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Vehicle Dynamics Analysis (Simulink / VI-Grade)

Advanced Motorsport Dynamics (ACHA6011)

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1 Introduction

Understanding vehicle dynamics is fundamental for optimising vehicle performance, whether the objective is maximising lap time in motorsport or ensuring stability and comfort in road applications. In motorsport, suspension behaviour influences driver confidence, tyre usage and cornering capability, while in road vehicles similar principles influence ride quality, platform control and safety. Effective suspension tuning therefore relies on understanding how individual parameters interact and identifying which changes have the greatest influence on vehicle behaviour.

This investigation uses two complementary modelling approaches: VI-Grade CarRealTime, providing a multibody representation of the supplied sports car model for dynamic manoeuvre and lap-simulation testing, and a spring–mass–damper model in Simulink, offering an analytical view of the same system to examine fundamental suspension responses. Together, these tools allow direct comparison between physical multibody behaviour and simplified theoretical predictions.

The study evaluates a series of suspension variations, including softer and stiffer springs, reduced and increased damping, and an optimised configuration derived from natural ride frequency and critical damping relationships. This structured approach enables assessment of how each parameter alters tyre load distribution, transient response and overall handling characteristics.

1.1 Aims

- To investigate the influence of suspension spring and damper characteristics on vehicle dynamic behaviour.
- To evaluate tyre load response and handling performance using VI-Grade multibody simulations.
- To compare analytical (Simulink) and multibody modelling approaches to assess their accuracy and applicability.

1.2 Objectives

- Conduct standard dynamic manoeuvres in VI-Grade, including acceleration, braking, sinusoidal steering and a flying lap at Pembrey.
- Implement a 4DOF Simulink model representing the same vehicle and suspension system.
- Simulate vehicle responses for multiple spring and damper configurations, including an idealised setup derived from ride-frequency targets.
- Quantify the effect of each setup on tyre loads, pitch/roll/heave motion and overall performance.

- Compare results from VI-Grade and Simulink to identify limitations, differences and correlations between modelling approaches.

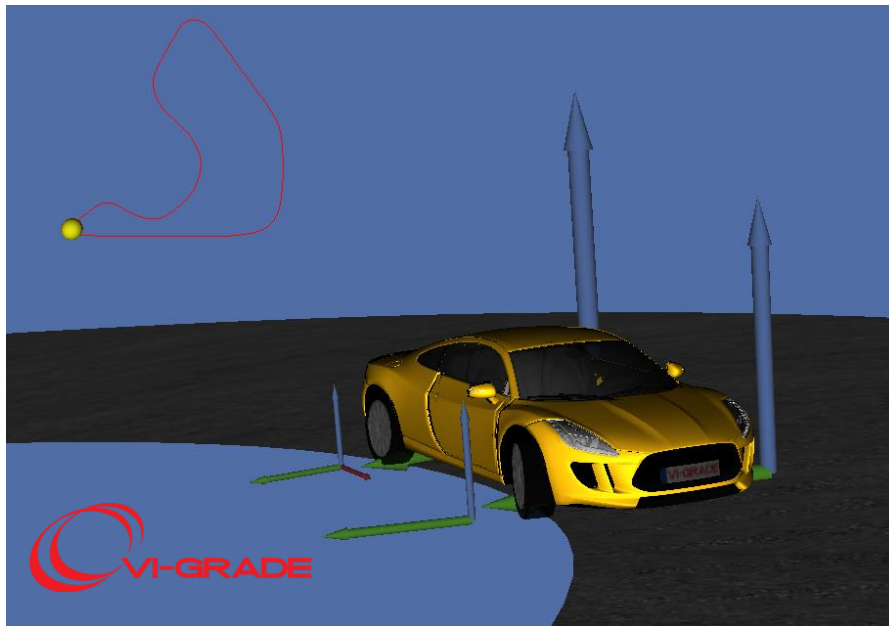


Figure 1 - VI-Grade sports car

2 Methodology

2.1 Overview of Approach

The investigation was carried out using two complementary modelling environments:

2.1.1 VI-Grade CarRealTime

This is a full multibody simulation environment used to evaluate transient vehicle behaviour under acceleration, braking, slalom and circuit-lap conditions.

Key reasons for using VI-Grade:

- Represents full suspension geometry and kinematics.
- Includes nonlinear tyre behaviour and load sensitivity (standard approach in industry and academic studies such as Milliken; Genta).
- Captures aero load changes and their influence on tyre forces.
- Provides realistic transient responses that simplified analytical models cannot reproduce.

2.1.2 Simulink 4DOF Half-Car Model

The 4 degrees of freedom model is physics-based analytical model developed to isolate and understand the core vertical dynamics of the vehicle.

Used because:

- It directly represents the fundamental mass–spring–damper behaviour taught in classical vehicle dynamics.
- Allows controlled assessment of heave, pitch, wheel hop and damping effects without geometric or nonlinear influences.
- Makes it possible to compare theoretical behaviour (ride frequency and damping ratio) against VI-Grade results.

2.1.3 Why both models are used

- VI-Grade gives high-fidelity behaviour based on full vehicle interactions.
- The 4DOF model reveals the underlying physical trends that govern ride and pitch.
- Comparing the two helps identify where simplified theory aligns with real vehicle behaviour and where multibody effects dominate.

2.2 Suspension setups and tests

To allow direct comparison, both environments were tested using the same suspension variations:

2.2.1.1 Spring setups

Table 1 - Spring setups

	-50%				50%		From Ride Frequency	
	Soft		Standard		Hard		Optimal	
	Linear rate (N/mm)	Free length (mm)	Linear rate (N/mm)	Free length (mm)	Linear rate (N/mm)	Free length (N/mm)	Linear rate (N/mm)	Free length (mm)
Front	47.5	344	95	344	142.5	344	53.678	344
Rear	52.5	182.5	105	182.5	157.5	182.5	74.111	182.5

2.2.1.2 Damper setups

Table 2 - Damper setups

		50%		Standard		150%		Optimal - Ride frequency	
		Velocity	Force	Velocity	Force	Velocity	Force	Velocity	Force
		x	y	x	y	x	y	x	y
FRONT	1	-1040	1590.908	-1040	3181.815	-1040	-4772.72	-1040	-4891.75
	2	-524	1323.368	-524	2646.735	-524	-3970.1	-524	-4069.12
	3	-393	1237.373	-393	2474.745	-393	-3712.12	-393	-3804.7
	4	-263	1127.490	-263	2254.980	-263	-3382.47	-263	-3466.83
	5	-131	-716.625	-131	1433.250	-131	-2149.88	-131	-2203.49
	6	-52	-229.320	-52	-458.640	-52	-687.96	-52	-705.118
	7	0	0.000	0	0.000	0	0	0	0
	8	52	361.900	52	723.800	52	1085.7	52	1112.777
	9	131	855.173	131	1710.345	131	2565.518	131	2629.501
	10	263	1356.810	263	2713.620	263	4070.43	263	4171.945
	11	393	1624.350	393	3248.700	393	4873.05	393	4994.582
	12	524	1901.445	524	3802.890	524	5704.335	524	5846.599
	13	1040	2275.000	1040	4550.000	1040	6825	1040	6995.213
REAR	1	-1040	-1477.27	-1040	-2954.54	-1040	-4431.81	-1040	-4612.89
	2	-520	-972.221	-520	-1944.44	-520	-2916.66	-520	-3035.83
	3	-390	-877.524	-390	-1755.05	-390	-2632.57	-390	-2740.13
	4	-260	-776.514	-260	-1553.03	-260	-2329.54	-260	-2424.72
	5	-130	-574.494	-130	-1148.99	-130	-1723.48	-130	-1793.9
	6	-50	-189.394	-50	-378.788	-50	-568.181	-50	-591.396
	7	0	0	0	0	0	0	0	0
	8	50	252	50	504	50	756	50	786.8882
	9	130	436.5075	130	873.015	130	1309.523	130	1363.026
	10	260	669.6422	260	1339.284	260	2008.927	260	2091.006

	11	390	828.3722	390	1656.744	390	2485.117	390	2586.652
	12	520	900	520	1800	520	2700	520	2810.315
	13	1040	1041.56	1040	2083.12	1040	3124.68	1040	3252.346

2.2.2 Optimal setup – calculations

Ride frequency was used to determine an optimised suspension setup, ensuring the vehicle operated within a target frequency window associated with improved transient response and stability. Using the measured corner masses, motion ratios and tyre stiffnesses, the effective wheel rates and corresponding natural frequencies were calculated for the sprung and unsprung masses. These values were then used to specify spring rates for the optimised configuration. The equations used are shown below.

A target ride-frequency window of 2.0–2.3 Hz at the front and 2.4–2.7 Hz at the rear was adopted for the optimised setup, consistent with commonly referenced guidance in vehicle dynamics literature (Milliken & Milliken, 1995) where higher rear frequencies help promote pitch stability and improved transient response in performance-oriented vehicles.

Wheel rate:

Equation 1 - Wheel rate

$$k_w = k_s \times MR^2$$

Ride frequency:

Equation 2 - Ride Frequency

$$f = \frac{1}{2\pi} \times \sqrt{\frac{k_w}{m_{sc}}}$$

K_w – wheel rate

m_{sc} – corner sprung mass

This equation was then reverse engineered to calculate the wheel rate and then from that the spring rate. The next step was to then match the damping to have a critically damped system.

A scaling factor was used to convert the standard damper curve,

Equation 3 - Damping scaling factor

$$\lambda = \frac{c_{crit}}{c_{current}}$$

The nearest positive force and velocity was used to approximate $c_{current}$.

Equation 4 - Current damping ratio

$$c_{current} \approx \frac{F_0}{v_0}$$

2.3 VI-Grade Modelling Approach

2.3.1 Vehicle Model

The investigation uses the standard sports car model supplied within VI-Grade CarRealTime, which provides fully defined suspension geometry, mass distribution and tyre properties. This model acts as a consistent baseline for all simulations. To isolate the influence of suspension tuning, only the spring and damper parameters were varied between test configurations; all other vehicle properties remained unchanged.

The key inputs underpinning the multibody analysis include sprung and unsprung masses, suspension and tyre vertical stiffnesses, CG location, axle distances a and b , and the motion ratios required to convert spring rates to wheel rates. These parameters define the vehicle's vertical dynamics and form the basis for both the VI-Grade simulations and the comparative 4DOF analytical model.

Table 3 - Vehicle values

Standard Vehicle Values		
Wheelbase	2740	m
ms	1360	kg
Ktf	200000	N/m
ksf	95000	N/m
musf	45.855	kg
mcf	340	kg
mscf	294.145	kg
kwf	35279.994	N/m
ktr	300000	N/m
ksr	105000	N/m
musr	51.302	kg
mcr	340	kg
mscr	288.69809	kg
kwr	61722.033	N/m
Ctf	50	N
Ctr	50	N
a	1.52	m
b	1.22	m

Alpha	0.8455	m
Beta	0.8455	m
Ixx	265	
Iyy	1040	
CG height	0.486	m
CG distance from front	1520	m
MRf	0.6094	
MRr	0.7667	

2.3.2 Setup Variations

VI-Grade defines damper behaviour using velocity-dependent force–velocity curves rather than a single damping coefficient. Each damper setup (50%, standard, 150% and the optimal configuration) is implemented by scaling or modifying the force–velocity table within the software. This allows the simulation to reflect how real dampers generate different forces at low-speed and high-speed shaft velocities.

The force–velocity curves for the standard and optimised setup:

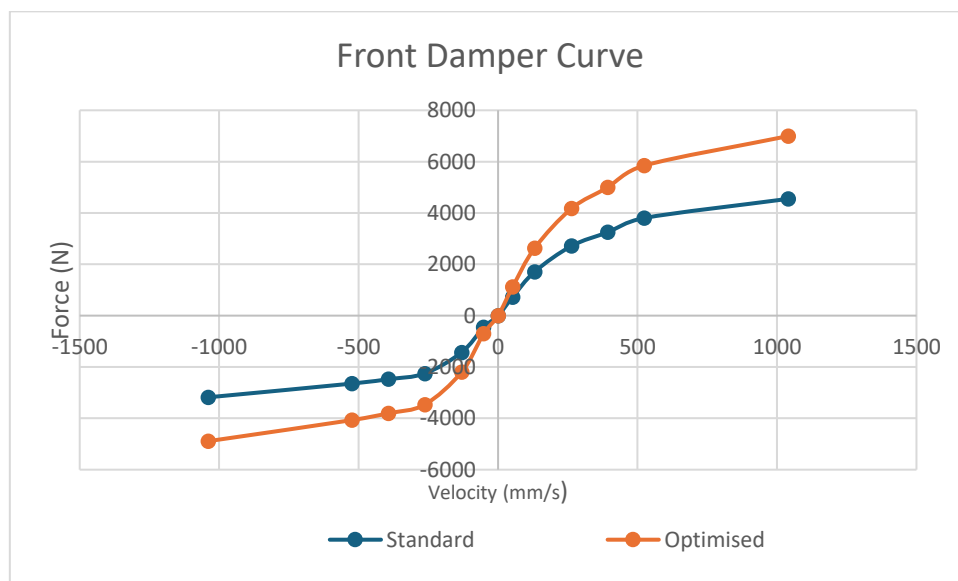


Figure 2 - Front Damper Curve

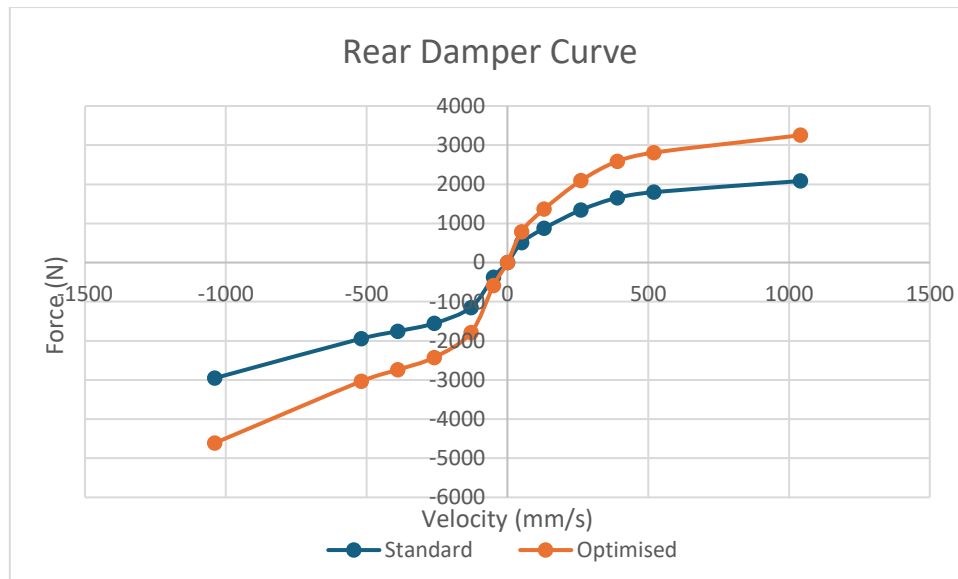


Figure 3 - Rear Damper Curve

2.3.3 Manoeuvres and Track

The vehicle was tested in four dynamic manoeuvres, each chosen to target a specific aspect of suspension behaviour:

- Straight-line acceleration**
 Used to assess pitch response, longitudinal load transfer and the influence of suspension stiffness/damping on rear-axle traction.
- Straight-line braking**
 Evaluates front load build-up, rear unloading, pitch stability and the effect of damping on how quickly the platform settles under heavy deceleration.
- Sin-steer (slalom) test**
 Provides a controlled lateral transient input for analysing yaw response, lateral load transfer, tyre utilisation and the effect of suspension changes on responsiveness.
- Maximum-performance lap (Pembrey)**
 Assesses combined-load behaviour, tyre load consistency across a full lap and the overall performance impact of each setup on lap time.

2.4 Spring Mass Damper systems

2.4.1 1 Degree of Freedom model

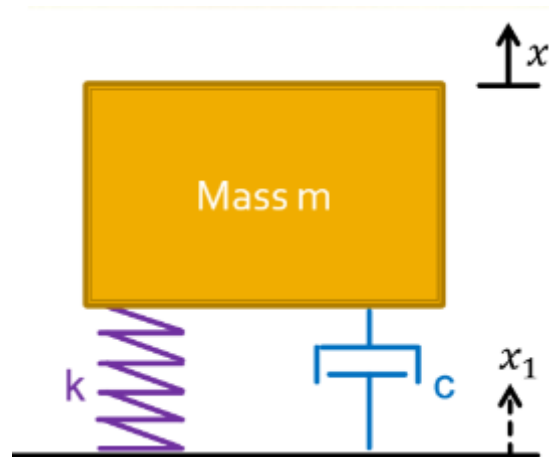


Figure 4 - 1 Degree of freedom model

Below are the equations used for the single degree of freedom model, these can then be developed for the 2 DOF model.

Equation 5 - 1 DOF equations

$$F = -kx - c\dot{x}$$

$$m\ddot{x} = -kx - c\dot{x}$$

$$m\ddot{x} + c\dot{x} + kx = 0$$

This model represents a single sprung mass moving vertically on a spring and damper and is used to illustrate the basic forced-oscillator behaviour that underpins all ride modelling.

2.4.2 2 Degrees of freedom model

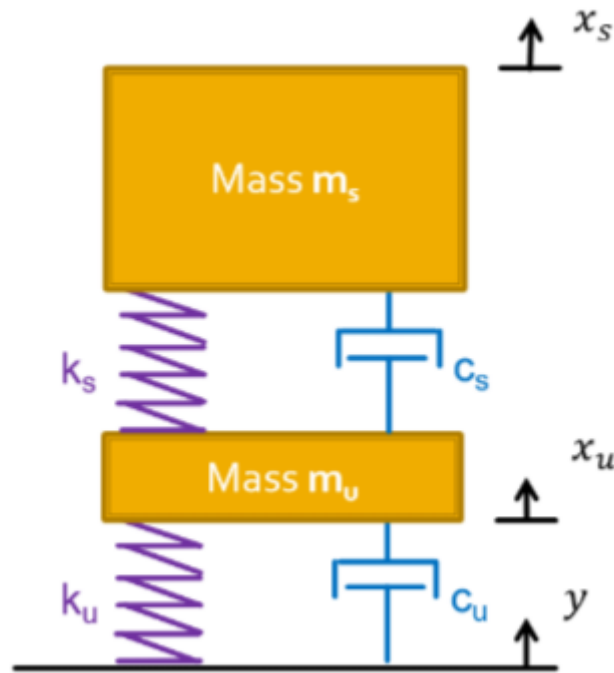


Figure 5 - 2 DOF model

The 2-DOF equations represent the vertical motion of the sprung mass and the unsprung mass, allowing the fundamental ride behaviour of a single wheel to be analysed.

Equation 6 - 2 DOF equations

$$\begin{aligned} -m_s \ddot{z} &= k_s(z - z_u) + C_s(\dot{z} - \dot{z}_s) \\ -m_s \ddot{z}_u &= k_u(z_u - y) + C_u(\dot{z}_u - \dot{y}) - k_s(z - z_u) + C_s(\dot{z} - \dot{z}_s) \end{aligned}$$

2.4.3 4 Degrees of Freedom

Working from the 1 and 2 degrees of freedom models, the 4DOF half-car model was developed to provide an analytical way of investigating how spring and damper changes influence the vehicle's heave and pitch behaviour. Unlike VI-Grade, which includes full suspension geometry, nonlinear tyres and aero maps, the 4DOF model isolates the fundamental vertical dynamics. This makes it ideal for comparing measured trends with classical ride-frequency and damping-ratio theory.

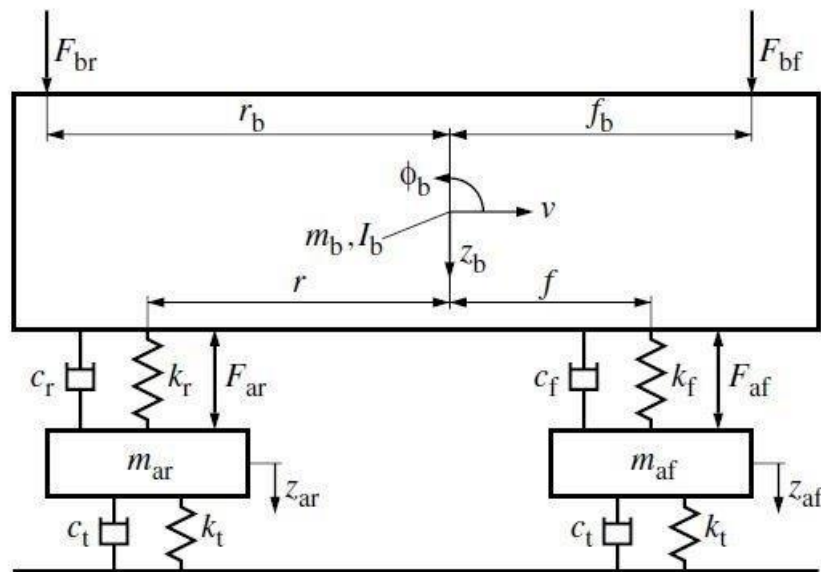


Figure 7 - Representation of a 4 degree of freedom model

2.4.4 Purpose of the Model

The model was used to:

- Examine the fundamental mass – spring- damper behaviour of the sprung and unsprung masses.
- Study how spring rate, damping coefficient and tyre stiffness affect **heave, pitch, and wheel forces**.
- Compare vertical responses with VI-Grade to identify where simplified theory holds and where multibody effects dominate.
- Provide a simplified platform for **bump** and **longitudinal acceleration** tests, where vertical load transfer is the dominant mechanism.

2.4.5 Model Assumptions

To focus purely on vertical behaviour, the following assumptions were applied:

- Two unsprung masses representing front and rear axles.
- Tyres modelled as vertical linear springs.
- No roll motion.
- Small-angle pitch assumption.
- Aero loads and longitudinal accelerations may be applied directly as external inputs.
- No compliance or bump stops effects.

These assumptions reduce complexity of the calculations while still capturing the key ride and pitch characteristics relevant to the VI-Grade tests.

2.4.6 Equations of Motion

The 4DOF half-car model uses the following generalised coordinates:

z_s — vertical displacement of the sprung mass CG (positive upwards)

θ — pitch angle about the CG (positive nose-up)

z_f — vertical displacement of the front unsprung mass (wheel centre, positive upwards)

z_r — vertical displacement of the rear unsprung mass (wheel centre, positive upwards)

Distances from the CG to each axle:

a — distance to front axle

b — distance to rear axle

Small-angle pitch is assumed, so the vertical positions of the sprung-mass attachment points at the front and rear suspensions are

$$y_f = z_s - a \theta, \quad y_r = z_s + b \theta$$

Road inputs at the front and rear tyres are denoted y_f and y_r .

Suspension and tyre forces are modelled as linear spring–damper elements:

$$F_{sf} = K_f(z_f - y_f) + C_f(\dot{z}_f - \dot{y}_f), F_{sr} = K_r(z_r - y_r) + C_r(\dot{z}_r - \dot{y}_r)$$

$$F_{tf} = K_{tf}(z_f - y_f) + C_{tf}(\dot{z}_f - \dot{y}_f), F_{tr} = K_{tr}(z_r - y_r) + C_{tr}(\dot{z}_r - \dot{y}_r)$$

By definition, F_{sf} and F_{sr} act upwards on the sprung mass, by Newton's third law they act downwards on the unsprung masses.

Substituting the attachment point expressions:

$$F_{sf} = K_f(z_f - z_s + a\theta) + C_f(\dot{z}_f - \dot{z}_s + a\dot{\theta})$$

$$F_{sr} = K_r(z_r - z_s - b\theta) + C_r(\dot{z}_r - \dot{z}_s - b\dot{\theta})$$

$$F_{tf} = K_{tf}(z_f - y_f) + C_{tf}(\dot{z}_f - \dot{y}_f)$$

$$F_{tr} = K_{tr}(z_r - y_r) + C_{tr}(\dot{z}_r - \dot{y}_r)$$

This then gives the 4 equations for each degree of freedom.

2.4.6.1 Sprung mass heave

Equation 6 - Sprung mass heave

$$-ms\ddot{z}_s = Kf(z_s - z_f - a\theta) + Kr(z_s - z_r + b\theta) + Cf(\dot{z}_s - \dot{z}_f - a\dot{\theta}) + Cr(\dot{z}_s - \dot{z}_r + b\dot{\theta})$$

2.4.6.2 Sprung mass pitch

Equation 7 - Sprung mass pitch

$$-I\ddot{\theta} = bKr(z_s - z_r + b\theta) - aKf(z_s - z_f - a\theta) + bCr(\dot{z}_s - \dot{z}_r + b\dot{\theta}) - aCf(\dot{z}_s - \dot{z}_f - a\dot{\theta})$$

2.4.6.3 Front unsprung mass:

Equation 8 - Front unsprung mass

$$-mfz''f = Ktf(zf - yf) - Kf(zs - zf - a\theta) + Ctf(\dot{z}f - \dot{y}f) - Cf(\dot{z}s - \dot{z}f - a\dot{\theta})$$

2.4.6.4 Rear unsprung mass:

Equation 9 - Rear unsprung mass

$$-mrz''r = Ktr(zr - yr) - Kr(zs - zr + b\theta) + Ctr(\dot{z}r - \dot{y}r) - Cr(\dot{z}s - \dot{z}r + b\dot{\theta})$$

2.4.7 Implementation in Simulink

These figures illustrate the Simulink implementation of the 4DOF model, including the pitch equation and the front sprung-mass subsystem. The remaining subsystem diagrams and equation blocks are provided in the Appendices for clarity and completeness.

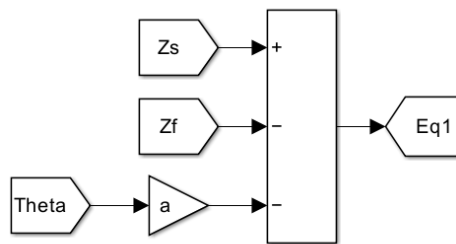


Figure 8 - Sub-equation for ease of model construction.

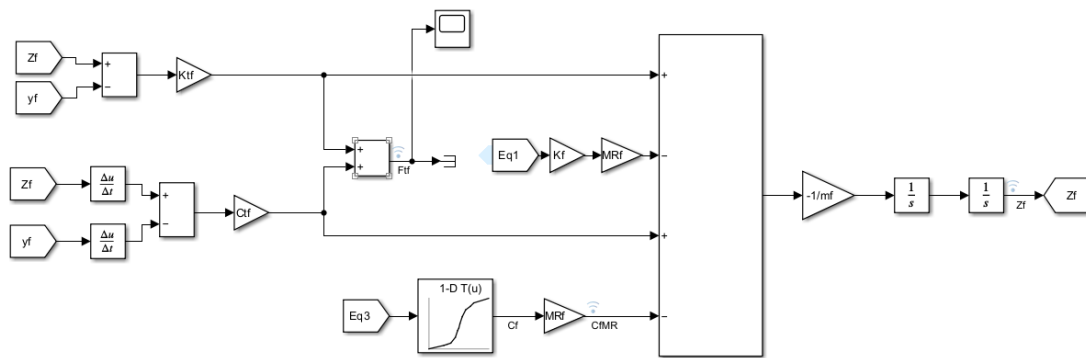


Figure 9 - Front unsprung mass, 4DOF model in Simulink

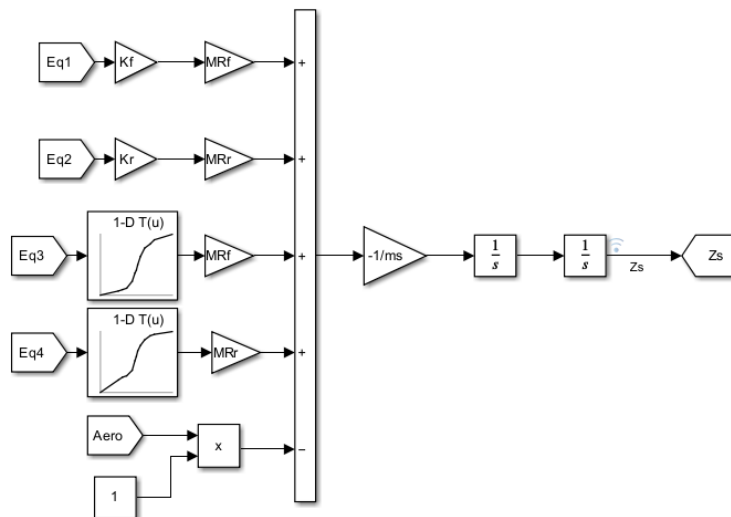


Figure 10 - Sprung mass, 4DOF model Simulink

2.4.8 Input Types

Two input types were applied to the 4DOF Simulink model:

1. Bump Input

- Implemented as a single bump, step, or sinusoidal vertical road disturbance applied at the tyres.
- Used to evaluate:
 - vertical oscillation and settling behaviour
 - the influence of spring and damper changes on damping performance

2. Acceleration Input

- Applied using longitudinal acceleration data taken directly from the VI-Grade straight-line tests.
- The corresponding **aero load variation** from VI-Grade was also imposed as an external vertical force acting on the sprung mass.
- Used to evaluate:
 - pitch response during acceleration and braking
 - how closely the analytical model matches the VI-Grade multibody behaviour, particularly under combined inertial and aerodynamic loading

2.5 7DOF Full-Car Model

A 7DOF full-car model was also developed alongside the half-car representation. This extended model includes heave, roll and pitch motions of the sprung mass, together with the vertical motion of all four unsprung masses, allowing a more complete representation of vehicle behaviour. The 7DOF model is used only for comparison on the lap tyre-force results to illustrate how a more comprehensive analytical model differs from the simpler 4DOF approach. The full model layout and supporting diagrams are provided in the Appendices, while the 4DOF half-car model remains the primary analytical model for Task 2.

3 Results

3.1 VI-Grade:

3.1.1 Straight-line acceleration

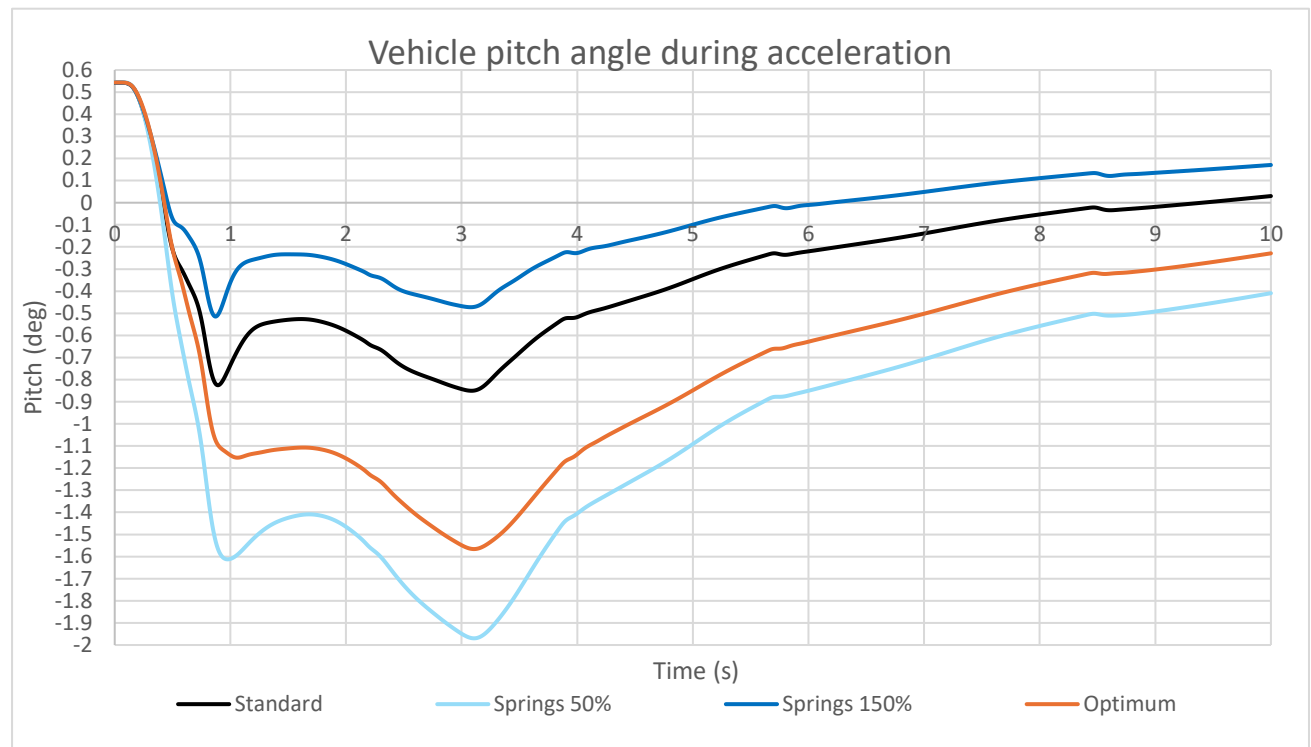


Figure 11 - Pitch angle during acceleration

All setups show pitch-down as load transfers rearwards. The 50% soft springs allow the greatest pitch ($\approx -1.9^\circ$) because the lower wheel rate cannot resist the rearward load shift, causing deeper suspension compression. The 50% stiff springs significantly reduce pitch amplitude, demonstrating the higher resistance to longitudinal load transfer, but also introduce a sharper transient due to the higher natural frequency. The optimal setup moderates both behaviours, reducing peak pitch without the harshness of the stiff configuration.

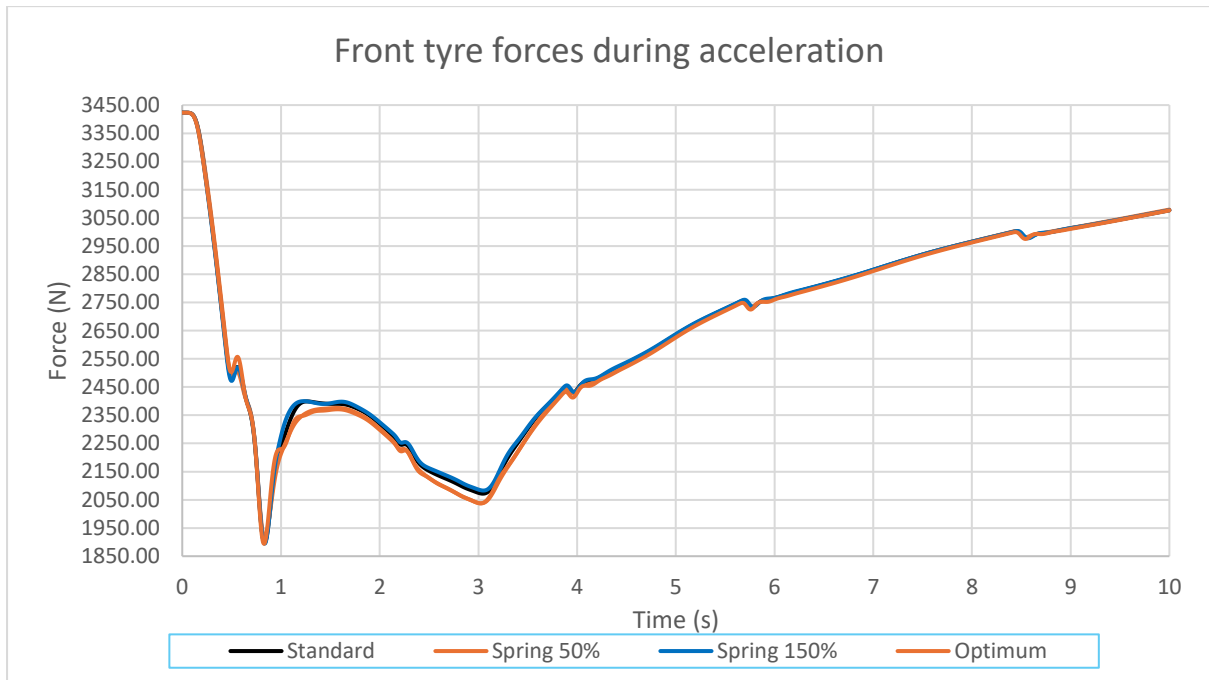


Figure 12 - Front tyre forces during acceleration

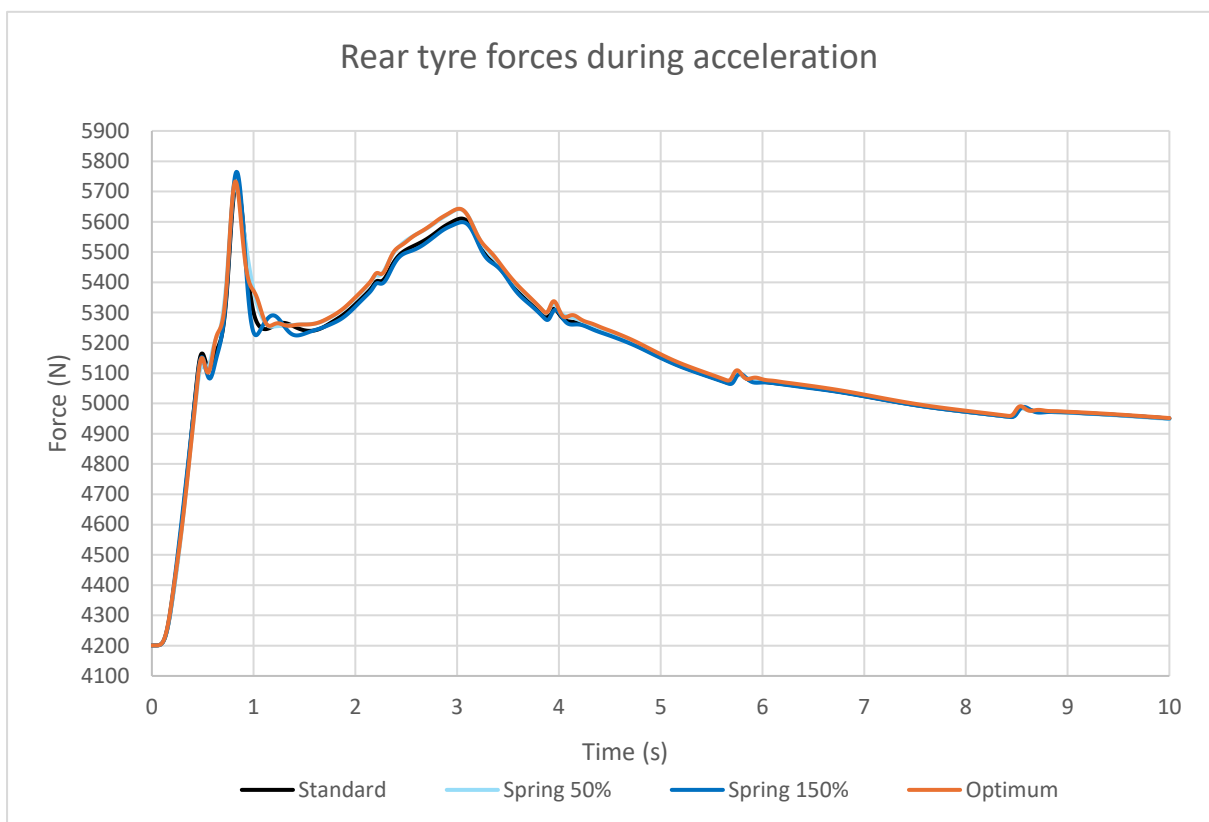


Figure 13 - Rear tyre forces during straight line acceleration

Tyre loads follow the pitch behaviour. Softer springs produce a larger reduction in front load and a higher rear-load peak because more weight transfers through suspension travel rather than through geometric load paths. Stiffer springs maintain front load more effectively during the transient, improving stability but at the cost of abrupt load changes. All setups converge to similar steady-state loads, showing that spring rate affects transients, not the final quasi-static distribution.

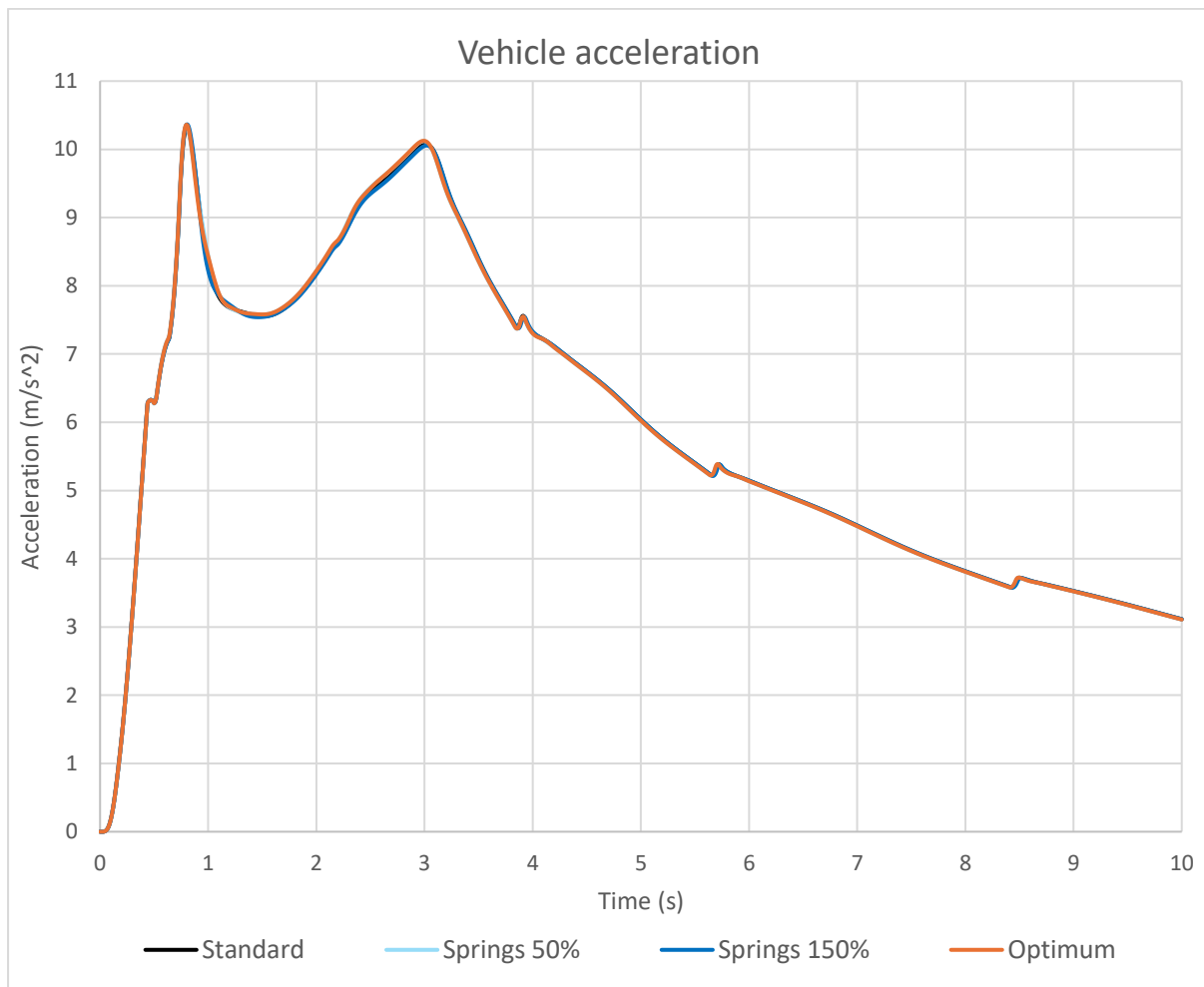


Figure 14 - Acceleration rate

Peak acceleration is effectively identical for all setups, confirming that traction is limited by tyre capability rather than suspension stiffness. The key effect of spring rate is therefore load distribution control, not absolute performance.

Softer springs increase pitch and tyre-load variation; stiffer springs improve platform control but with harsher transients. The optimal ride-frequency setup provides the best balance.

Table 4 - 0-100km/h times

	Time (s)	
Std	3.57	100.1155
Sprg50	3.56	100.1174
Sprg150	3.57	100.0231
Damp50	3.57	100.1656
Damp50	3.57	100.0983
Optimum	3.56	100.0972

As seen in the table above the 0–100 km/h times across all setups showed only small variation. The stiffened spring setup (Sprg150) delivered the fastest time, while the soft-damper configuration produced the slowest. The optimised setup matched the best-performing cases, indicating that ride-frequency-based tuning does not compromise straight-line acceleration performance.

3.1.2 Straight-line braking

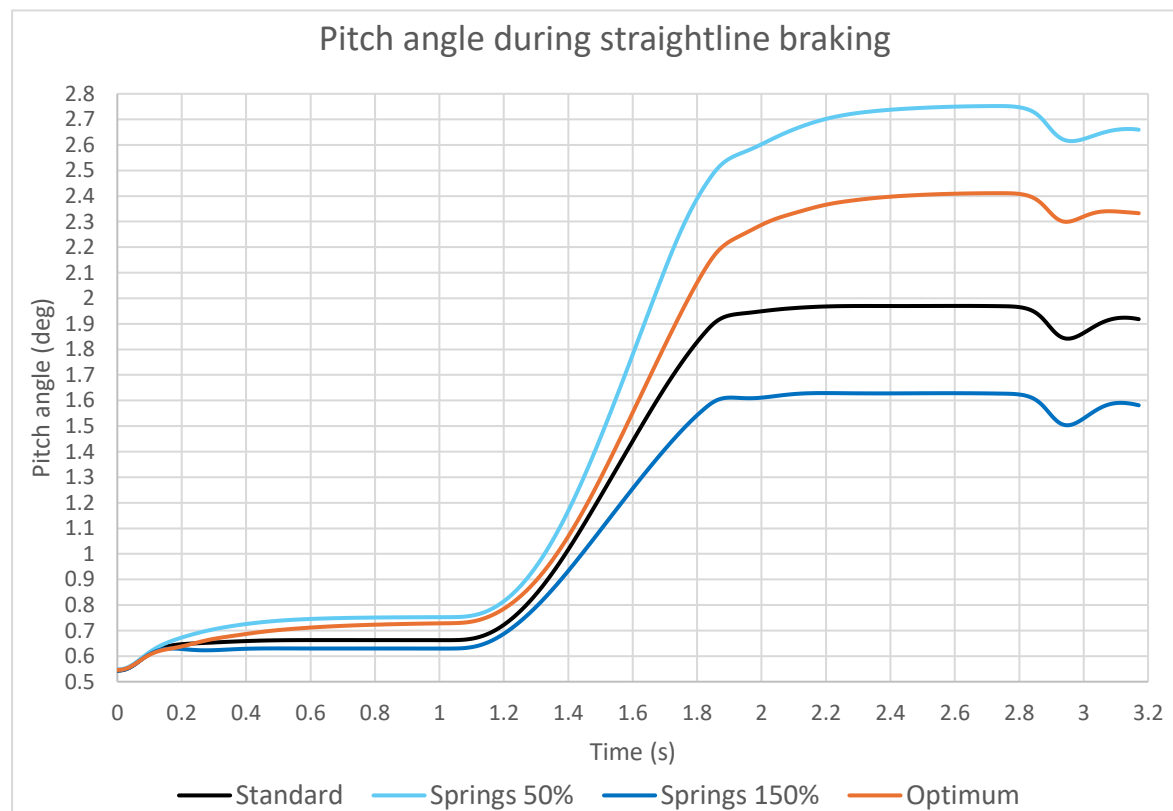


Figure 15 - Pitch Angle during straight line braking

All setups show forward pitch under braking, with the 50% soft springs producing the largest dive ($\approx 2.7^\circ$) due to reduced front suspension support. The 50% stiffer springs limit pitch to around 1.6° , giving better platform control but a sharper transient. The optimal setup reduces peak pitch relative to standard while maintaining a smoother response.

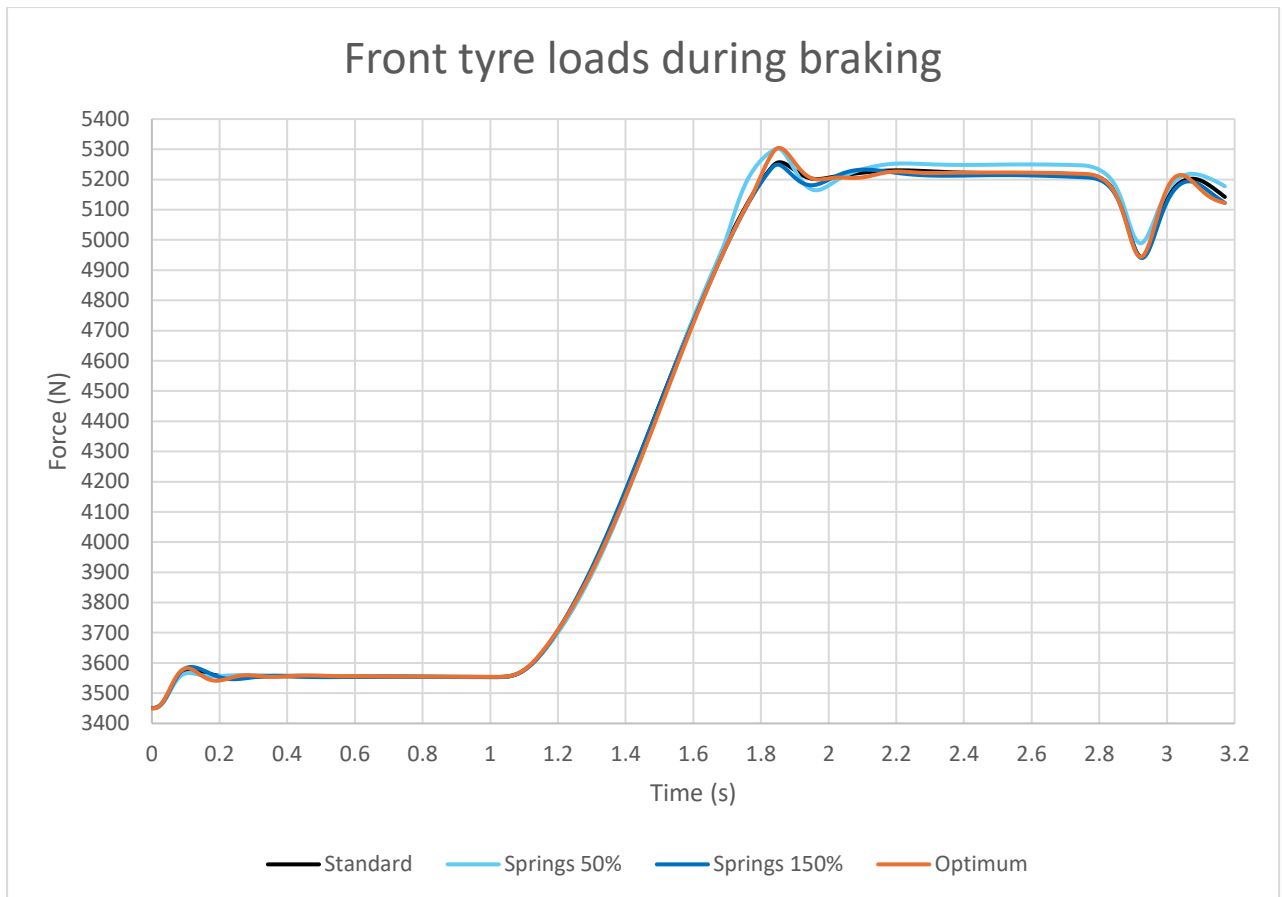


Figure 16 - Rear tyre loads during braking

Front tyre loads rise rapidly as weight transfers forward. Stiffer springs reach their peak load fastest due to reduced pitch travel, while soft springs create a larger but slower load build-up. Steady-state front loads are similar across all setups, indicating spring stiffness mainly affects the transient, not the final distribution.

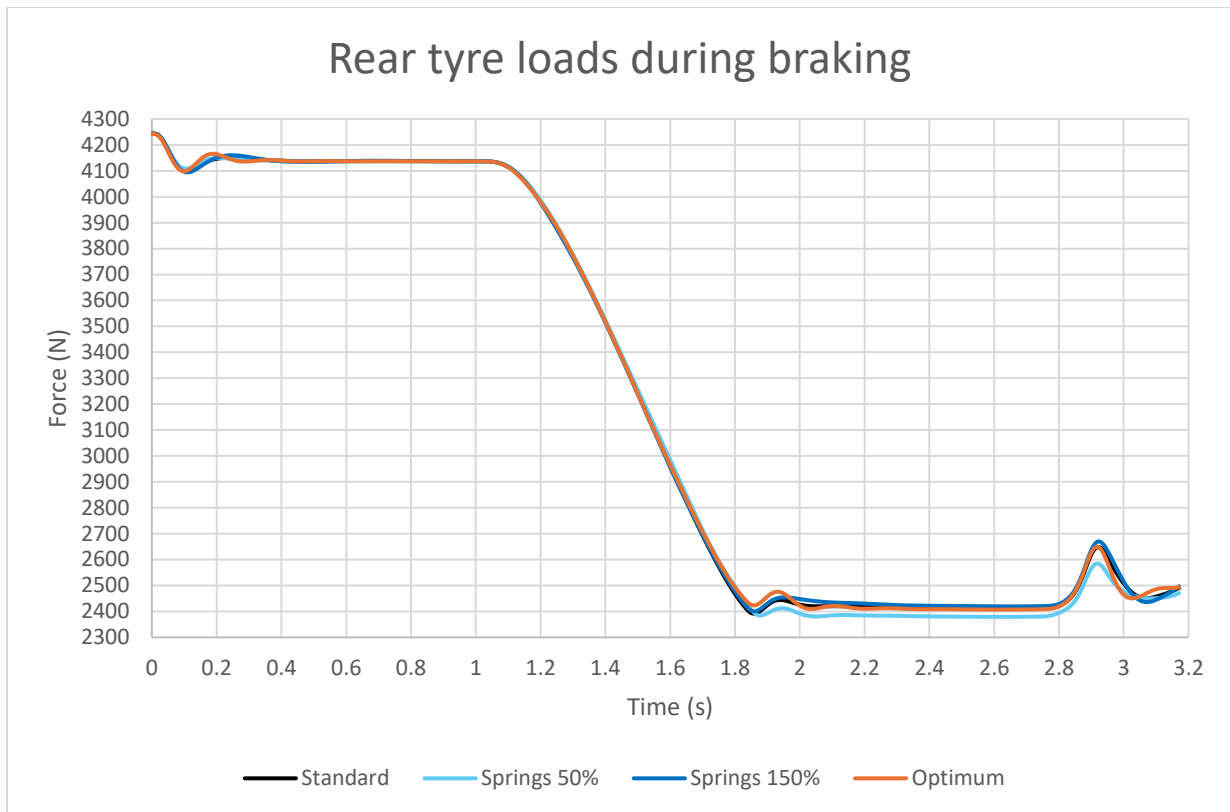


Figure 17 - Rear Tyre loads during braking

Rear loads fall in line with pitch behaviour: the soft setup shows the largest reduction, while the stiff setup maintains more rear load during the transient. All configurations converge once deceleration stabilises.

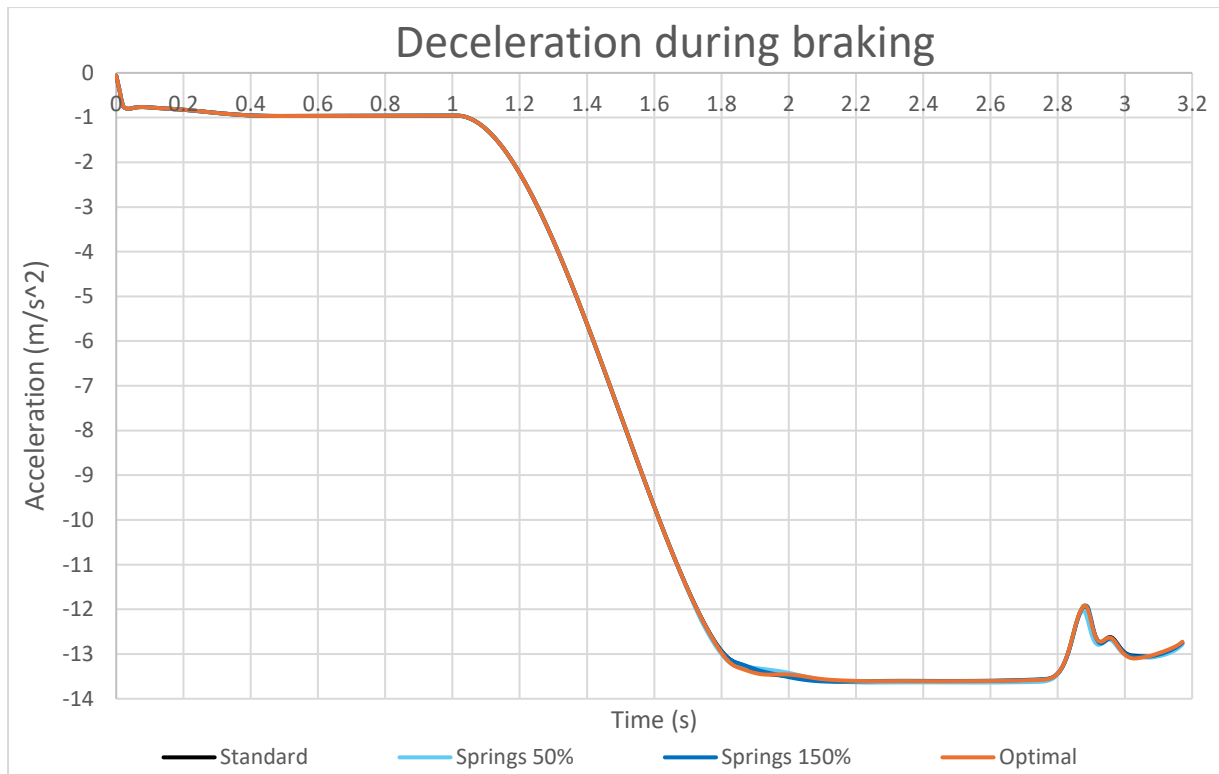


Figure 18 - Deceleration during braking

Peak braking deceleration is effectively unchanged across all setups, showing that suspension stiffness influences platform motion and transient load distribution, but does not significantly affect overall braking capability.

Braking performance is governed by tyre and brake characteristics, while spring stiffness shapes the transient dive and load transfer. The soft setup produces excessive pitch; the stiff setup offers control but with harsher transients; and the optimal ride-frequency setup provides the most balanced response.

3.1.3 Sin steer / slalom test

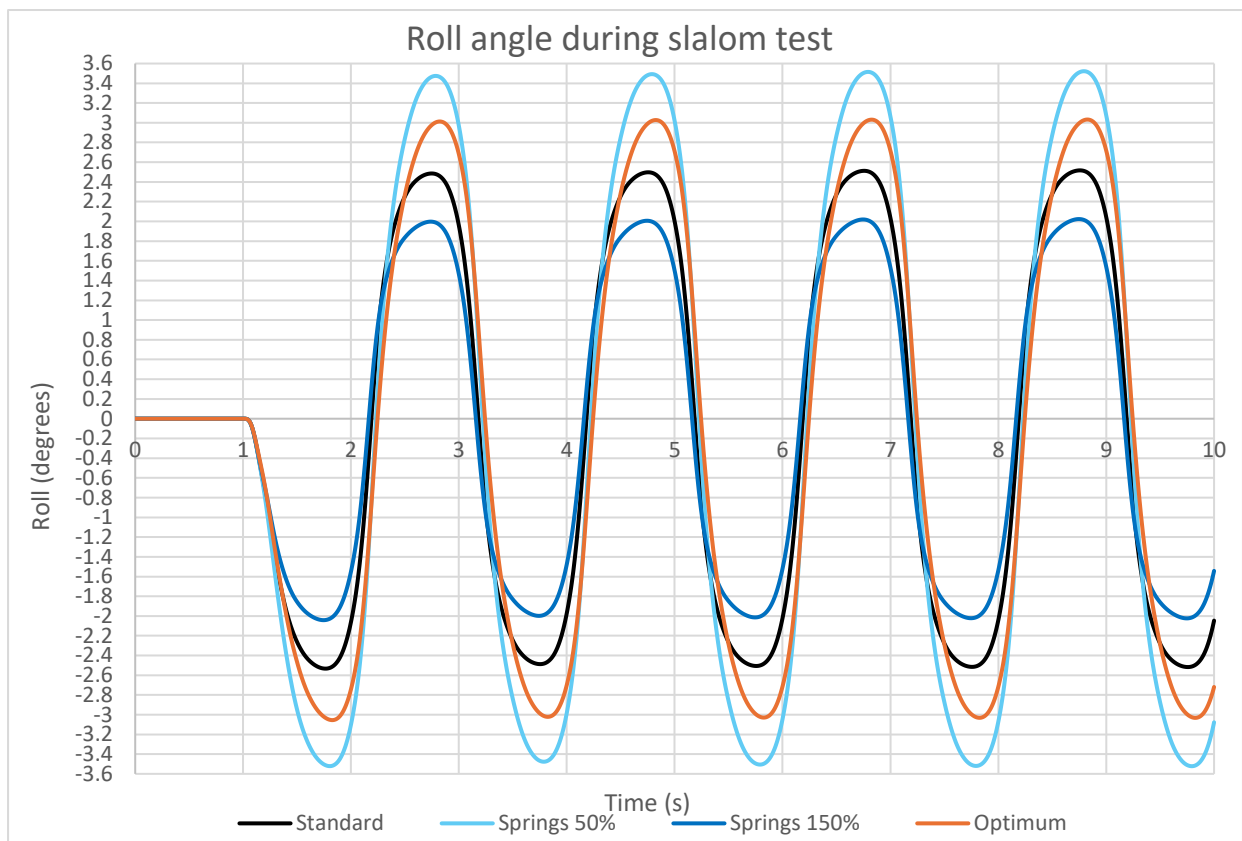


Figure 19 - Roll angle during Sin Steer test

The roll angle plot shows clear differences in lateral load transfer behaviour. The 50% soft springs produce the largest roll magnitudes, as expected from their lower roll stiffness, resulting in greater lateral weight transfer variation. The 50% stiff springs show noticeably reduced roll amplitude, indicating higher platform control but also sharper reversals as the vehicle transitions between cones. The optimal setup reduces peak roll relative to the standard configuration, but without the abrupt, high-frequency response observed in the stiff setup, suggesting improved transient balance.

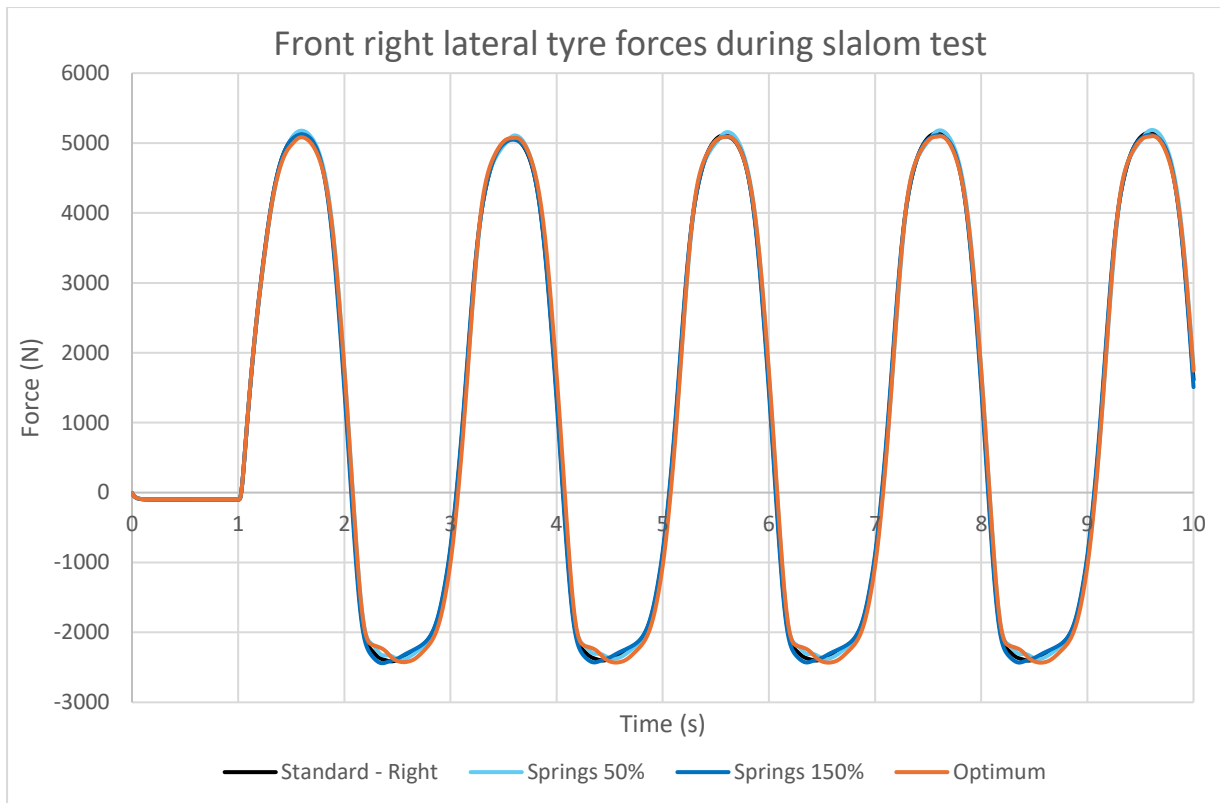


Figure 20 - Front lateral tyre loads during slalom test

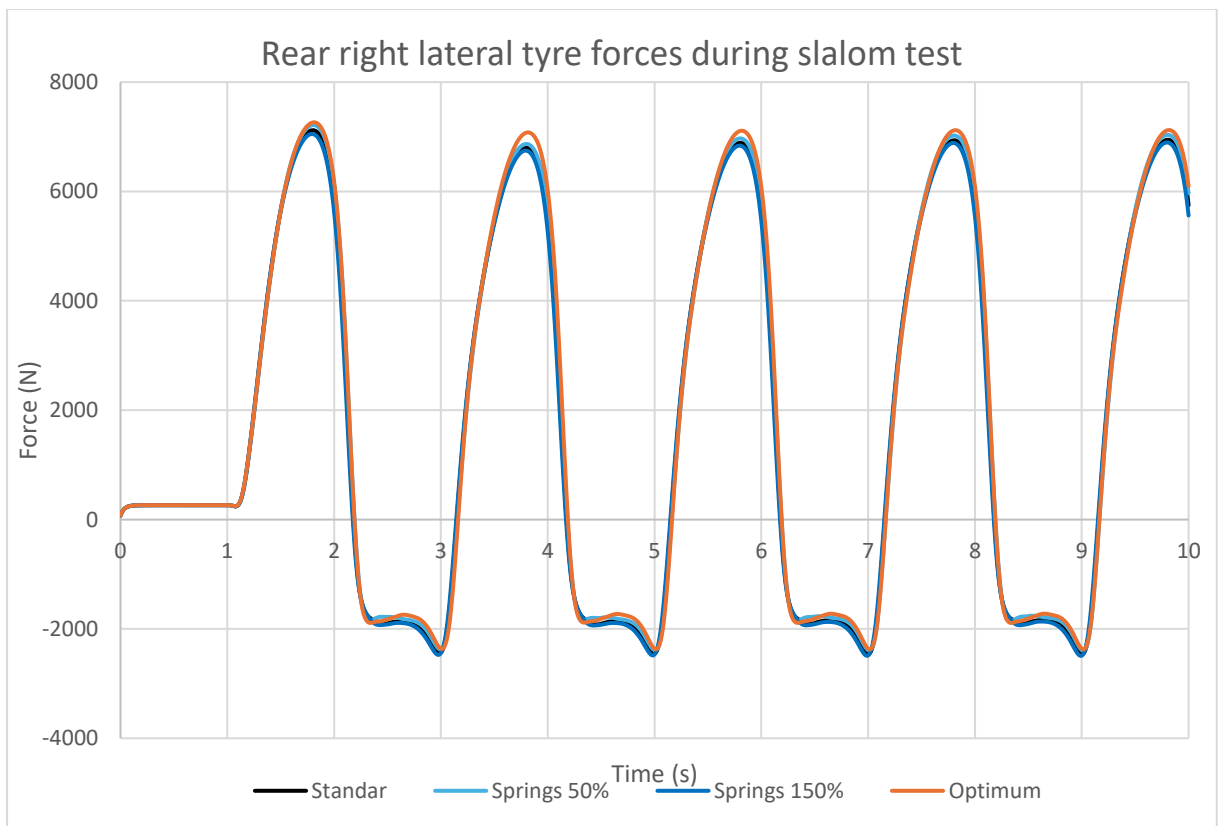


Figure 21 - Rear lateral tyre loads during slalom test

Front- and rear-right lateral force traces follow the roll behaviour: the soft setup shows the largest peak forces and largest negative troughs, consistent with its high roll angle and increased load transfer onto the outside tyres. The stiff setup generates lower peak forces due to reduced roll but exhibits faster transient load changes. The optimum configuration again sits between these extremes, providing a more consistent lateral force profile with reduced oscillation. This indicates improved tyre utilisation, as highly fluctuating vertical load reduces available lateral grip.

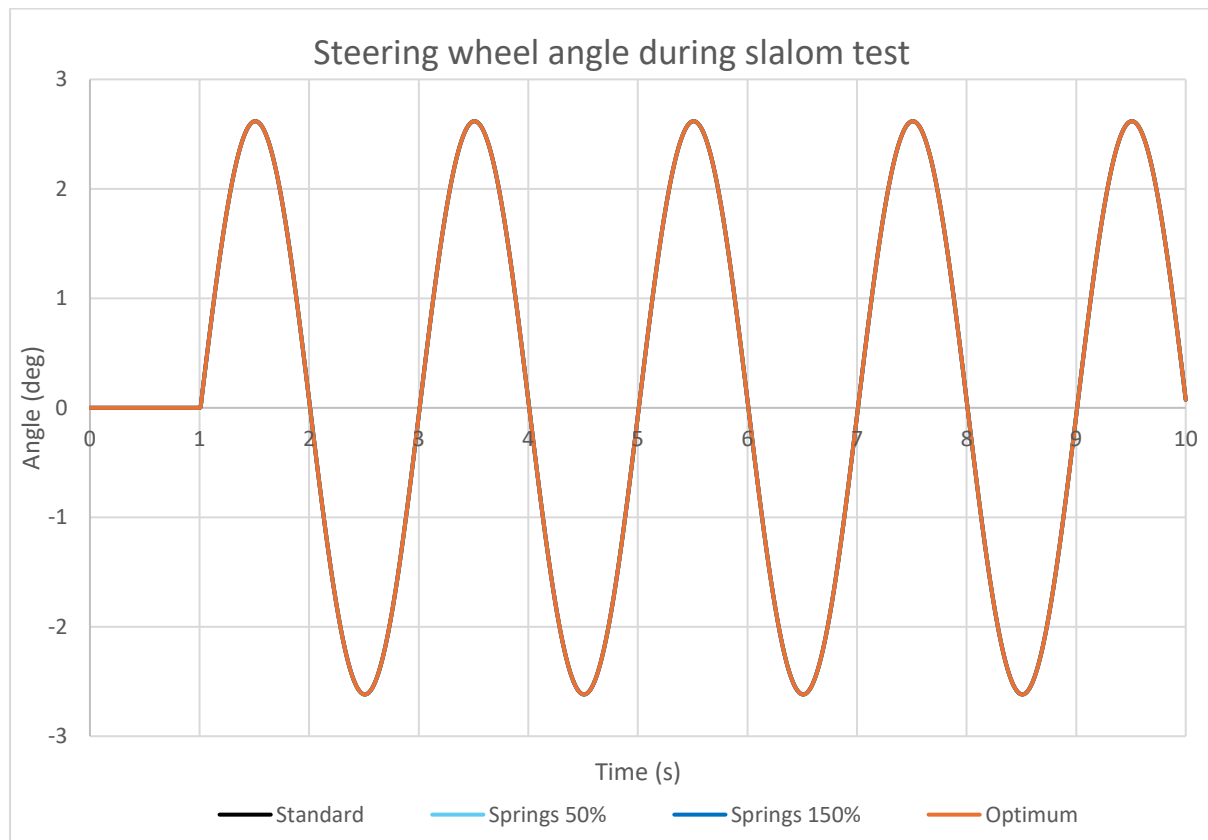


Figure 22 - Steering angle input, slalom test

The steering angle graph shows the correlation between the equal steering input of each test.

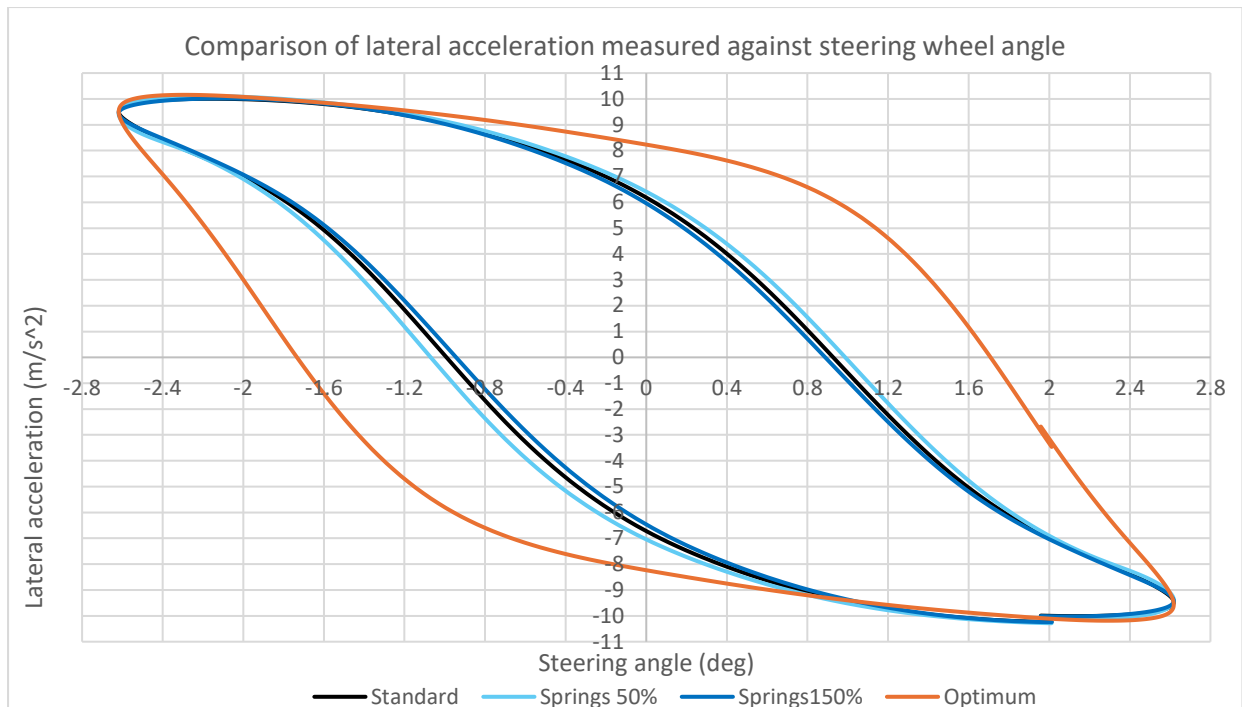


Figure 23 - Lateral acceleration measured against steering angle

The lateral acceleration–steering plot reveals the effect of suspension stiffness on handling response. Softer springs require more steering angle to produce the same lateral acceleration (reduced gain), indicating increased understeer. The stiff setup delivers the highest gain but also shows a more abrupt response, consistent with reduced compliance. The optimum setup offers higher lateral acceleration for a given steering input than standard, but with smoother curvature, demonstrating improved transient response without excessive harshness.

3.1.4 Max performance lap

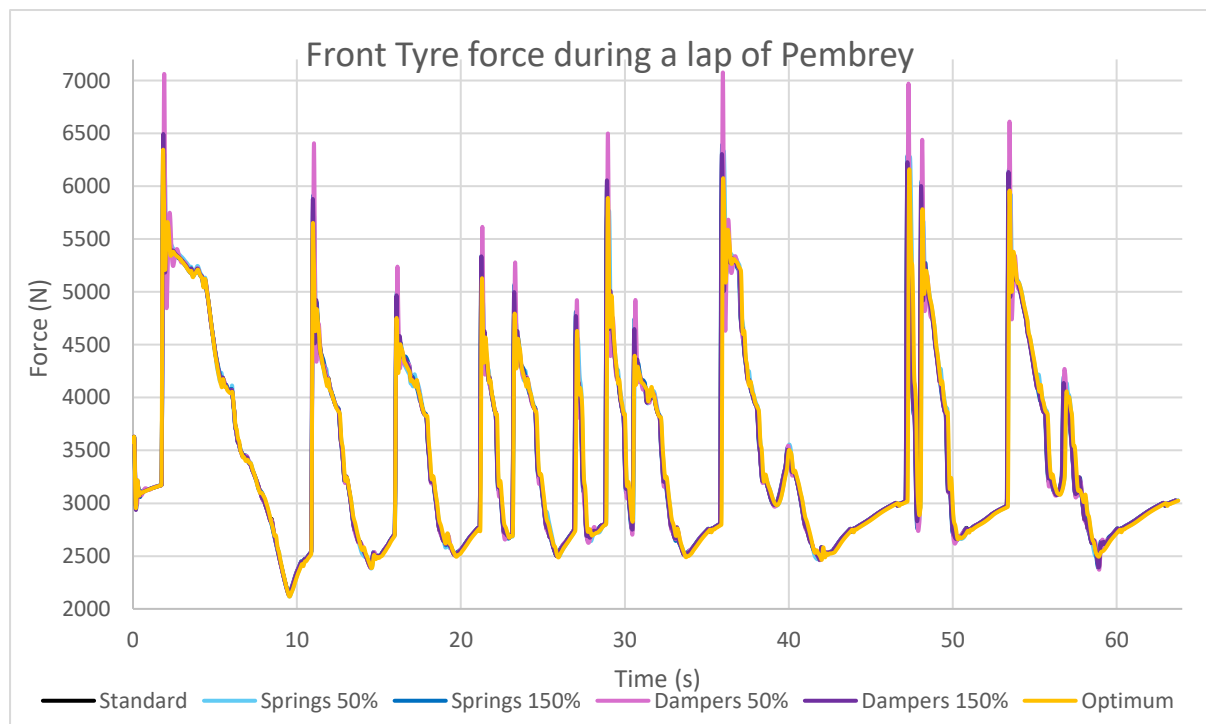


Figure 24 - Front tyre forces over a full lap

The full-lap tyre-load plot shows that all setups experience the same braking and cornering events, confirming consistent lap structure. Because the overlays are too dense to compare meaningfully, a detailed analysis is carried out in **Turn 1**, where load transfer differences between setups—and between VI-Grade and the 4DOF model—are much clearer.

Table 5 - Lap times

Lap times	Time (s)
Std	63.601
Sprg50	63.731
Sprg150	63.581
Damp50	63.601
Damp50	63.591
Optimum	63.761

Lap-time differences between setups are small (<0.2 s), showing that suspension changes mainly affect load transfer behaviour rather than outright pace. The **stiff-spring setup** is marginally quickest due to reduced platform motion, while the **optimal setup** prioritises stability over peak performance. The **soft setup** is slowest, reflecting greater dynamic load variation.

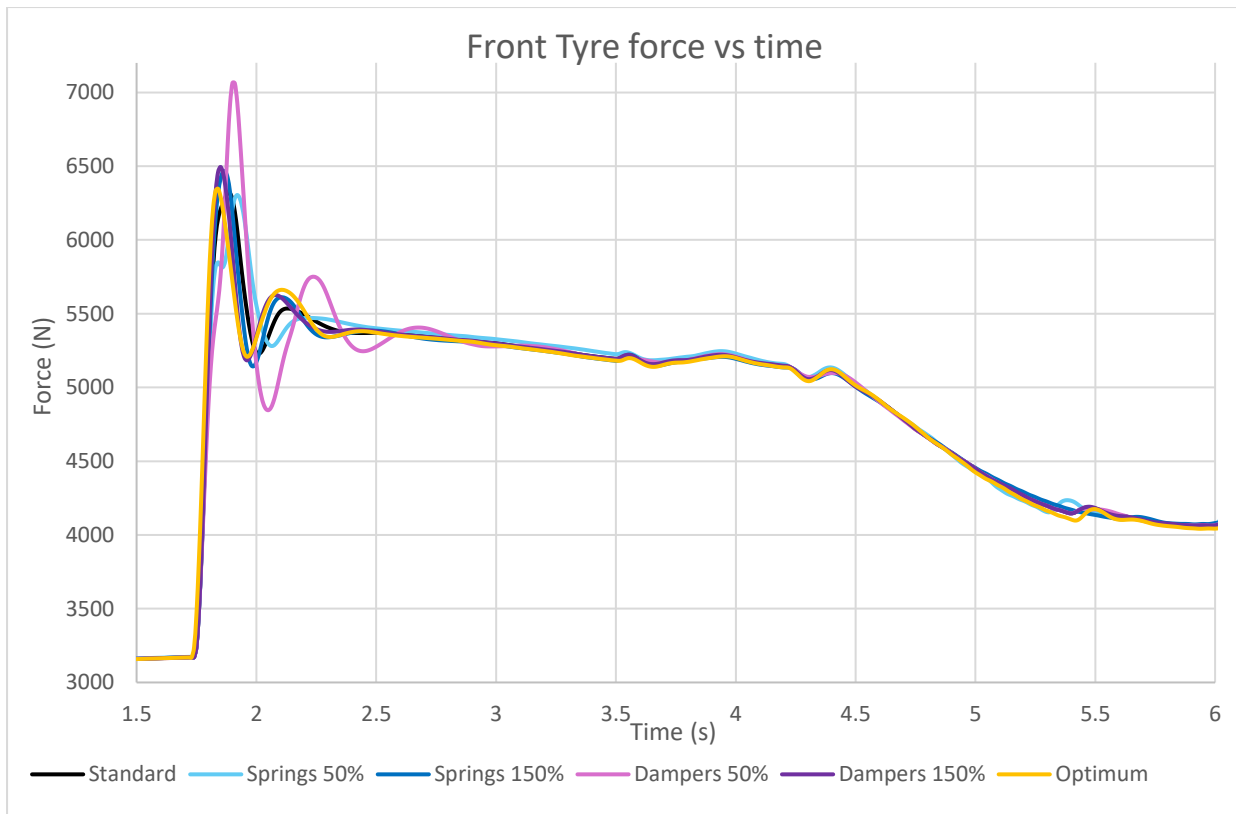


Figure 25 - Front tyre forces, turn 1 Pembrey

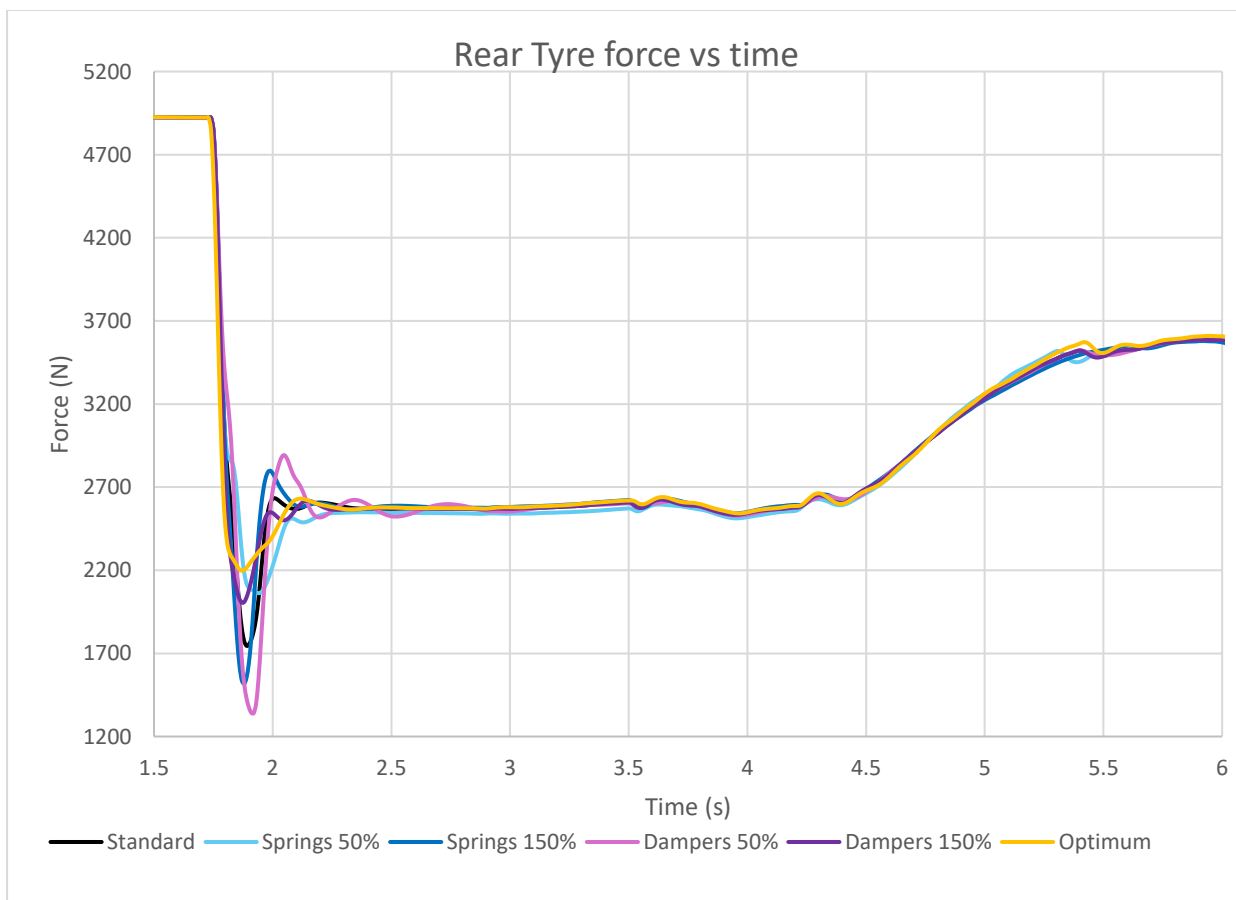


Figure 26 - Rear Tyre forces, turn 1 Pembrey

Turn 1 combines heavy braking followed by initial lateral loading, making it an effective point to compare how each suspension setup manages transient load transfer. The front-axle traces show that the 50% soft springs produce the largest overshoot in peak load and the slowest settling, reflecting lower wheel rates and greater pitch/roll compliance. In contrast, the stiff-spring setup reduces the magnitude of both the initial spike and the follow-up oscillations, consistent with higher vertical stiffness resisting rapid load transfer. The optimal setup again sits between these extremes, offering reduced peak-to-peak variation and quicker stabilisation, indicating more efficient load control and improved tyre utilisation.

Rear-axle behaviour is complementary: softer springs generate deeper load troughs during braking and turn-in, while stiffer springs reduce the depth but introduce sharper, high-frequency oscillations. The optimal configuration provides the smoothest recovery towards steady-state rear load, suggesting a better balance between compliance and platform control. These patterns align with earlier manoeuvre results: excessive compliance increases dynamic load variability, while excessive stiffness transfers load too abruptly. The optimal ride-frequency configuration manages Turn-1 transients more consistently, which is beneficial for corner-entry stability and grip.

3.2 Simulink Models

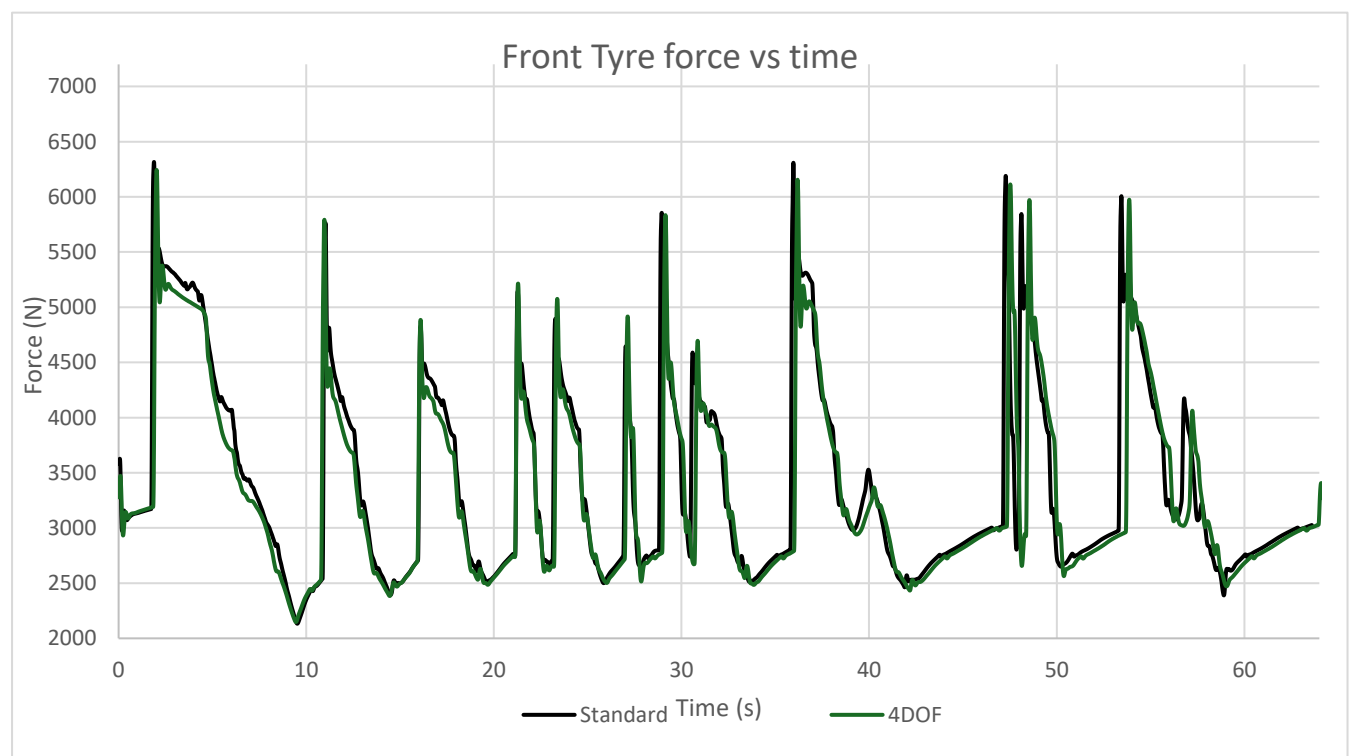


Figure 27 - Front tyre forces - 4DOF vs VI-Grade

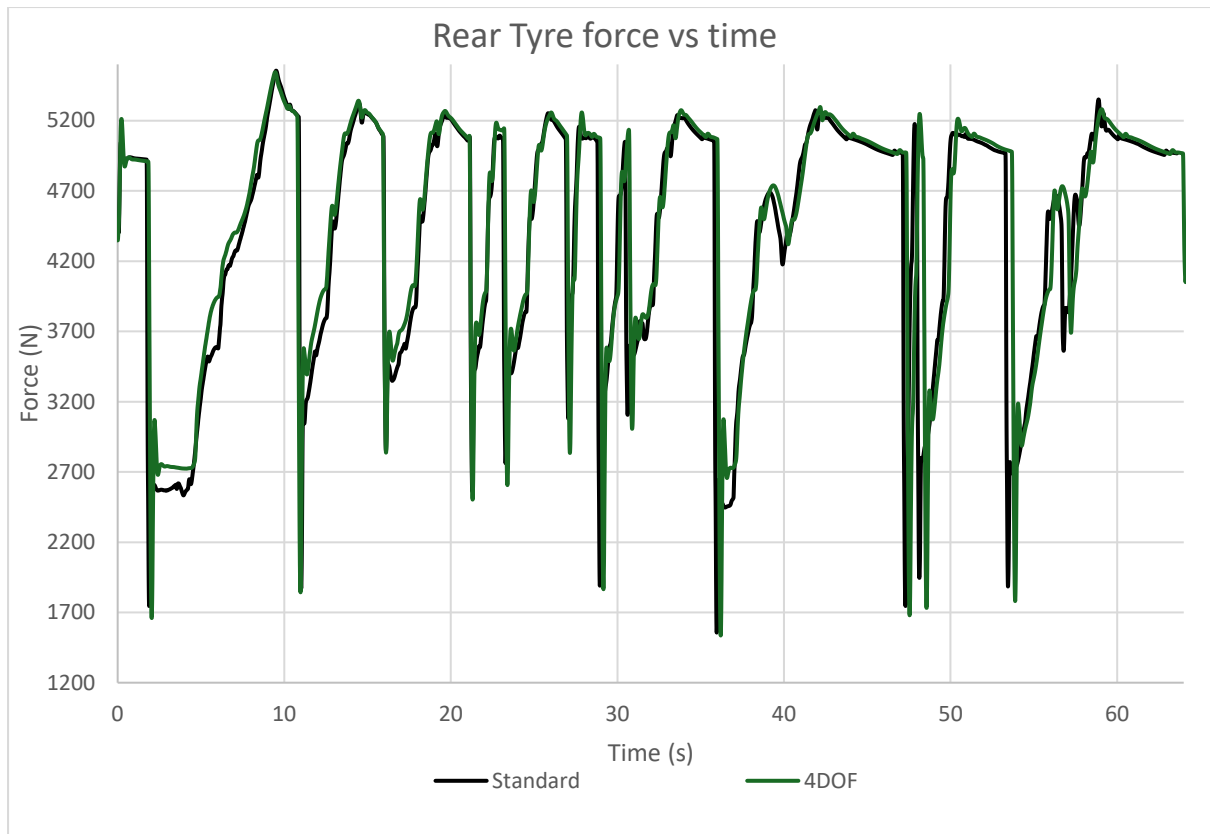


Figure 28 - Rear Tyre forces comparison - 4DOF vs VI-Grade

The above figures compare the VI-Grade standard setup with the 4DOF model using identical inputs, including variable damping and aero loads. The 4DOF model reproduces the overall shape and timing of major load-transfer events, confirming that the simplified mass–spring–damper structure captures the fundamental heave and pitch dynamics. However, the 4DOF outputs show larger oscillations and sharper load transitions, particularly after braking and corner-exit events. This over-prediction arises from the absence of suspension kinematics, tyre load sensitivity, lateral compliance and roll dynamics, all of which in VI-Grade dissipate energy and smooth transient responses. The comparison shows that the 4DOF model is effective for identifying trends but cannot match the absolute load magnitudes or high-frequency behaviour of the multibody simulation.

4 Discussion, Conclusions and Further Development

4.1.1 4.1 Discussion

Across all manoeuvres, suspension stiffness was found to influence transient load transfer far more than steady-state performance. Softer springs increased pitch and roll motion, producing larger dynamic variations in tyre load that reduced stability during braking and corner entry. Stiffer springs reduced these motions but introduced sharper, high-frequency load changes. The ride-frequency-based “optimal” setup consistently provided the smoothest transient response and most stable axle-load distribution, although this did not always translate into the fastest lap time due to the circuit’s emphasis on long straights rather than combined-load corners.

Comparison with the 4DOF model showed that while simplified analytical models capture directional trends, they over-predict oscillation amplitude and cannot reproduce the damping and smoothing seen in VI-Grade. This is expected given the absence of suspension geometry, tyre nonlinearities, roll dynamics and load-sensitive vertical stiffness.

4.1.2 4.2 Conclusions

Suspension tuning primarily shapes the dynamics of load transfer, not the absolute achievable performance. Softer setups degrade stability through excessive motion, while overly stiff setups sharpen transients. The optimal configuration, derived from ride-frequency targets, achieved the most balanced behaviour and improved predictability, though gains in lap time remained small.

4.1.3 4.3 Further Development – 7DOF Model

A 7DOF full-vehicle model was developed to extend the analytical approach. Including heave, roll, pitch and four unsprung masses allow more realistic coupling between longitudinal and lateral dynamics. Although not used for full analysis here, the 7DOF model provides a platform for future work such as validating roll-coupled load transfer, improving alignment with VI-Grade, and assessing suspension changes under combined-load conditions.

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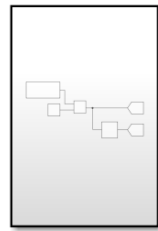
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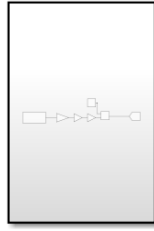
Sanjay Sharma, M. C. V. P. P. J., 2020. Modal and frequency response characteristics of vehicle. *Scholarly*, 810(1).

6 Appendices

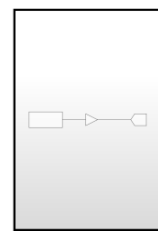
6.1 4 Degree of Freedom Model



Road Input



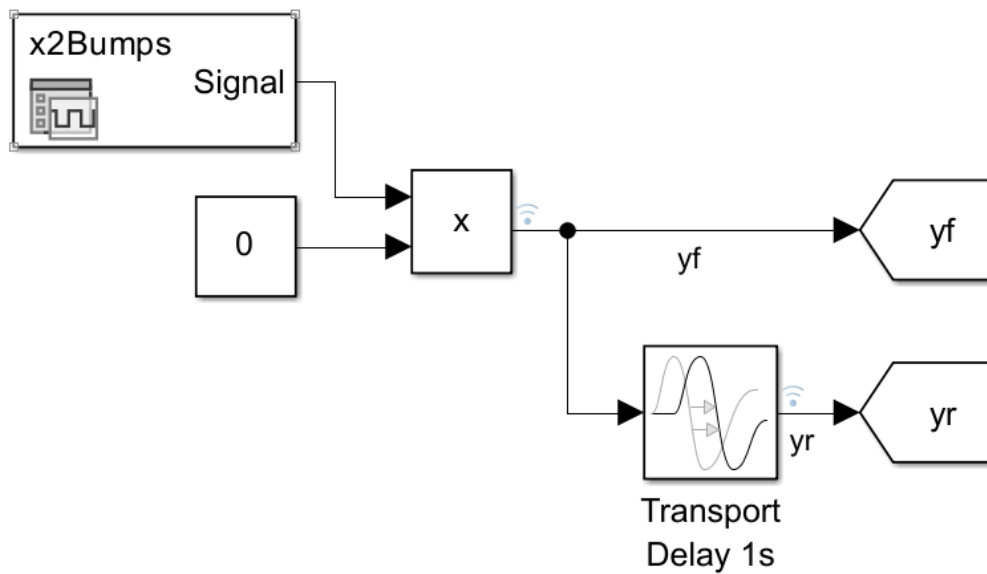
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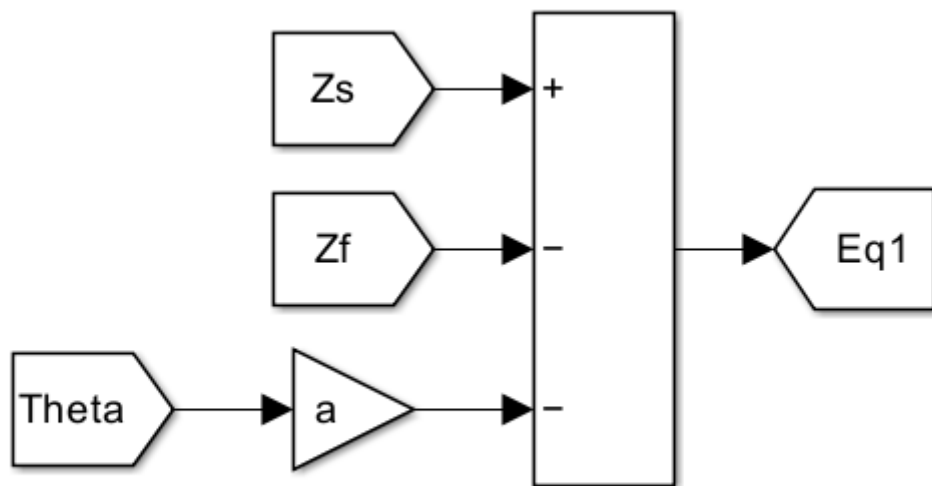
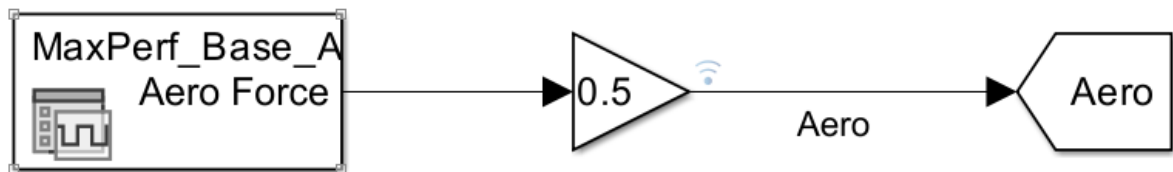
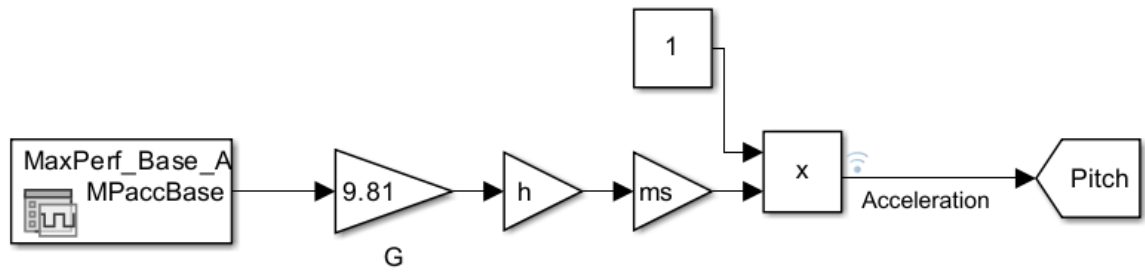


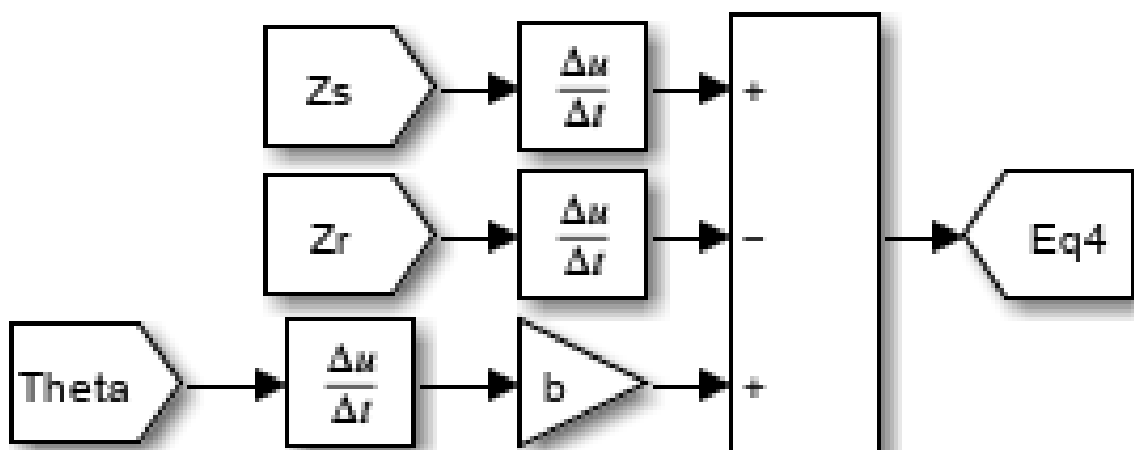
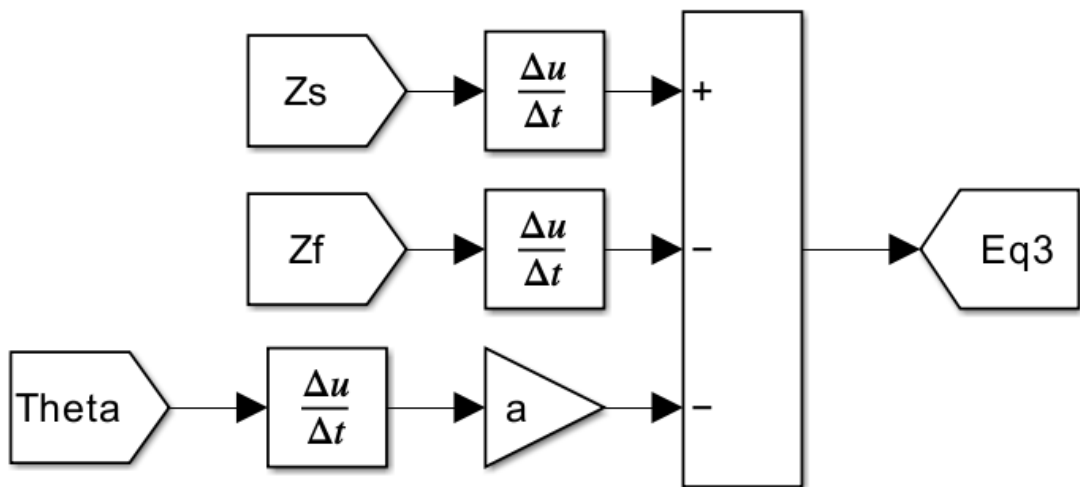
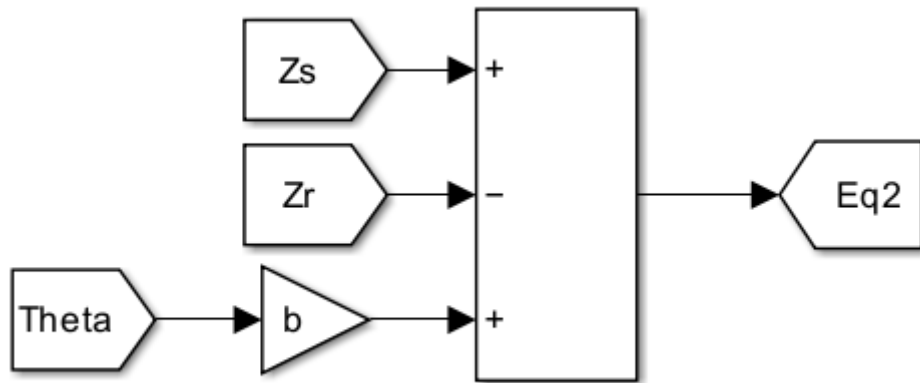
Aero Force

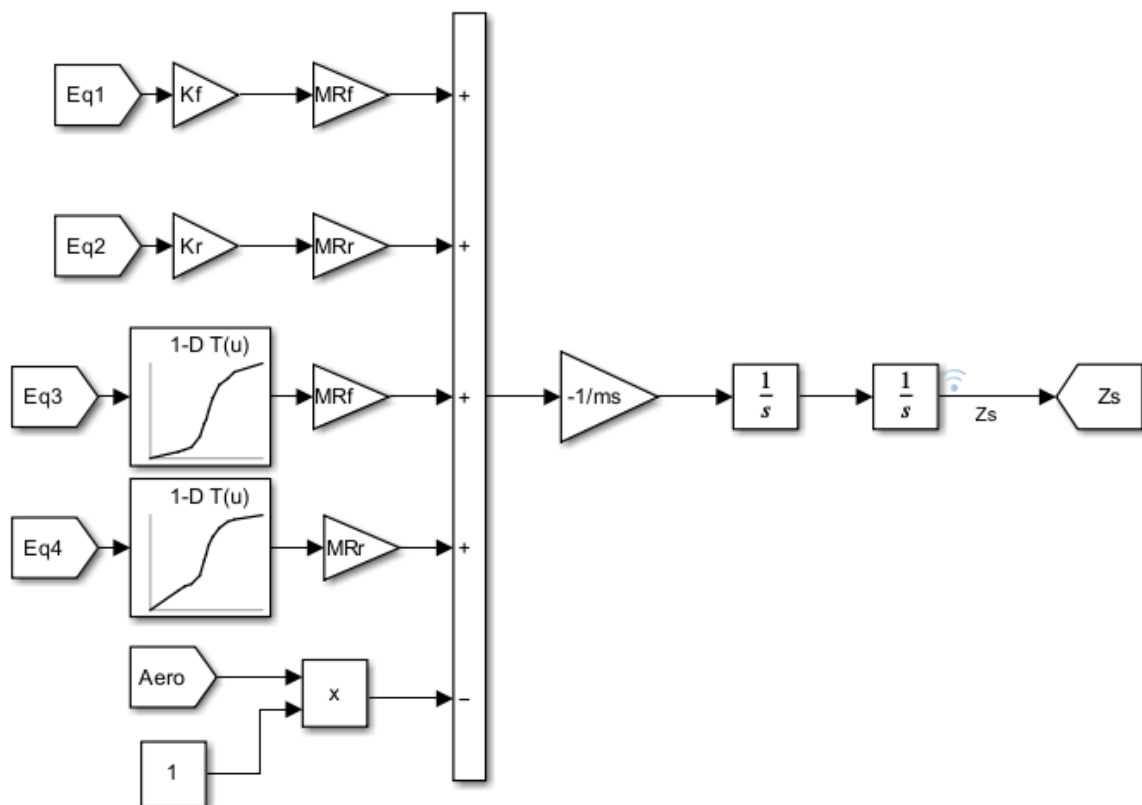
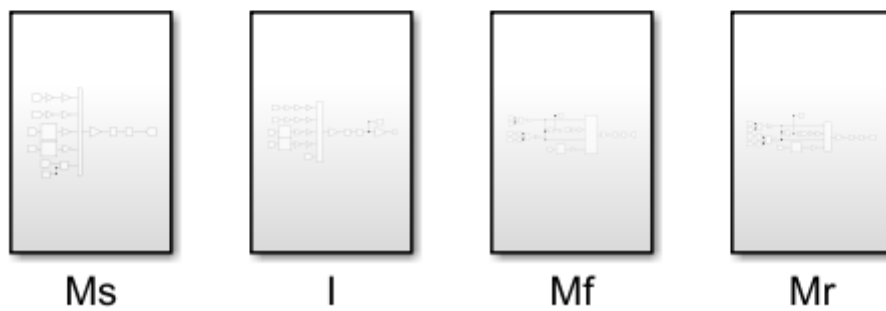


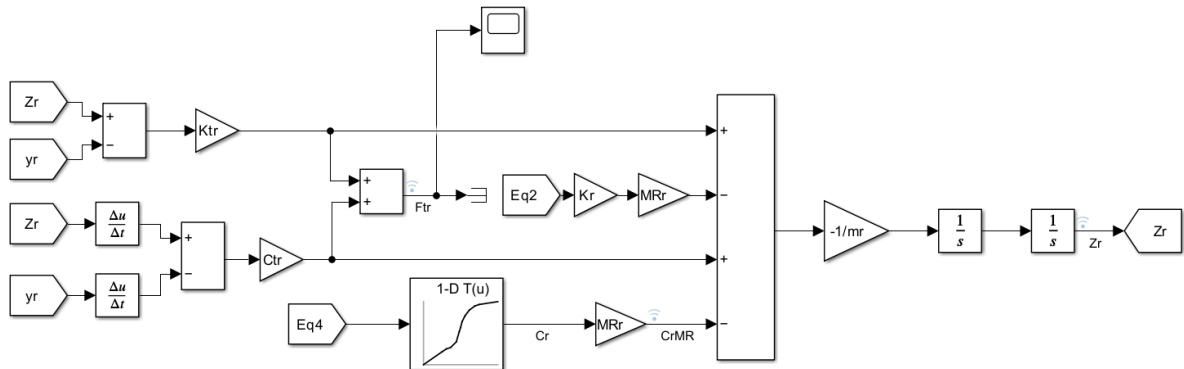
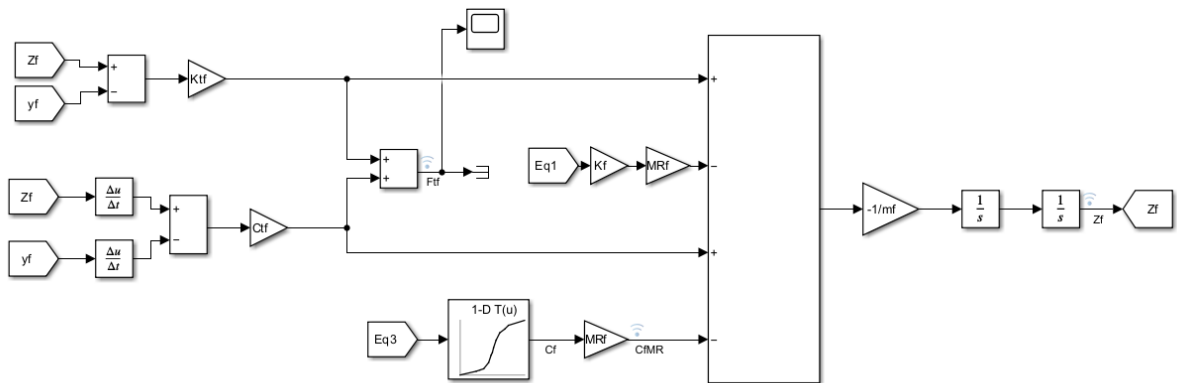
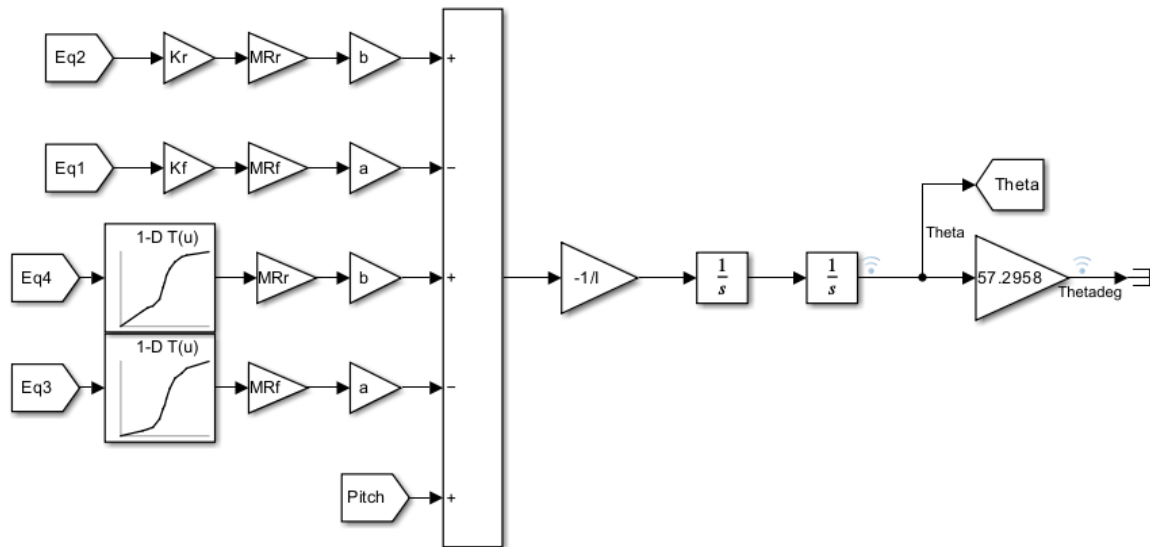
Equations of Motion

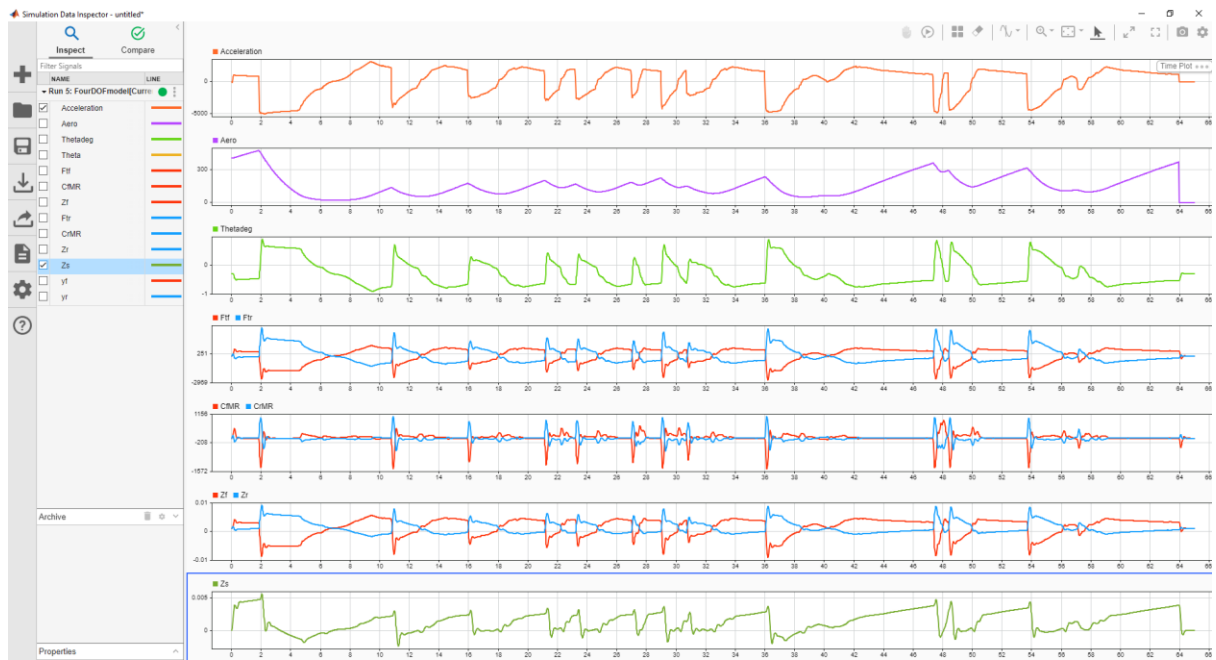




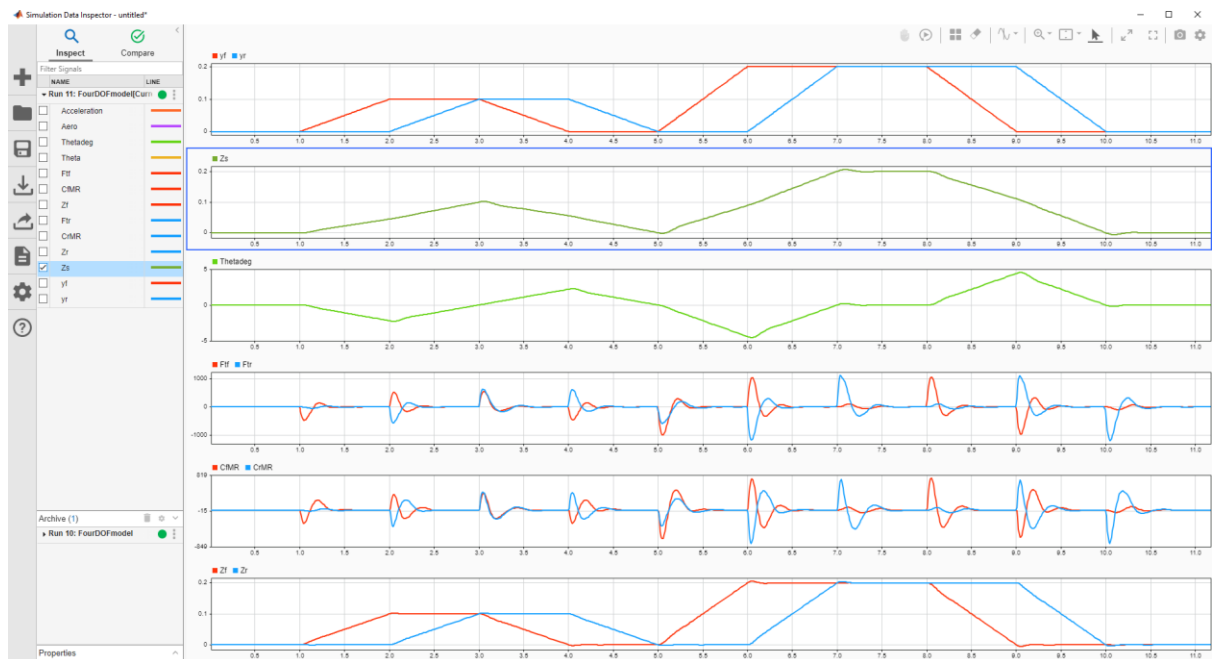








Acceleration input



Bump input

6.2 7 Degree of Freedom

