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1 Abstract

This report investigates how suspension stiffness and damping influence vehicle heave, pitch and tyre loads using both VI-Grade and a 4-DOF half-car Simulink model. VI-Grade provides a high-accuracy reference for dynamic behaviour, while the mathematical model assesses how simplified representations capture vertical and pitch responses. Simulated bump, acceleration and lap-based manoeuvres highlight the effects of spring-damper variations and the limitations of linear modelling. Comparison shows that the half-car model reproduces key trends but overpredicts load transfer, demonstrating the importance of validation when applying reduced-order suspension models.

2 Introduction

The suspension system is a crucial aspect of vehicle dynamics directly affecting handling, stability and ride comfort. Its main role is to manage the interaction between the vehicle body and the road surface, maintaining optimal tyre contact to maximise grip. In motorsport and high-performance vehicles, suspension tuning becomes especially important as it has a significant impact on traction, cornering ability and driver confidence. An accurate suspension model is therefore essential for predicting vehicle behaviour and guiding setup or design decisions across road and race applications.

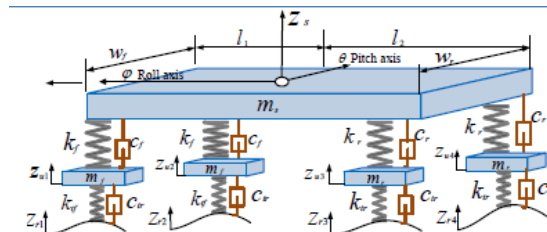
This assignment investigates the dynamic behaviour of a vehicle model in response to changes in suspension setup. VI-Grade is used to assess the performance and tyre loads during dynamic manoeuvres, allowing analysis of how the suspension reacts under different configurations. In parallel, a half-car model is developed in Simulink using a system of equations representing spring, damper and mass interactions. Results from the Simulink model are compared against the VI-Grade simulation data to evaluate the accuracy of the mathematical representation. This comparison helps to determine how well simplified models replicate real-world vehicle behaviour and supports further optimisation of suspension design and tuning.

3 Literature review

The purpose of vehicle suspension systems is to control the ride comfort, handling stability, load transfer and overall body motion. In motorsport applications, vehicle dynamics focuses on how the vehicle responds to driver inputs and external forces such as aerodynamic loads, road disturbances and inertial effects. To understand this behaviour-and to assess the accuracy of simulation tools-numerous studies have examined suspension dynamics using both simplified and full-vehicle models.

Suspension analysis often begins with simplified systems such as the quarter-car and half-car models, which effectively capture the vertical behaviour of sprung and unsprung masses and are useful for initial validation. However, these models cannot fully replicate real vehicle behaviour: the quarter-car model lacks pitch and roll effects, while the half-car model cannot capture full lateral dynamics.

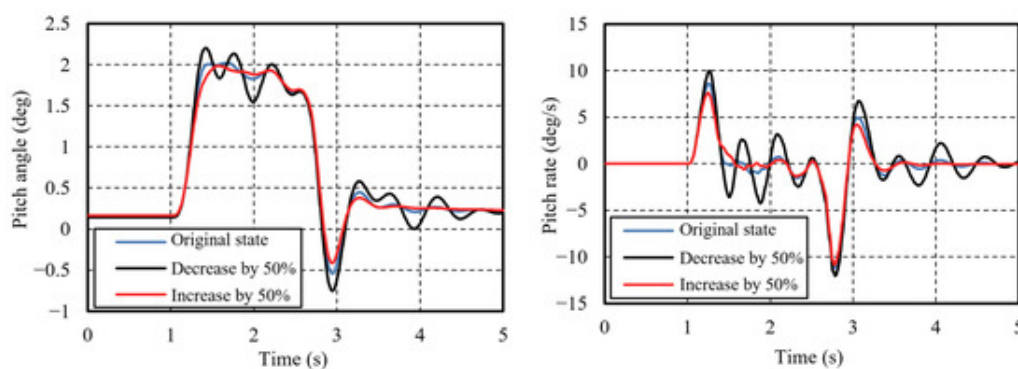
More advanced research therefore employs full-vehicle models. Hamed et al. (2018) uses a seven-degree-of-freedom model incorporating bounce, pitch, roll and individual wheel dynamics. Their study investigates how variations in spring stiffness affect ride and handling performance and validates the simulation using experimental data, highlighting the importance of accurate parameter selection.



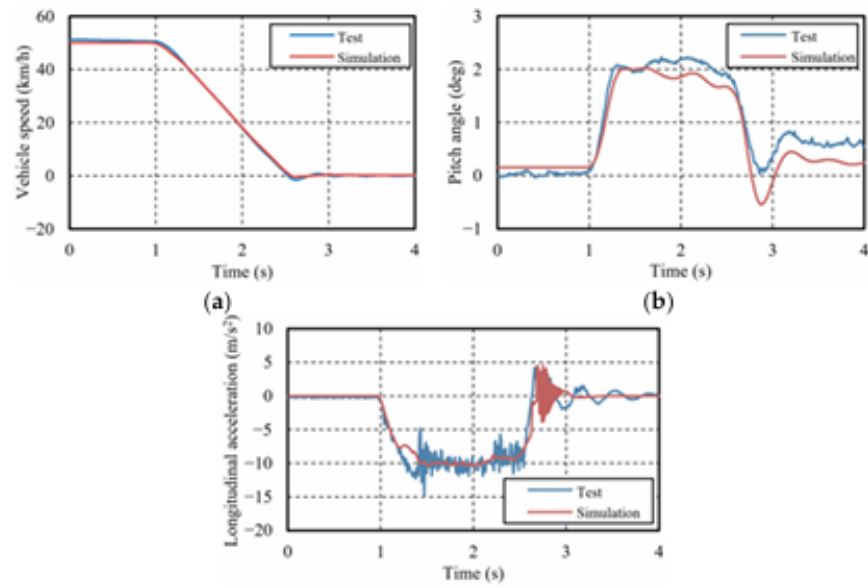
[Figure 1] 7DOF Spring-Mass-Damper system (Hamed et al., 2018)

Additional insight comes from semi-active suspension control research. Samaroo et al.(2025) compares passive, active, and semi-active damping strategies using a quarter-car model, showing that passive systems offer limited adaptability whereas controlled damping can substantially improve ride comfort and transient response. Although simplified, the results reinforce the fundamental relationships between stiffness, damping and body motion that underpin the half-car model used in this report.

Wang et al. (2025) emphasises the need for simulation validation. Their comparison of a semi-active suspension model with braking-test data shows close agreement in pitch angle, speed and longitudinal acceleration, with the simulation under-predicting peak values by 8-9%. This demonstrates both the strengths and limitations of suspension models and provides useful context for interpreting the results presented in this project.



[Figure 2] Pitch changes with Spring-Stiffness (Wang, Wen and Xie, 2025)



[Figure 3] Real test data vs Simulation (Wang, Wen and Xie, 2025)

4 VI-Grade modelling

4.1 Vehicle Data & Setup

VI-Grade is a simulation software that provides a full vehicle representation modelling nonlinear suspension kinematics, load-dependant tyre behaviour and chassis compliance effects. In this report VI-Grade is used as a reference model to evaluate how suspension setup changes influence the dynamic response and performance of a race vehicle. The results obtained provide benchmark data which will later be compared with predictions from a 4DOF Simulink model.

A baseline vehicle was selected from the VI-Grade database (race car package), providing a validated starting point with representative geometry, mass distribution and spring and damper properties. This data will be modified to implement a series of different spring and damper rates to study their effects on vertical and pitch dynamics. The primary parameters extracted from VI-Grade and required for the 4-DOF Simulink model are summarised in table...

[Table 1] VI-Grade 'race-car' parameters

Parameter	Value	Unit
Mass	1052	kg
CoG height	381.16	mm
Wheelbase	2712.9	mm
Distance from front wheels to CoG (a)	1230	mm
Distance from rear wheels to CoG (b)	1482.9	mm
Inertia (I)	138.03	kgm ²
Unsprung Mass Front (mf)	34	kg
Unsprung Mass Rear (mr)	41	kg
Spring Rate Front (kf)	202.3	N/mm
Spring Rate Rear (kr)	100.2	N/mm

The baseline suspension configuration consisted of supplied front and rear spring and damper rates. To evaluate system sensitivity, several variations of these baseline values were applied. Anti-roll bar rates were kept constant due to model constraints in the 4DOF model, meaning the analysis focuses primarily on vertical and pitch behaviour rather than lateral load transfer or roll dynamics. Damping characteristics are defined in VI-Grade as force-velocity curves, therefore, damper variations were implemented by applying multipliers to the baseline force-velocity profile.

Setups 1-6 consist of front and rear spring rate variations to investigate the effect of global stiffness changes on pitch angle, ride response and tyre load fluctuations.

Setups 7-10 alter damping levels by scaling the baseline damper curves, enabling analysis of how increased or reduced damping influences transient behaviour and ride comfort.

[Table 2] Setups 1-6: Spring stiffness changes

	Setup 1	Setup 2	Setup 3	Setup 4	Setup 5	Setup 6
Baseline	Soft springs	Stiff springs	Front soft	Rear soft	Front stiff	Rear stiff
(N/mm)	50%	150%	Front 50%	Rear 50%	Front 150%	Rear 150%
202.3	101.15	303.45	101.15	202.3	303.45	202.3
100.2	50.1	150.3	100.2	50.1	100.2	150.3

[Table 3] Setups 7-10: Damper changes

		Setup 7		Setup 8		Setup 9		Setup 10	
Baseline		Underdamped		Light underdamped		Lightly overdamped		Overdamped	
Force(N)		50%		80%		120%		150%	
F	R	F	R	F	R	F	R	F	R
-2111	-1390	-1055.5	-695	-1688.8	-1112	-2533.2	-1668	-3166.5	-2085
-671	-359	-335.5	-179.5	-536.8	-287.2	-805.2	-430.8	-1006.5	-538.5
0	0	0	0	0	0	0	0	0	0
533	249	266.5	124.5	426.4	199.2	639.6	298.8	799.5	373.5
1557	1052	778.5	526	1245.6	841.6	1868.4	1262.4	2335.5	1578

[Table 4] Setup 11-15: Spring and damper changes, Setup 15: Spring change

Setup 11		Setup 12		Setup 13		Setup 14		Setup 15
Soft ride		Extra Sport		Hatchback style		Max Extreme		
50%		150%		10%		200%		10%
101.15		303.45		20.23		404.6		20.23
50.1		150.3		10.02		200.4		10.02
F	R	F	R	F	R	F	R	
-1055.5	-695	-3166.5	-2085	-211.1	-139	-4222	-2780	
-335.5	-179.5	-1006.5	-538.5	-67.1	-35.9	-1342	-718	
0	0	0	0	0	0	0	0	
266.5	124.5	799.5	373.5	53.3	24.9	1066	498	
778.5	526	2335.5	1578	155.7	105.2	3114	2104	

Setups 11-14 extend this analysis by applying more extreme spring and damper combinations, allowing a wider range of handling behaviours to be explored. Setup 15 was also included to observe the model's response at very low stiffness levels, providing understanding of how the system behaves under minimal spring stiffness.

Initial simulations revealed that the baseline vehicle had significantly higher suspension stiffness than values typically reported in the literature review for other vehicles. This high stiffness limited the observable variation in outputs, particularly in straight line acceleration, where pitch angle remained small and tyre force changes were negligible. This led to the decision to introduce a small variation of lower spring rates (Eshtaiwi, Ahmed and Barood, 2022).

This approach to setups ensures that the comparisons between VI-Grade and Simulink are based on physically meaningful changes in suspension properties, enabling clearer interpretation of model limitations.

4.2 Manoeuvres

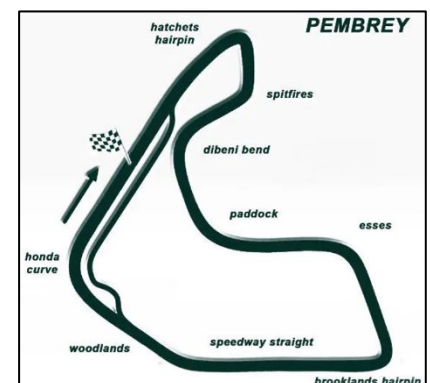
4.2.1 Straight Line Acceleration

A straight line acceleration test was used to assess how each setup influences longitudinal dynamics, specifically pitch response and vertical load transfer. Although acceleration performance is mainly limited by powertrain output and tyre grip, suspension stiffness and damping determine how the chassis pitches under throttle and how quickly it settles. Softer springs increase rearward pitch and load transfer, while stiffer springs reduce pitch magnitude but increase transfer rate. Higher damping limits overshoot and shortens transient response.

Because this manoeuvre involves no lateral forces, it provides a clean reference for validating the 4DOF half-car model. This model's pitch and vertical load predictions can be directly compared to VI-Grade results under the same acceleration profile.

4.2.2 Lap Simulation-Pembrey

A flying lap of Pembrey was completed using the baseline and modified suspension setups to evaluate performance under realistic, combined-load conditions. Unlike straight-line acceleration, the lap introduces simultaneous breaking, cornering and acceleration events, generating tyre loads and chassis motion that isolated tests cannot produce. The same data-collection process was applied as before, allowing changes in spring and damper settings to be compared across an entire lap. This provides insight into how each setup influences stability, load transfer and overall track performance when subjected to real-world dynamic demands.



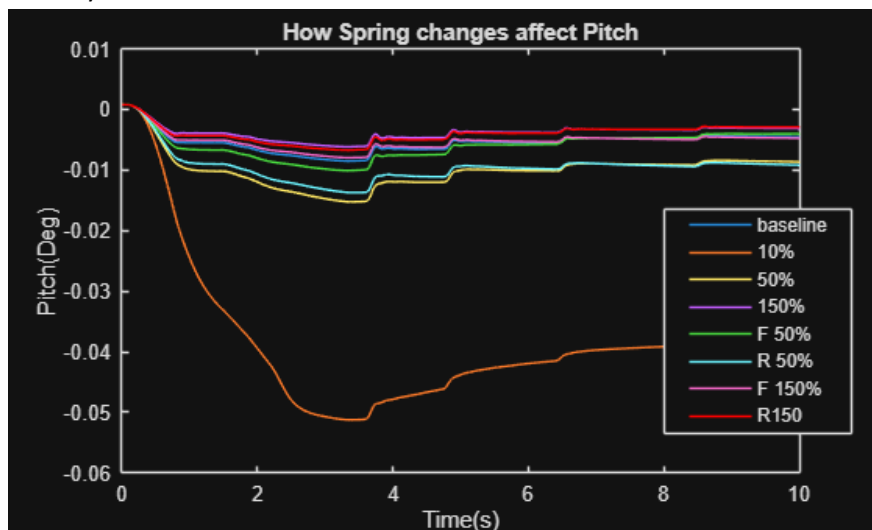
[Figure 4] Pembrey circuit (Stanley, 2019)

4.3 Results & Analysis

4.3.1 Straight Line Acceleration

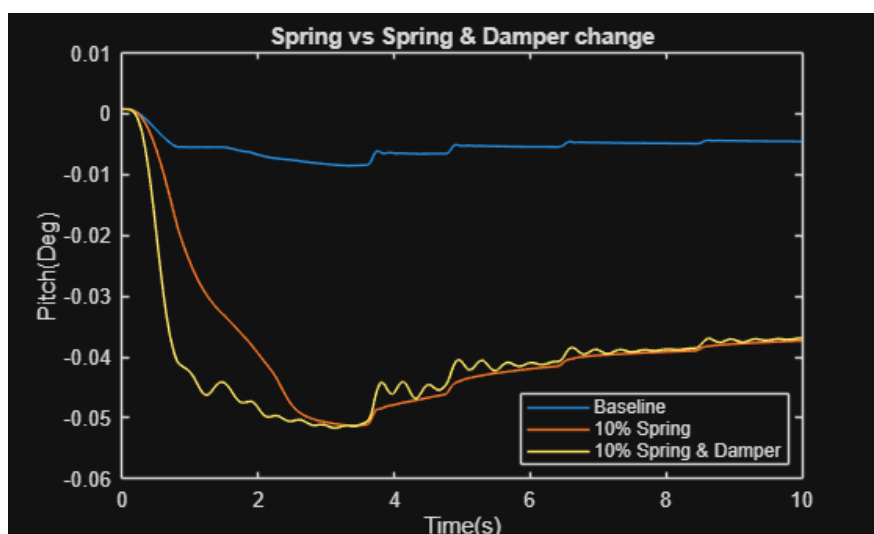
The following results analyse the vehicle's dynamic response during straight line acceleration, assessing the influence of spring and damper changes on pitch behaviour under load transfer.

The graph in figure 5 shows how the varying spring stiffness has little effect on pitching in straight line acceleration because the baseline springs are already very stiff, limiting suspension travel and chassis rotation. The sharp initial pitch changes come from the rapid rearward weight transfer at launch. Softer springs show greater pitch deviation since they allow more movement under load, with the 10% reduction producing the largest change, though still partly controlled by the baseline dampers. In motorsport, higher spring rates are preferred as reduced body motion maintains aerodynamic stability.

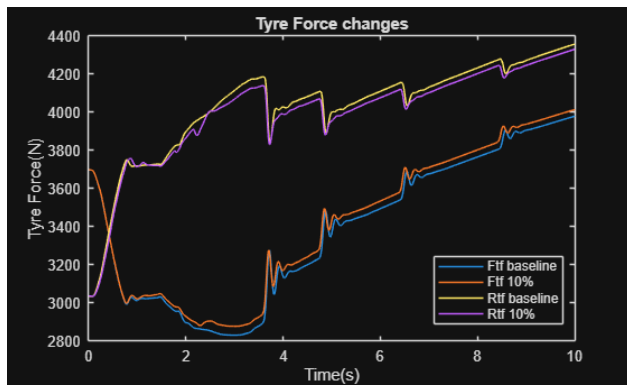


[Figure 5] How Pitch changes with Spring rates

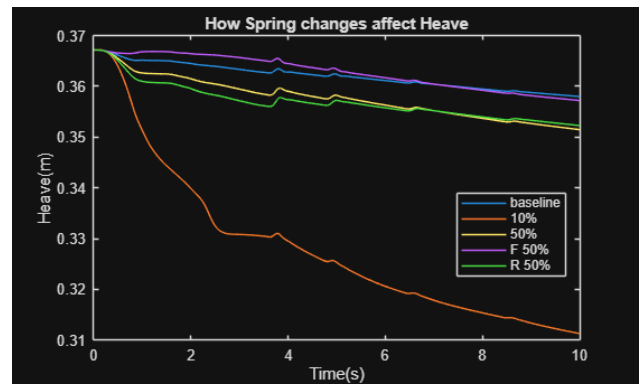
Setups 11-14 in section 4.1 were tested in a straight-line simulation. The stiffer setups behaved similarly to the spring-only tests, so the focus is on the 10% changes. The graph in figure 6 compares reducing only the springs to reducing both springs and dampers by 10%. A large oscillation around 4s indicates setup 13 is underdamped, allowing continued chassis rotation instead of rapid stabilisation (as shown in the baseline setup). This shows how strongly damping controls transient motion and why higher spring and damper rates improve performance and stability.



[Figure 6] How Pitch changes with Spring vs Spring and Dampers



[Figure 7] Tyre force changes with spring rates



[Figure 8] Spring change affects on Heave

Figure 7 shows how spring stiffness affects tyre forces during acceleration. A rapid load transfer to the rear occurs before the springs and dampers attempt to stabilise under constant acceleration. Differences between the baseline and 10% setup are small because the dampers remain unchanged, absorbing most of the additional chassis motion caused by the softer springs

Figure 8 illustrates how spring stiffness influences heave motion. Reducing the springs to 10% causes a noticeably larger drop in heave, while 50% reductions allow faster vertical movement. Under acceleration, rearward load transfer dominates, increasing rear compression and producing a greater overall heave decrease regardless of front stiffness.

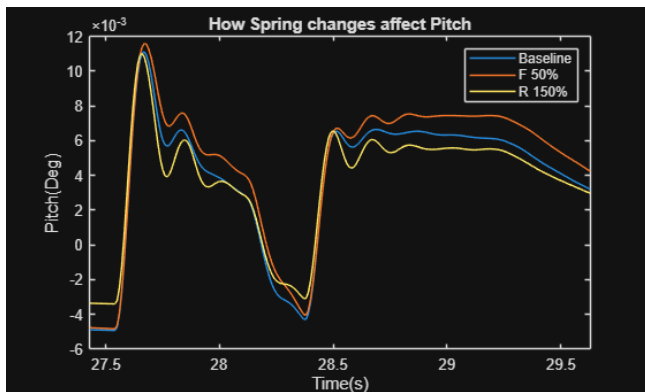
4.3.2 Lap Simulation

The lap simulation results provide a broader assessment of how spring and damper changes influence the vehicle's dynamic behaviour around a circuit, including cornering balance, tyre loads, and lap time.

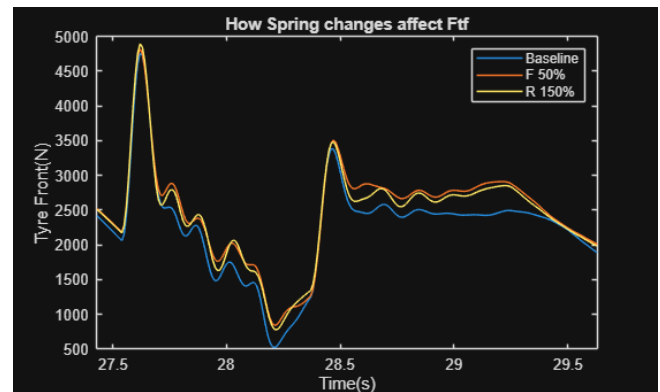
The table below shows the performance of each setup around Pembrey and the lap-time difference from the baseline. The quickest laps came from setup 3 (50% softer front springs) and setup 6 (50% stiffer rear), both producing a more balanced front-rear stiffness. This suggests the car favours a neutral setup. Damping adjustments had minimal influence on lap-time whereas large spring-rate imbalances, particularly a much stiffer front, significantly worsened performance.

[Table 5] Lap times for each setup

Setup	Lap Time	Difference vs Baseline(s)
Baseline	56.931	-
Setup 1	56.951	0.02
Setup 2	56.941	0.01
Setup 3	56.891	-0.04
Setup 4	57.161	0.23
Setup 5	57.081	0.15
Setup 6	56.891	-0.04
Setup 7	56.981	0.05
Setup 8	56.931	0
Setup 9	56.941	0.01
Setup 10	56.951	0.02
Setup 11	56.941	0.01
Setup 12	56.931	0
Setup 13	59.141	2.21
Setup 14	56.931	0
Setup 15	57.131	0.2

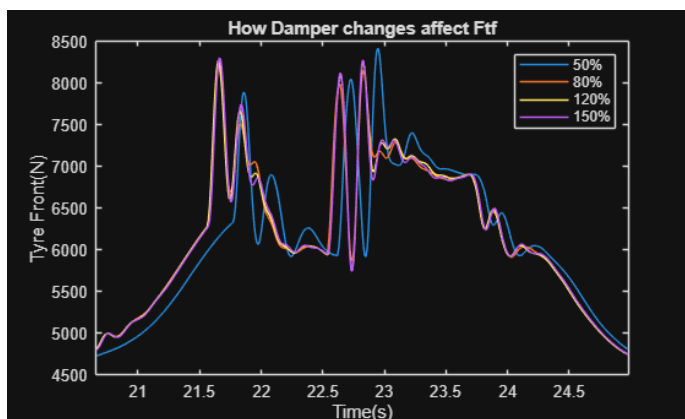


[Figure 9] Spring change affects on pitch around Pembrey

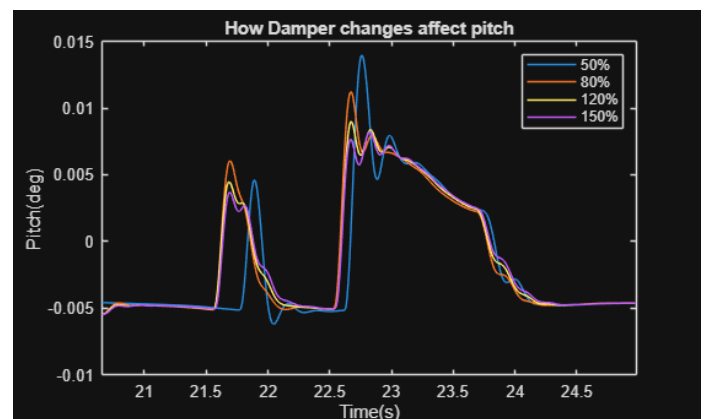


[Figure 10] Spring change affects on front-tyre force around Pembrey

Looking at the fastest laps, the graphs in figure 9 and 10 show how pitching and front-tyre forces change through esses corner. Just after 27.5s the brakes are applied, generating forward pitch and transferring load to the front-axle. The softer springs produce greater pitch and higher front-tyre forces than the baseline setup, yet still achieves a slightly quicker lap (0.04s faster). In contrast, stiffening the rear by 50% reduces pitch, while increasing front-tyre forces relative to baseline. Some compliance is important for driver feedback, but overall a stiffer setup limits body motion and generally improves cornering performance-especially in a car with a very low centre-of-gravity and mass.



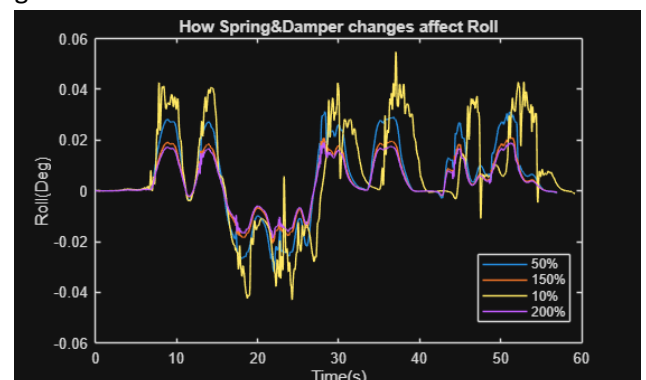
[Figure 11] Damper change affects on Front-tyre force



[Figure 12] How damper rates affect pitch

The graphs in figure 11 and 12 are used to show how damper changes affect front-tyre forces and pitch on paddock corner (figure 4). The biggest difference is with 50% damping rate. This setup has much later breaking, causing a delay in front-tyre force and forward pitch compared to the others. The small difference in change between the other setups suggests that damping below 80% of baseline causes a prolonged settling of ride height and less ride comfort, as the orange line follows the same trend as 120% and 150% but with smaller damping in some areas.

Figure 13 shows how different springs and damper settings affect the car's roll angle over time. The stiffer setups (200%) create sharper, more pronounced roll movements, while softer setups (10 or 50%) produce smoother, smaller changes. This further proves that suspension stiffness clearly changes handling and shows exactly how much and how quickly the car rolls during the lap, especially with softer springs (10%).



[Figure 13] Suspension affects on Roll

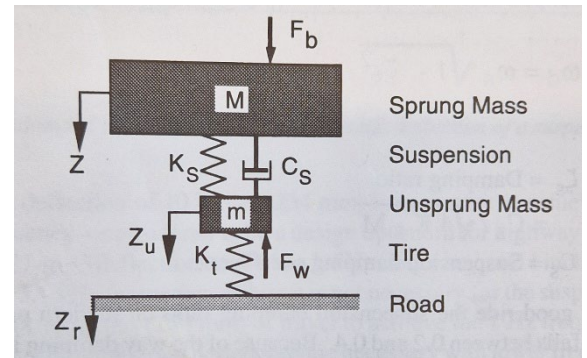
5 4DOF Half-Car Model

5.1 Equations of Motion Derivation

This section derives the equations of motion for the 4-DOF (degrees of freedom) half-car model, defining all coordinates, assumptions and forces used in later Simulink implementation.

The four degrees of freedom considered are:

1. Sprung mass heave (z_s)
2. Front unsprung mass vertical displacement (z_f)
3. Rear unsprung mass vertical displacement (z_r)
4. Pitch angle of the sprung mass (θ)



[Figure 14] Quarter-car model (Gillespie, 1994)

A quarter-car model (Figure 14) is commonly used to represent vertical suspension behaviour, however it cannot describe pitch motion. The half-car model extends the quarter-car system by including pitch dynamics, enabling analysis of longitudinal load transfer between the front and rear axles. To simplify the derivation, the following assumptions are made:

- Small-angle pitch approximation: $\sin\theta \approx \theta$, $\cos\theta \approx 1$.
- Linear suspension springs.
- Constant tyre stiffness (200kN/m) and tyre damping (16Ns/m).
- Left and right assumed symmetry.
- Matched road input (left and right).
- No roll dynamics.

These assumptions allow the system to be modelled using linear differential equations suitable for simulation and control analysis.

[Table 7] Coordinates and Definitions

Symbol	Description	Unit
z_s	Vertical displacement of sprung mass	m
z_f, z_r	Vertical displacement of front and rear unsprung mass	m
y_f, y_r	Road inputs (front and rear)	m
k_f, k_r	Suspension stiffness (front and rear)	N/m
c_f, c_r	Suspension damping coefficients	Ns/m
k_{tf}, k_{tr}	Tyre stiffness	N/m
c_{tf}, c_{tr}	Tyre damping	Ns/m
a, b	Distances from CoG to front and rear axles	m
I	Pitch moment of inertia about CoG	kgm ²
θ	Pitch angle about CoG	Degrees

[Table 8] Elements and extensions for Force equations

Element	Extension
k_f	$z_s - z_f - a\theta$
k_r	$z_s - z_r + b\theta$
k_{tf}	$z_f - y_f$
k_{tr}	$z_r - y_r$
c_f	$\dot{z}_s - \dot{z}_f - a\dot{\theta}$
c_r	$\dot{z}_s - \dot{z}_r + b\dot{\theta}$
c_{tf}	$\dot{z}_f - \dot{y}_f$
c_{tr}	$\dot{z}_r - \dot{y}_r$

5.1.1 Force definitions

The suspension and tyre forces are expressed below using spring-damper relationships. These forces act on the sprung and unsprung masses and form the basis of the dynamic equations.

Suspension Forces:

$$F_f = -k_f(z_s - z_f + a\theta) - c_f(\dot{z}_s - \dot{z}_f + a\dot{\theta}) \quad [\text{Equation 1}]$$

$$F_r = -k_r(z_s - z_r + b\theta) - c_r(\dot{z}_s - \dot{z}_r + b\dot{\theta}) \quad [\text{Equation 2}]$$

Tyre Forces:

$$F_{tf} = -k_{tf}(z_f - y_f) - c_{tf}(\dot{z}_f - \dot{y}_f) \quad [\text{Equation 3}]$$

$$F_{tr} = -k_{tr}(z_r - y_r) - c_{tr}(\dot{z}_r - \dot{y}_r) \quad [\text{Equation 4}]$$

5.1.2 Equations of Motion

Upwards is taken as positive. Newton's Second Law is applied in the vertical direction for each mass, consistent with the approach used in literature (Jonjo and Nyalloma, 2020):

$$F = ma \therefore F = m\ddot{z} \quad [\text{Equation 5}]$$

Sprung mass Equation:

$$m_s\ddot{z}_s = -(F_{sf} + F_{sr}) \quad [\text{Equation 6}]$$

Unsprung Mass Equations:

$$m_f\ddot{z}_f = F_{sf} - F_{tf} \quad [\text{Equation 7}]$$

$$m_r\ddot{z}_r = F_{sr} - F_{tr} \quad [\text{Equation 8}]$$

Pitch moment equation:

Using: $T = I\alpha \therefore T = I\ddot{\theta}$

$$I\ddot{\theta} = aF_f - bF_r \quad [\text{Equation 10}]$$

The front suspension force produces a nose-down moment about the centre of gravity, while the rear force produces a nose-up moment. Their difference governs pitch acceleration.

Sprung mass vertical forces:

$$-m_s\ddot{z}_s = k_f(z_s - z_f - a\theta) + k_r(z_s - z_r + b\theta) + c_f(\dot{z}_s - \dot{z}_f - a\dot{\theta}) + c_r(\dot{z}_s - \dot{z}_r + b\dot{\theta}) \quad [\text{Equation 11}]$$

Pitching of sprung mass:

$$-I\ddot{\theta} = bk_r(z_s - z_r + b\theta) - ak_f(z_s - z_f - a\theta) + bc_r(\dot{z}_s - \dot{z}_r + b\dot{\theta}) + c_f(\dot{z}_s - \dot{z}_f - a\dot{\theta}) \quad [\text{Equation 12}]$$

Unsprung masses front and rear:

$$-m_f\ddot{z}_f = k_{tf}(z_f - y_f) - k_f(z_s - z_f - a\theta) + c_{tf}(\dot{z}_f - \dot{y}_f) - c_f(\dot{z}_s - \dot{z}_f - a\dot{\theta}) \quad [\text{Equation 13}]$$

$$-m_r\ddot{z}_r = k_{tr}(z_r - y_r) - k_r(z_s - z_r + b\theta) + c_{tr}(\dot{z}_r - \dot{y}_r) - c_r(\dot{z}_s - \dot{z}_r + b\dot{\theta}) \quad [\text{Equation 14}]$$

5.2 Simulink Model

5.2.1 Model overview

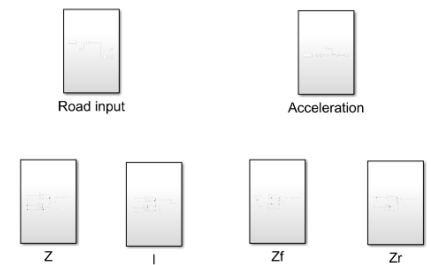
A 4DOF half-car model was developed in Simulink to simulate suspension dynamics such as heave and pitch. It represents values such as sprung mass, front and rear unsprung mass, and pitch rotation using a system of linear equations taken from the derivation in the section above. The relevant data from VI-Grade was inputted into a MATLAB workspace for Simulink to receive. For a half-car system, the total vehicle mass and inertia values are halved to represent a symmetric left/right split and to remain consistent with the model assumptions.

5.2.2 Model development process

To support the development of the 4DOF model, two simplified validation models were created first. A 1DOF heave model was used to confirm the sprung mass natural frequency and damping behaviour, while a 2DOF quarter-car model verified tyre-suspension interaction and basic vertical dynamics. These steps ensured the final 4DOF system behaved with expected suspension responses. (Both models and outputs shown in appendix).

5.2.3 Implementing Equations into Simulink

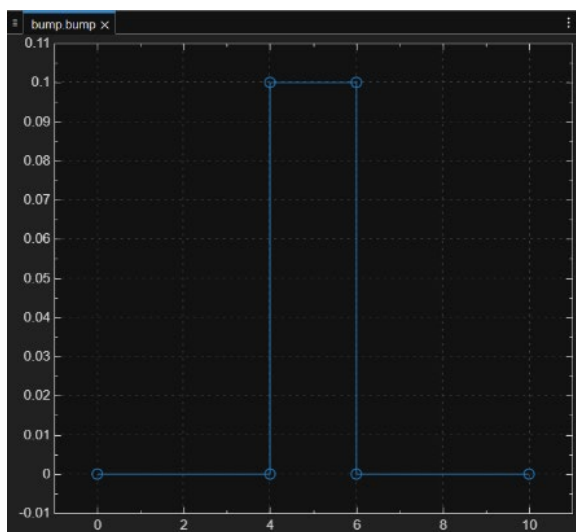
The Simulink model is built around the equations of motion derived in section 5.1. Each degree of freedom is modelled as a second order differential equation using summation blocks for total force (per corner axle), integrator blocks (for acceleration to displacement), and gain blocks for parameter values (taken from VI-Grade). Front and rear suspension subsystems share an identical architecture with differences only in parameter values and the necessary sign conventions from pitch rotation. Suspension forces were modelled using linear spring stiffness values, while damper forces were derived from the vi grade damper curves.



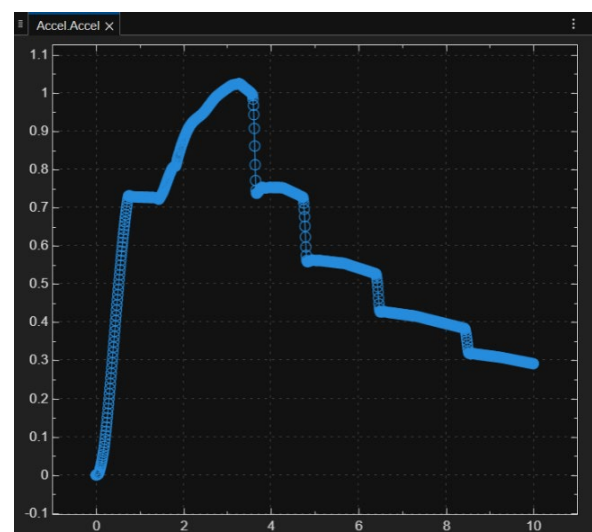
[Figure 15] Simulink layout

5.2.4 Road & Acceleration input

Road disturbance and acceleration inputs and aerodynamic forces were generated using a signal editor block. Front and rear downforce values were extracted from VI-Grade and applied as speed-dependant vertical loads on the sprung mass. All input data points were imported from MATLAB and VI-Grade. Road input is applied to front unsprung mass, with a delay of 1s to the rear, while longitudinal acceleration acts on the sprung mass and generates pitch response. Below shows the bump and acceleration input graphs.



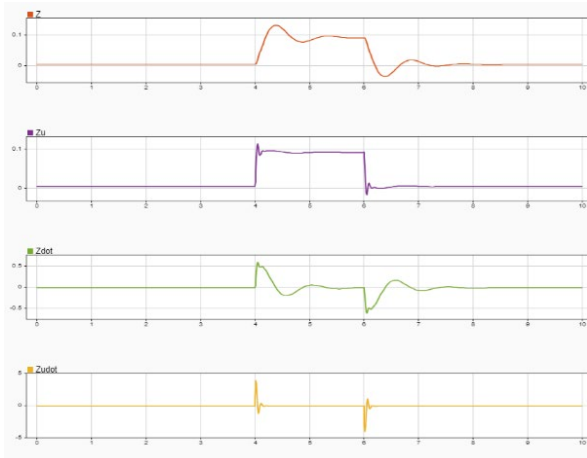
[Figure 16] Simulink bump input



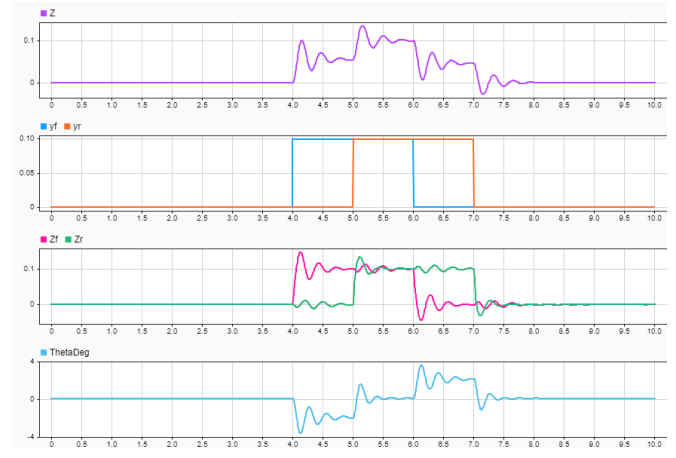
[Figure 17] Acceleration input from VI-Grade

5.3 Bump Results

When a vehicle passes over a bump, the tyres rise rapidly, causing an initial rearward pitch, followed by a forward pitch recovery (Deac et al., 2019). Both front and rear heave increase briefly as the suspension absorbs the input. The 2DOF model output in figure 18 shows the single axle response, while the 4DOF model in Figure 19 captures the full front-rear interaction. The oscillations represent spring-damper energy dissipation. This behaviour aligns with Gillespie's observation that 'high jounce damping increases transmitted forces', whereas rebound damping is needed to dissipate stored spring energy (Gillespie, 1994).



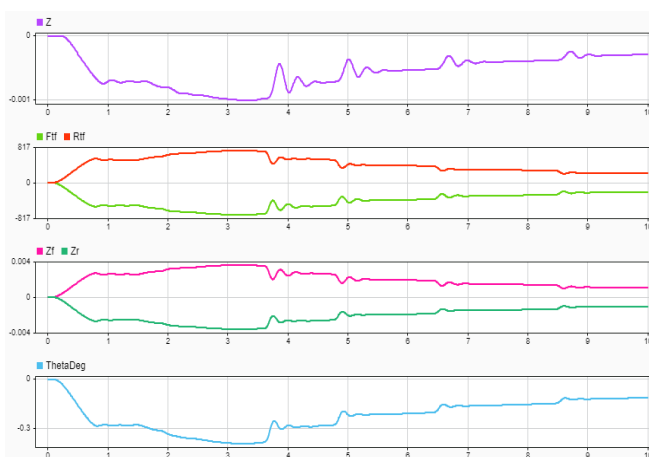
[Figure 18] 2DOF bump response



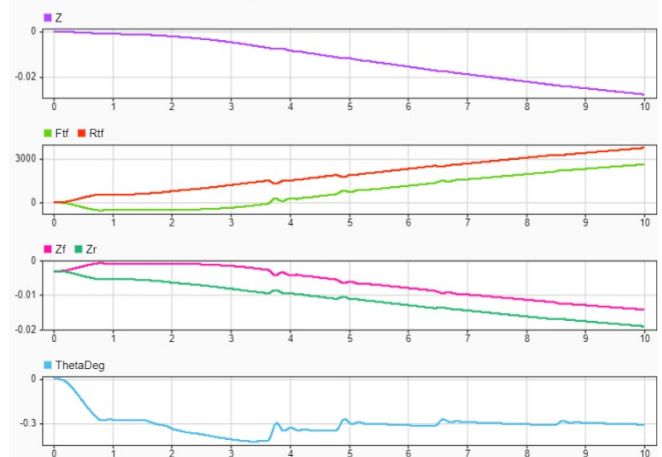
[Figure 19] 4DOF bump response

5.4 Acceleration Outputs

Figure 20 shows the baseline acceleration response (from figure 17) where the vehicle pitches nose-up and load transfers rearward, increasing rear-tyre force while reducing front force. In figure 21, the addition of downforce alters this behaviour: heave reduces steadily, pitch remains more controlled, and tyre forces shift less dramatically. This aerodynamic load counteracts squat, maintaining a flatter attitude and reducing the extent of front unloading during acceleration.

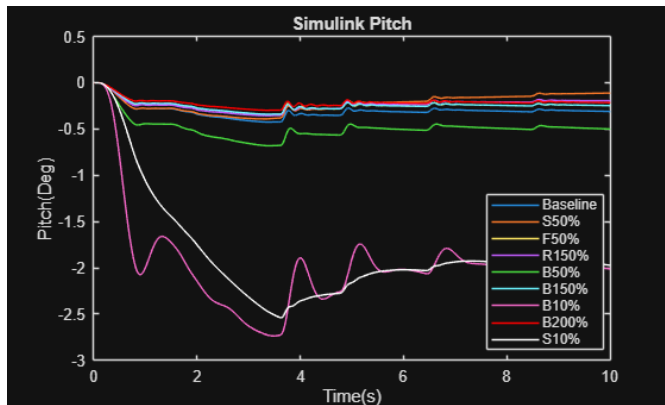


[Figure 20] Acceleration response without aero

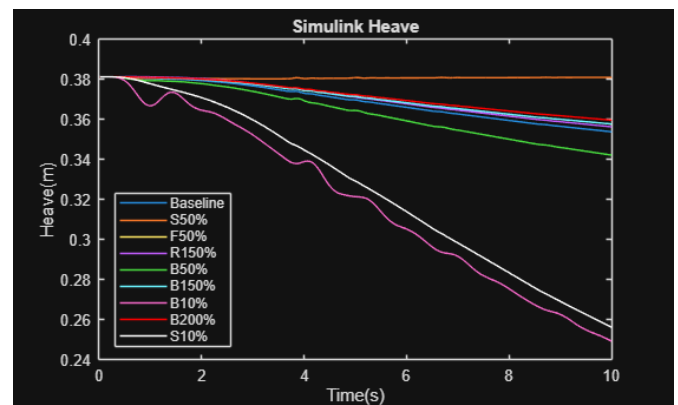


[Figure 21] Acceleration response with aero

Figures 22 and 23 show the corresponding pitch and heave outputs from Simulink, which will be compared to the VI-Grade results in the next section. A selection of suspension setups with the most noticeable influence on vertical and pitch behaviour were also simulated. Reducing both spring and damper rates by 10% produced larger and more prolonged pitch oscillations, as seen by the increased amplitude and slower settling time in the pitch plot. This is consistent with reduced stiffness and lower energy dissipation. Similarly, reducing front spring stiffness resulted in a clear increase in front heave, as shown by the upwards shift of heave curves, which reflects the reduced resistance to vertical motion under pitch-inducing acceleration.



[Figure 22] Simulink Pitch responses with various setups

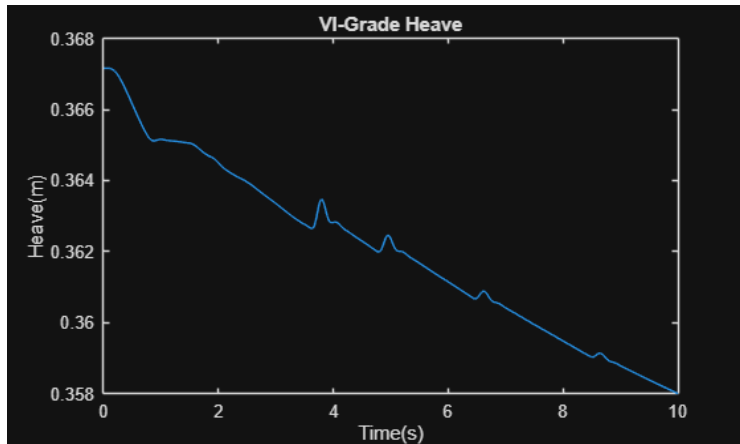


[Figure 23] Simulink Heave responses with various setups

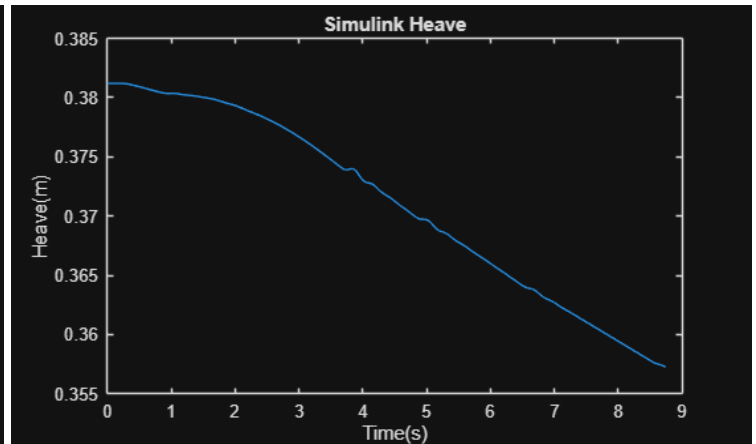
6 Comparison & Discussion

6.1 Data comparison

This section shows a comparison of data extracted from VI-Grade against a created Simulink model.

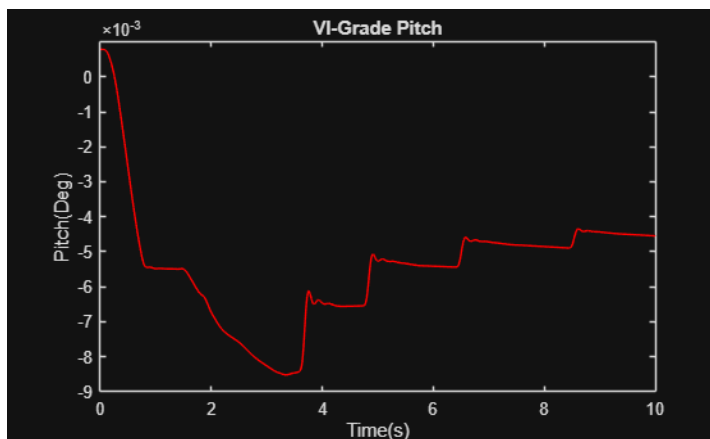


[Figure 24] VI-Grade Heave

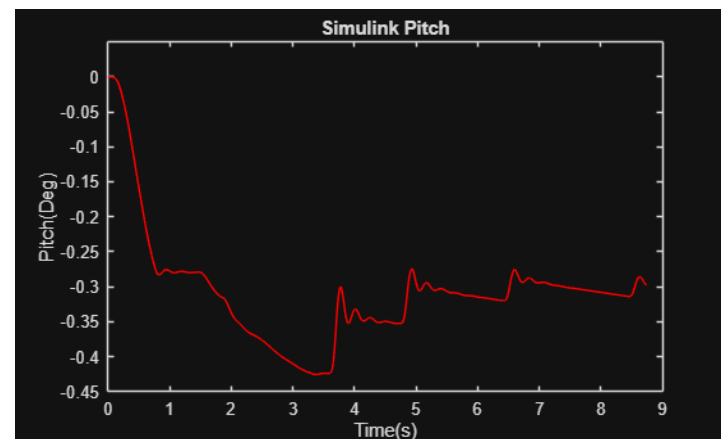


[Figure 25] Simulink Heave

The graphs above show a comparison of VI-Grade and Simulink model heave responses. Both show a common trend of initial height reduction followed by a gradual recovery as the suspension settles. The magnitude of the heave reduction in Simulink is larger, indicating that the linear spring-damper model produces a stronger vertical response than the more detailed VI-Grade model. Both signals display a repeating damping effect throughout the 4-9s region, although the Simulink oscillations decay slightly faster. The mismatch in initial heave is expected because Simulink begins from zero initial conditions; however, the static ride height of 0.3812m was added to the Simulink output to counteract this and improve accuracy. The remaining time difference therefore reflects model simplifications rather than an error in implementation.

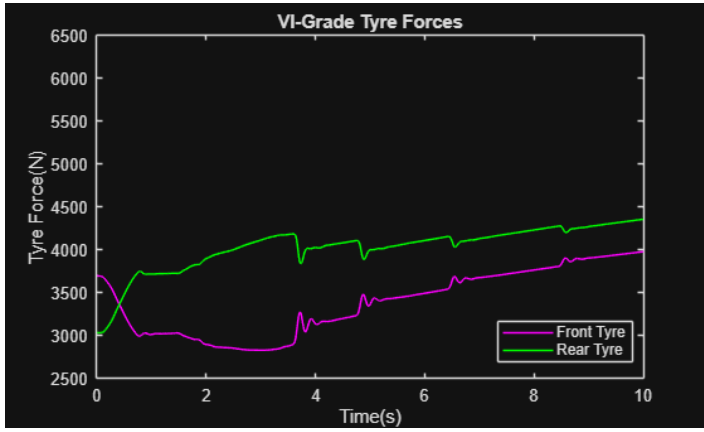


[Figure 26] VI-Grade Pitch

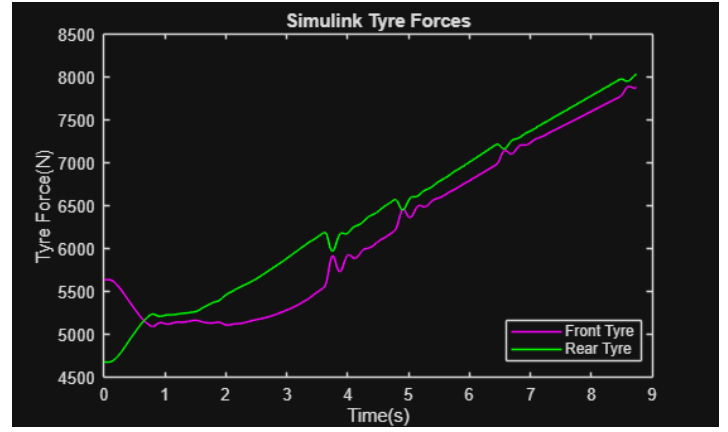


[Figure 27] Simulink Pitch

Figure 26 and 27 show the comparison of pitch response to longitudinal acceleration. The overall trend of the graphs matches quite well, however Simulink scale is off. When scaled appropriately (Simulink $\times 0.02$), both models exhibit an almost identical pitch trend: an initial rapid negative rotation due to longitudinal weight transfer, followed by several declining oscillations. Both datasets show the same sequence of local minimum and maximum, indicating that the half-car Simulink model captures the dominant pitch dynamics correctly. The remaining differences are likely due to unmodelled effects such as chassis compliance, spring preload, and the coupled roll-pitch behaviour that would be captured in a 7-dof model, but is absent in the simplified half-car model.



[Figure 28] VI-Grade Tyre Forces



[Figure 29] Simulink Forces

To compare tyre forces directly, the static front-rear weight distribution (Table 9) was added to the Simulink output so both models began with comparable preload. After correction, Simulink and VI-Grade show similar trends though Simulink consistently overpredicts load transfer. This is likely due to simplified vertical dynamics, missing suspension non-linearities and tyre relaxation. Before preload correction, Simulink tyre forces started at zero because the model lacked initial static deflection and reported only dynamic, not total tyre load.

[Table 9] Load Distribution

Quantity	Front Axle	Rear Axle	Units
CoG distance from axle	1.23	1.4829	m
Load distribution	0.546611	0.453389	-
Static Mass	575.03	476.97	kg
Static Load	5641.09	4679.03	N
Total vehicle mass	1052	-	kg

Overall, the simplified Simulink model successfully reproduces the main heave and pitch behaviours seen in VI-Grade, although differences in amplitude and tyre force prediction highlights limitations of linear modelling and zero-preload initial conditions.

6.2 Model Limitations

The main limitations of the 4DOF model arise from simplified tyre modelling (Romano, 2025) and zero static initial conditions. The tyres are represented as linear springs, neglecting tyre damping, relaxation behaviour and nonlinear stiffness, which reduces accuracy under transient load transfer. The half-car structure also excludes roll and lateral weight transfer, meaning it cannot fully replicate the dynamics observed in full-vehicle models, as highlighted in literature. VI-Grade itself contains modelling assumptions but remains more complete due to the detailed tyre, suspension and chassis representation.

7 Conclusion

The comparison between the 4-DOF Simulink model and the VI-Grade simulation highlights key relationships between spring stiffness, damping, pitch motion and vertical load transfer. Simulink provides useful insight into how individual parameter changes influence vehicle behaviour, while VI-Grade captures the more complex coupled dynamics excluded from the simplified model. The best-performing lap occurred with either a 50% reduction in front spring stiffness or a 50% increase in rear stiffness from the baseline (202.3kN/m front, 100.2kN/m rear). Both cases indicate that the car performs best with a more balanced pitch response and reduced front-to-rear load transfer.

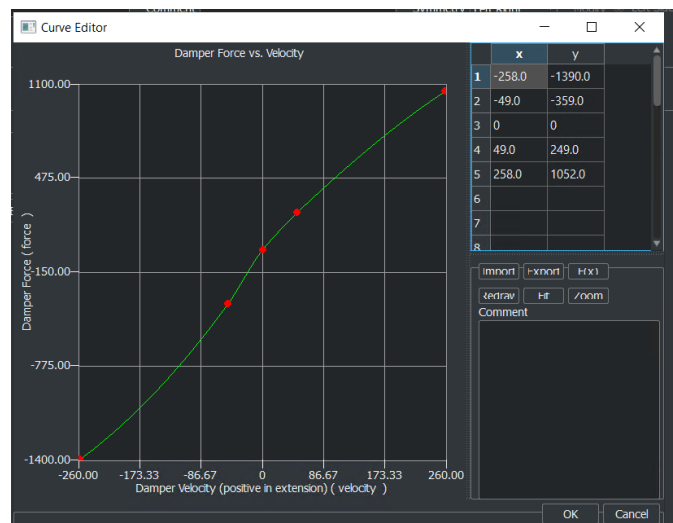
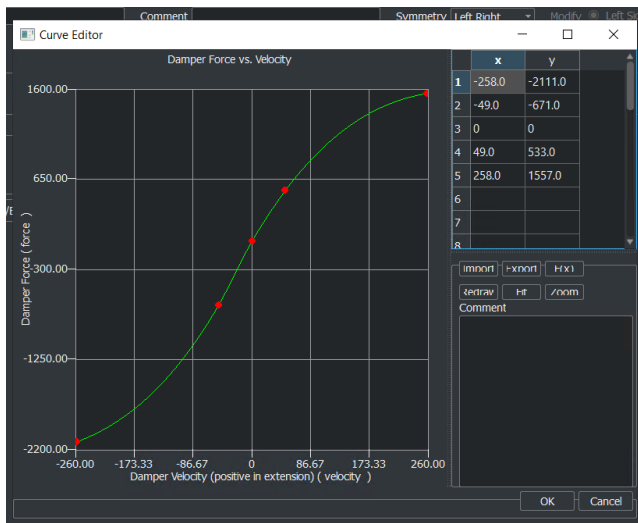
However, the 4-DOF model is limited by linear tyre assumptions, zero-preload initial conditions and the absence of roll effects. Future work should incorporate a 7DOF model (Palli, Sharma and Rao, 2017) to capture nonlinear and combined-load vehicle behaviour.

8 References

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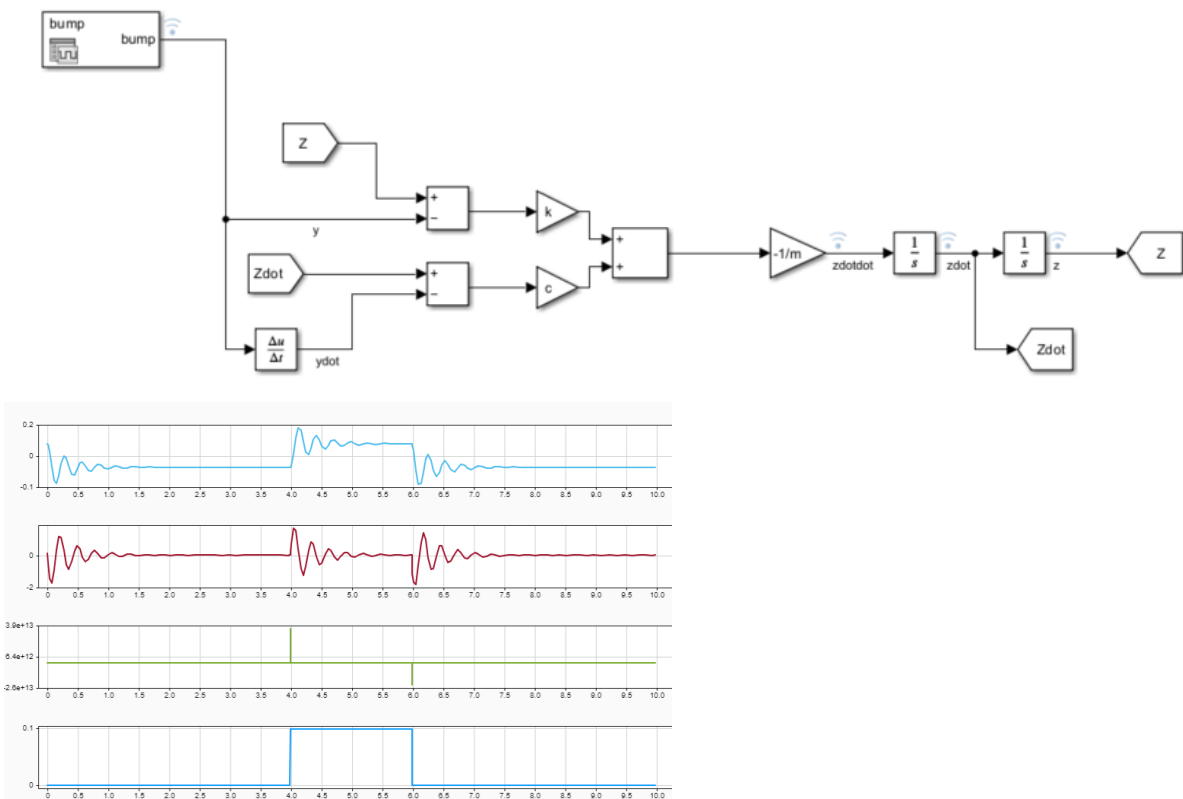
Appendix A: VI-grade

Baseline Damper curves:

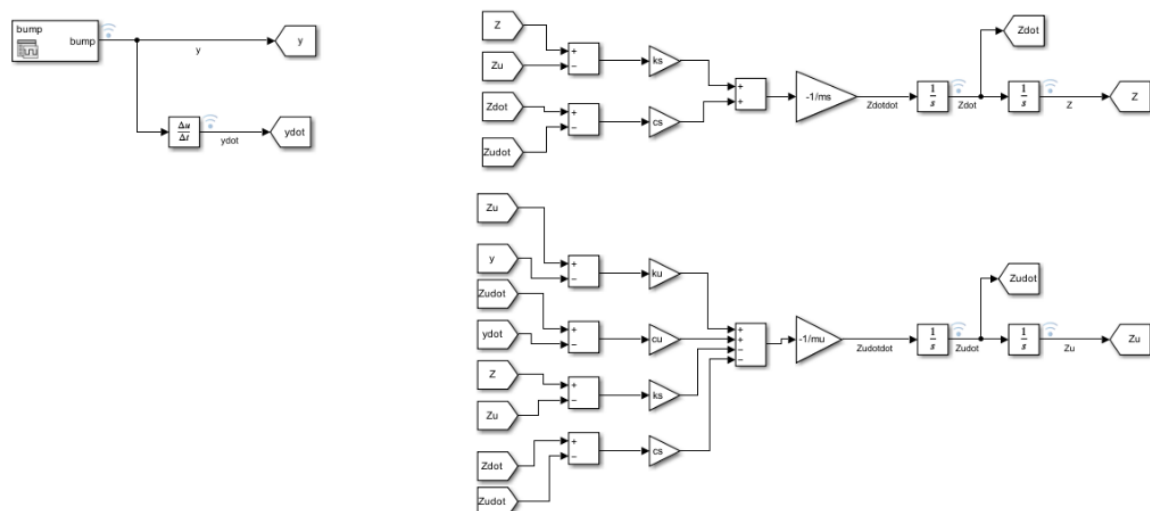


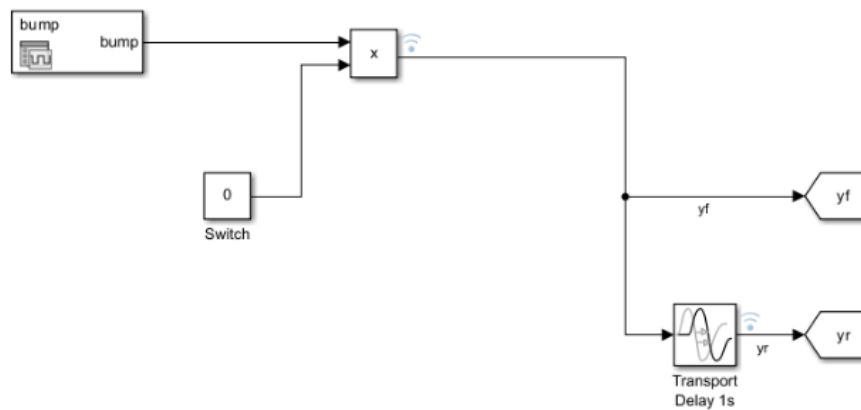
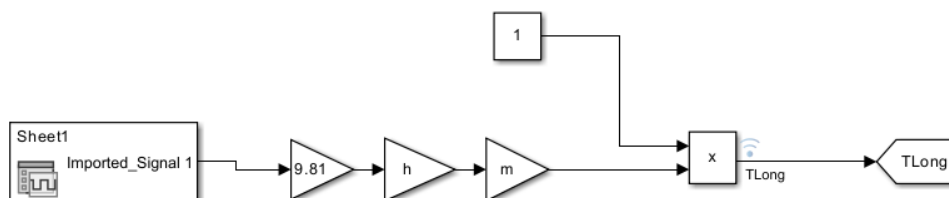
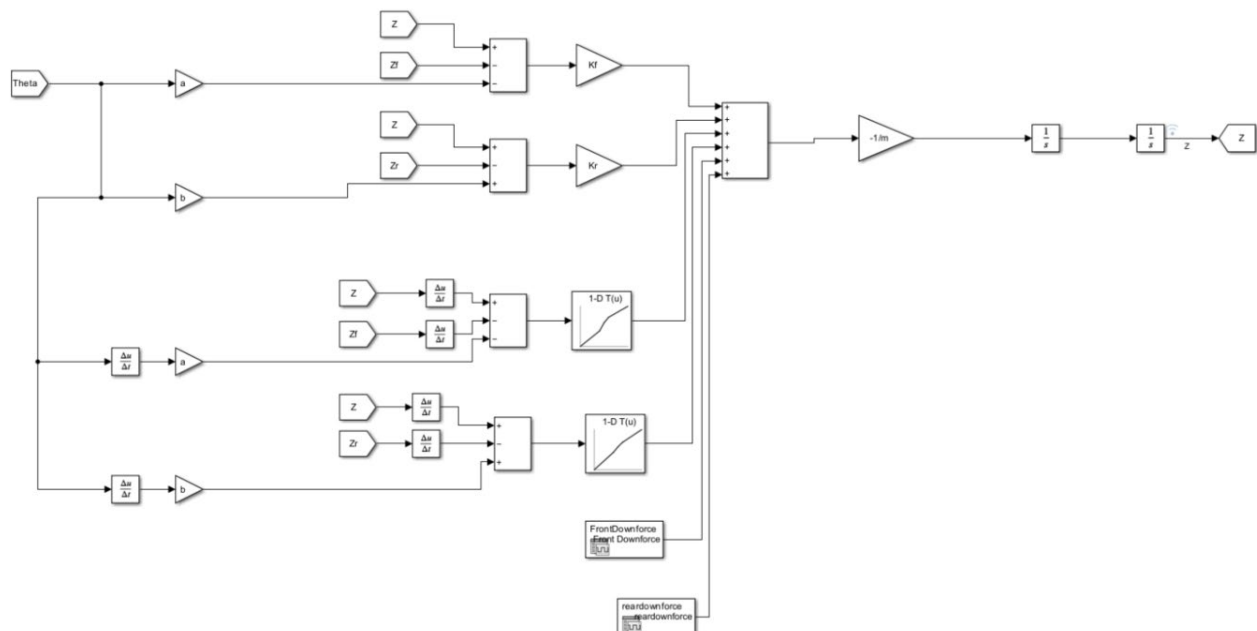
Appendix B: Simulink

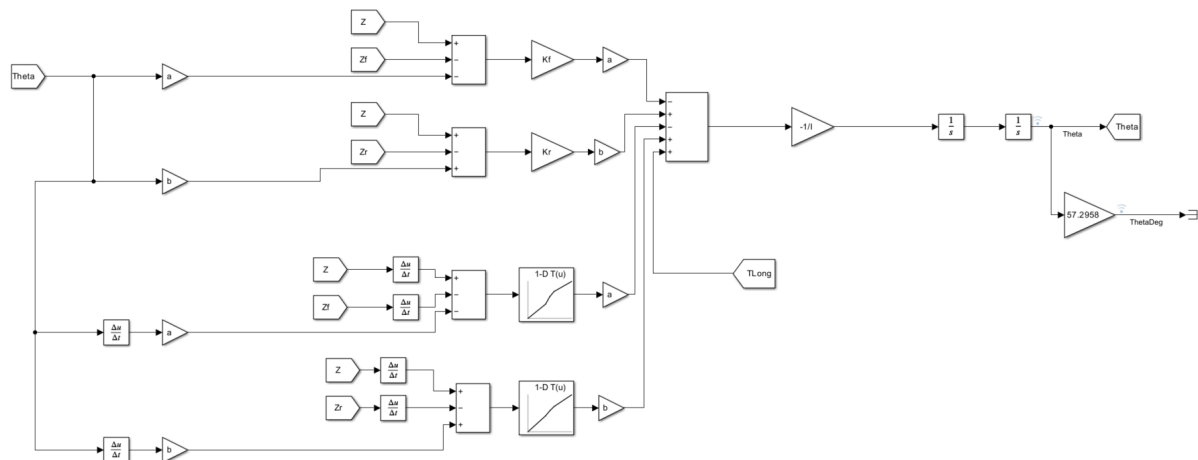
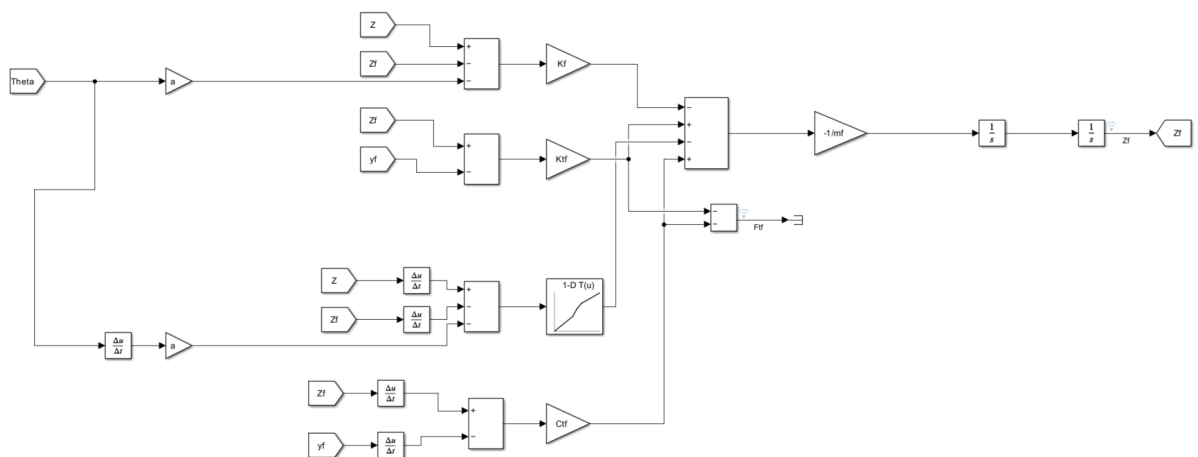
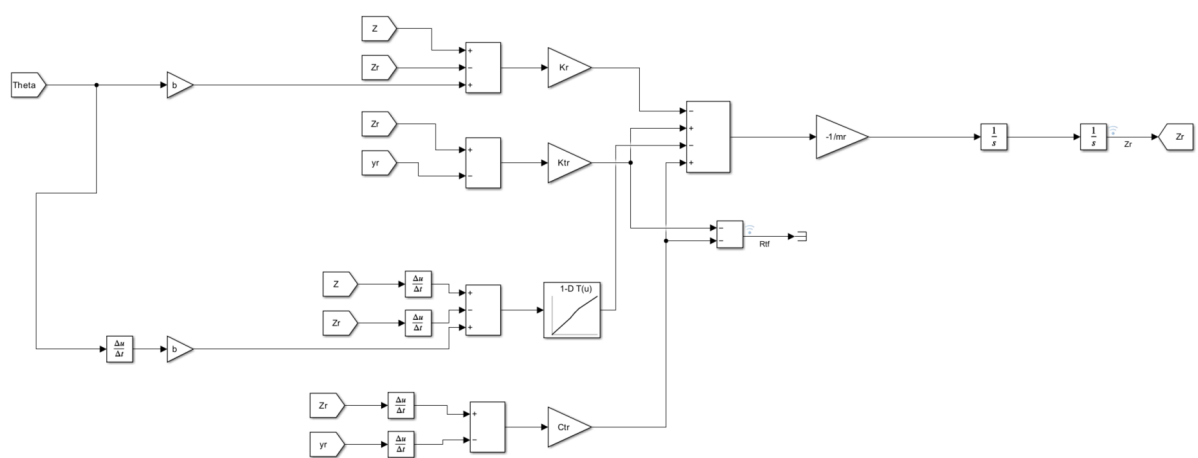
1DOF: model and output



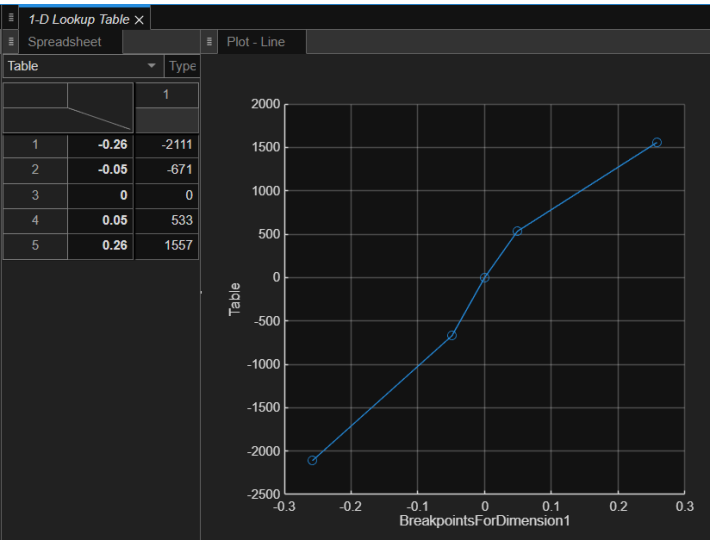
2DOF Model:



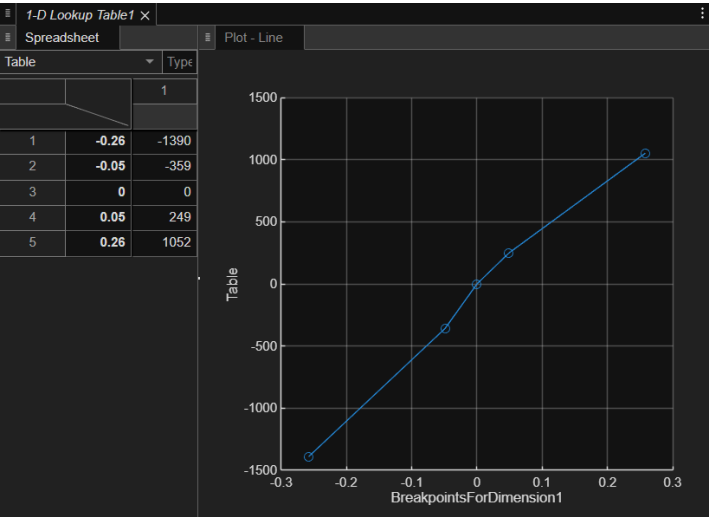
Bump/Road input:Acceleration:Sprung mass:

Inertia:Unsprung mass front:Unsprung mass rear:

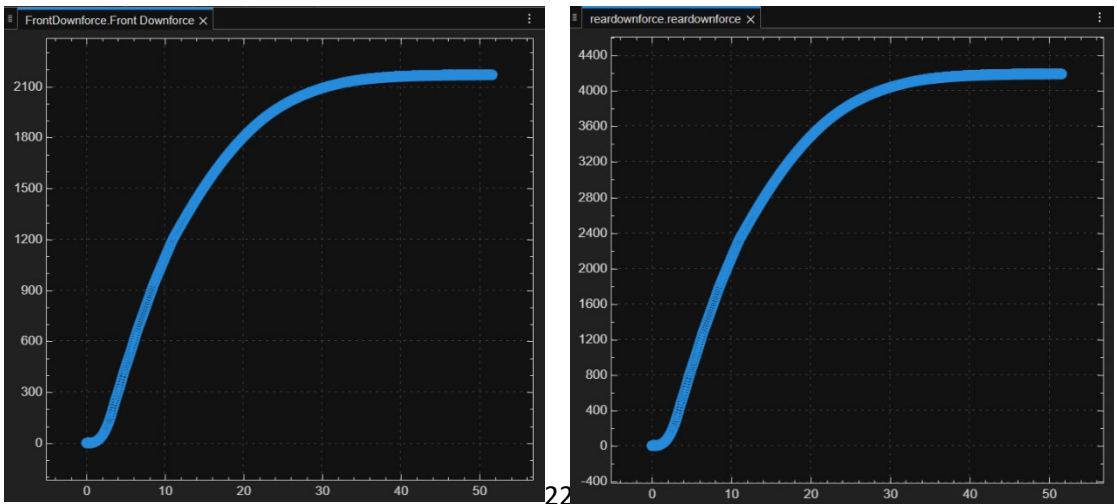
Front variable damping: x-axis velocity (divided by 1000 from VI-Grade for mm/s to m/s)



Rear variable damping: x-axis velocity (divided by 1000 from VI-Grade for mm/s to m/s)



Front and rear downforce curves from VI-Grade (baseline setup)



Appendix C: MATLAB Code

Straight line Pembrey:

```
% change in springs
% Setups 1-8: 1=baseline 2=10% springs

h=plot(Time(1:1001,1:8), Pitch(1:1001,1:8));
h(8).Color = 'r';
xlabel('Time(s)')
ylabel('Pitch(Deg)')
title('How Spring changes affect Pitch')
legend(['baseline', "10%", "50%", "150%", "F 50%", "R 50%", "F 150%", "R150"], "Position", [0.7513 0.1919 0.2145, 0.4288])

Maxpitch = max(Pitch(:,1:8))
Minpitch = min(Pitch(:,1:8))

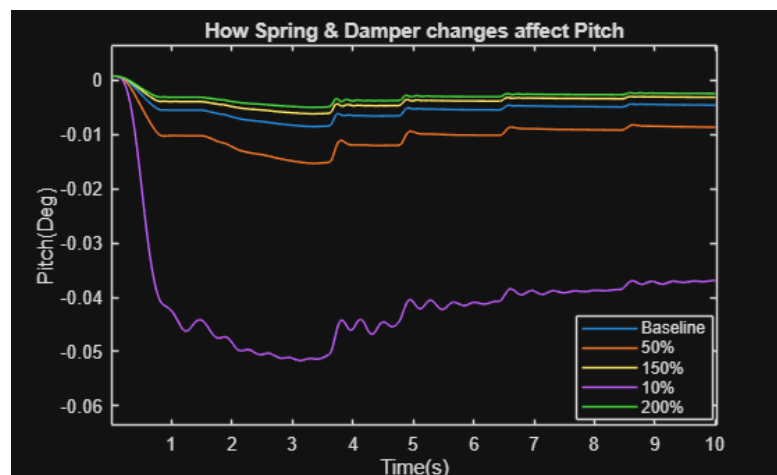
plot(Time(1:1001,1:8),Heave(1:1001,1:8))
xlabel('Time')
ylabel('Heave')
title('How Spring changes affect Heave')
legend('baseline','10%','50%', '150%', 'F 50%', 'R 50%', 'F 150%', 'R 150%')
```

```
%pitch changes
plot(Time(1:1001,[1 2 15]), Pitch(1:1001,[1 2 15]));
xlabel('Time(s)')
ylabel('Pitch(Deg)')
title('Spring vs Spring & Damper change')
legend('Baseline', '10% Spring', '10% Spring & Damper')

%%difference between baseline and 10% spring rates
plot(Time(1:1001,1:2), TyreF(1:1001,1:2))
title('Tyre Force changes')
xlabel('Time(s)')
ylabel('Tyre Force(N)')
hold on
plot(Time(1:1001,1:2), TyreR(1:1001,1:2))
legend(['Ftf baseline', "Ftf 10%", "Rtf baseline", "Rtf 10%"], "Position", [0.6813 0.1625 0.2054, 0.1779])
hold off
```

```
%both damper and springs

plot(Time(1:1001,[1,13,14,15,16]), Pitch(1:1001,[1,13,14,15,16]))
legend
xlabel('Time(s)')
ylabel('Pitch(Deg)')
title('How Spring & Damper changes affect Pitch')
legend(["Baseline", "50%", "150%", "10%", "200%"], "Position", [0.7399 0.1495 0.1468, 0.1779])
```



VI-Grade & Simulink comparison:

```
%comparison
plot(SimTime(1:1145,1), SimHeave(1:1145,1))
xlabel('Time(s)')
ylabel('Heave(m)')
title('Simulink Heave')

plot(Time(1:1001, 1), Heave(1:1001, 1))
xlabel('Time(s)')
ylabel('Heave(m)')
title('VI-Grade Heave')

plot(SimTime(1:1145,1), SimPitch(1:1145,1),'r')
xlabel('Time(s)')
ylabel('Pitch(Deg)')
title('Simulink Pitch')

plot(Time(1:1001, 1), Pitch(1:1001, 1),'r')
xlabel('Time(s)')
ylabel('Pitch(Deg)')
title('VI-Grade Pitch')
```

```
plot(SimTime(1:1145,1), SimTyreF(1:1145,1),'m')
xlabel('Time(s)')
ylabel('Tyre Force(N)')
title('Simulink Tyre Forces')
hold on
plot(SimTime(1:1145,1), SimTyreR(1:1145,1),'g')
legend(["Front Tyre", "Rear Tyre"], "Position", [0.6992 0.1572 0.1911, 0.0941])
hold off

plot(Time(1:1001, 1), TyreF(1:1001, 1),'m')
xlabel('Time(s)')
ylabel('Tyre Force(N)')
title('VI-Grade Tyre Forces')
hold on
plot(Time(1:1001,1), TyreR(1:1001,1),'g')
legend(["Front Tyre", "Rear Tyre"], "Position", [0.6992 0.1572 0.1911, 0.0941])
ylim([2500 6500])
hold off
```

Pembrey data:

```

files_Pembrey = ["PemBaseline.csv" , "Pem10%.csv", "PemSetup1.csv", ...
    "PemSetup2.csv" , "PemSetup3.csv" , "PemSetup4.csv" , "PemSetup5.csv" , ...
    "PemSetup6.csv" , "PemSetup7.csv" , "PemSetup8.csv" , ...
    "PemSetup9.csv" , "PemSetup10.csv", "PemSetup11.csv" , ...
    "PemSetup12.csv", "PemSetup13.csv", "PemSetup14.csv"];

num_files = numel (files_Pembrey);

for cnt = 1:num_files
    Pem = readmatrix(files_Pembrey(cnt));

    Time(1:length(Pem),cnt) = Pem(:,1);
    Velocity(1:length(Pem),cnt) = Pem(:,763);
    Accel_long(1:length(Pem),cnt) = (Pem(:,82)).*9.81;
    Accel_lat(1:length(Pem),cnt) = (Pem(:,83)).*9.81;
    Pitch(1:length(Pem),cnt) = Pem(:,91);
    Roll(1:length(Pem),cnt) = Pem(:,93);
    TyreF(1:length(Pem),cnt) = Pem(:,679);
    TyreR(1:length(Pem),cnt) = Pem(:,680);
end

Time(Time == 0) = NaN;
Velocity(Velocity == 0) = NaN;

```

```

%damper changes
plot(Time(:,9:12), Pitch(:,9:12))
legend
xlabel('Time(s)')
ylabel('Pitch(deg)')
title('How Damper changes affect pitch')
xlim([20.66 24.98])
legend('50%', '80%', '120%', '150%')

plot(Time(:,9:12), TyreF(:,9:12))
xlabel('Time(s)')
ylabel('Tyre Front(N)')
title('How Damper changes affect Ftf')
xlim([20.66 24.98])
legend('50%', '80%', '120%', '150%')

```

```

%roll
plot(Time(:,13:16), Roll(:,13:16))
legend
xlabel('Time(s)')
ylabel('Roll(Deg)')
title('How Spring&Damper changes affect Roll')
legend(["50%", "150%", "10%", "200%"], "Position", [0.7399 0.1744 0.1468, 0.1779])

```