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Multi-Model Verification of a Pushrod-Actuated Double-Wishbone Suspension for Formula Racing Applications

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Abstract

Pushrod-actuated double-wishbone suspensions offer substantial kinematic freedom, but the mechanical advantage of the pushrod-rocker linkage can make spring selection misleading when damper stiffness is interpreted directly as vertical wheel stiffness. This study develops a verification-oriented multi-model framework that connects constrained suspension kinematics, MATLAB/Simulink damper-axis force calculations, MSC Adams/Car kinematics-and-compliance outputs, geometry-dependent rate transformations, and a two-degree-of-freedom dynamic screening model through shared physical variables. Front and rear damper spring rates are 39.181 and 56.517 N/mm, while damper-to-wheel motion ratios of 0.8967 and 0.6700 produce installed wheel rates of 31.504 and 25.371 N/mm, respectively. Thus, the softer front damper produces a 24.2% higher vertical wheel rate because of the rocker geometry. Setting the motion ratio to unity would overestimate the installed-rate frequency estimate by 11.5% at the front and 49.3% at the rear. For a 30 mm raised-cosine road input, peak sprung accelerations are 1.145 g and 1.002 g for the front- and rear-rate parameter sets. Independent fixed-step classical fourth-order Runge-Kutta calculations agree with a tight-tolerance adaptive Runge-Kutta reference to within 0.011% for the selected peak metrics at a 2 ms step. The framework establishes an auditable numerical verification baseline and identifies steering hardpoints as the priority for subsequent geometric refinement and physical validation.

Keywords: multibody system dynamics, double-wishbone suspension, pushrod actuation, motion ratio, MSC Adams/Car, numerical verification

1 Introduction

Formula racing vehicles are purpose-built single-seaters whose suspension systems must provide precise tyre-load management and controllable wheel orientation within severe packaging constraints. Current FIA formula categories are governed by category-specific technical regulations, providing a representative professional single-seater context for the suspension architecture considered here [1]. Unequal-length double-wishbone suspensions are attractive in formula racing vehicles because the hardpoints can be selected to control camber, toe, steering-axis geometry and roll-centre migration while satisfying severe packaging constraints [2–5]. Pushrod actuation additionally moves the spring-damper unit inboard and permits the wheel-to-damper displacement transformation to be tuned by the rocker geometry.

The same geometric freedom creates an important modelling problem. A damper spring rate is not, in general, the vertical wheel rate: the transformation depends on the instantaneous motion ratio, and the square of the motion ratio enters the stiffness relationship under an ideal, lossless linkage. Classic racing-vehicle formulations already emphasize the importance of instantaneous motion ratio [3, 14]. More recent work has developed analytical and multibody descriptions of double-wishbone kinematics [5–7], equivalent reduced-order models [8], virtual formula-racing development environments [9], full-vehicle multibody set-up optimization [10], heave-roll decoupling [11], and automated suspension design loops [12]. Recent work also shows that nonlinear double-wishbone geometry can be solved efficiently enough to support optimization rather than relying only on small-displacement assumptions [7].

For formula-racing development programmes, however, design evidence is often distributed across analytical calculations, MATLAB/Simulink models, Adams/Car K&C results and reduced-order dynamic models. These model levels may use consistent physics yet differ in coordinates, units and implicit assumptions. The practical research question addressed here is therefore not the invention of a new suspension topology, but whether the available model levels can be made mutually consistent and whether the geometry-dependent rate transformation materially changes the design interpretation.

This paper develops a verification-oriented multi-model workflow for a pushrod-actuated double-wishbone formula-racing suspension. The contributions are fourfold. First, a common signal interface is defined for wheel travel, dependent coordinates, damper travel, damper-axis force, K&C outputs, motion ratio and installed wheel rate. Second, the force/rate calculations are independently verified with consistent SI units. Third, the quarter-car screening model is subjected to solution verification by comparing adaptive RK45 integration with fixed-step classical RK4. Fourth, a motion-ratio ablation and bounded sensitivity study quantify how strongly rocker geometry affects installed-rate conclusions. The analysis is intentionally separated from physical validation: no claim is made that the current numerical study substitutes for measured hardpoints, damper dynamometer data, tyre stiffness measurements or synchronized track telemetry.

2 Suspension model and multi-model methodology

2.1 System architecture and shared variables

Figure 1(a) shows the pushrod-actuated double-wishbone topology considered in the study. Vertical wheel motion is transmitted through the upright and wishbones to a pushrod, which rotates a rocker and compresses the inboard spring-damper. Figure 1(b) shows the information flow used to connect the model levels. The analytical branch defines the constrained coordinates; the MATLAB/Simulink branch supplies the damper-axis loading used for spring-rate calculation; exported Adams/Car K&C outputs provide wheel-travel-dependent steering trends; the motion-ratio transformation converts damper stiffness to vertical wheel stiffness; and the resulting installed rate parameterizes the reduced-order dynamic model.

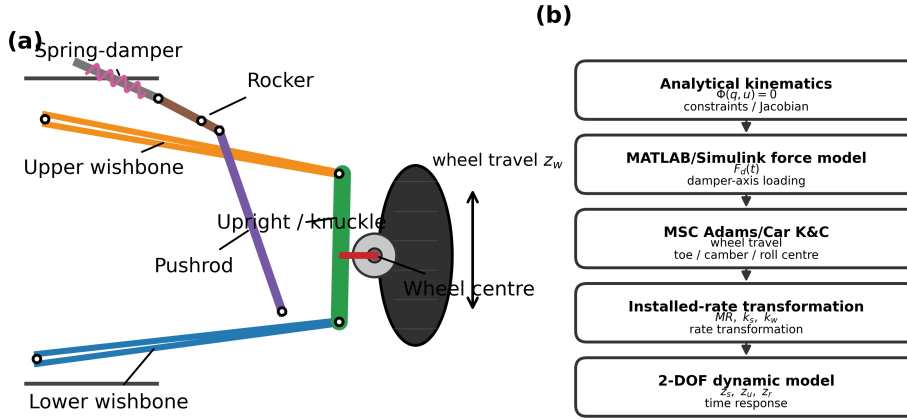


Fig. 1 Pushrod-actuated double-wishbone suspension and the multi-model analysis workflow. (a) Programmatically redrawn kinematic diagram based on the suspension geometry, with the principal links, rocker, spring-damper, upright, wheel and chassis pivots identified. (b) Model levels connected by shared physical variables rather than a live solver-to-solver interface

The workflow is a sequential multi-model analysis rather than a runtime Adams Controls co-simulation. The native Adams database and plant export were not available for this study; consequently, Adams/Car is represented by the available K&C outputs from the vehicle-development model. This terminology is important because live bidirectional exchange between Adams and MATLAB/Simulink requires an Adams Controls plant interface [16, 17].

2.2 Constrained kinematics

Let $\mathbf{q}$ contain the dependent generalized coordinates of the suspension representation and let u denote a prescribed wheel- or control-arm-travel input. The closed-loop geometry is expressed as

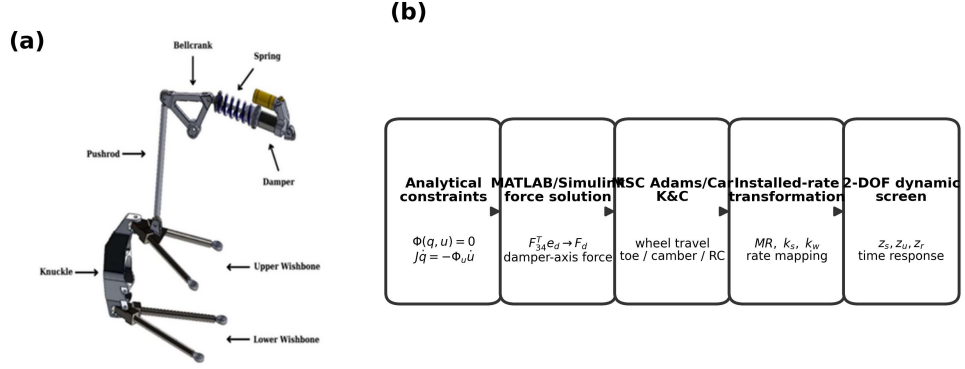


Fig. 2 Pushrod-actuated double-wishbone suspension assembly from the project evidence and programmatically redrawn sequential multi-model verification architecture. The present implementation exchanges quantities offline; a native Adams Controls bidirectional run is the next re-execution step when original model files become available

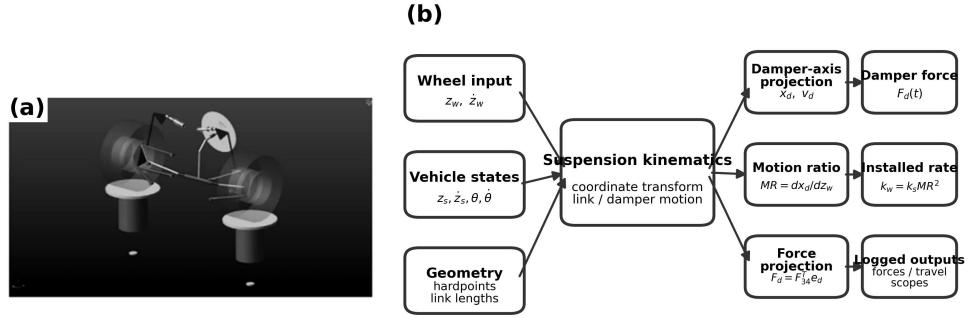


Fig. 3 Computational models used in the coupled suspension analysis: (a) MSC Adams/Car front half-car view cropped from the author-provided project evidence, with the embedded legacy figure caption removed, and (b) a programmatically redrawn MATLAB/Simulink-style force-analysis block diagram based on the documented signal flow. The two branches remain sequentially coupled through shared physical variables; native runtime exchange files were not available

$$\Phi(\mathbf{q}, \mathbf{u}) = 0.$$

The constraint Jacobian is

$$\mathbf{J}(\mathbf{q}, \mathbf{u}) = \frac{\partial \Phi}{\partial \mathbf{q}}.$$

A locally full-rank $\mathbf{J}$ is required for a unique velocity solution. Differentiating the constraint relation gives

$$\mathbf{J}\dot{\mathbf{q}} = -\Phi_u \dot{u}.$$

A second differentiation yields

$$\mathbf{J}\ddot{\mathbf{q}} = \gamma, \quad \gamma = -\left(\dot{\mathbf{J}}\dot{\mathbf{q}} + \frac{d\Phi_u}{dt}\dot{u} + \Phi_u\ddot{u}\right).$$

This form retains the prescribed input explicitly and avoids treating a time-varying wheel input as a homogeneous constraint.

2.3 Constrained dynamics and damper-axis force

For a Lagrange-multiplier multibody formulation, the accelerations and constraint reactions may be written as

$$\begin{bmatrix} \mathbf{M} & \mathbf{J}^T \\ \mathbf{J} & \mathbf{0} \end{bmatrix} \begin{bmatrix} \ddot{\mathbf{q}} \\ \lambda \end{bmatrix} = \begin{bmatrix} \mathbf{Q} \\ \gamma \end{bmatrix}.$$

Here $\mathbf{M}$ is the mass matrix, $\mathbf{Q}$ contains applied and inertial generalized forces, and λ contains the constraint multipliers [4, 6]. If $\mathbf{F}_{34}$ is the reaction transmitted into the spring-damper linkage and $\mathbf{e}_d$ is the unit vector along the damper axis, the useful axial load is

$$F_d(t) = \mathbf{F}_{34}^T(t)\mathbf{e}_d.$$

The projection is necessary because the magnitude of a joint reaction is not generally equal to the axial load carried by the spring-damper.

2.4 Spring rate, motion ratio and installed wheel rate

Over an approximately linear interval, the secant spring rate is

$$k_s = \frac{\Delta F_d}{\Delta x_d},$$

where x_d is damper-axis displacement. For a nonlinear spring or progressive rocker, the local tangent dF_d/dx_d should be used instead. The instantaneous damper-to-wheel motion ratio is defined as

$$MR = \frac{dx_d}{dz_w}, \quad k_w = k_s MR^2,$$

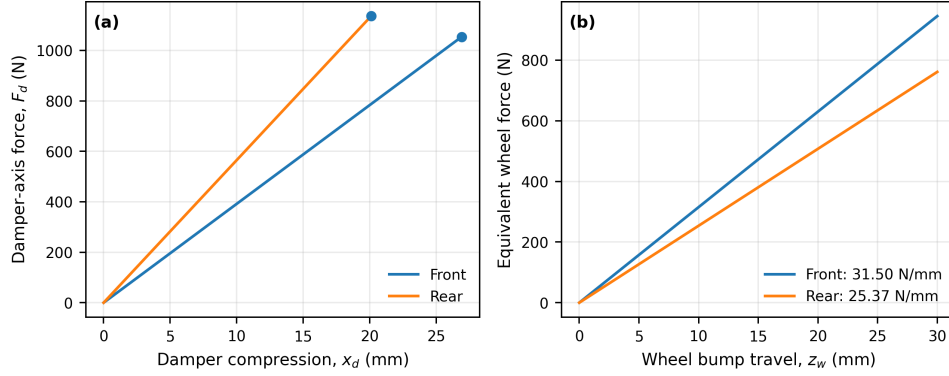
where z_w is vertical wheel travel and k_w is the installed vertical wheel rate. The squared term follows from virtual work: displacement is scaled by MR , while force is transformed by the reciprocal mechanical advantage [2–4, 14]. The present analysis uses a secant motion ratio from ride height to +30 mm bump because continuous damper-displacement data were not available.

The model force/rate quantities and the resulting installed rates are summarized in Table 1. The front damper compression is inferred from the available peak force and spring rate; the rear damper compression is obtained directly from the model spring-length change. All conversions use 1 lb/in = 0.175126835 N/mm.

Figure 4 illustrates why the transformation matters: although the rear damper spring is substantially stiffer, the front geometry produces the larger vertical wheel rate.

Table 1 Damper-axis quantities and geometry-dependent installed rates

Quantity	Front	Rear	Basis
Peak damper-axis force	1054 N	1136 N	Model output
Damper compression at peak	26.901 mm	20.100 mm	Derived/model output
Damper spring rate	39.181 N/mm	56.517 N/mm	Verified calculation
Damper spring rate	223.73 lb/in	322.72 lb/in	Verified calculation
Damper-to-wheel motion ratio	0.8967	0.6700	Secant geometry
Installed wheel rate	31.504 N/mm	25.371 N/mm	Motion-ratio transformation

**Fig. 4** Damper-axis spring-rate relationship and the corresponding installed wheel-rate transformation

2.5 Adams/Car K&C response and steering behaviour

The vehicle-development model used static equilibrium, parallel wheel travel for heave and opposite wheel travel for roll. These are standard K&C excitations for extracting toe, camber, wheel rate and roll-centre behaviour in Adams/Car [16]. The available toe/steer plot did not include an exported numerical channel; therefore three clearly readable anchors at -30, 0 and +30 mm wheel travel were digitized as -1.3, 0 and +2.0 deg. The lowest-order polynomial through these anchors is

$$\delta_{toe}(z_w) = 3.8889 \times 10^{-4} z_w^2 + 0.055 z_w,$$

with z_w in millimetres and δ_{toe} in degrees. A ± 0.07 deg band is used only to communicate approximate plot-read uncertainty; it is not a statistical confidence interval. Figure 5 shows the resulting response.

2.6 Reduced-order dynamic model and solution verification

A linear two-degree-of-freedom quarter-car model is used as a screening layer to translate the installed wheel rates into a time-domain response. The equations are

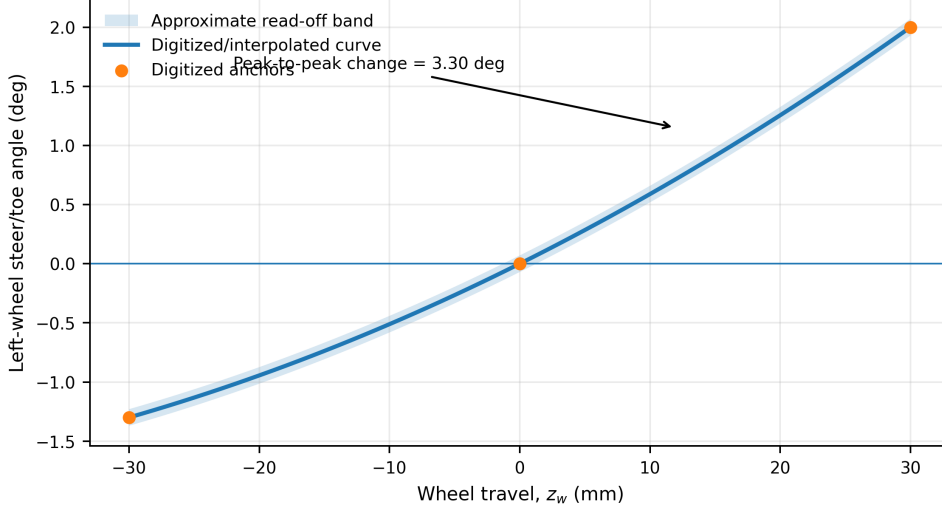


Fig. 5 Digitized left-wheel toe/steer response from the available Adams/Car K&C plot. The 3.30 deg peak-to-peak change over ± 30 mm is sufficiently large to require hardpoint refinement

$$m_s \ddot{z}_s + c_s (\dot{z}_s - \dot{z}_u) + k_w (z_s - z_u) = 0,$$

$$m_u \ddot{z}_u + c_s (\dot{z}_u - \dot{z}_s) + k_w (z_u - z_s) + k_t (z_u - z_r) = 0.$$

The damping coefficient is defined from the chosen damping ratio as

$$c_s = 2\zeta \sqrt{k_w m_s}.$$

The road displacement is a raised-cosine pulse,

$$z_r(t) = \begin{cases} \frac{h}{2} \left[1 - \cos \left(\frac{2\pi(t-t_0)}{T} \right) \right], & t_0 \leq t \leq t_0 + T, \\ 0, & \text{otherwise.} \end{cases} \quad (1)$$

with $h = 30$ mm, $T = 0.25$ s and $t_0 = 0.25$ s. Table 2 separates measured/model-derived quantities from screening assumptions. The 980 N static corner load is taken from the Adams/Car equilibrium output. The unsprung mass, tyre vertical rate and damping ratio should be replaced by measured values before physical validation.

The reference solution is obtained with adaptive RK45 using relative and absolute tolerances of 10^{-9} and 10^{-11} , respectively. Numerical solution verification is performed independently with classical fixed-step RK4 at 2.0, 1.0 and 0.5 ms. Stability of the continuous linear model is checked by confirming that every eigenvalue of the state matrix has a negative real part. This distinction follows ASME VVUQ terminology: verification addresses correct solution and implementation of the mathematical model, whereas validation addresses agreement with the physical system for an intended context of use [18, 19].

Table 2 Parameters of the dynamic screening model

Parameter	Value	Status
Static corner load	980 N	Adams/Car equilibrium output
Unsprung corner mass	15.0 kg	Screening assumption
Sprung corner mass	84.93 kg	Derived from static load
Tyre vertical rate	200 N/mm	Screening assumption
Damping ratio	0.35	Screening assumption
Road input	30 mm; 0.25 s	Defined raised-cosine pulse

3 Results

3.1 Installed-rate consistency

The rear spring calculation gives $1136/(198.4 - 178.3) = 56.517$ N/mm, equivalent to 322.72 lb/in. The front value of 223.73 lb/in converts to 39.181 N/mm; together with the 1054 N peak force it implies 26.901 mm damper compression. The corresponding motion ratios are 0.8967 front and 0.6700 rear. Consequently, the installed front wheel rate is 31.504 N/mm and the rear wheel rate is 25.371 N/mm. The front rate is therefore 24.2% larger at the wheel even though the front damper spring itself is 30.7% softer than the rear damper spring.

A deliberately naive $MR = 1$ baseline quantifies the importance of the rocker transformation. Using the damper spring rate directly as wheel rate would increase the installed-rate frequency estimate from 3.065 to 3.418 Hz at the front (+11.5%) and from 2.751 to 4.106 Hz at the rear (+49.3%). The rear result is particularly important because the smaller motion ratio changes the qualitative interpretation of which axle is dynamically stiffer.

3.2 Steering response

The digitized toe/steer response changes from approximately -1.3 deg at 30 mm rebound to +2.0 deg at 30 mm bump, a 3.30 deg full-range variation. For a racing vehicle this magnitude is not negligible because it changes the steer angle demanded of the tyre as vertical load and suspension travel vary. The result identifies the steering/tie-rod hardpoints as the highest-priority geometric refinement. Because the response was digitized from a raster plot, geometry release should be based on re-exported left- and right-wheel toe channels from the native K&C model rather than on the digitized curve alone.

3.3 Dynamic screening response

Figure 6 compares the two installed-rate sets under the common 30 mm raised-cosine input. The front-rate set produces the higher peak sprung acceleration and dynamic tyre-force component, consistent with its higher installed stiffness. Table 3 reports the principal response metrics.

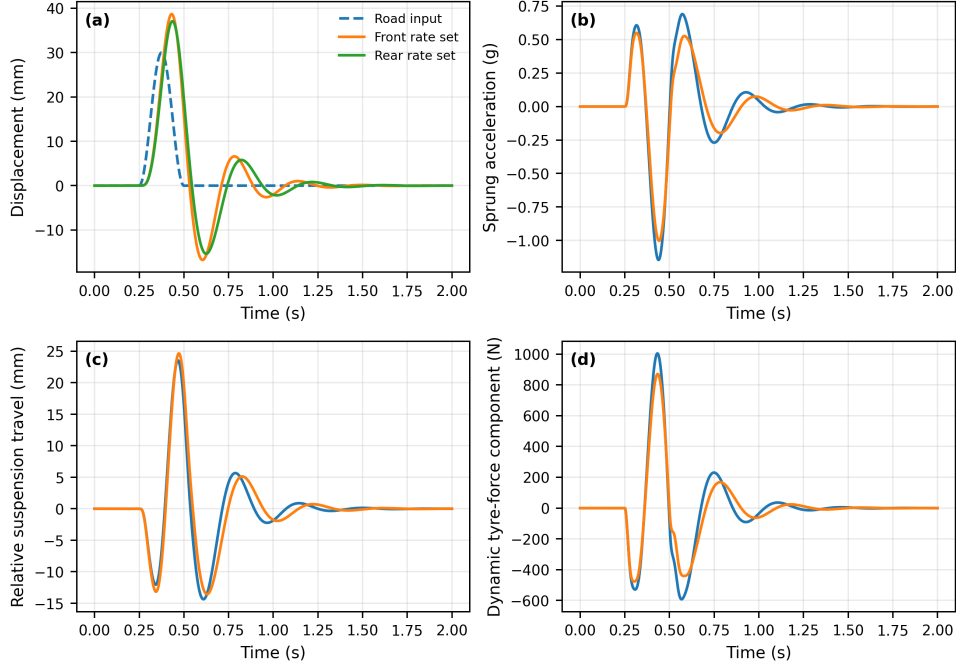


Fig. 6 Two-degree-of-freedom dynamic screening response for the common 30 mm raised-cosine road input. Front and rear labels refer to installed-rate parameter sets, not separate full-vehicle axle models

Table 3 Dynamic screening metrics

Metric	Front-rate set	Rear-rate set
Body-mode frequency	3.065 Hz	2.751 Hz
Damping coefficient	1145 N s/m	1028 N s/m
Peak sprung acceleration	1.145 g	1.002 g
Peak suspension travel	23.54 mm	24.65 mm
Peak dynamic tyre-force component	1004 N	870 N
Settling time	1.400 s	1.364 s

These values are comparative screening outputs rather than predictions of complete vehicle behaviour. A real formula racing car couples the four corners through chassis heave, pitch and roll, anti-roll bars, aerodynamic loading, steering compliance and nonlinear tyres.

3.4 Numerical convergence and bounded sensitivity

The fixed-step RK4 solutions converge closely to the tight-tolerance RK45 reference. At a 2.0 ms step, the largest relative error among the two rate sets is 0.0109% for peak sprung acceleration and 0.0106% for peak suspension travel. At 1.0 ms the

corresponding maximum errors fall to 0.0046% and 0.0031%, and at 0.5 ms the selected metrics are unchanged to the displayed precision. Table 4 and Figure 7(a) therefore support numerical solution stability over the tested step sizes.

Table 4 Fixed-step RK4 convergence relative to the tight-tolerance RK45 reference

RK4 step	Max error: peak acceleration	Max error: peak travel
2.0 ms	0.0109%	0.0106%
1.0 ms	0.0046%	0.0031%
0.5 ms	< 0.00001%	< 0.00001%

Wheel rate varies linearly with k_s and quadratically with MR . Over a deterministic screening range of $\pm 15\%$ in motion ratio and $\pm 10\%$ in spring rate, the installed-rate frequency estimate spans 2.47-3.70 Hz for the front-rate set and 2.22-3.32 Hz for the rear-rate set. These ranges are engineering sensitivity envelopes, not probabilistic confidence intervals, because distributions of hardpoint position, spring tolerance, corner mass and tyre stiffness are not available.

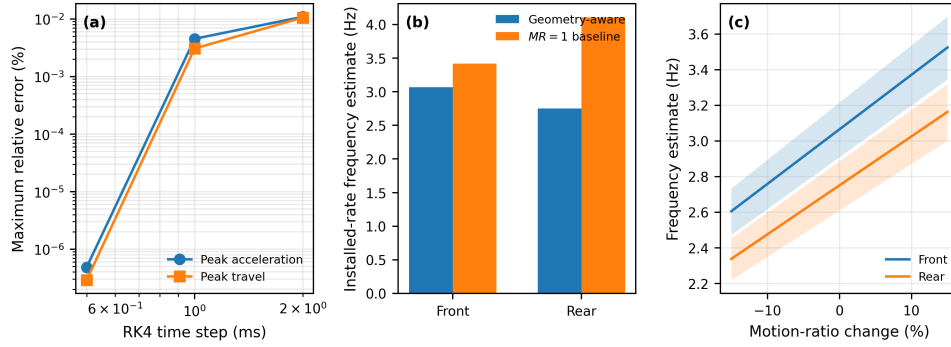


Fig. 7 Verification and sensitivity analyses. (a) Fixed-step RK4 error relative to the tight-tolerance RK45 solution. (b) Motion-ratio ablation. (c) Frequency sensitivity to motion-ratio variation; shaded regions also include $\pm 10\%$ spring-rate variation

4 Discussion

The results expose a design issue that is easy to obscure when damper components and wheel behaviour are discussed with the same stiffness terminology. The front damper spring is softer than the rear spring, yet the front wheel rate is higher because the front motion ratio is closer to unity. The $MR = 1$ ablation makes this distinction quantitative and shows that geometry cannot be treated as a minor correction, especially at the

rear. This finding is consistent with competition-vehicle modelling approaches that use instantaneous motion ratios rather than catalogue spring rate alone [3, 14].

The steering result is the second important design outcome. The approximately 3.30 deg toe/steer change across the available travel envelope is large relative to the objective of maintaining predictable tyre slip demand. Double-wishbone design studies commonly optimize hardpoints to control wheel-alignment variation [5, 7, 13]. Here the available curve indicates that steering hardpoints should be included explicitly in the next optimization loop rather than treating bump steer as already acceptable. A practical update would vary the inner and outer tie-rod locations while constraining rack packaging and Ackermann behaviour, and would use the maximum absolute toe change or root-mean-square toe deviation over the intended wheel-travel range as an objective.

The multi-model workflow is deliberately narrower than a full virtual prototype. Cherian et al. [6] compared analytical and ADAMS suspension models with experimental K&C evidence; Balena et al. [10] developed a complete formula-style multibody racing vehicle for set-up optimization; Panchal and Bastiaan [11] examined heave-roll decoupling; and Borowski and Rudolph [12] demonstrated much deeper digital continuity and automated optimization. The present contribution is instead a verification layer that keeps coordinates, units and rate transformations consistent across model fidelities and quantifies the error introduced when the rocker geometry is ignored. This is directly relevant during design development, where component selection is often made before a fully correlated vehicle model is available.

There are three main limitations. First, the native Adams/Car database and numerical hardpoint table were not available, so the current study cannot re-run the K&C model or compute a point-by-point analytical-to-Adams error norm. Second, the quarter-car layer uses a declared unsprung mass, tyre vertical stiffness and damping ratio; these parameters are suitable for comparative screening but not component sign-off. Third, the toe curve is digitized from an exported plot and therefore carries read-off uncertainty. These limitations define the next validation stage rather than invalidating the internal numerical checks.

The most valuable continuation is a closed verification-and-validation loop. A common CAD baseline should supply hardpoints, masses and inertias to both the analytical model and Adams/Car. Parallel and opposite wheel travel should then be exported numerically at identical displacement points, allowing toe, camber, roll-centre height, damper travel and wheel rate to be compared by maximum error and root-mean-square error. Measured corner masses, tyre vertical stiffness and damper force-velocity curves should replace screening assumptions. Finally, calibrated damper displacement, steering angle, wheel-speed and inertial channels should be synchronized during a low-risk track manoeuvre. Such data would permit physical validation in the ASME VVUQ sense rather than only verification of numerical consistency [18, 19].

5 Conclusions

A verification-oriented multi-model analysis of a pushrod-actuated double-wishbone formula-racing suspension has been presented. The analysis links constrained kinematics,

damper-axis force calculation, Adams/Car K&C evidence, geometry-dependent rate transformation and a reduced-order dynamic model without implying unavailable live Adams-MATLAB coupling.

The front and rear damper spring rates are 39.181 and 56.517 N/mm, while the corresponding motion ratios of 0.8967 and 0.6700 yield installed wheel rates of 31.504 and 25.371 N/mm. The resulting front wheel rate is 24.2% higher despite the softer front damper spring. Ignoring motion ratio would overestimate the installed-rate frequency estimate by 11.5% at the front and 49.3% at the rear. Independent RK4 integration verifies the quarter-car peak metrics to within 0.011% of the tight-tolerance RK45 reference at a 2 ms step.

The digitized Adams/Car steering trace changes by approximately 3.30 deg over +/-30 mm wheel travel and therefore identifies tie-rod and steering hardpoints as the principal geometric refinement. The dynamic results are intentionally treated as comparative screening rather than physical vehicle validation. The next engineering milestone is numerical re-execution from a common hardpoint dataset followed by component measurements and synchronized track correlation.

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Statements and Declarations

Author contributions. Ogul Ertugrul: investigation, suspension model development, simulation, data curation, visualization, and writing - original draft. Caglar Uyulan: conceptualization, methodology, software verification, formal analysis, supervision, visualization, and writing - review and editing. Both authors approved the submitted manuscript.

Competing interests. The authors declare no financial or non-financial interests that are directly or indirectly related to the work submitted for publication.

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Data availability. The numerical quantities and parameter values required to reproduce the algebraic rate calculations and the two-degree-of-freedom screening analysis are reported in the article, including the digitized toe-response anchors used in the steering analysis. The study also relies on author-provided project evidence from the vehicle-development model. The native Adams/Car database, Adams Controls plant export, and raw track telemetry were not available for the present study and therefore cannot be deposited or re-executed from the supplied evidence.

Materials availability. Not applicable.

Code availability. The governing equations, numerical parameters, integration settings, and verification tolerances required to reproduce the reported calculations are provided in the article. No separately archived source-code package accompanies this submission.

Ethics approval. Not applicable. This computational engineering study does not involve human participants, human data, human tissue, or animals.

Consent to participate. Not applicable.

Consent for publication. Not applicable.

Use of generative AI. OpenAI ChatGPT was used during manuscript preparation to assist with language restructuring, reference-format checking, and preparation of reproducible plotting/document-generation code. The authors reviewed and verified the technical statements, equations, calculations, references, and figures and take full responsibility for the final manuscript. No generative AI system was used to create or alter primary research data. No generative-AI-created images are included in this submission; schematic and flow-diagram artwork was redrawn programmatically from author-provided model evidence and documented quantities.

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