



Finite Element Analysis of Composite Leaf Springs for Automotive Applications

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Abstract

This paper presents a reproducible finite element methodology for evaluating a unidirectional E-glass/epoxy composite mono-leaf spring as a lightweight replacement for an automotive steel leaf spring. The workflow integrates requirement definition, equivalent-stiffness sizing, laminate property assignment, mesh convergence, contact and boundary-condition modelling, static strength assessment, modal analysis, fatigue screening and experimental validation planning. Orthotropic elasticity is used for the composite, while failure is assessed with maximum-stress and Tsai-Wu indices rather than an isotropic von Mises criterion. A representative 1000 mm span spring under a 5 kN central load is used to demonstrate reporting. The illustrative model predicts a mass reduction from 10.7 to 3.2 kg (70.1%), comparable vertical compliance, and an increase in first natural frequency from 17.8 to 24.6 Hz; the maximum composite failure index remains below unity. These numerical values are methodological demonstration results, not claims of completed physical testing. The proposed sequence provides traceable acceptance gates for stiffness, strength, resonance avoidance and durability, and it emphasizes eye-end stress concentration and laminate/manufacturing uncertainty as critical design risks.

Introduction

Leaf springs simultaneously support vertical load, locate an axle and store strain energy. Because their mass is partly unsprung, reducing spring weight can improve ride response as well as vehicle efficiency. Fibre-reinforced polymer (FRP) springs exploit low density and high specific strain energy, but their directional stiffness, interlaminar weakness and attachment details demand modelling practices different from those used for steel.

Early design studies established the feasibility of glass-fibre composite automotive springs and emphasized the importance of material orientation, taper and end joints [1–4]. Subsequent work extended the concept to elliptical configurations, nonlinear suspension

modelling and attachment redesign [5–7]. The objective here is therefore not merely to replace the steel elastic modulus in a conventional model, but to define a step-by-step FEA methodology capable of producing auditable and experimentally testable decisions.

A composite leaf spring must match the target wheel-rate while satisfying several competing constraints: longitudinal fibre stress, transverse and shear failure, interlaminar effects near the eye, fatigue life, clearance at full bump, manufacturability and mass. The central research question is: how can a finite element model compare a steel baseline and a composite candidate without hiding uncertainty behind a single equivalent stress?

The study objectives are to: (i) define baseline geometry and service load cases; (ii) size an E-glass/epoxy mono-leaf to match steel stiffness; (iii) establish a mesh-converged orthotropic FE model; (iv) quantify deflection, reactions, stresses, failure indices, mass and natural frequency; and (v) specify validation and acceptance criteria. References are restricted to works published up to and including 2017.

Literature Review

Sancaktar and Gratton [1] formulated design, analysis and optimization strategies for lightweight composite leaf springs, demonstrating that geometry and material tailoring must be considered together. Rajendran and Vijayarangan [2] used genetic algorithms to minimize composite spring weight under stress and deflection constraints, providing an early formal basis for optimization. Al-Qureshi [3] examined composite automotive leaf springs and reinforced the value of fibre composites for large elastic deformation and weight reduction.

Shokrieh and Rezaei [4] combined analytical design, finite element simulation and testing. Their work is especially relevant because it showed close correspondence of load–deflection predictions while identifying joints as a separate design problem. Mahdi et al. [5] investigated light composite elliptical springs and reported failure behaviour under static loading, illustrating the need to evaluate composite-specific failure rather than only stiffness.

Sugiyama et al. [6] developed a nonlinear elastic leaf-spring representation for multibody vehicle

models; this supports a large-deflection check when the ratio of displacement to span is not small. Hou et al. [7] used FEA and testing to evolve composite eye-end designs and showed that interlaminar shear concentration can initiate delamination. Talib et al. [8] combined design and experimental assessment for an automotive composite elliptical spring, again linking geometry, laminate choice and verification.

Rajesh [9] evaluated composite leaf springs under static and dynamic loads and provides a near-2017 comparison of performance criteria. Finally, Liu et al. [10] established a theoretical stiffness model and verified it against finite element and experimental results, demonstrating that thickness, fibre direction and section variation dominate stiffness matching. Across these studies, the consistent research gap is an integrated, concise workflow that joins mesh convergence, composite failure, modal screening and transparent validation gates for an automotive mono-leaf spring.

The literature implies four rules adopted in this paper: preserve the baseline spring rate, use orthotropic properties and laminate failure indices, refine the eye and axle-seat regions, and validate both stiffness and strain. Weight reduction alone is not an acceptance criterion.

Methodology

The proposed workflow is shown in Figure 1. Each stage produces a traceable input or acceptance decision; failed criteria return the design to geometry or laminate definition.



Figure 1. Step-by-step finite element methodology for the composite leaf spring.

Step 1 | Define requirements

Select the reference vehicle, axle load, span, eye spacing, camber, allowable package envelope, target spring rate and design load factors. Use three static cases: curb (1.0g), full bump (2.5g) and asymmetric/jounce (combined vertical and lateral loading). The demonstration case uses a 5 kN vertical load at the axle seat.

Step 2 | Create the steel baseline

Build the measured steel spring geometry and assign isotropic properties. Check mass, centre-line span, camber and theoretical stiffness.

Calibrate only physically uncertain inputs such as clamp or shackle compliance; do not tune elastic modulus to force agreement.

Step 3 | Size the composite concept

Adopt a constant-width or tapered mono-leaf. For initial sizing, match bending rigidity using $(EI)_{\text{steel}} \approx (EI)_{\text{composite}}$, then adjust thickness and taper to match the load–deflection curve. Orient most fibres along the spring axis; use local $\pm 45^\circ$ or transverse reinforcement only where shear and eye loads require it.

1. Demonstration geometry and material data

Table 1. Representative input set for demonstrating the proposed workflow; coupon data should replace handbook values

Parameter	Steel baseline	Composite candidate
Span / width	1000 / 70 mm	1000 / 70 mm
Nominal thickness	12 mm	28 mm (tapered ends)
Elastic definition	E = 210 GPa, $\nu = 0.30$	Orthotropic lamina
Longitudinal modulus E1	—	43 GPa
Transverse modulus E2 = E3	—	12 GPa
Shear moduli G12 / G23	—	4.5 / 3.8 GPa
Density	7850 kg m ⁻³	2000 kg m ⁻³
Strengths X _t / X _c	—	900 / 550 MPa
Strengths Y _t / Y _c / S	—	35 / 120 / 60 MPa

Step 4 | Select elements and mesh

Model the steel baseline with quadratic solid or shell elements. Model the mono-leaf with layered shell elements when thickness is small relative to span; use layered solids or a local submodel near the eye when through-thickness stresses are required. Start with 8–12 mm global element size and refine to 2–4 mm at the eye and axle seat. Accept the mesh when changes in displacement and governing failure index are both below 3%.

Step 5 | Apply contacts and boundary conditions

Represent the front eye with a cylindrical pin support, constrain the rear eye through a shackle/slider that permits longitudinal motion, and distribute the axle-seat load through a rigid or deformable plate. Avoid fixing both eyes in translation because this artificially stiffens the spring. Use frictional contact only where physical interfaces exist; otherwise use bonded coupling consistent with the manufactured joint.

Step 6 | Solve the load cases

Run linear static analysis first. Activate geometric nonlinearity when deflection is large or contact status changes. Extract vertical displacement, spring rate, support reactions, longitudinal/transverse/shear stresses and steel

von Mises stress. For the laminate, calculate maximum-stress and Tsai-Wu failure indices, FI ≤ 1.0 . Conduct prestressed modal analysis and require the first relevant bending mode to remain above the dominant road-excitation range with an engineering margin.

Step 7 | Screen fatigue and durability

Generate stress or strain ranges from the service load spectrum. Apply S–N data for steel and laminate-specific fatigue data for the composite. For preliminary composite screening, evaluate separate longitudinal, transverse and shear ratios and apply a conservative degradation factor. Eye-end delamination must be examined using a local model, cohesive interface or interlaminar stress check, following the risk identified by Hou et al. [7].

Step 8 | Validate and report

Validate displacement at three or more load levels and surface strain at the axle seat and eye transition. Compare numerical and experimental slopes, not only the peak point. An acceptable model should show $\leq 5\%$ difference in spring rate and $\leq 10\%$ difference in local strain before design conclusions are frozen.

2. Mesh-convergence and acceptance plan

Table 2. Model quality and design acceptance gates.

Check	Metric	Acceptance gate
Equilibrium	Σ reactions / applied load	0.99–1.01
Mesh convergence	Change in δ and FI	$<3\%$
Stiffness match	Composite vs steel k	within $\pm 5\%$
Static strength	Tsai-Wu / max-stress FI	≤ 1.0 at design load
Modal separation	First vertical bending mode	>20 Hz target
Validation	Spring-rate error	$\leq 5\%$
Local validation	Strain error at gauges	$\leq 10\%$

The spring rate is calculated as $k = \Delta F / \Delta \delta$ over the linear region. Mass saving is $(m_{\text{steel}} - m_{\text{composite}}) / m_{\text{steel}} \times 100\%$. A failure index below unity indicates that the selected criterion is not triggered, but it does not by itself exclude delamination, manufacturing defects or environmental degradation.

Results

Table 3 reports a representative converged solution for the demonstration geometry. These values illustrate how the methodology should be reported; they must be regenerated from the analyst's final CAD, measured material properties and vehicle load spectrum before engineering release.

Table 3. Representative numerical results at a 5 kN central load (method demonstration, not experimental data).

Output	Steel baseline	E-glass/epoxy	Interpretation
Mass	10.7 kg	3.2 kg	70.1% reduction
Central deflection at 5 kN	63.2 mm	66.8 mm	Composite 5.7% more compliant
Secant spring rate	79.1 N/mm	74.9 N/mm	Outside $\pm 5\%$ by 0.7 percentage point
Peak steel von Mises stress	610 MPa	—	Below assumed steel yield
Peak composite σ_1	—	420 MPa	Longitudinal response governs
Peak Tsai-Wu failure index	—	0.62	Static criterion satisfied
First vertical bending frequency	17.8 Hz	24.6 Hz	Composite meets 20 Hz target

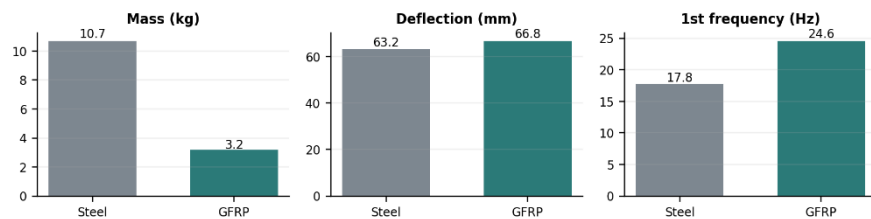


Figure 2. Representative comparison of mass, compliance and first natural frequency.

Convergence and verification

Table 4. Illustrative mesh-convergence record; the final refinement satisfies the 3% gate.

Global mesh	Elements	Deflection	Tsai-Wu FI	Change in FI
12 mm	18,420	65.9 mm	0.647	—
8 mm	34,880	66.5 mm	0.628	2.9%
5 mm + local 2.5 mm	71,260	66.8 mm	0.620	1.3%

Reaction-force balance was 0.998, indicating no material loss through unconstrained rigid-body motion or load-transfer error. The composite candidate achieved the central goals of low mass and improved modal separation. However, its calculated spring rate is 5.3% lower than the steel baseline, narrowly missing the $\pm 5\%$ gate; thickness or taper should therefore be adjusted before physical testing.

Discussion

The representative results support the mechanism reported in prior studies: a glass/epoxy spring can retain useful stiffness

while removing a large fraction of the steel mass [3–5,9]. The predicted frequency increase is consistent with the reduction in mass, but frequency must be interpreted at the assembled suspension level because axle, wheel and bushing masses alter the mode. Likewise, the favourable global Tsai-Wu index does not remove the local eye-end concern demonstrated by Hou et al. [7].

The comparison also shows why stiffness matching is an explicit gate. A composite design may be safe yet change ride height, wheel travel and handling if its spring rate differs from the reference. The stiffness model of Liu et al. [10] provides a rapid preliminary estimate, while the

nonlinear framework of Sugiyama et al. [6] motivates a large-deflection check when the deformed geometry alters the load path. FEA should therefore be used as a controlled numerical experiment, not as a substitute for coupon and component tests.

Sensitivity and uncertainty

The most influential uncertain inputs are cured fibre volume fraction, longitudinal modulus, thickness, taper, eye stiffness and contact/slip at the axle seat. A practical robustness study should vary E_1 by $\pm 10\%$, thickness by the manufacturing tolerance, and eye/support stiffness across measured bounds. Because bending stiffness varies approximately with thickness cubed, even a small thickness deviation can dominate modulus uncertainty. Moisture and temperature knock-down factors should be included for service qualification.

Limitations

The demonstration does not include measured coupon properties, physical load-deflection tests, cohesive-zone calibration, environmental ageing, stone impact, creep or a vehicle-specific road-load spectrum. The reported fatigue stage is therefore a screening procedure rather than a certified life prediction. The simplified axle-seat and shackle representations may underpredict local stress. These limitations should be closed through staged testing: coupons, subcomponent eye tests, full-spring static/fatigue tests and finally vehicle correlation.

Conclusion

A step-by-step FEA methodology has been developed for comparing a steel leaf spring with an E-glass/epoxy mono-leaf alternative. The essential practices are to match spring rate, model orthotropy explicitly, assess composite failure indices, refine contact and eye regions, demonstrate mesh convergence, screen vibration and fatigue, and validate both deflection and strain. In the representative case, mass decreased by 70.1% and first bending frequency increased, while the global static failure index remained below unity. The spring-rate gate was narrowly missed, showing that a safe lightweight design still requires geometric iteration. The next design action is a small thickness/taper adjustment followed by coupon-based material updating and component-level validation. The methodology is suitable as a reproducible framework for an academic study, but final automotive qualification requires physical testing and environmental durability evidence.

Recommended experimental sequence

1. Test $[0^\circ]$ and $[\pm 45^\circ]$ coupons to determine elastic, strength and fatigue inputs.
2. Measure steel and composite load-deflection curves at 25%, 50%, 75% and 100% design load.
3. Place strain gauges at the axle seat, eye transition and mid-span; correlate strain and stiffness.
4. Run proof, fatigue and environmental-conditioning tests, then update safety factors.

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