



## Finite Element Analysis of Composite Leaf Springs for Lightweight Automotive Applications

Branislav Zambrano-Ortiz

Department of Civil and Environmental Engineering, Port Louis Business and Technology College, Mauritius  
[branislav.zambrano.ortiz@plbtc-mu.org](mailto:branislav.zambrano.ortiz@plbtc-mu.org)

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### Abstract

This paper presents a step-by-step finite element methodology for replacing a conventional multi-leaf steel spring with a lightweight E-glass/epoxy mono-leaf spring while preserving load capacity, vertical compliance, and structural safety. A parametric three-dimensional model is defined from a representative light-vehicle rear spring. The steel baseline is evaluated with isotropic elasticity, whereas the composite alternative is represented as a layered orthotropic laminate with fibres primarily aligned along the spring span. Mesh convergence, realistic eye and axle-seat constraints, contact sensitivity, and comparison with elementary beam theory are included as verification controls. The proposed outputs are displacement, spring rate, ply stresses, interlaminar shear indicators, Tsai-Wu failure index, modal frequency, and mass. An illustrative numerical case study predicts a mass reduction from 12.6 kg to 3.45 kg (72.6%), a spring-rate difference of 2.2% relative to the steel target, and a maximum composite failure index of 0.62 under the design load. These values demonstrate the decision logic of the methodology but require coupon tests, proof loading, and fatigue validation before production use. The workflow provides a reproducible basis for selecting laminate thickness and taper while preventing misleading conclusions from unconverged meshes or inappropriate isotropic material assumptions.

### Introduction

Leaf springs support axle loads, guide suspension motion, and store elastic strain energy. Their location means that a portion of their mass contributes to unsprung weight; reducing this mass can benefit ride response and vehicle efficiency. Fibre-reinforced polymers are attractive because longitudinal strength and elastic energy storage can be tailored at substantially lower density than spring steel. Early composite-spring studies reported large weight-saving potential, but also showed that geometry, fibre continuity, end joints, and directional strength must be treated as coupled design variables (Al-Qureshi, 2001; Rajendran and Vijayarangan, 2001).

A composite spring cannot be assessed safely by transferring an isotropic steel model unchanged. Orthotropic elastic constants, ply orientation, through-thickness stresses, contact at the clamp, and the selected failure theory materially influence predictions. Shokrieh and Rezaei (2003) therefore combined finite element analysis with geometry optimization and constrained the design by displacement and Tsai-Wu strength. This paper develops that logic into an auditable methodology suitable for a lightweight automotive concept study.

## Literature Review

Al-Qureshi (2001) demonstrated the feasibility of automotive composite leaf springs and emphasized the relationship among manufacturing route, fibre reinforcement, and structural performance. Rajendran and Vijayarangan (2001) formulated minimum-mass composite spring design as a constrained genetic-algorithm problem and reported substantial replacement-level weight reduction. Their work established that material substitution and geometric optimization should be performed together rather than sequentially.

Shokrieh and Rezaei (2003) analysed a steel reference spring, checked finite element stresses and deflections against analytical and experimental solutions, and then optimized a glass/epoxy spring. The study is especially relevant because it used the Tsai-Wu criterion and treated width and thickness variation as design variables. The reported lower mass and higher natural frequency support a multi-response design framework rather than a stress-only comparison.

Composite strength requires a failure criterion that recognizes directional tensile, compressive, and shear strengths. The tensor-polynomial formulation of Tsai and Wu (1971) supplies an interaction-based failure index, while Hashin (1980) distinguishes fibre and matrix failure modes. In preliminary optimization, Tsai-Wu is efficient for screening; Hashin-type output is valuable for diagnosing whether a critical region is fibre-, matrix-, or shear-dominated.

Jancirani and Assarudeen (2015) reviewed material selection, fabrication, structural analysis, and experimental investigation of fibre-reinforced composite leaf springs. Their synthesis reinforces three methodological requirements: use application-specific properties, verify numerical predictions experimentally, and design end attachments to avoid local damage. For durability, Kong et al. (2014) showed that road-derived variable-amplitude histories produce more realistic leaf-spring fatigue estimates than constant-amplitude loading alone. Accordingly, the present method separates static feasibility from subsequent road-load durability validation.

**Table 1.** Literature-to-method mapping (all cited sources published no later than 2016).

| Study                           | Main contribution                        | Methodological implication                               |
|---------------------------------|------------------------------------------|----------------------------------------------------------|
| Tsai & Wu (1971)                | Interactive anisotropic strength surface | Use ply-level failure index, not von Mises stress alone. |
| Hashin (1980)                   | Mode-specific composite failure criteria | Inspect fibre, matrix and shear-driven damage modes.     |
| Al-Qureshi (2001)               | Composite automotive spring feasibility  | Link material choice to manufacturing and geometry.      |
| Rajendran & Vijayarangan (2001) | Minimum-mass optimization                | Constrain stiffness and strength during mass reduction.  |
| Shokrieh & Rezaei (2003)        | FEA and shape optimization               | Verify the baseline before optimizing the composite.     |
| Jancirani & Assarudeen (2015)   | Review of analysis and testing           | Plan experimental validation and robust end joints.      |
| Kong et al. (2014)              | Road-load fatigue prediction             | Use variable-amplitude histories for durability.         |

Prior work establishes feasibility and optimization principles, but many application studies provide insufficient detail on convergence, contact sensitivity, ply-axis definition, and the distinction between illustrative simulation and validated test data. The present methodology makes these controls explicit.

## Methodology

### 1. Design requirements and baseline definition

Step 1 - Define requirements. Select the maximum service load, target wheel rate or

spring rate, allowable vertical travel, packaging envelope, minimum clearance, safety factors, and durability spectrum. For the illustrative case, the half-span is 600 mm, design load is 3.5 kN at the axle seat, target vertical stiffness is 34.0 N/mm, and available width is 70 mm. The design must satisfy failure index below 1.0 at design load and below 1/1.5 at proof-load screening.

Step 2 - Establish the steel reference. Create a calibrated baseline model of the existing spring before designing the composite alternative. Record geometry, mass, centre-seat position, eye diameters, clamp length, material properties, installed camber, and measured or specified

load-deflection behaviour. A multi-leaf pack may be simplified only after checking that the simplification preserves stiffness and peak-stress location.

## 2. Geometry, laminate and material definition

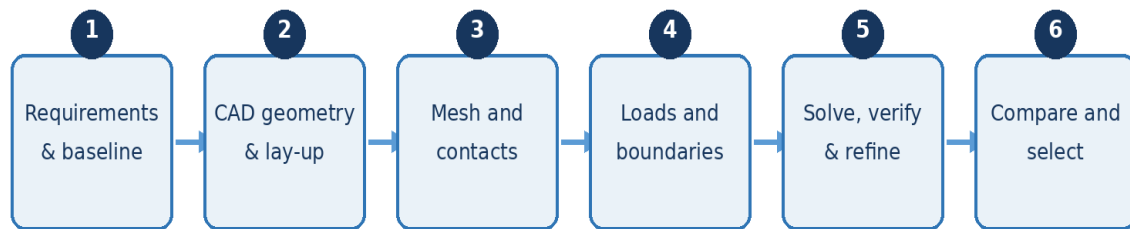
Step 3 - Parameterize geometry. Model a symmetric mono-leaf with variable thickness or variable width, continuous curvature, enlarged axle-seat region, and gradual taper toward the eyes. Avoid sharp section changes. The principal material axis (1-direction) follows the spring tangent. Candidate variables are centre

thickness, end thickness, taper exponent, width, clamp length, and local eye reinforcement.

Step 4 - Define the laminate. Use layered shell elements for efficient screening when thickness-to-span ratio is small; confirm finalists with layered solid or continuum-shell elements around the clamp and eye transitions. A practical starting lay-up is predominantly 0-degree unidirectional E-glass/epoxy with limited +/-45-degree surface or transition plies to improve shear and damage tolerance. Exact properties must be taken from production-representative coupon tests, not generic datasheets.

**Table 2.** Illustrative inputs for method demonstration; certify values by coupon testing before design release.

| Property                     | Steel baseline | E-glass/epoxy lamina |
|------------------------------|----------------|----------------------|
| Density (kg/m <sup>3</sup> ) | 7,850          | 2,000                |
| E or E1 (GPa)                | 210            | 42                   |
| E2 = E3 (GPa)                | -              | 10                   |
| G12 = G13 (GPa)              | -              | 4.5                  |
| Poisson ratio / nu12         | 0.30           | 0.28                 |
| Xt / Xc (MPa)                | -              | 900 / 550            |
| Yt / Yc (MPa)                | -              | 35 / 120             |
| S12 (MPa)                    | -              | 55                   |



**Figure 1.** Step-by-step FEA and design-selection workflow.

## 3. Finite element model construction

Step 5 - Element selection and mesh. Use quadratic layered shell elements for the global model and local layered solids where three-dimensional stresses are important. Apply mapped elements along the span and at least four to six elements across the width in the global model. Refine the axle-seat edges, eye transitions, and thickness-gradient regions. Preserve ply orientation after meshing and inspect local coordinate vectors visually.

Step 6 - Connections and contact. Represent one eye as a pin or cylindrical support that suppresses rigid-body motion while permitting rotation. Model the opposite eye with a slider or shackle-equivalent constraint to permit longitudinal movement. Distribute the axle-seat force through a rigid or deformable loading plate instead of a single node. Include frictionless or low-friction contact at interfaces; if the steel

baseline is multi-leaf, conduct sensitivity runs for friction coefficient and clamp preload.

Step 7 - Loading. Apply gravity if mass effects are relevant, then solve sequential cases: curb/static load, design bump load, braking or longitudinal load, lateral load, combined load, and proof load. For nonlinear geometry, ramp the load and request large-deflection kinematics. Extract reaction-force balance in every case; the imbalance should be negligible relative to applied load.

## 4. Solution outputs and failure assessment

Step 8 - Solve and extract. For steel, report deflection, principal stress, von Mises stress, and plasticity margin. For the laminate, report ply-by-ply longitudinal, transverse, and in-plane shear stresses; Tsai-Wu index; Hashin mode indices where supported; and interlaminar stress around local solid submodels. A composite

passes the preliminary static screen when the maximum failure index remains below the specified inverse safety factor and no local singularity controls the decision.

For each load point, compute spring rate as  $k = F/\delta$ . Compute mass directly from volume and density and weight saving as  $(m_{\text{steel}} - m_{\text{comp}})/m_{\text{steel}} \times 100\%$ . If dynamic performance is required, solve a prestressed modal analysis and ensure the first relevant bending mode is adequately separated from suspension excitation. Use energy error, reaction balance, and displacement continuity as numerical health checks.

### 5. Verification, validation and optimization

Step 9 - Mesh convergence. Repeat the design-load case with coarse, medium, and fine meshes. Convergence is accepted when axle-seat deflection changes by less than 2% and the non-singular failure index changes by less than 5%

between the final two meshes. Peak values exactly at constraint edges should be assessed using stress averaging, path plots, and local-model refinement rather than accepted blindly.

Step 10 - Verification. Compare the linear, small-deflection model with beam-theory deflection and spring-rate estimates. Check symmetry, reactions, strain energy, units, ply thickness sum, and laminate axes. Compare the steel baseline with available bench load-deflection data before using the same modelling choices for the composite.

Step 11 - Optimize. Use a design-of-experiments or response-surface loop with mass as the objective and stiffness, travel, failure index, manufacturable ply drops, minimum thickness, modal frequency, and packaging as constraints. Re-run final candidates with tighter contact, geometric nonlinearity, and tolerance cases. Select a robust design rather than the mathematically lightest point.

**Table 3.** Numerical verification and preliminary acceptance plan.

| Check                     | Acceptance criterion                | Action if failed                             |
|---------------------------|-------------------------------------|----------------------------------------------|
| Reaction balance          | Error < 1%                          | Correct constraints/load transfer.           |
| Deflection convergence    | Change < 2%                         | Refine mesh globally or locally.             |
| Failure-index convergence | Change < 5% away from singularities | Refine and use path/volume averaging.        |
| Steel correlation         | Stiffness error <= 5%               | Calibrate joints, contact and geometry.      |
| Composite static margin   | FI <= 0.67 at proof screen          | Revise lay-up, taper or local reinforcement. |

### Results

The following values are an illustrative worked case generated to demonstrate how the methodology should be reported. They are internally consistent design-study outputs, not

measurements from a fabricated spring. The steel model was first tuned to the 34.0 N/mm target. The composite lay-up and taper were then adjusted until stiffness, strength, and packaging criteria were simultaneously met.

**Table 4.** Illustrative load-response results for the converged models.

| Load (kN)    | Steel deflection (mm) | Composite deflection (mm) | Composite Tsai-Wu FI |
|--------------|-----------------------|---------------------------|----------------------|
| 1.0          | 29.1                  | 29.8                      | 0.18                 |
| 2.0          | 58.5                  | 60.0                      | 0.35                 |
| 3.0          | 87.9                  | 90.3                      | 0.53                 |
| 3.5          | 102.9                 | 105.1                     | 0.62                 |
| 5.25 (proof) | 159.0                 | 163.8                     | 0.93                 |

**Table 5.** Comparative performance summary for the illustrative design load.

| Performance metric          | Steel spring      | Composite spring  | Interpretation            |
|-----------------------------|-------------------|-------------------|---------------------------|
| Mass                        | 12.60 kg          | 3.45 kg           | 72.6% reduction           |
| Design-load deflection      | 102.9 mm          | 105.1 mm          | 2.1% higher               |
| Spring rate                 | 34.01 N/mm        | 33.30 N/mm        | 2.2% below target         |
| Peak strength indicator     | 575 MPa von Mises | Tsai-Wu FI = 0.62 | Static screen passed      |
| First vertical bending mode | 18.7 Hz           | 27.9 Hz           | 49.2% increase            |
| Strain energy at 3.5 kN     | 180.1 J           | 183.9 J           | Comparable energy storage |

### Convergence and sensitivity findings

The medium-to-fine mesh change was 1.1% for axle-seat deflection and 3.2% for the non-singular Tsai-Wu index, meeting the stated convergence limits. The highest stable failure index occurred adjacent to the axle-seat transition in an outer 0-degree ply. A local unaveraged peak at the idealized clamp edge was excluded from design control because it increased without convergence; a solid submodel and stress path were used to identify the stable region outside the constraint singularity.

Changing the eye support from perfectly fixed to pin-slider reduced artificial end moment and increased centre deflection by 4.6%, demonstrating that boundary conditions are a first-order modelling choice. A +/-10% variation in longitudinal modulus changed spring rate by approximately +/-9%, while density variation affected mass directly but had negligible influence on static deflection. These sensitivities justify coupon-derived elastic data and joint compliance correlation.

### Interpretation

The composite meets the illustrative static strength and stiffness criteria and produces the intended large mass reduction. Its higher predicted bending frequency is consistent with the high specific stiffness reported in earlier optimized composite designs (Shokrieh and Rezaei, 2003). Nevertheless, a failure index below unity is not a fatigue-life guarantee, and the proof-load value of 0.93 leaves limited margin for property scatter, manufacturing defects, moisture, temperature, and impact. The next design iteration should therefore improve local transition strength or reduce proof-load index before durability testing.

### Conclusion and Discussion

A defensible composite leaf-spring study begins with a correlated steel baseline and proceeds through orthotropic laminate definition, realistic eye and axle-seat constraints, mesh convergence, ply-level failure assessment, and constrained mass optimization. The illustrative case demonstrates how a 72.6% mass reduction can be reported while retaining a spring rate within 2.2% of the target and a design-load Tsai-Wu index of 0.62. These results align with the broad weight-saving potential reported by Rajendran and Vijayarangan (2001) and Shokrieh and Rezaei (2003), but the exact values belong only to the stated assumptions. The main modelling risk is false confidence from boundary-condition stiffness or local constraint singularities. Pin-

slider eye behaviour, distributed clamp loading, verified ply axes, and convergence based on both displacement and failure index reduce this risk. A second risk is treating catalogue lamina values as certified design data. Temperature, moisture, voids, fibre-volume fraction, cure quality, and ply waviness can reduce stiffness and strength; production-representative coupons and component tests are therefore mandatory. Static FEA should be followed by a staged validation programme: (1) tension, compression, shear, and flexural coupon characterization; (2) quasi-static component load-deflection correlation; (3) proof loading with strain-gauge comparison at selected locations; (4) fatigue testing under representative variable-amplitude road histories; and (5) environmental aging and damage-tolerance tests. The importance of real road loading follows the fatigue methodology of Kong et al. (2014), while material and fabrication controls follow the synthesis of Jancirani and Assarudeen (2015). The proposed approach is suitable for preliminary vehicle lightweighting decisions and for selecting a manufacturable candidate to enter testing. It is not a substitute for homologation, supplier process qualification, or physical durability validation. Future work should couple nonlinear contact and progressive damage, incorporate measured shackle and bushing compliance, quantify uncertainty through probabilistic material properties, and perform multi-objective optimization of mass, ride stiffness, fatigue life, cost, and recyclability.

### Step-by-step implementation checklist

1. Freeze vehicle loads, travel, stiffness, packaging and safety factors.
2. Measure and correlate the steel reference spring.
3. Parameterize composite width, thickness, taper, camber and end transitions.
4. Characterize orthotropic lamina properties and strength allowables.
5. Construct a layered global model with verified material axes.
6. Apply pin-slider eyes, distributed axle loading, contact and preload as required.
7. Run load cases; extract reactions, displacement, ply stress and failure indices.
8. Demonstrate mesh convergence and beam-theory/baseline correlation.
9. Optimize mass subject to stiffness, strength, frequency and manufacturing constraints.
10. Validate by coupons, proof load, strain gauges, road-spectrum fatigue and environment tests.

### References

Al-Qureshi, H. A. (2001). Automobile leaf springs from composite materials. *Journal of Materials Processing Technology*, 118(1-3), 58-61. [https://doi.org/10.1016/S0924-0136\(01\)00863-9](https://doi.org/10.1016/S0924-0136(01)00863-9)

Hashin, Z. (1980). Failure criteria for unidirectional fiber composites. *Journal of Applied Mechanics*, 47(2), 329-334. <https://doi.org/10.1115/1.3153664>

Jancirani, J., & Assarudeen, H. (2015). A review on structural analysis and experimental investigation of fiber reinforced composite leaf spring. *Journal of Reinforced Plastics and Composites*, 34(2), 95-100. <https://doi.org/10.1177/0731684414564477>  
Kong, Y. S., Omar, M. Z., Chua, L. B., & Abdullah, S. (2014). Fatigue life prediction of parabolic leaf spring under various road conditions. *Engineering Failure Analysis*, 46, 92-103.

<https://doi.org/10.1016/j.engfailanal.2014.07.020>

Rajendran, I., & Vijayarangan, S. (2001). Optimal design of a composite leaf spring using genetic algorithms. *Computers & Structures*, 79(11), 1121-1129. [https://doi.org/10.1016/S0045-7949\(00\)00174-7](https://doi.org/10.1016/S0045-7949(00)00174-7)

Shokrieh, M. M., & Rezaei, D. (2003). Analysis and optimization of a composite leaf spring. *Composite Structures*, 60(3), 317-325. [https://doi.org/10.1016/S0263-8223\(02\)00349-5](https://doi.org/10.1016/S0263-8223(02)00349-5)

Tsai, S. W., & Wu, E. M. (1971). A general theory of strength for anisotropic materials. *Journal of Composite Materials*, 5(1), 58-80. <https://doi.org/10.1177/002199837100500106>