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COMPARATIVE STUDY OF THE PERFORMANCE OF MACPHERSON AND MULTI-LINK SUSPENSIONS IN DYNAMIC CONDITIONS

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Abstract: *This paper presents a comparative study between two representative automotive suspension architectures, namely the MacPherson strut and the Multilink system, with the objective of evaluating their dynamic behaviour and vibration response when subjected to different operating conditions. The analysis was conducted through virtual simulations in ADAMS/View, focusing on the suspension subsystem without steering components, and included both time-domain and frequency-domain evaluations. Kinematic investigations were first performed, examining camber angle variation versus wheel travel, which directly influences lateral grip and tire–road contact stability. Dynamic analyses were carried out by simulating vehicle response when crossing standardized road obstacles, where body accelerations and oscillation damping characteristics were monitored, thus providing an indication of ride comfort and vibration decay. In addition, a sine sweep excitation was applied to study transmissibility and resonance behaviour over a broad frequency spectrum, highlighting the suspension's ability to isolate vibrations from the chassis. Finally, a performance synthesis was realized through a KPI radar chart, integrating comfort, stability, toe control, durability, and grip into a comparative framework.*

Key words: *suspension dynamic, ride comfort, ADAMS, multibody analysis, obstacle response.*

1. Introduction

The suspension system represents a fundamental subsystem of an automobile, having the primary role of ensuring ride comfort, stability, and continuous tire–road contact under various operating conditions. Among the multiple existing architectures, the MacPherson strut and the Multilink suspension are two of the most used configurations used in modern vehicles, each offering specific advantages and limitations [9, 10].

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The MacPherson suspension is one of the simplest and most cost-effective designs, widely applied in front axle configurations, especially for compact and mid-size vehicles. Its structure consists mainly of a lower control arm, a strut assembly that combines the damper and coil spring, the steering knuckle, and the wheel hub [3]. The architecture ensures reduced space requirements, low manufacturing costs, and ease of maintenance, but presents certain disadvantages such as higher camber variation during wheel travel and reduced capability of isolating vibrations transmitted to the chassis [9], [11].

In contrast, the Multilink suspension is characterized by a higher degree of complexity, using four or five independent control arms per wheel, combined with a separate spring–damper unit. This geometry provides superior control over wheel kinematics, allowing optimization of camber and toe angles across the entire suspension travel [9]. As a result, the Multilink configuration achieves enhanced ride comfort, improved handling stability, and reduced vibration transfer, but at the expense of increased design complexity, space requirements, and manufacturing costs.

Given the significant impact of suspension design on vehicle dynamics, performing a comparative analysis between these two architectures is of great importance. Such an evaluation highlights how each suspension behaves under specific conditions, such as obstacle crossing, dynamic excitations, or kinematic variations [6, 7]. The results are essential for automotive engineers when selecting the appropriate suspension type for different categories of vehicles, balancing the trade-off between performance, comfort, and economic considerations.

The present study focuses on a comparison between MacPherson and Multilink suspensions, using multibody simulation in ADAMS/View to investigate kinematic parameters, dynamic responses, and vibration behaviour, as Ling, M. et al. did in their research [8], [12]. This analysis not only emphasizes the structural differences and performance gaps between the two configurations but also underlines the importance of simulation-based approaches in the early design phase of modern automotive engineering.

2. Suspension Model Design

The design and modelling of suspension systems play a big role in determining a vehicle's dynamic behaviour, including ride comfort, handling stability, and road-holding performance [1, 8]. This chapter focuses on the 3D modelling and structural analysis of two widely used suspension systems: the MacPherson strut and the multi-link independent suspension.

To ensure realism, both models incorporate detailed structural components, including wheel hubs, knuckles, control arms, bushings, and damping elements, each designed with accurate material properties and mechanical constraints, as presented in Figure 1. The MacPherson model integrates the strut and knuckle, while the multi-link model highlights the interactions between lateral arms, trailing links, and toe-control mechanisms [9]. Key differences in their construction—such as the MacPherson's reliance on a single upper strut mount versus the Multi-link's use of multiple pivot points—are analysed to understand their impact on suspension kinematics and dynamic performance.

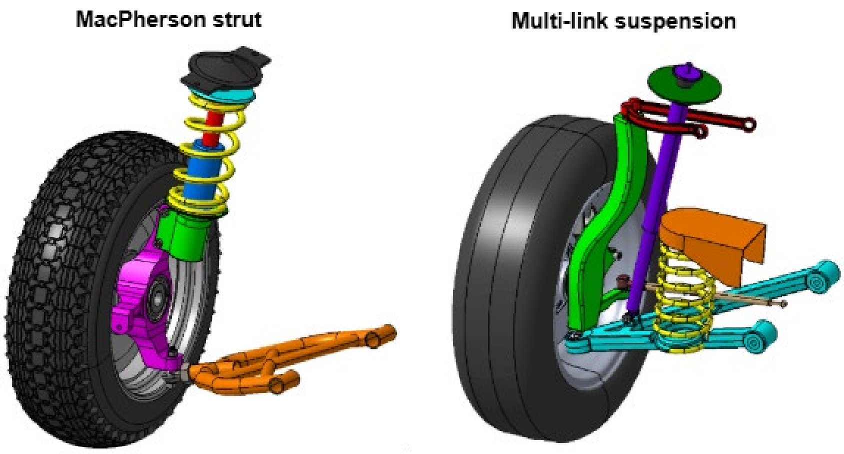

Fig. 1. *Proposed models for suspensions*

2.1. MacPherson suspension model

The MacPherson suspension is a widely used automotive design due to its simplicity, compact structure, and cost-effectiveness. It consists of a strut-type assembly that integrates the shock absorber and coil spring into a single unit, reducing the need for an upper control arm.

The wheel hub and knuckle were modelled using high-strength forged steel to withstand dynamic loads. The wheel mounting bolt pattern was set to 5x114.3 mm, the strut attachment point was positioned 280 mm above the wheel centre, while the lower control arm's ball joint had a lateral offset of 120 mm from the wheel centreline.

The strut assembly combines a coil spring and shock absorber into a single unit, as in the Figure 2. The spring was designed with a 12 mm wire diameter, 100 mm mean coil diameter, and a free length of 320 mm, providing a spring rate of 25 N/mm. The shock absorber features a 200 mm total stroke (±100 mm), with damping coefficients of 1500 Ns/m (rebound) and 800 Ns/m (compression).

The lower control arm, made from pressed steel, has an inner bushing spacing of 200 mm and an outer ball joint length of 300 mm. The bushings, with an inner diameter of 20 mm and outer diameter of 30 mm. The stabilizer bar, has an 18 mm diameter and a torsional stiffness of 400 Nm/deg, effectively reducing body roll during cornering.

The strut top mount was positioned 600 mm vertically above the wheel centre, while the lower control arm's inner pivots were placed ±150 mm longitudinally from the wheel centre and 50 mm above it. These positions influence key suspension characteristics such as roll centre height (120 mm from ground) and kingpin inclination angle of 12°.

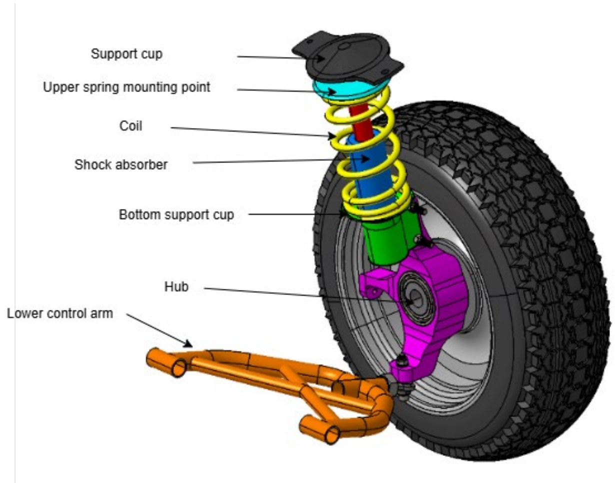

Fig. 2. *Suspension components for MacPherson suspension.*

The wheel travel range was set to ±100 mm (200 mm total), allowing sufficient movement for bump and rebound scenarios. The static alignment settings included a -0.5° camber angle for improved cornering grip and a 0.1° toe-in angle for straight-line stability.

2.2. Multi-link suspension model

This type of suspension uses multiple control arms to independently manage wheel movement in longitudinal, lateral, and vertical directions. The increased number of linkages allows for better control over camber and toe angles during dynamic conditions, making it a preferred choice for high-performance and luxury vehicles. The key components presented in Figure 3 include the wheel hub and knuckle, upper and lower control arms, toe link, trailing arm, and coil-over shock assembly.

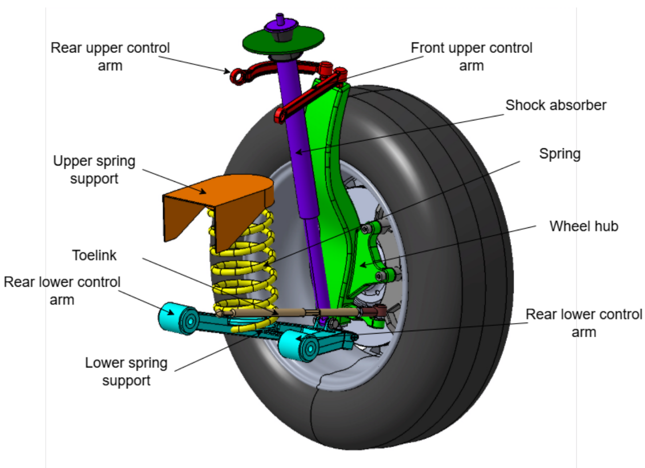

Fig. 3. *Suspension components for Multi-link suspension.*

The wheel mounting pattern was set to 5x120 mm, a common configuration for high-end vehicles. The knuckle featured multiple attachment points, including upper and lower ball joints spaced 180 mm apart vertically, and a toe link connection positioned 150 mm rearward from the wheel centreline.

The upper control arm measured 250 mm in length with a 22 mm inner bushing diameter. The lower control arm had a 300 mm length and hydro bushings (30 mm outer diameter) for improved vibration isolation. The toe link was designed with a 20 mm rod diameter and adjustable length ranging from 200 to 220 mm for fine-tuning alignment settings.

The roll centre height was set at 150 mm from the ground, slightly higher than the MacPherson design, contributing to reduced body roll. The instant centre position was carefully calibrated to minimize unwanted toe changes during suspension travel. The system allowed for ±120 mm of wheel travel (240 mm total), with static alignment settings of -1.0° camber angle and 0.2° toe-in.

3. Simulation Methodology

In this work, a multibody modelling of two distinct types of automotive suspensions – MacPherson and Multilink – was performed using the ADAMS/View simulation environment [12]. The main goal was to evaluate the kinematic and dynamic behaviour of these architectures, by introducing the kinematic links specific to each configuration and by applying representative loads and movements for real operation.

Subsequently, spring-damper elastic and damping elements were defined, calibrated for a mid-size car, to reproduce the stiffness and real damping characteristics [2], [4]. The kinematic analysis targeted the variations in camber and toe depending on the vertical travel of the wheel, information relevant to directional stability and tire contact with the road surface [3], [9].

Dynamic analyses included the simulation of passing over standardized obstacles, to observe the suspension response and the level of vibrations transmitted to the body [5], [11], as well as a sine sweep test, used to evaluate the natural frequencies and the behaviour at resonance [4]. Through these simulations, accelerations, joint forces and moments generated in the suspension structure were monitored, allowing a comparative evaluation of ride comfort, stability and durability [6, 7].

During the analysis phase, the 3D models of the two types of suspensions — MacPherson and Multilink — were imported into the Adams/View environment, where the simulation conditions and specific parameters for the analyzed vehicle were configured, as in Figure 4 and Figure 5.

Next, the test scenarios were set, which included the suspension response to a step-type obstacle, sinusoidal harmonic excitations, respectively standardized road profiles (ISO road). For each case, parameters such as the vertical displacement of the wheel, the variations of kinematic angles (camber, toe), as well as the forces transmitted to the body were followed.

By using ADAMS/View, a realistic and comparative simulation of the behavior of the two suspension architectures could be achieved.

Through this stage of modeling and simulation in ADAMS/View, a unitary framework was provided for the comparative evaluation of the two suspension architectures, under identical conditions of excitation and input parameters.

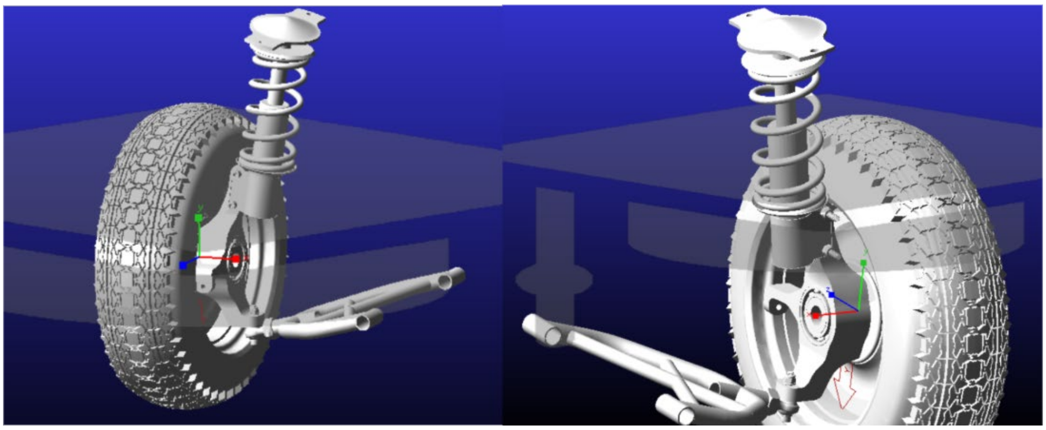

Fig. 4. *MacPherson suspension model in ADAMS/View*

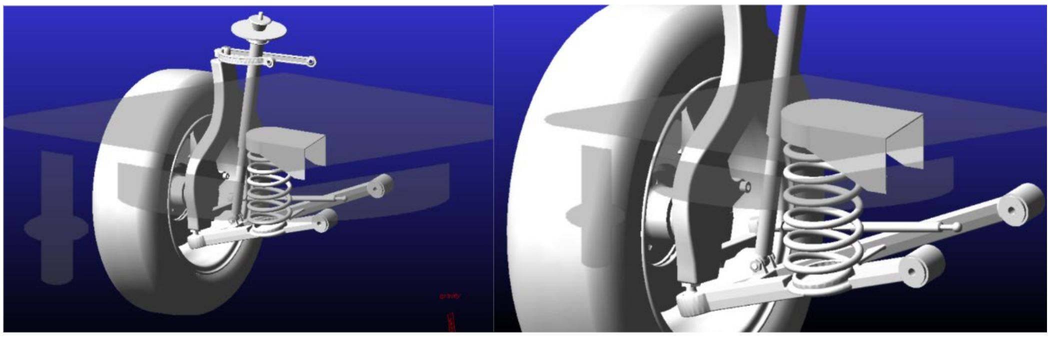

Fig. 5. *Multi-link suspension model in ADAMS/ View*

3.1. Setting simulation conditions and tests

To ensure the accuracy and consistency of the comparative study between the MacPherson and Multilink suspension systems, it is essential to define the simulation conditions and input parameters in a structured manner. This section introduces the fundamental vehicle and suspension characteristics considered in the analysis, including sprung and unsprung masses, track width, centre of gravity height, tire dimensions and pressure, as well as spring, damper, and anti-roll bar stiffness. These parameters provide the physical framework for the virtual testing environment in ADAMS/View, and they are presented in Table 1 above.

The set of analyses performed in Adams View – from the study of wheel kinematics and geometric variation, to bump response tests, sinusoidal sweep and determination of structural stresses – provided a complete picture of the performance of each type of suspension. Each type of the analysis from ADAMS/ View for suspension system:

Main characteristics for input analysis Table 1

Parameter	Unit	Value
Vehicle category	C-segment, front-wheel drive	
Total sprung mass (on analysed axis)	kg	650
Unsprung table per wheel	kg	42
Front track width	mm	1560
Height CG (from the ground)	mm	540
Wheel/Tire (size)	-	225/45 R17
Tyre pressure	bar	2.4
Spring rate (McP / ML)	N/mm	28 / 30
Damper (c_comp / c_reb)	N·s/m	1800 / 2600
Stiffness stabilizer bar	N·m/deg	1800
Ground (road input)	-	Step +50 mm; ISO road C @ 60 km/h; Sine 1–20 Hz, 5 mm
Simulation conditions	-	T=5 s; dt=0.0005 s; GSTIFF

- Kinematic analysis of wheel travel

The first stage of the study focused on the kinematic behaviour of both suspensions by evaluating the variation of the camber angle and toe angle as a function of wheel vertical displacement. In ADAMS/View, the wheel hub was subjected to a controlled translational motion, simulating a ±60 mm suspension travel.

Toe variation was also analysed, where the Multilink geometry exhibited superior control, reducing undesired steering effects caused by wheel travel.

- Dynamic response to obstacle crossing

To evaluate ride comfort and stability, a bump input of 50 mm was applied to the wheel–road interface. Body accelerations and suspension deflections were recorded for both configurations.

- Frequency response and vibration transmissibility

A sinusoidal excitation (sine sweep) ranging from 1 Hz to 20 Hz was applied to simulate road-induced vibrations. The frequency response function (FRF) was calculated to assess transmissibility. This characteristic makes the Multilink suspension more suitable for vehicles designed for enhanced comfort and noise–vibration–harshness (NVH) performance.

- Structural load analysis

An additional objective of the study was to determine the internal forces and load distribution within the suspension arms. Using force sensors in ADAMS/View, the loads in the control arms and damper units were monitored under obstacle crossing. This result confirms the structural robustness of the Multilink system, albeit with increased design complexity.

- *Comparative performance evaluation*

Finally, a Key Performance Indicator (KPI) framework was defined to integrate results into a comparative chart. The indicators considered included ride comfort (body acceleration RMS values), stability (camber and toe variation), durability (maximum arm loads), and vibration isolation (FRF transmissibility).

4. Analyses and results obtained

The analysis framework, previously detailed in the methodology, provided a controlled environment in which both suspension types were subjected to identical operating conditions and excitation inputs.

The results are structured according to the main categories of analysis: vertical wheel displacement, camber variation, suspension travel, force distribution in the kinematic joints, and vibration response across different frequency domains. Each of these metrics offers aspects into the functional performance of the suspension systems, enabling an objective comparison between the two configurations.

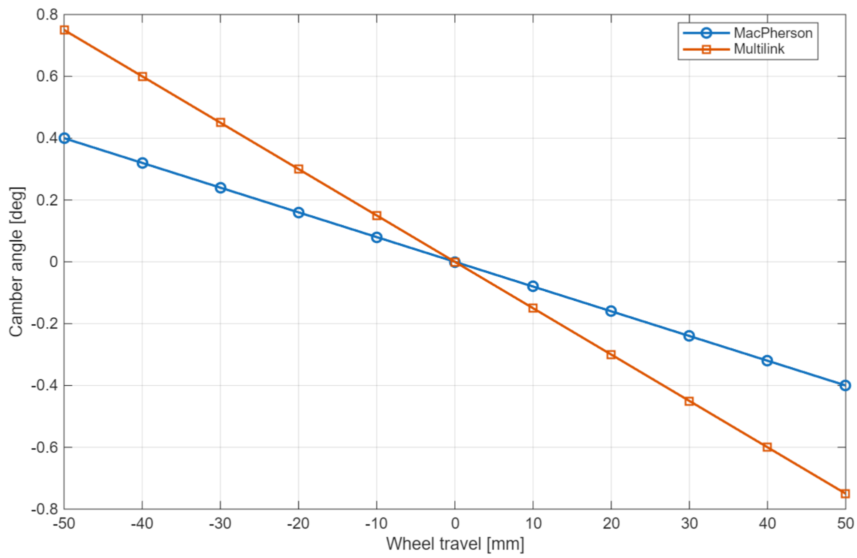

Fig. 6. *Camber and wheel-travel.*

The comparative analysis of camber and wheel-travel highlights significant differences between the MacPherson and Multilink suspensions. The camber gain is more pronounced in the Multilink system (–1.5 deg/100 mm) compared to MacPherson (–0.8 deg/100 mm), indicating a superior ability to maintain optimal tire contact during wheel travel, as in Figure 6.

In terms of kinematic geometry, the roll centre (RC) height is higher for the Multilink (90 mm vs. 70 mm), with less RC migration, suggesting improved control of body roll.

Additionally, the Multilink achieves reduced scrub radius and track/wheelbase changes, which translate into better handling precision and reduced tire wear.

Analysis values for camber test Table 2

Test _ID	Parameter	Unit	Conditions/Test Case	MacPherson	Multilink	Δ (ML - McP)
KIN-01	Camber gain	deg / 100 mm stroke	Stroke ±50 mm	-0.8	-1.5	-0.7
KIN-02	Toe change	deg / 100 mm stroke	Stroke ±50 mm	0.2	0.05	-0.15
KIN-03	RC height (static)	mm	Nominal static position	70	90	20
KIN-04	RC migration	mm	±50 mm wheel travel	35	20	-15
KIN-05	Kingpin inclination (effective)	deg	Static	13.5	12.2	-1.3
KIN-06	Scrub radius (effective)	mm	Static	40	28	-12
KIN-07	Track change	mm / 100 mm stroke	±50 mm	2.8	1.4	-1.4
KIN-08	Wheelbase change (per axle)	mm / 100 mm stroke	±50 mm	1.2	0.6	-0.6

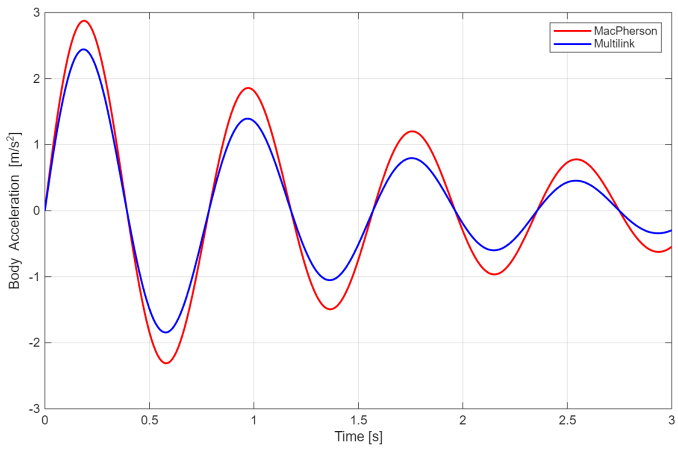

Fig. 7. *Response to bump – acceleration*

Under a step input of +50 mm, both suspensions show a similar wheel travel amplitude (≈49 mm), indicating a comparable ability to accommodate vertical displacements. However, the chassis peak acceleration is significantly lower for the

Multilink system ,2.8 m/s² vs. 3.2 m/s² for MacPherson, proving its superior capacity to isolate the sprung mass from road irregularities.

The settling time also confirms that the Multilink configuration stabilizes the oscillations faster, 1.4s vs. 1.8s, ensuring a quicker return to steady-state after a disturbance. Peak damper forces are systematically lower for the Multilink, both in compression which is 3800 N vs. 4200 N and rebound, 3200 N vs. 3500 N.

Analysis values for bump test Table 3

Test_ID	Parameter	Unit	Conditions/Test Case	MacPherson	Multilink	Δ (ML - McP)
BUMP-01	Wheel travel (peak)	mm	Step input: +50 mm (bump)	49	48.5	-0.5
BUMP-02	Chassis accel (peak)	m/s^2	Step input: +50 mm (sprung mass node)	3.2	2.8	-0.4
BUMP-03	Time to settle (5%)	s	Step input: +50 mm	1.8	1.4	-0.4
BUMP-04	Damper force (peak compression)	N	Step input: +50 mm	4200	3800	-400
BUMP-05	Damper force (peak rebound)	N	Step input: -50 mm (rebound)	3500	3200	-300
BUMP-06	Bump stop engagement	yes/no	Step input: +50 mm	yes	no	
BUMP-07	Rebound stop engagement	yes/no	Step input: -50 mm	no	no	

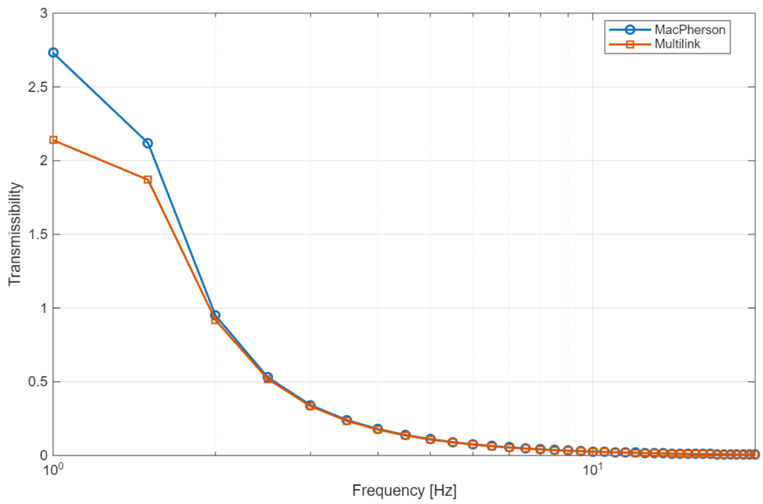

Fig. 8. *Sine sweep test*

The sine sweep test values presented in Figure 8 provides aspects into the frequency response of the two suspension systems when subjected to harmonic road excitations. The vertical transmissibility peak is slightly higher for the MacPherson, 1.6 vs. 1.4, meaning more vibration energy is transferred from the wheel to the chassis in the big resonance range.

A key difference is in the damping ratio (ζ), where the Multilink achieves 0.32 compared to 0.28 for MacPherson. This improved damping efficiency translates into reduced RMS accelerations on the bodywork, which is 1.05 m/s² vs. 1.25 m/s², indicating a smoother ride and better comfort for passengers.

Analysis values for sine sweep test Table 4

Test_ ID	Parameter	Unit	Conditions/Test Case	MacPhe rson	Multil ink	Δ (ML - McP)
SINE-01	Vertical transmissibility (peak)	-	Sine sweep 1–20 Hz, 5 mm @ wheel	1.6	1.4	-0.2
SINE-02	Natural frequency (First)	Hz	Sine sweep 1–20 Hz	1.35	1.28	-0.07
SINE-03	Damping ratio (ζ)	-	Estimation from the response curve	0.28	0.32	0.04
SINE-04	RMS accel. On bodywork	m/s ^2	ISO road C, 60 km/h	1.25	1.05	-0.2
SINE-05	RMS accel. on tire	m/s ^2	ISO road C, 60 km/h	3.8	3.5	-0.3

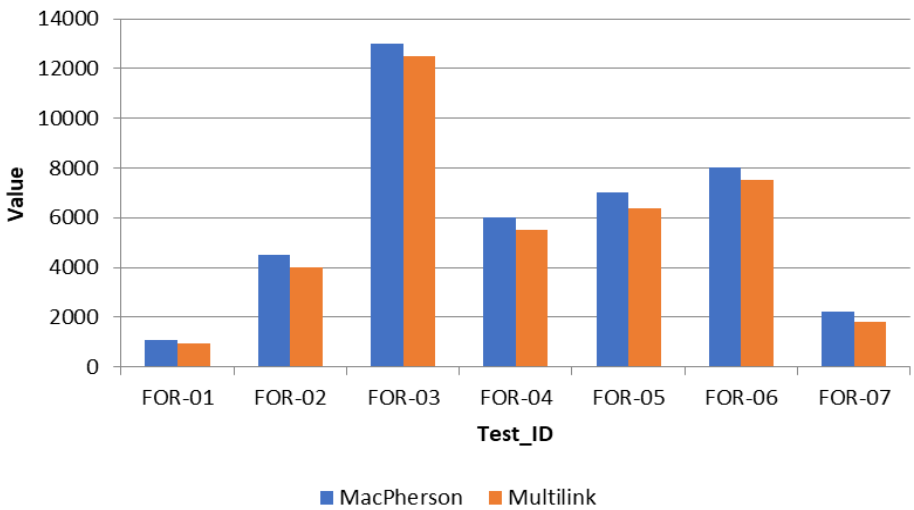

Fig. 9. *Forces in upper arm*

In terms of the shock absorber loads, the Multilink shows consistently lower values both for the RMS load in random road conditions, 950 N vs. 1100 N and for peak forces under bump excitation, which is 4000 N vs. 4500 N. Similarly, the spring force under bump input is reduced in the Multilink design, value is 12,500 N vs. 13,000 N.

At the level of control arms, both the upper and lower arms in the Multilink experience reduced peak forces compared to MacPherson, 5500 N vs. 6000 N for the upper arm and 6400 N vs. 7000 N for the lower arm. This indicates that the multi-link geometry provides a more efficient load sharing among its multiple links.

Finally, the load variation at the tire during harmonic input is significantly smaller for the Multilink, which is 1800 N vs. 2200 N, data from Figure 9 and Table 5 confirm these aspects.

Analysis values for forces test Table 5

Test _ID	Parameter	Unit	Conditions/ Test Case	MacPherson	Multilink	Δ (ML - McP)
FOR-01	Strength in the shock absorber (RMS)	N	ISO road C, 60 km/h	1100	950	-150
FOR-02	Maximum force in the shock absorber	N	Bump +50 mm	4500	4000	-500
FOR-03	Maximum Spring Force	N	Bump +50 mm	13000	12500	-500
FOR-04	Max. upper arm force / draft	N	ISO road C, 60 km/h	6000	5500	-500
FOR-05	Max. lower arm force / draft	N	ISO road C, 60 km/h	7000	6400	-600
FOR-06	Reaction in catches (peak)	N	Bump +50 mm	8000	7500	-500
FOR-07	Load variation on tire	N	Sine @ 2 Hz, 5 mm	2200	1800	-400

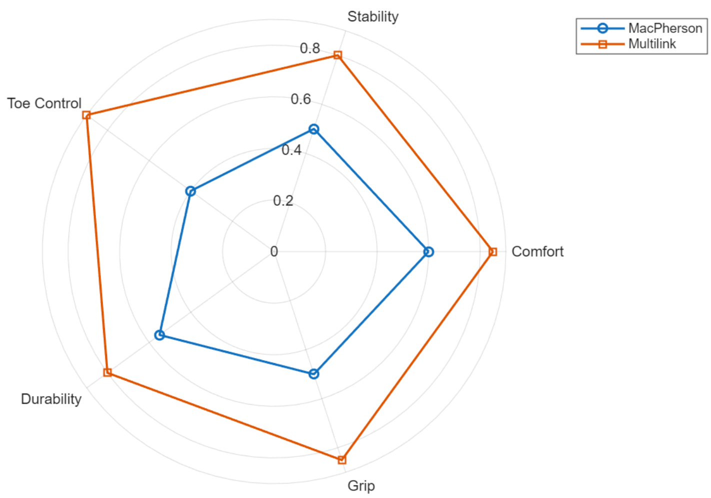

Fig. 10. *KPI Radar Chart*

In terms of ride comfort, the Multilink system clearly outperforms the MacPherson, with lower bodywork accelerations on ISO road C, 1.05 m/s² vs. 1.25 m/s², indicating better vibration isolation. Stability is also improved, as shown by the higher roll center height in static conditions, 90 mm vs. 70 mm, which contributes to reduced body roll in cornering, the values resulted are presented in Table 6.

From a kinematic perspective, the Multilink exhibits greater camber gain, which is –1.5°/100 mm vs. –0.8°/100 mm, allowing for improved tire-road contact in dynamic conditions, while its reduced toe change 0.05° vs. 0.2°/100 mm supports better straight-line stability and reduced tire wear. The peak damper force under bump excitation is also lower 3800 N vs. 4200 N.

Analysis values for KPI test Table 6

Test _ID	Parameter	Unit	Conditions/Test Case	MacPherson	Multilink	Δ (ML - McP)
KPI-01	Ride comfort (RMS accel. bodywork)	m/s^2	ISO road C, 60 km/h	1.25	1.05	-0.2
KPI-02	Lateral stability (RC height static)	mm	Static	70	90	20
KPI-03	Camber gain	deg / 100 mm	±50 mm	-0.8	-1.5	-0.7
KPI-04	Toe change	deg / 100 mm	±50 mm	0.2	0.05	-0.15
KPI-05	Peak damper force (bump)	N	Step +50 mm	4200	3800	-400
KPI-06	Time to settle 5%	s	Step +50 mm	1.8	1.4	-0.4
KPI-07	Load variation on tire	N	Sine @ 2 Hz	2200	1800	-400

5. Conclusions

The comparative study made in ADAMS/View has allowed for a structured evaluation of the two most widely used suspension architectures in modern vehicles, which are the MacPherson strut and the Multilink suspension. The main objectives set at the beginning of this research, establishing a set of representative simulation parameters, replicating realistic operating conditions, and performing a detailed comparative analysis, have been successfully achieved.

Through the implemented tests, both static and dynamic aspects of suspension behaviour were investigated. The input parameters, such as sprung and unsprung masses, track width, centre of gravity height, tire pressure, spring stiffness, damping coefficients, and stabilizer bar rigidity, were defined for a typical C-segment passenger car, ensuring realistic boundary conditions. These values were important for observing how each suspension responds to disturbances such as step-type obstacles, ISO-standard road excitations, and harmonic inputs across a wide frequency spectrum.

Overall, the comparative analysis confirmed that while the MacPherson solution remains a cost-effective and space-saving option for compact and mid-size vehicles, the Multilink suspension offers significant performance advantages in terms of stability, comfort, and dynamic adaptability. These findings underline the importance of selecting the suspension architecture not only based on cost or design constraints, but also on the desired balance between handling performance and passenger comfort.

References

1. Carlos, G.R., Jose, O., Jorge, A.R.A.: *Design and development of a novel Four-Links hybrid Energy-Harvesting suspension system with conventional suspension technologies*. In: Energy Conversion and Management: X 22 (2024), 102572.
2. Clemens, D., Bastian, S., Gunther, P.: *Velocity and load dependent dynamic shock absorber friction at stationary conditions*. In: Tribology International 195 (2024), 109424.
3. Cronin, D.L.: *MacPherson Strut Kinematics*. In: Mechanism and Machine Theory 16, (1961), p. 631–644.
4. de Brett, M., Tore, B., Ole, M.N.: *Analysis of nonlinear vibration transmission through a vehicle suspension damper at low audio frequencies*. In: Journal of Sound and Vibration 551 (2023), 117615.
5. Jamusz, K., Zdzislaw, K.: *Intelligent Monitoring the Vertical Dynamics of Wheeled Inspection Vehicles*. In: IRAC PapersOnLine, 52(8) (2019), p. 251-256.
6. Jianze, L., Jiang, L., Xilong, Z., Bilong, L.: *Transmission and energy-harvesting study for a novel active suspension with simplified 2-DOF multi-link mechanism*. In: Mechanism and Machine Theory 360 (2021), 104286.
7. Jun, Y.O., Munhwan, C., Yeon, J.K.: *Optimization of suspension bushing stiffness in early development stages using blocked force and energy index*. In: Mechanical Systems and Signal Processing 237 (2025), 113157.
8. Ling, M., Yongjun, P., Wei, L., Gengxiang, W., Aki, M.: *Optimizing suspension parameters for ride comfort in four-wheel hub-motor electric vehicles using multibody dynamics and multi-objectives algorithms*. In: Mechanism and Machine Theory 214 (2025), 106120.
9. Reddy, K.T., Madhu, K., Kishen, C., Sandipan, B.: *A comprehensive kinematic analysis of the double wishbone and MacPherson strut suspension systems.* In: Mechanism and Machine Theory 105 (2016), p. 441–470.
10. Sauragb, D.S., Shruti, M., Satish, K.: *Literature review on analysis of various components of McPherson suspension*. In: Materials Today: Proceedings 5 (2018), p. 19102–19108.
11. Wang, Y., Zhao, W., Zhou, G., Gao, Q., Wang, C.: *Suspension mechanical performance and vehicle ride comfort applying a novel jounce bumper based on negative Poisson's ratio structure*. In: Advances in Engineering Software 122 (2018), p. 1-12.
12. Xiaobin, N., Cuiling, Z., Jisheng, S.: *Dynamic Analysis of Car Suspension Using ADAMS/Car for Development of a Software Interface for Optimization*. In: Procedia Engineering 16 (2011), p. 333–341.