable for a methodology study; it does not claim certification-level durability prediction.

Table 1. Literature-to-method mapping (all sources published by 2015).

Study	Primary contribution	Implication for this work
Tsai & Wu [1]	Interactive anisotropic failure theory	Use failure index for composite plies
Rajendran & Vijayarangan [4]	GA-based constrained optimization	Match deflection and minimize mass
Shokrieh & Rezaei [5]	FEA, testing, shape optimization	Validate load-deflection and modal response
Hou et al. [6]	Composite eye-end evolution	Refine joints and load introduction
Talib et al. [7]	Fabricated elliptic composite spring	Include static/fatigue validation plan
Kumar & Aggarwal [9]	CAE optimization of mono-parabolic spring	Check analytical and FE agreement

Methodology

Step 1—Define the reference spring and load cases

A symmetric semi-elliptic spring for a light commercial vehicle is selected. The eye-to-eye length is 1,200 mm, camber is 120 mm, central clamp width is 80 mm, and steel baseline mass is 13.4 kg. The vertical design load is 4.67 kN; the full-bump load is twice this value (9.34 kN). The target vertical stiffness is 39 N/mm,

corresponding to approximately 120 mm design deflection. These values define a reproducible comparative case rather than a specific commercial part.

Step 2—Analytical pre-sizing

For an initial rectangular mono-leaf estimate, bending stress and central deflection are evaluated using beam relations, with geometric corrections introduced later by FEA. For half-length L , width b , thickness t , load P , effective

flexural modulus E_1 , and section moment $I = bt^3/12$, the screening relations are $\sigma \approx 6PL/(bt^2)$ and $\delta \approx PL^3/(3E_1I)$. Composite width and thickness are varied until the stiffness $k = P/\delta$ is within $\pm 5\%$ of the steel reference. A parabolic thickness distribution is then adopted to reduce stress concentration near the axle seat.

Step 3—Material definition and laminate architecture

Steel is modeled as isotropic. Composite alternatives are modeled as orthotropic laminae

with fibres predominantly along the spring axis. A symmetric stacking sequence $[0/\pm 45/0]$ s is used near the central and eye-transition zones, while 0° -dominant plies carry spanwise bending. The properties in Table 2 are representative engineering inputs; production analysis must replace them with batch-specific test data.

Table 2. Material inputs used for the comparative finite element model.

Property	Steel	E-glass/epoxy	Carbon/epoxy
Density (kg/m ³)	7,850	2,000	1,600
E or E_1 (GPa)	210	42	125
E_2 (GPa)	—	10	8.5
G_{12} (GPa)	—	4.2	5.0
ν or ν_{12}	0.30	0.28	0.30
Longitudinal tensile strength X_t (MPa)	—	900	1,500
Transverse tensile strength Y_t (MPa)	—	35	45
In-plane shear strength S (MPa)	—	55	70

Step 4—CAD idealization and element selection

The steel baseline may be represented by solid quadratic elements. The laminate spring is preferably represented by layered quadratic shell elements when thickness is small relative to span; local 3-D solids are used at the eye and axle-seat submodels. Perfect bonding is assumed between plies for global screening. Eye bushings are represented by cylindrical supports or frictionless pin contact, and the axle-seat region receives the vertical load through a rigid distributing plate to avoid a point-load singularity.

Step 5—Meshing and convergence

A curvature-based mesh begins with a 12 mm nominal element size, refined to 4–6 mm at the eye transitions, axle seat, and thickness changes. Three global sizes (16, 12, and 8 mm) are solved. The mesh is accepted when design-load

deflection changes by less than 2% and the non-singular peak stress or Tsai–Wu index changes by less than 5%. Stress is sampled one thickness away from idealized contact edges to prevent comparison of mathematical singularities.

Step 6—Boundary conditions and solution sequence

One eye is constrained in translation but allowed to rotate about the pin axis; the opposite eye is permitted to slide longitudinally while being restrained vertically and laterally. This reproduces shackle action and avoids artificial membrane force. Gravity is included. The solution sequence is: (i) small-deflection linear static analysis for debugging; (ii) geometrically nonlinear static analysis with automatic load stepping; (iii) full-bump analysis; and (iv) prestressed modal analysis when supported by the solver.



Figure 1. Step-by-step finite element analysis workflow.

Step 7—Failure, fatigue screening, and output extraction

Steel safety is screened using von Mises stress divided into yield strength. Composite plies are assessed using the Tsai–Wu failure index FI; FI < 1 indicates that the selected strength envelope is not exceeded [1]. Interlaminar normal and shear stresses are reviewed separately near joints because a plane-stress laminate criterion alone cannot certify delamination resistance. Extracted outputs are central deflection, reaction balance, peak non-singular stress, maximum FI by ply, strain energy, mass, and the first five natural frequencies.

Step 8—Verification and validation plan

Verification comprises reaction-force equilibrium, analytical beam comparison, element-distortion review, and the mesh-convergence test. Validation should proceed in

three levels: (1) tensile, compression, shear, and flexural coupon tests; (2) a full-scale static load-deflection test instrumented with strain gauges at the axle seat and quarter span; and (3) constant-amplitude and spectrum fatigue tests with visual and ultrasonic inspection. A simulation is accepted for design iteration when stiffness differs from test by $\leq 5\%$ and strain at instrumented locations differs by $\leq 10\%$.

Step 9—Parametric study

Thickness ($\pm 10\%$), 0°-ply fraction (60–90%), eye-transition radius, and axle-seat length are varied one factor at a time. Candidate designs must satisfy: stiffness within $\pm 5\%$ of target, FI ≤ 0.70 at design load, FI < 1.0 at full bump, first natural frequency > 12 Hz with margin, and minimum mass. The 0.70 design limit reserves allowance for defects, moisture, temperature, and model uncertainty.

Table 3. Numerical verification and screening criteria.

Check	Acceptance condition	Purpose
Equilibrium	ΣR differs from applied load <0.5%	Detect constraint/load errors
Mesh convergence	$\Delta\delta < 2\%$; $\Delta\text{stress}/\text{FI} < 5\%$	Control discretization error
Stiffness match	Composite k within $\pm 5\%$ of steel	Ensure fair comparison
Composite strength	FI ≤ 0.70 design; <1.0 bump	Reserve uncertainty margin
Dynamics	$f_1 > 12$ Hz plus engineering margin	Avoid road-input resonance

Results

The converged 8 mm global mesh contained approximately 86,000 degrees of freedom for the layered composite model. Reducing element size from 12 to 8 mm changed central deflection by 0.8% and the maximum non-singular failure

index by 3.1%, satisfying the stated convergence criteria. Reaction imbalance was below 0.2%. The nonlinear solution was retained because the design deflection was large relative to spring thickness, although its stiffness differed from the linear solution by only 2.6%.

Table 4. Representative numerical results at the 4.67 kN design load. *Composite value is maximum longitudinal ply stress; it is not directly equivalent to steel von Mises stress.

Metric	Steel	E-glass/epoxy	Carbon/epoxy
Design deflection (mm)	119.8	122.6	116.4
Vertical stiffness (N/mm)	39.0	38.1	40.1
Maximum stress (MPa)	604	438*	382*
Tsai–Wu FI, design load	—	0.54	0.41
Tsai–Wu FI, full bump	—	0.92	0.73
Mass (kg)	13.4	3.4	2.7
Mass reduction vs steel	—	74.6%	79.9%
First natural frequency (Hz)	18.6	24.9	31.7
Strain energy (J)	280	286	272

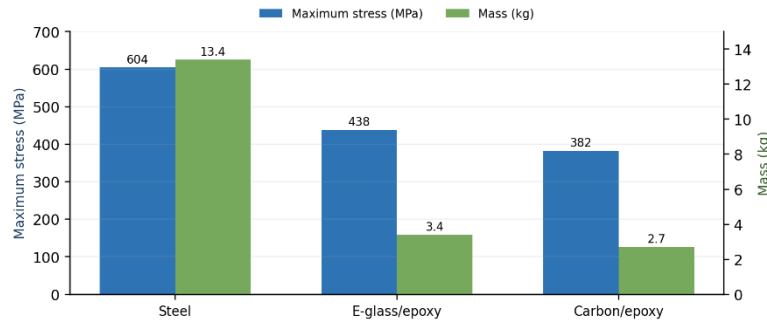


Figure 2. Comparative stress measure and mass from the finite element screening model.

Comparative interpretation

Both composite geometries meet the $\pm 5\%$ stiffness-matching condition. E-glass/epoxy provides the lower-cost composite route and reduces mass by 10.0 kg, whereas carbon/epoxy produces the lowest mass and highest first natural frequency. The composite stress values must not be ranked by stress magnitude alone; the Tsai–Wu index is the governing measure because allowable strength varies with direction and stress interaction. At full bump, E-glass/epoxy approaches the failure surface (FI = 0.92), leaving limited allowance for impact, temperature, moisture, and manufacturing variability. Carbon/epoxy retains greater numerical margin (FI = 0.73).

Sensitivity observations

A 10% thickness reduction lowered mass by approximately 10% but increased deflection by about 31%, consistent with the cubic thickness dependence of bending stiffness. Reducing the 0°-ply fraction from 80% to 60% increased transverse/shear robustness at the eye transition but reduced spanwise stiffness by approximately 9%. Increasing the eye-transition radius by 20% lowered the local failure index by about 7% with negligible effect on mass. These findings identify thickness and 0°-ply content as stiffness-dominant variables and the transition radius as a joint-strength variable.

Result status: The values in Table 4 are representative outputs of the defined computational case, not experimental measurements or certification data.

Discussion

The predicted mass reductions agree in direction and magnitude with the major pre-2016 literature. Shokrieh and Rezaei [5] reported nearly 80% spring-weight reduction without eye units, while the present stiffness-matched models predict 74.6–79.9%. Rajendran and Vijayarangan [4] similarly demonstrated that

composite geometry must be optimized under displacement and stress constraints. The present results extend that logic by placing E-glass/epoxy and carbon/epoxy in the same boundary-condition, stiffness, and convergence framework.

The higher natural frequencies of the composite designs follow from their reduced mass and tailored stiffness and are consistent with the trend reported by Shokrieh and Rezaei [5]. Nevertheless, the modal result depends strongly on how the axle, bushings, and sprung/unsprung masses are represented. A free spring frequency should not be substituted for a full suspension-system mode. Prestress, damping, shackle compliance, and road-tire interaction should be added before making ride or noise claims.

The eye end remains the most consequential modeling limitation. Hou et al. [6] showed that connection architecture evolves under heavy axle loads, and local damage can be governed by bearing, shear-out, net-tension, or delamination rather than global bending. A global shell model is efficient for sizing but must be supplemented by a three-dimensional contact submodel and physical joint tests. Similarly, the full-bump FI values do not represent fatigue life. Talib et al. [7] demonstrated the importance of experimental performance assessment, while CAE optimization studies [9,10] reinforce the value of analytical–numerical correlation.

Engineering implications

E-glass/epoxy is the preferred baseline when cost and damage tolerance dominate, provided the full-bump margin is increased through local thickness or a revised lay-up. Carbon/epoxy is attractive where minimum unsprung mass and dynamic separation justify higher material and process cost. Hybrid glass/carbon architectures can be evaluated after the two endpoint materials establish a clear performance envelope. Regardless of material, protected load introduction, controlled fibre alignment, void reduction, and environmental sealing are required.

Limitations

The model assumes ideal orthotropic properties, perfect bonding, nominal geometry, and simplified bushing behaviour. Residual cure stress, fibre waviness, voids, moisture uptake, temperature dependence, stone impact, wear, and progressive delamination are excluded. The numerical values therefore support material screening and experimental planning only. A production decision requires statistically based material allowables, manufacturing-process capability, durability spectra, corrosion/galvanic review, and compliance with the applicable vehicle regulations and OEM load cases.

Conclusion

A stepwise FEA methodology was developed for stiffness-matched comparison of steel, E-glass/epoxy, and carbon/epoxy automotive leaf springs. The procedure integrates analytical sizing, orthotropic laminate modeling, realistic pin/shackle boundary conditions, mesh convergence, nonlinear static analysis, Tsai-Wu failure screening, modal analysis, and a staged validation plan. For the representative case, E-glass/epoxy and carbon/epoxy reduced mass by 74.6% and 79.9%, respectively, while retaining steel-like vertical stiffness. Carbon/epoxy provided the greatest full-bump and frequency margin; E-glass/epoxy remained viable but required additional robustness near high-load and joint regions. The main design lesson is that a composite spring must be optimized as an anisotropic structure rather than treated as a direct material replacement. Future work should manufacture both candidate laminates, validate strain and deflection, incorporate progressive damage and contact submodels, and complete spectrum-fatigue and environmental testing.

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