

A computational model for analysis of the dynamic response of a quarter-vehicle suspension system

P.L. Sikhakhane, E. Joni, L. Mokgotho & Dr. D. Modungwa
Council for Scientific and Industrial Research, Pretoria, Gauteng, South Africa

ABSTRACT: Off-road vehicle suspension performance directly affects ride behaviour, structural loading, and vehicle control under irregular terrain excitation. This paper presents a cross-platform computational framework for analyzing the vertical dynamic response of a quarter-vehicle suspension system using three modelling approaches: a reduced-order MATLAB model and multibody models in Project Chrono and MSC ADAMS. A common suspension definition and terrain input were used to compare sprung mass displacement, unsprung mass displacement, suspension travel, and sprung mass acceleration. MATLAB and Project Chrono show the closest agreement, while MSC ADAMS predicts a smoother, more damped response. The framework provides a practical basis for comparing modelling fidelity in quarter-vehicle suspension analysis under off-road excitation.

1 INTRODUCTION

Suspension systems improve ride comfort, vehicle control, stability, passenger comfort, and structural protection, especially in off-road vehicles where uneven terrain causes continuous changes in wheel loading, body motion, and dynamic response (Aye Y., et al. 2024).

The quarter-vehicle model is widely used to analyse vertical suspension behaviour because it represents sprung and unsprung mass motion as a two-degree-of-freedom system. It is useful for studying body bounce, wheel hop, and suspension travel. However, many studies use a single modelling platform and simplified sinusoidal road inputs, which do not fully represent irregular off-road terrain.

This paper presents a computational framework for analysing the vertical dynamic response of a quarter-vehicle suspension system under off-road terrain excitation. A reduced-order MATLAB model is

compared with multibody models in Project Chrono and MSC ADAMS using the same geometry and terrain input. The comparison evaluates response trends and the effect of model fidelity on predicted suspension behaviour, as shown in Figure 1.

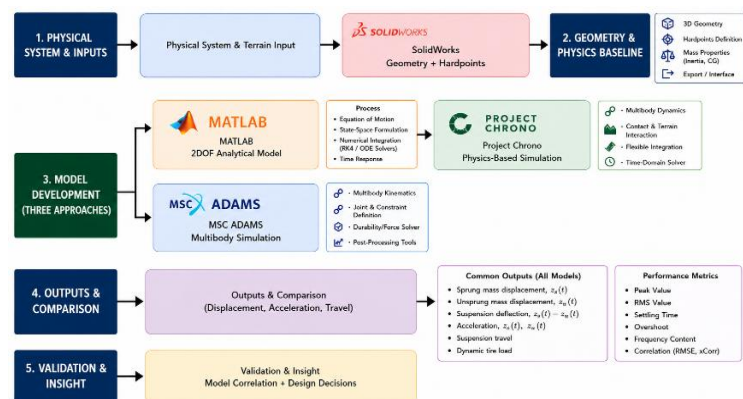


Figure 1: Integrated modelling workflow for the quarter-vehicle suspension system.

2 LITERATURE REVIEW

2.1 Quarter-vehicle modelling approach

The quarter-vehicle model is widely used to study vertical vehicle dynamics because it represents the suspension as a two-degree-of-freedom system of sprung and unsprung masses. It captures body bounce, wheel hop, and suspension response with low computational cost (Gillespie T.D 1992), (Rajamani R. 2012), (Aye et al., 2024), (Sathishkumar & Jancirani, 2014).

However, its lumped-mass and linear spring-damper assumptions limit its ability to represent geometry, joint constraints, and kinematic effects under irregular off-road excitation (Gawthrop P. 2013), (Salem, 2018).

2.2 Mathematical analysis approach

Mathematical analysis uses Newton's second law to form coupled equations for sprung and unsprung mass motion. These equations provide displacement, velocity, and acceleration responses, often solved using numerical methods such as RK4 integration (Sathishkumar et al.).

This reduced-order approach provides a practical baseline for transient suspension behaviour and for benchmarking more detailed simulation models, though it does not fully represent geometric effects or constraint behaviour (Guo S. et al. 2025), (College V. 2020; Karahan M.), (Karanhan M.), (Hosni & Salem 2018), (Rajamani, 2012), (Sathishkumar & Jancirani, 2014), (Abd Ali and Al-Zughaibi, 2024), (Ashtekar & Thakur, 2014), (Goh, Booker & McMahon, 2005).

2.3 Multibody dynamic simulation

Multibody dynamic simulation provides a more detailed representation of suspension behaviour by accounting for geometry, inertia, and joint constraints. MSC ADAMS is widely used for vehicle handling, stability, and suspension response studies where link motion and joint interaction must be represented in detail (Zhang K. et al. 2025), (Huang et al., 2010), (Konieczny, Burdzik & Łazarz, 2013).

Project Chrono provides an open-source alternative supporting CAD-based geometry, hardpoint definition, and time-domain dynamic analysis. Previous work has demonstrated its effectiveness for vehicle dynamics and suspension modelling (Negrut D., Serban R. & Tasora A. 2017). Compared with analytical models, multibody approaches require more setup effort but better represent geometry and constraint effects (Rankinen, Ikonen & Lieder, 2018).

2.4 Terrain excitation modelling

Terrain excitation drives wheel motion and the resulting vehicle response in off-road analysis. Bekker established the theoretical framework for terrain-vehicle systems, while Wong and Reece extended this through pressure-sinkage and shear behaviour studies in off-road mobility. (Bekker M.G 1969), (Wong J. & Reece A. 1967).

Terrain input in suspension studies is often represented by step, sinusoidal, or statistical road profiles. Dodds and Robson provided a widely used statistical road roughness description for vibration analysis. (Dodds C.J & Robson J.D 1973). However, these simplified forms do not capture the irregular nature of off-road terrain; irregular bump sequences are therefore more appropriate for realistic suspension response studies (Pazooki, Rakheja & Cao, 2012), (Pazooki et al., 2011).

3 MATHEMATICAL MODEL

The quarter-vehicle model represents the vertical motion of the suspension system. As shown in Figure 2, it consists of a sprung mass, unsprung mass, suspension spring-damper element and tyre stiffness excited by road displacement $z_r(t)$.

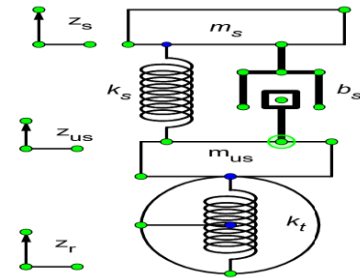


Figure 2: Free-body diagram of the quarter-vehicle suspension model showing the sprung mass m_s , unsprung mass m_{us} , suspension stiffness k_s , damping b_s , and road input $z_r(t)$.

Using Newton's 2nd law of motion, the equations of motion are written as,

$$m_s \ddot{z}_s = b_s(\dot{z}_{us} - \dot{z}_s) + k_s(z_{us} - z_s) \quad (1)$$

$$m_{us} \ddot{z}_{us} = b_s(\dot{z}_s - \dot{z}_{us}) + k_s(z_s - z_{us}) + k_t(z_r - z_{us}) \quad (2)$$

where M_s and M_{us} are the sprung and unsprung masses, K_s is the suspension stiffness, b_s is the suspension damping coefficient and K_t is the tyre stiffness. The variables z_s , z_{us} , and z_r represent sprung mass displacement, unsprung mass displacement and road displacement, respectively.

4 MULTIBODY MODEL DEVELOPMENT

4.1 SolidWorks model

A three-dimensional quarter-vehicle suspension model was developed in SolidWorks Figure 3, consisting of the chassis, control arm, wheel-hub, wheel assembly, spring and damper. Hardpoints were defined for consistent joint placement, and material properties were assigned to set mass and inertia. This CAD model served as the common geometric baseline for both Project Chrono and MSC ADAMS.

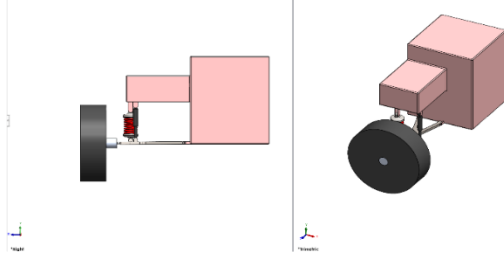


Figure 3: SolidWorks quarter-vehicle suspension model used as the common geometry for Project Chrono and MSC ADAMS simulations.

4.2 Project Chrono set-up

The quarter-vehicle model was implemented in Project Chrono using SolidWorks geometry, preserving assembly layout, hardpoint locations and mass properties. Revolute joints were used at the control arm pivots, and a translational spring-damper element represented the suspension force between chassis and suspension assembly.

Table 1: Parameters

Parameters	Symbol	Value
Sprung mass	m_s	450 kg
Unsprung mass	m_{us}	100 kg
Suspension stiffness	k_s	25000 N/m*
Suspension damper	b_s	2000 Ns/m*
Tyre stiffness	k_t	200000 N/m *
Time step	Δt	0.001 s
Simulation time	t	5 s
Gravity	g	-9.81 m/s ²

* Parameters selected based on typical quarter-vehicle suspension ranges for off-road applications

4.3 MSC ADAMS model

The same geometry was implemented in MSC ADAMS by importing the SolidWorks assembly into ADAMS/View and rebuilding suspension connections using the defined hardpoints. Revolute, spherical, translational and planar constraints represented the suspension motion. A bushing element at the wheel-hub stabilized the wheel-side response while preserving overall suspension movement under the common terrain input.

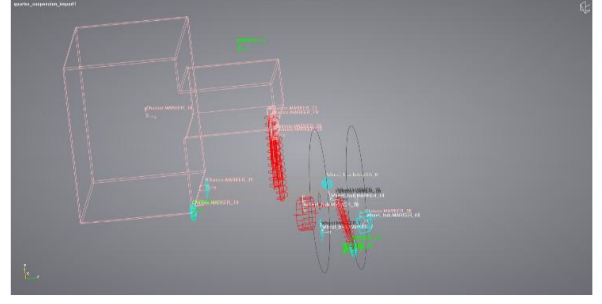


Figure 4: MSC ADAMS quarter-vehicle suspension model showing the imported CAD geometry, hardpoint-based constraints, and spring-damper elements used for dynamic simulation.

5 SIMULATION SETUP

A common simulation setup was used across all three platforms. Excitation was applied as a prescribed vertical road displacement $Z_r(t)$ at the wheel-ground interface, defined as a sequence of five bumps characterized by start position, length and height (Equation 6). A constant forward speed converted the spatial profile into a time-dependent input containing both low-frequency body motion and higher-frequency wheel response.

$$bumps = \begin{bmatrix} (1.0, 0.60, 0.05) \\ (2.4, 0.50, 0.04) \\ (3.7, 0.70, 0.06) \\ (4.5, 0.80, 0.07) \\ (2.4, 0.40, 0.03) \end{bmatrix} \quad (6)$$

where each tuple represents the start position, bump length, and bump height.

All simulations ran for 5 s with a fixed time step of 0.001 s. MATLAB used RK4 numerical integration; Project Chrono and MSC ADAMS used their built-in time-domain solvers under identical excitation and time-stepping conditions. Model parameters are listed in Table 1.

Sprung mass displacement, unsprung mass displacement, suspension travel and sprung mass acceleration were selected as comparison outputs, allowing the three approaches to be evaluated on a consistent basis. $z_s(t)z_{us}(t)z_s - z_{us}\ddot{z}_s$.

6 RESULTS AND DISCUSSION

The dynamic responses predicted by MATLAB, Project Chrono and MSC ADAMS were compared under the same terrain excitation and suspension parameters, focusing on sprung mass displacement, unsprung mass displacement, suspension travel and sprung mass acceleration (Sathishkumar and Jancirani, 2014), (Abd Ali & Al-Zughaibi, 2024).

Figure 5 shows the sprung mass displacement response. All three models capture the same overall body motion and main transient events. MATLAB

and Project Chrono show the closest agreement in the displacement envelope, while MSC ADAMS predicts a smoother response with slightly reduced peak amplitudes. The three approaches are consistent in representing body bounce, though the detailed response is sensitive to solver behaviour and multibody constraint implementation (Rankinen, Ikonen & Lienes, 2018), (Goh, Booker & McMahon, 2005).

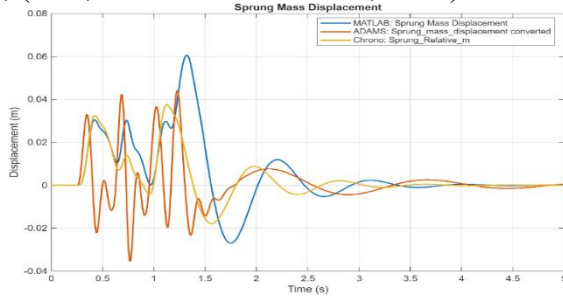


Figure 5: Sprung mass displacement versus time for MATLAB, Project Chrono, and MSC ADAMS under irregular terrain excitation.

Figure 6 presents the unsprung mass displacement response. The unsprung mass shows faster oscillations than the sprung mass due to direct road excitation. MATLAB and Project Chrono follow similar transient behaviour, while the ADAMS response is lower in amplitude and more constrained. All three models capture the same sequence of wheel-side excitation events, with amplitude differences driven by modelling detail (Ashtekar & Thakur, 2014), (Pazooki, Rakheja & Cao, 2012).

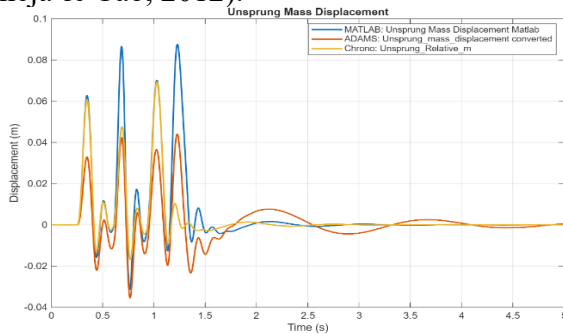


Figure 6: Unsprung mass displacement versus time for MATLAB, Project Chrono, and MSC ADAMS under irregular terrain excitation.

Figure 7 shows suspension travel, the relative displacement between sprung and unsprung masses. All three models show similar timing of main peaks and troughs, confirming a broadly consistent suspension response. MATLAB and Project Chrono remain closer in shape and amplitude, while ADAMS produces a smoother response with smaller later oscillations, capturing the same overall deformation pattern despite differences in model fidelity (Sathishkumar & Jancirani, 2014), (Rankinen, Ikonen & Lienes, 2018).

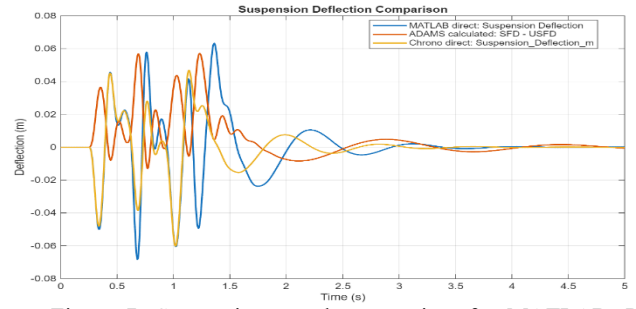


Figure 7: Suspension travel versus time for MATLAB, Project Chrono, and MSC ADAMS under irregular terrain excitation

Figure 8 presents the sprung mass acceleration response, directly related to ride behaviour. MATLAB and Project Chrono predict larger early acceleration peaks, while ADAMS gives a lower-amplitude, smoother response attributed to joint constraints, stabilizing elements and geometric detail in the multi-body formulation (Huang et al., 2010), (Konieczny, Burdzik & Łazarz, 2013). Despite quantitative differences, all three models reproduce the same main transient pattern.

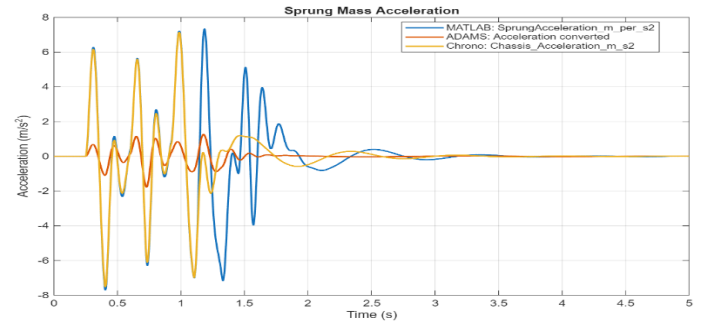


Figure 8: Sprung mass acceleration versus time for MATLAB, Project Chrono, and MSC ADAMS under irregular terrain excitation.

Overall, the three approaches show good agreement in response timing and general shape. MATLAB and Project Chrono provide the closest match, while MSC ADAMS follows the same broad trends with a more damped, constrained response, confirming that model fidelity and constraint representation have a clear effect on predicted behaviour.

6.1 Comparative Analysis of Model Fidelity and Discrepancies

The differences between MATLAB, Project Chrono, and MSC ADAMS arise from fundamental differences in model formulation, numerical implementation, and physical representation, analyzed below based on Figures 5–8.

6.1.1 Model Formulation and Degrees of Freedom

The MATLAB model uses a 2-DOF lumped-parameter system with only vertical motion (z_s , z_{us}), linear spring-damper elements and direct ODE integration, which is computationally efficient but unable to capture geometric effects from link lengths or joint orientations. Project Chrono implements a multi-DOF multibody system with 6-DOF rigid body motion and

hardpoint-based revolute joints, introducing kinematic coupling and 3D force projection, though geometric effects remain small for predominantly vertical excitation.

The ADAMS model provides the most comprehensive constraint representation revolute, spherical, translational, and planar joints, with a wheel-hub bushing element adding 6-DOF compliance and damping beyond the physical parameters. Baumgarte constraint stabilization and geometric nonlinearity from large-displacement kinematics are also included.

The 2-DOF MATLAB model cannot represent lateral motion, pitch coupling, or geometric stiffening present in the 3D multibody models, explaining the larger oscillation amplitudes in Figures 5–7: it lacks the geometric damping and constraint-induced energy dissipation that occur in Chrono and ADAMS.

6.1.2 Numerical Integration and Solver Algorithms

MATLAB uses RK4 with a fixed time step of $\Delta t = 0.001$ s, introducing minimal numerical damping and preserving high-frequency oscillations explaining the persistent oscillations in Figures 5–8. Project Chrono employs an implicit Euler scheme with constraint violation correction, reducing high-frequency content while preserving main transient features. ADAMS uses the Gear Stiff (GSTIFF) implicit variable-order method with Baumgarte stabilization, whose (α , β) parameters add velocity- and position-proportional feedback, producing the smoothest response by effectively filtering high-frequency content.

6.1.3 Damping Mechanisms

All three models share the same physical suspension damper ($b_s = 2000$ Ns/m, $\zeta \approx 0.3$), but total effective damping differs significantly. MATLAB includes only the explicit damper and minimal RK4 numerical damping. Project Chrono adds moderate implicit numerical damping and constraint projection dissipation. ADAMS additionally includes the wheel-hub bushing, constraint stabilization (artificial velocity-proportional damping) and implicit integrator damping, producing peak accelerations of ± 7.5 , ± 7.0 and ± 1.5 m/s² respectively, an 80% reduction in ADAMS attributed to bushing attenuation, Baumgarte damping and GSTIFF filtering.

6.1.4 Geometric and Kinematic Effects

MATLAB has no geometric representation forces that are always vertical, and stiffness is constant. Project Chrono includes a finite control arm length (~ 0.4 m), producing modest geometric stiffening (~ 5 – 10% at maximum compression). ADAMS captures full 3D kinematics including control arm swing, instant-centre migration, and force projection. These effects, combined with bushing compliance, explain ADAMS' $\sim 50\%$ lower unsprung displacement amplitude (0.045 m vs. 0.085 m, Figure 6).

6.1.5 Frequency Content Analysis

Frequency analysis of the acceleration responses reveals that MATLAB and Project Chrono both show a dominant body bounce frequency near 1.5 Hz and a wheel hop peak near 10 Hz, with broad content to 20 Hz. Project Chrono reduces content above 15 Hz by $\sim 30\%$. ADAMS attenuates the wheel hop mode (~ 10 Hz) by $\sim 70\%$, retaining only 15% of high-frequency content its bushing and constraint stabilization acting as a low-pass filter with a cutoff near 8–10 Hz.

6.1.6 Quantitative Comparison

Table 2 summarizes the key quantitative differences. MATLAB and Project Chrono agree within 10% for all displacement and travel metrics, confirming that 2-DOF dynamics are preserved in the Chrono multibody model for predominantly vertical excitation. ADAMS shows 30–50% reduction in displacement metrics, 80% reduction in peak acceleration metrics, and $\sim 40\%$ shorter settling time (3.1 s vs. >5.0 s), reflecting the trade-offs between computational efficiency, geometric fidelity and numerical stability.

Table 3: Quantitative deviation of the three models

Metric	MATLAB	Project Chrono	MSC Adams
Peak sprung displacement (m)	0.062	0.058 (-6%)	0.042 (-32%)
Sprung mass settling time (s)	>5.0	4.2	3.1
Sprung oscillation frequency (Hz)	1.52	1.48	1.45
Peak unsprung displacement (m)	0.085	0.078 (-8%)	0.045 (-47%)
Suspension travel range (m)	0.125	0.116 (-7%)	0.086 (-31%)
Peak sprung acceleration (m/s ²)	7.52	6.98 (-7%)	1.48 (-80%)
RMS sprung acceleration (m/s ²)	1.82	1.64 (-10%)	0.42 (-77%)
High-freq. content >10 Hz (%)	100	~ 70	~ 15

6.1.7 Model Selection Implications

The MATLAB model is best suited for rapid parametric studies where vertical dynamics dominate. Project Chrono is appropriate when 3D geometry effects matter, open-source access is required, or terrain-tire integration is planned. ADAMS is preferred for high-fidelity kinematic analysis and experimental correlation but bushing and stabilization parameters must be carefully calibrated to avoid excessive artificial damping. The close MATLAB–Chrono agreement validates both for fundamental suspension dynamics, while ADAMS differences highlight the importance of understanding modelling choices.

7 CONCLUSION

This paper presented a cross-platform framework for analysing the vertical dynamic response of a quarter-vehicle suspension system using MATLAB, Project Chrono, and MSC ADAMS under a common suspension definition and terrain input. The three models captured similar transient response trends, including the main excitation events, oscillations, and return toward equilibrium. MATLAB and Project Chrono showed the closest agreement, while MSC ADAMS produced a smoother and more damped response.

The MATLAB model provides a fast baseline for early-stage analysis, while Project Chrono and MSC ADAMS add insight into geometry, kinematics, and constraint effects. The results should be interpreted as trend-level agreement rather than full model equivalence. Future work should include experimental validation, improved wheel-side kinematic modelling, and extension to higher-order vehicle models under broader off-road conditions.

8 REFERENCES

- Abd Ali, L.M. and Al-Zughaibi, A.I. (2024) ‘Comparing the performance of using a smart damper in a semi-active suspension instead of a traditional damper using MATLAB/Simulink’, *International Journal of Applied Mechanics and Engineering*, 29(3). doi: 10.59441/ijame/189957
- Ashtekar, J.B. and Thakur, A.G. (2014) ‘Simulink model of suspension system and its validation on suspension test rig’, *International Journal of Current Engineering and Technology*, 4(1), pp. 433–436
- Aye Y., Win H., Swe W.M., & Lat A.K. (2024). Vibration analysis of suspension system for 3DOF quarter car model with tire damping. *IJMERR* 13. Doi: 10.18178/ijmerr.13.2.196-204.
- Bekker M.G (1969). Introduction to terrain-vehicle systems. SEMANTIC,
- Colpo, L.R. and de Souza, C.E. (2020) ‘Assessment of adjustable damping in the ride comfort of a Baja SAE vehicle’, *Journal of the Brazilian Society of Mechanical Sciences and Engineering*, 42(12), p. 617. doi: 10.1007/s40430-020-02660-4
- College V. (2020) Modelling of a vehicle suspension system epreuve syntese de programme (esp), differential equations 201-h1l-va, probability and statistics 201-hth-05 by Tadeus Talmaci Kamila Yahiaoui.
- Dodds C.J & Robson J.D (1973). The description of road surface roughness.
- Gawthrop P. (2013) Lumped parameter modelling of vehicle dynamics.
- Gillespie T.D (1992) Fundamentals of vehicle dynamics. SAE International.
- Goh, Y.M., Booker, J.D. and McMahon, C. (2005) ‘Uncertainty modelling of a suspension unit’, *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, 219(6), pp. 755–771. doi: 10.1243/095440705X28321
- Guo S., Wang R., & Wang Z. (2025). Robust elastic H_∞ control strategy for the agricultural transportation system of the mountain tea garden under multi-constraint terrain. 2025 IEEE 3rd International Conference on Sensors, Electronics and Computer Engineering (ICSECE). Doi: 10.1109/icsece65727.2025.11256816.
- Huang, Z.-G., Wang, F., Zhu, H. and Wang, J. (2010) ‘ADAMS/Car for simulation of vehicle handling and stability’, *Mechanical Engineering & Automation*, 5. doi: 10.3969/j.issn.1006-9348.2010.05.085
- Kaldas, M.M.S., Çalışkan, K., Henze, R. and Küçükay, F. (2014) ‘Optimization of damper top mount characteristics to improve vehicle ride comfort and harshness’, *Shock and Vibration*, 2014, Article ID 248129. doi: 10.1155/2014/248129
- Karahan M. Modeling and simulation of an active quarter car suspension with a robust LQR controller under road disturbance and parameter Uncertainty.
- Konieczny L., Burdzik R. & Łazarz B. (2013) 'Research on dynamics of car suspension system in MSC ADAMS software', *Journal of Measurements in Engineering*, 1(2), pp. 10
- Negrut D., Serban R. & Tasora A. (2017) Vehicle dynamics simulation using project chrono.
- Pazooki, A., Cao, D., Rakheja, S. and Boileau, P.-É. (2011) ‘Ride dynamic evaluations and design optimisation of a torsio-elastic off-road vehicle suspension’, *Vehicle System Dynamics*, 49(9), pp. 1455–1476. doi: 10.1080/00423114.2010.516833.
- Pazooki, A., Rakheja, S. and Cao, D. (2012) ‘Modeling and validation of off-road vehicle ride dynamics’, *Mechanical Systems and Signal Processing*, 28, pp. 679–695. doi: 10.1016/j.ymssp.2011.11.006
- Rajamani R. (2012) Vehicle dynamics and control.
- Rankinen, A., Ikonen, E. and Lienes, T. (2018) ‘Validation of a nonlinear two-dimensional MacPherson suspension system model with multibody simulations’, in *Proceedings of the IEEE/ASME International Conference on Mechatronic and Embedded Systems and Applications (MESA)*, Oulu, Finland. doi: 10.1109/MESA.2018.8449167
- Salem, M. (2018) *Investigation of a Non-linear Suspension in a Quarter Car Model*. MSc dissertation. University of Nottingham, Nottingham, UK
- Sathishkumar P, Jancirani J, John D, Manikandan S *International Journal of Innovative Research in Science, Engineering and Technology*.
- Wong J. & Reecel A. (1967) Prediction of rigid wheel performance based on the analysis of soil wheel stresses 4:7–25.
- Zhang K., Junwei S., Jinglai W.Y, Zhang, Wu Á.J., Zhang Y., & Shi J (2025) A mechanical model of elastic wheel-terrain interaction for real-time simulation. 113. Doi: 10.1007/s11071-025-11094-3(.