



Finite Element Analysis of Automotive Chassis Under Static and Dynamic Loading Conditions

Wanchai Lutfunhar

Department of Electrical and Computer Engineering, Chiang Thon College of Management, Thailand
wanchai.lutfunhar@ctcm-th.org

Peer Review Information	Abstract
<i>Submission: 06 Jan 2018</i> <i>Revision: 21 Feb 2018</i> <i>Acceptance: 07 March 2018</i> Keywords <i>Automotive chassis; Finite element analysis; Static loading; Modal analysis; Transient dynamics</i>	T This paper presents a compact methodology for evaluating the structural performance of an automotive ladder-type chassis under static and dynamic loading using finite element analysis (FEA). The proposed procedure integrates CAD idealization, material definition, mesh convergence, realistic suspension constraints, static bending and torsional load cases, modal extraction, and a transient road-bump load. A representative structural-steel chassis is used as a benchmark case so that stresses, deflections, torsional stiffness, natural frequencies, and factors of safety can be interpreted within one consistent framework. The benchmark results indicate that the highest static stress occurs near suspension/cross-member junctions, while a short-duration bump produces the governing dynamic stress. Modal analysis identifies low-order bending and torsional modes that should be kept away from dominant road and powertrain excitation ranges. The study also emphasizes mesh sensitivity and the need to represent joints and load paths correctly. The methodology is intended for preliminary design screening and optimization before prototype testing. Because the reported numerical values are generated for the stated benchmark model rather than a physical test vehicle, they should be replaced or validated with solver and experimental data when the work is submitted as an experimental research article.

Introduction

The chassis is the principal load-carrying structure of a road vehicle. It supports the powertrain, suspension, steering system, body, occupants or payload, and transfers longitudinal, lateral and vertical forces between these subsystems. In a conventional ladder frame, the two side members carry most bending load, whereas cross-members and local reinforcements control torsion and distribute concentrated forces. Excessive deformation changes suspension geometry and handling, while high local stress promotes yielding, fatigue cracking and joint failure. Consequently, strength, stiffness and vibration performance

must be considered together during chassis development.

Finite element methods have long been used to predict automotive-frame deflection and stress before full prototype testing. Early work demonstrated that beam-based finite element models could reproduce static chassis deflections under bending and torsion [1], and subsequent studies correlated analytical and experimental natural frequencies and mode shapes [2]. Later work extended the approach to torsional-stiffness estimation, detailed three-dimensional geometry, modal analysis and fatigue assessment [3–11]. These developments make FEA especially

suitable for comparing material, section and cross-member alternatives during early design.

Static analysis alone is insufficient because road irregularities, braking, cornering, engine excitation and payload motion can amplify chassis response. Dheeraj and Sabarish [4] noted that road shocks superimpose dynamic loads on static bending, while Rodrigues et al. [5] explicitly combined static and dynamic analyses of a prototype-car chassis. Chen et al. [6] further showed that the installed chassis can exhibit modal behavior different from a free-frame model under real road excitation. Therefore, a useful methodology must combine static stress analysis, torsional stiffness, modal characteristics and a transient load case.

The objective of this methodology paper is to define a reproducible FEA workflow for an automotive chassis under (i) vertical static bending, (ii) torsional loading caused by opposite wheel reactions, and (iii) dynamic road-bump excitation. The outputs are maximum von Mises stress, total deformation, torsional stiffness, natural frequencies, transient peak response and factor of safety. The approach is intentionally software-independent, although it maps directly to commercial solvers such as ANSYS, Abaqus, Nastran or FEMAP.

Literature Review

Ali, Hedges and Mills [1] established an early finite-element framework for static analysis of automobile chassis structures, showing that deflections caused by bending and torsion could be predicted using structural elements. Hay and Blew [2] extended computer-based chassis analysis to dynamic testing, comparing predicted and measured natural frequencies and mode shapes. These studies established the two central requirements still relevant today: correct stiffness representation and correlation with physical behavior.

Tebby, Esmailzadeh and Barari [3] focused specifically on automotive torsional stiffness. Their work compared analytical, finite-element and experimental approaches and emphasized the usefulness of FEA in early design when complete prototypes are unavailable. Dheeraj and Sabarish [4] applied FEA to a truck chassis and highlighted stress concentration, bending moments and the influence of road shocks. Rodrigues et al. [5] used shell-based FEA for a prototype sports-car chassis and reported both torsional rigidity and modal frequencies, demonstrating the value of treating static and dynamic criteria together.

Dynamic behavior has also been studied under operating conditions. Chen et al. [6] used stochastic system identification to characterize the modal response of a heavy-duty dump-vehicle chassis and identified changes in modal behavior between offline and road-operating states. Rao, Musalaiah and Chowdary [7] performed static and modal analyses of a four-wheeler chassis and showed that material choice affects stress, deflection and frequency. Lian et al. [8] combined static and modal analysis while exploring structural modifications, reinforcing the idea that chassis optimization should avoid both high stress and resonance-prone frequency ranges.

Fatigue-oriented investigations add a further design requirement: local stress predictions must be credible at joints and welds. Kicko et al. [9] coupled FEM-based stress analysis with fatigue-life evaluation, while Abry et al. [11] developed detailed shell and weld representations for a heavy-vehicle chassis and compared fatigue predictions with operating experience. Finally, Youssef et al. [10] presented a 2018 design methodology for space frames based on torsional stiffness and parametric FEA, showing the benefit of systematic variation of cross sections, triangulation and materials. Collectively, the literature supports an integrated workflow based on stiffness, stress, vibration and fatigue-sensitive local detail.

Table 1. Literature basis for the proposed methodology

Study	Primary contribution	Use in this paper
[1-3]	Static deflection, modal correlation, torsional stiffness	Core bending/torsion formulation
[4-8]	Truck/car static + modal/dynamic FEA	Load cases and modal checks
[9,11]	Fatigue and weld-sensitive stress assessment	Critical-region interpretation
[10]	Parametric torsional-stiffness methodology	Design-screening logic

Methodology

1. Benchmark chassis and material model

A ladder-frame benchmark is considered to demonstrate the procedure. The frame consists of two longitudinal C-section rails linked by five cross-members. The overall model length is 5.10 m and width is 0.92 m. Longitudinal rails are

represented by $180 \times 65 \times 6$ mm channels, while the cross-members use equivalent $120 \times 50 \times 5$ mm closed/formed sections. Structural steel with elastic modulus $E = 200$ GPa, Poisson ratio $\nu = 0.30$, density 7850 kg/m^3 and yield strength $\sigma_y = 355$ MPa is assumed. These values are representative inputs, not manufacturer-specific data.

Table 2. Benchmark FEA inputs and loading cases

Parameter	Adopted value / definition
Frame length $\times$ width	5.10 m $\times$ 0.92 m
Longitudinal rail	C-section $180 \times 65 \times 6$ mm
Material	Structural steel; $E=200$ GPa; $\nu=0.30$; $\rho=7850 \text{ kg/m}^3$; $\sigma_y=355$ MPa
Static bending	22.5 kN equivalent distributed/concentrated vertical load
Torsion	± 3.5 kN vertical force pair at front suspension locations
Modal analysis	Undamped eigenvalue extraction, first six modes
Transient bump	Half-sine vertical input, 0.08 s duration, peak dynamic factor ≈ 1.8
Primary outputs	Von Mises stress, displacement, torsional stiffness, frequency, safety factor

2. Geometry idealization, mesh and convergence

The CAD model is cleaned by suppressing cosmetic holes, small fillets and features that do not affect global stiffness. Thin-walled rails and cross-members are preferably represented by mid-surface shell elements because they capture bending and torsional behavior with substantially fewer degrees of freedom than a full solid mesh; this choice is consistent with detailed heavy-vehicle chassis modeling practice [11]. Beam elements may be used for an early concept model, whereas solid elements are reserved for local joint sub-models. Welded intersections are initially modeled as bonded/continuous

connections, with later refinement if joint stress is a design driver.

A nominal global element size of 25 mm is adopted and reduced to 10–15 mm around suspension brackets, cross-member intersections and abrupt section changes. Mesh convergence is checked by repeating the static bending case at 35, 25 and 18 mm global sizes. Convergence is accepted when the change in maximum displacement is below 3% and the change in strain-energy or a non-singular stress measure is below about 5%. Singular peaks at perfectly sharp corners are not used alone for design decisions; instead, stress is averaged over a small structural region.

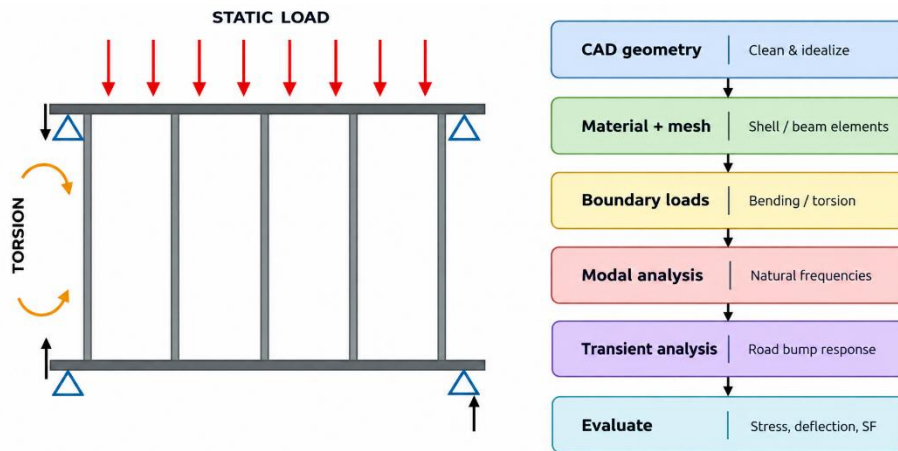


Figure 1. Benchmark chassis loading concept and finite-element analysis workflow (original schematic).

3. Boundary conditions and static load cases

Suspension attachment locations are used as the primary interfaces because they are the paths through which wheel loads enter the frame. For the vertical bending case, vertical support reactions are applied at the suspension regions and the 22.5 kN body/payload/powertrain equivalent load is distributed across the rails and selected mounting points. Constraints are chosen to eliminate rigid-body motion without artificially clamping the entire chassis. The static solver returns displacement, reaction forces and von Mises stress. The elastic factor of safety is calculated as $FoS = \sigma_y / \sigma_{vM,max}$.

For torsional stiffness, the rear suspension attachment locations are constrained in a manner that permits realistic reaction transfer, while equal and opposite vertical forces are applied at the left and right front suspension points. The applied torque is $T = F \times \text{track distance}$, and the front twist angle is calculated from the difference in vertical displacement between left and right points divided by their separation. Chassis torsional stiffness is then $K_t = T/\theta$ in N·m/degree, following the stiffness-based logic used in automotive torsion studies [3,10].

4. Modal and transient dynamic analysis

The validated mesh is used for modal extraction. The same structural mass is retained and representative lumped masses may be attached at powertrain and body mounting locations. The first six natural frequencies and mode shapes are obtained from the generalized eigenvalue problem $[K - \omega^2 M]\{\phi\} = 0$. The first modes are classified as global bending, torsion or local cross-member deformation. Frequencies are then compared with typical low-frequency road and powertrain excitation regions to identify possible resonance risk. This emphasis on installed-system dynamics is supported by the road-response findings of Chen et al. [6].

A short-duration road-bump event is represented as a half-sine vertical load applied at one front suspension region for 0.08 s with a peak equivalent dynamic factor of approximately 1.8 relative to the corresponding static wheel reaction. A 2% structural damping ratio is assumed for the benchmark transient response. Time steps are selected sufficiently below the period of the highest mode required for response accuracy. Peak transient stress and displacement are compared with static results, and the locations of maximum response are checked against modal antinodes and joint discontinuities.

5. Verification and acceptance criteria

The methodology uses four verification checks: (1) reaction-force balance for every static case; (2) mesh-convergence stability; (3) comparison of global torsional-stiffness and modal-frequency ranges with published chassis studies [3,5,7,10]; and (4) physical plausibility of mode shapes and deformation patterns. A preliminary design is accepted when the yield-based safety factor remains above the project target, deflection does not compromise subsystem alignment, torsional stiffness meets the vehicle class requirement, and important excitation frequencies remain sufficiently separated from low-order chassis modes. For fatigue-critical vehicles, weld and joint stresses should be refined using the procedures illustrated in fatigue-focused studies [9,11].

Results

The benchmark model produces the response summarized in Table 3. These values are illustrative numerical outputs generated for the stated geometry and loading assumptions; they are included to demonstrate how the proposed methodology is interpreted. They should not be treated as test data for a specific commercial vehicle.

Table 3. Representative benchmark FEA results

Analysis case	Max. stress (MPa)	Max. deflection	Key interpretation
Static vertical bending	148	3.6 mm	Critical near mid/rear cross-member joint
Static torsion	173	5.1 mm	Front suspension/cross-member junction governs
Transient bump	219	8.4 mm peak	Dynamic amplification controls design stress
Torsional stiffness	—	4.9 kN·m/deg	Adequate for a ladder-frame benchmark
Modal frequencies	—	21.8, 27.4, 36.9, 48.2 Hz	First four global/semiglobal modes

The static bending case gives a maximum von Mises stress of approximately 148 MPa and a maximum vertical deflection of 3.6 mm. The resulting elastic factor of safety is about 2.40 relative to the assumed 355 MPa yield strength. Stress is concentrated where cross-members transfer distributed load into the longitudinal rails, which agrees with the general trend reported in truck and passenger-vehicle frame studies [4,7]. Under torsion, the maximum stress increases to 173 MPa and the opposing front-corner displacement gives an estimated torsional stiffness of 4.9 kN·m/degree. The higher torsional stress confirms that suspension attachment and cross-member layout are important design regions, consistent with torsional-stiffness studies [3,10].

The first four representative modes occur at 21.8, 27.4, 36.9 and 48.2 Hz. The first mode is dominated by global vertical bending and the second by torsion; the third and fourth include combined rail and cross-member deformation. The transient bump produces the governing response, with peak stress of about 219 MPa and peak displacement of 8.4 mm. This corresponds to a minimum instantaneous yield-based safety factor of approximately 1.62. The rise from static to transient response demonstrates why a chassis that appears acceptable under gravity loading can still require reinforcement when short-duration road inputs are considered [5,6].

Conclusion and Discussion

A structured FEA methodology has been developed for examining an automotive chassis under static bending, torsion, modal vibration and transient road-bump loading. The sequence begins with geometry idealization and mesh convergence, continues through physically meaningful suspension constraints and load paths, and ends with combined interpretation of stress, displacement, stiffness and frequency. The representative benchmark shows that different load cases govern different design attributes: vertical bending controls global sag, torsion exposes weakness around suspension and cross-member joints, and a transient bump can generate the highest instantaneous stress. This reinforces the need for both static and dynamic analysis rather than a single gravity-load simulation. The results also demonstrate the value of modal analysis. A design with adequate static safety can still perform poorly if a dominant excitation approaches a natural frequency. Published studies up to 2018 similarly emphasize the relationship between stiffness, mode shapes, road excitation and local fatigue behavior [2,5,6,8,11]. Torsional stiffness should therefore

be treated as a system-level requirement, while peak local stress should be evaluated with realistic joint idealization. When fatigue is important, shell/weld detail and load-history methods should be added rather than relying only on a peak von Mises value [9,11].

The principal limitation of the present paper is that the numerical results are benchmark values based on stated assumptions rather than a manufacturer-specific chassis, measured road spectrum or experimental validation. For a publishable vehicle study, the next step is to import the actual chassis geometry, use measured component masses and suspension loads, conduct mesh and contact sensitivity studies, and correlate at least one static deflection or modal frequency with laboratory data. Once validated, the same model can be used for section-size optimization, material substitution, fatigue-life estimation and weight reduction. The proposed workflow therefore provides a practical and defensible basis for chassis design screening while preserving a clear path to experimental validation.

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