



University of Wales Trinity Saint David

Wales Institute of Science & Art

Vehicle Dynamics Analysis (Simulink / VI-Grade)

Advanced Motorsport Dynamics

ACHA6011

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Contents

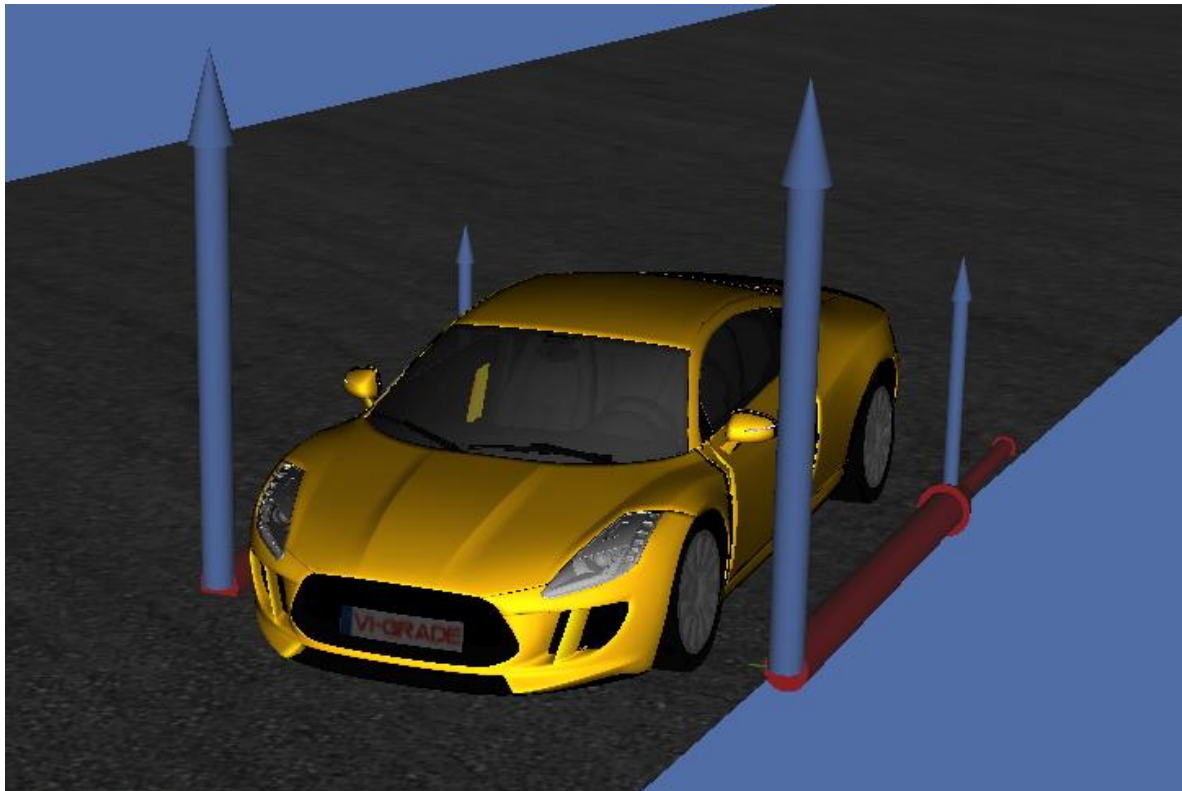
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Word Count: 3080

1 Introduction

Computer modelling tools have become incredibly useful to evaluate vehicle dynamics, handling and overall performance of automotive and motorsport-based applications. Through this investigation I aim to assess how suspension spring and damping parameters influence vehicle ride and handling performance through both multi-body simulation (VI-Grade CarRealTime) and analytical modelling (Simulink). This report will begin by analysing the response of the VI-Grade Sports Car to different setups during a range of standard dynamic manoeuvres and track performance. The next focus of the report will be deriving equations of motion to develop a 4 degree of freedom (4DOF) spring mass damper Simulink model. Various bump inputs will be used to simulate ride performance before developing the model to include acceleration and aerodynamic force inputs. The report will conclude by comparing VI-Grade manoeuvre and lap simulation results to my Simulink 4DOF model and a developed 7 degree of freedom model.

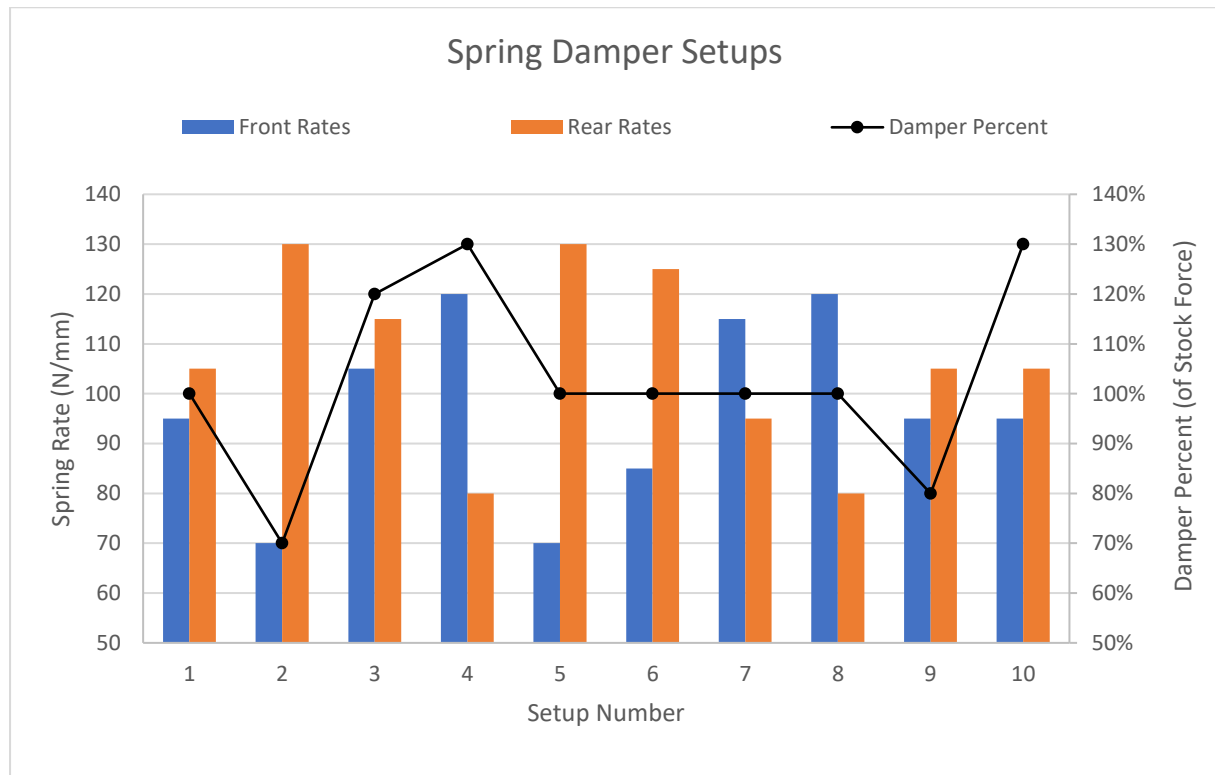


(Figure 1: VI-Grade Sports Car Model)

2 Vehicle Modelling using VI-Grade CarRealTime

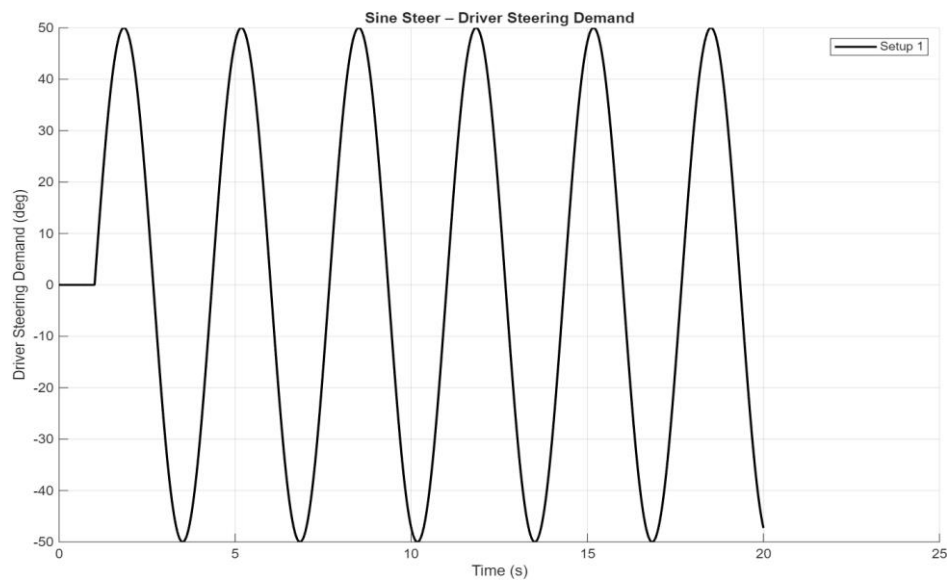
2.1 Standard Dynamic Manoeuvres

I will be testing the VI-Grade sports car through a sine steer, constant radius cornering and acceleration test. The stock car (Setup 1 below) as well as 9 other setups were tested to evaluate spring-rate, damper-rate and combined spring and damper alterations.



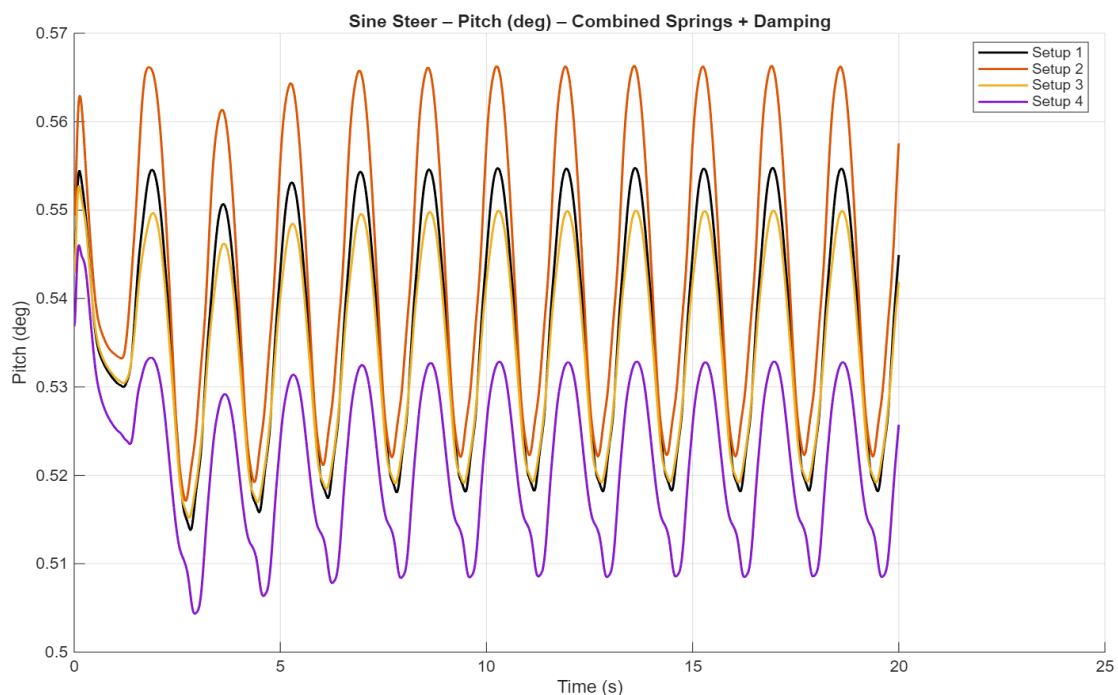
(Graph 1: VI-Grade Spring Damper Setups – References Appendix A)

2.1.1 Sine Steer Test



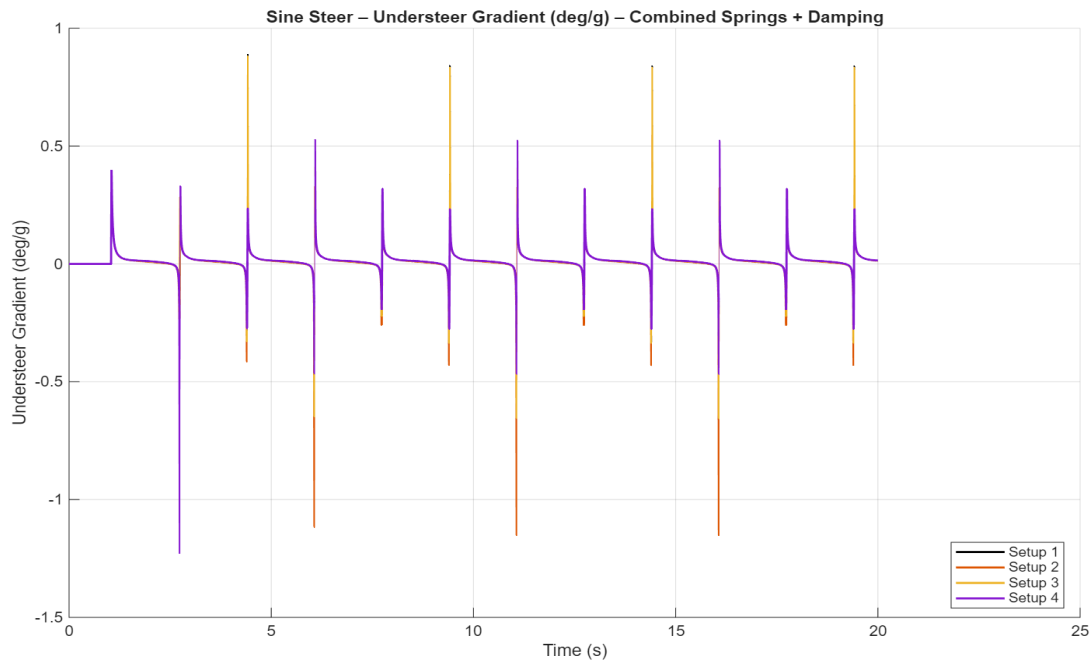
(Graph 2: Driver Steering Demand vs Test Time)

Graph 2 details the sine steer test steering movement, this will be useful to evaluate the transient capabilities and weight transfer effects with constant steering movements. Setup values are detailed in Appendix B. Other layouts of this test are used within industry (Euro NCAP 2011).



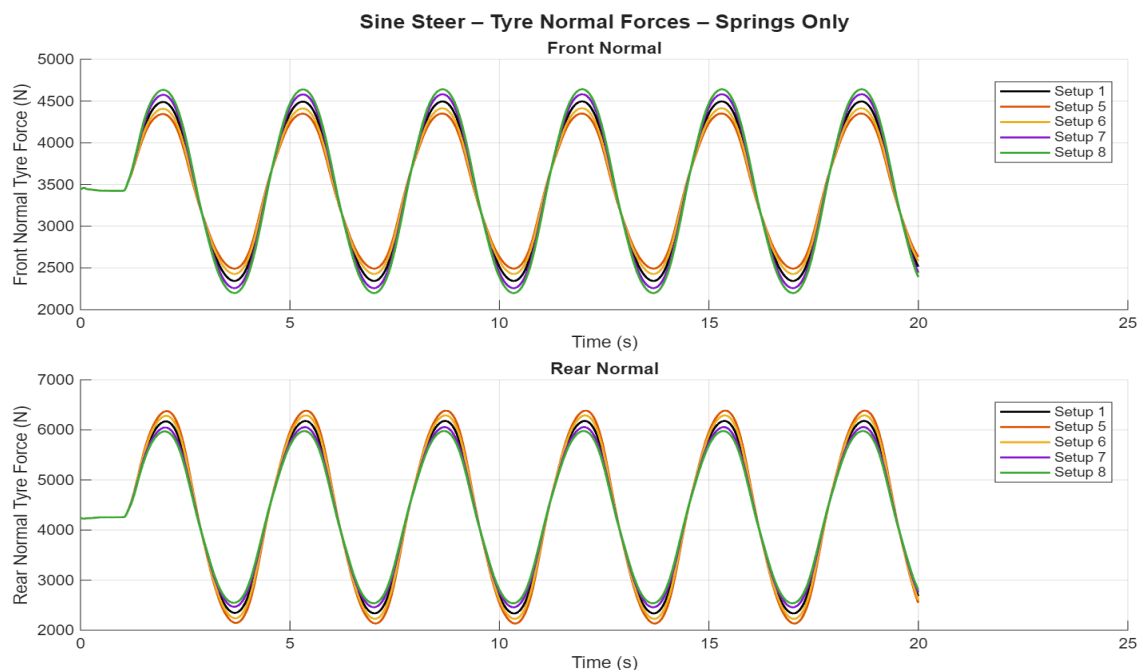
(Graph 3: Combined Setup Pitch vs Test Time)

Graph 3 presents the result for pitch, showing the front to rear balance of the car changing through different combined spring and damper setups. Setup 2 featured much softer front springs relative to the rear, and this shows to increase forward pitch over stock (Setup 1). This logic is also proven through Setup 4 providing more rear pitch for a softer rear rate.



(Graph 4: Combined Setup Understeer Gradient vs Test Time)

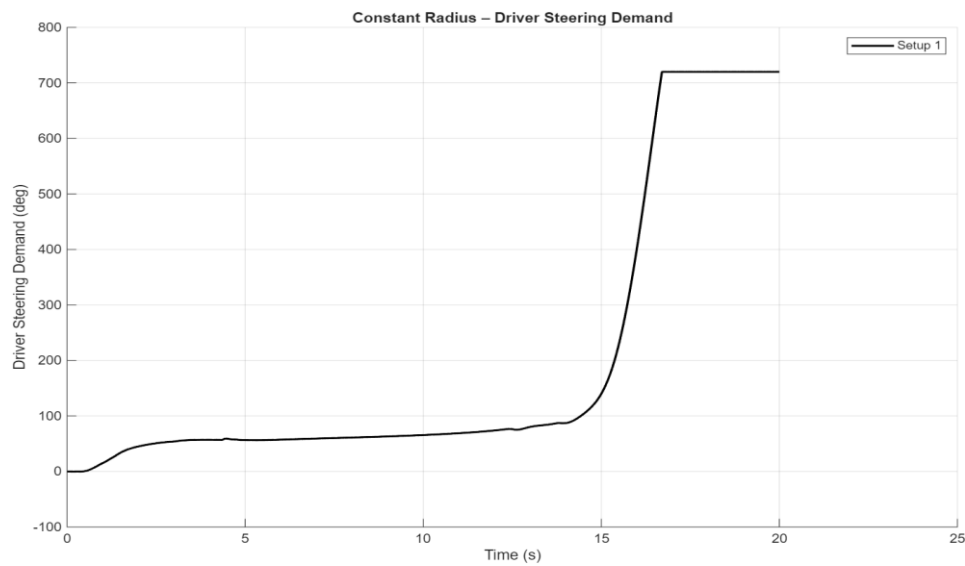
The results of pitch motion under transient loading can also be seen in graph 4. Other than the slight initial oversteer spike (presented as negative understeer) for Setup 4, it appears to have smaller fluctuation between understeer and oversteer suggesting a more stable front to rear balance. The stiffest rear springs (setup 2) showed large oversteer snaps with minimal understeer. Setup 1 and setup 3, which has similar spring rates and balance to baseline, tended to understeer (Segers, J. 2014).



(Graph 5: Spring Rate Setup Tyre Normal Forces vs Test Time)

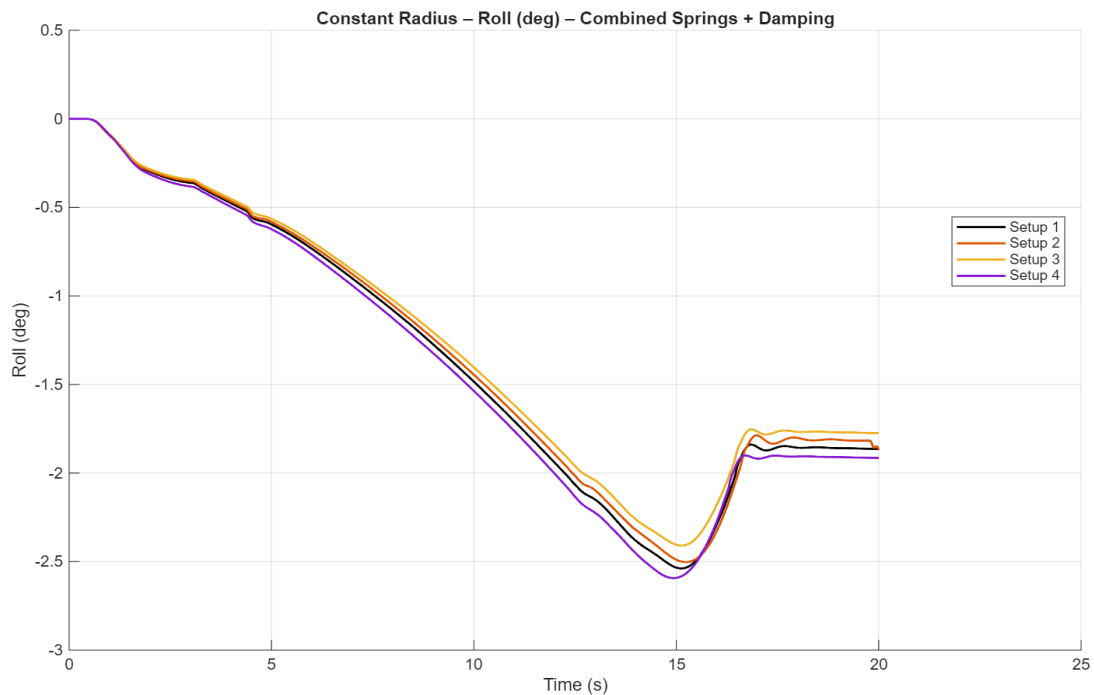
Graph 5 shows how the spring rate changes have affected the overall tyre forces, confirming that the softer rear (Setup 8) increased front loading and reduced rear loading, which was a better balance for this test. The damper only setups (9-10) provided unnoticeable differences to understeer and tyre loads (see Appendix C & D).

2.1.2 Constant Radius Cornering Test



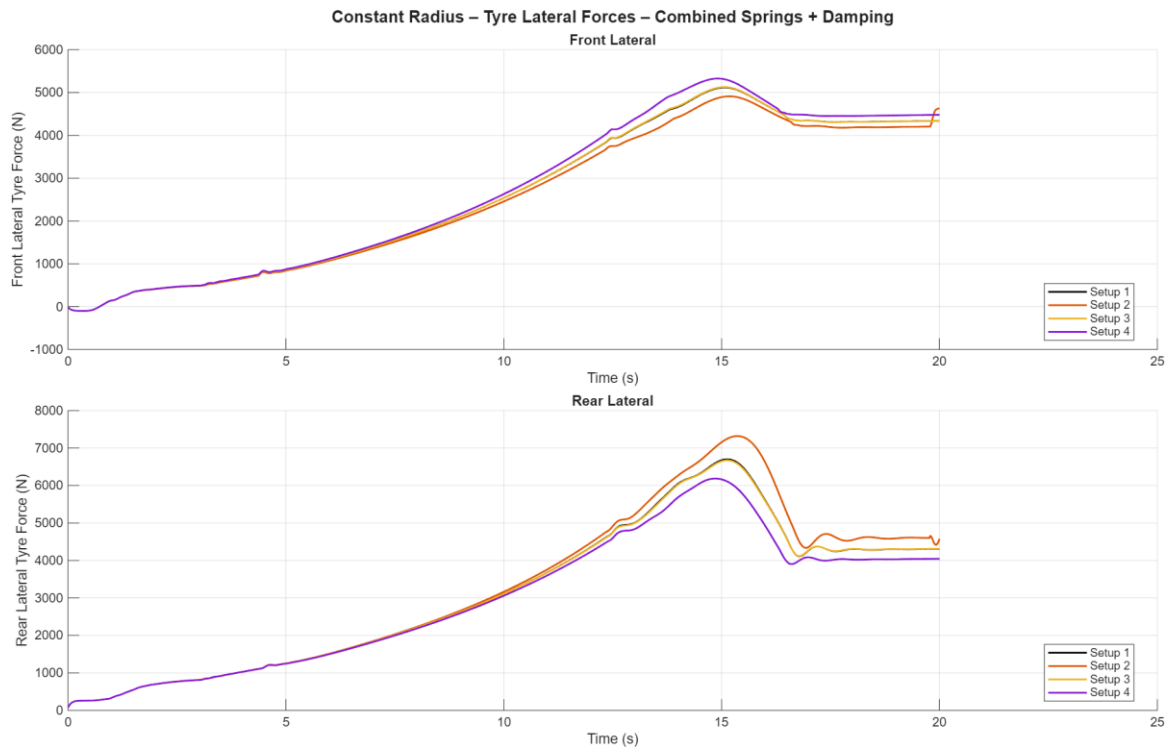
(Graph 6: Driver Steering Demand vs Test Time)

Graph 6 explains the steering input for the constant radius cornering test (Huang, H.-H. and Tsai, M.-J. 2019). This is a steady state test, and the car turns only left for the duration. The car was setup to accelerate gradually to transfer weight longitudinally (see setup values in Appendix E). Past 15 seconds there is an understeer moment to find the peak force of each setup before the tyres slip.



(Graph 7: Combined Setup Roll vs Test Time)

Graph 7 details the resistance against roll for each of the combined setup changes. Setup 4 provides the most roll peaking at over 2.5 degrees, shown on the graph negative, as left-hand turning produces negative values in VI-Grade. Setup 3 is only slightly stiffer than the baseline but along with a 20% damping force increase, provides the least roll and a more controlled transition to understeer as the roll migrates less in the last 5 seconds of the test.



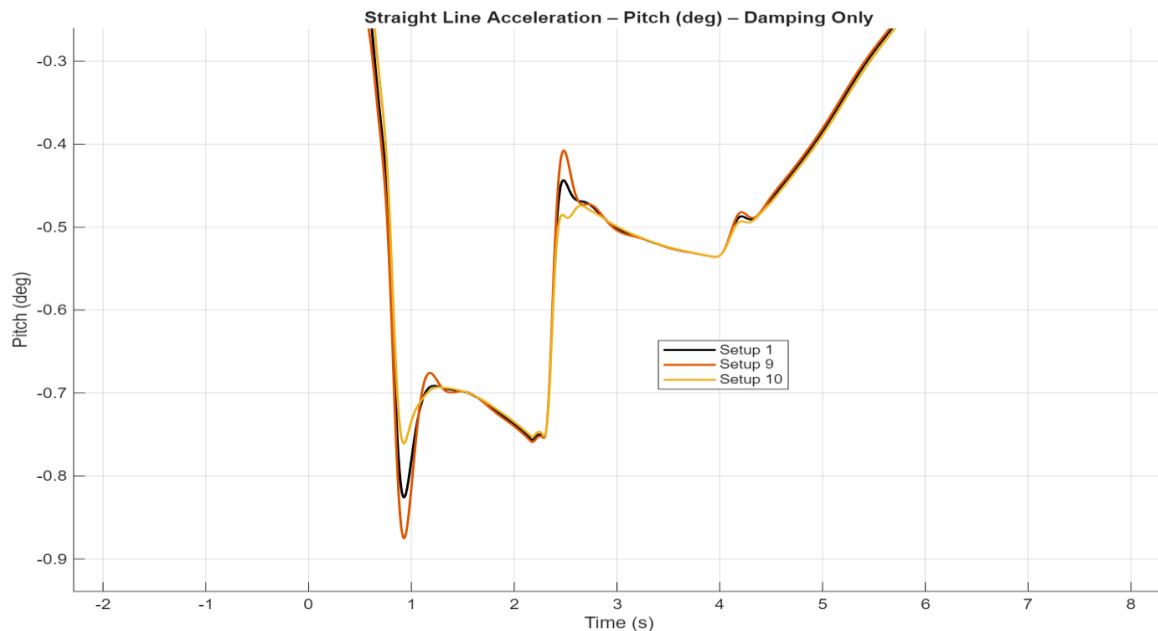
(Graph 8: Combined Setup Lateral Tyre Forces vs Test Time)

The results shown in graph 8 details the limit of the tyres for each setup. Interestingly there wasn't complete correlation between resistance to roll and lateral tyre forces, as setup 3 was more resistive to roll than setup 1 but provided almost identical tyre forces. This is a good performance characteristic as the extra spring stiffness and damper force increase has reduced the roll but retained lateral loading on the front and rear (Guiggiani, M. 2018).

Once again, damper rate changes alone made small differences, so the assumption can be made that combined adjustment is the most effective change for constant radius cornering (see Appendix F & G). (Lambert, A. 2021).

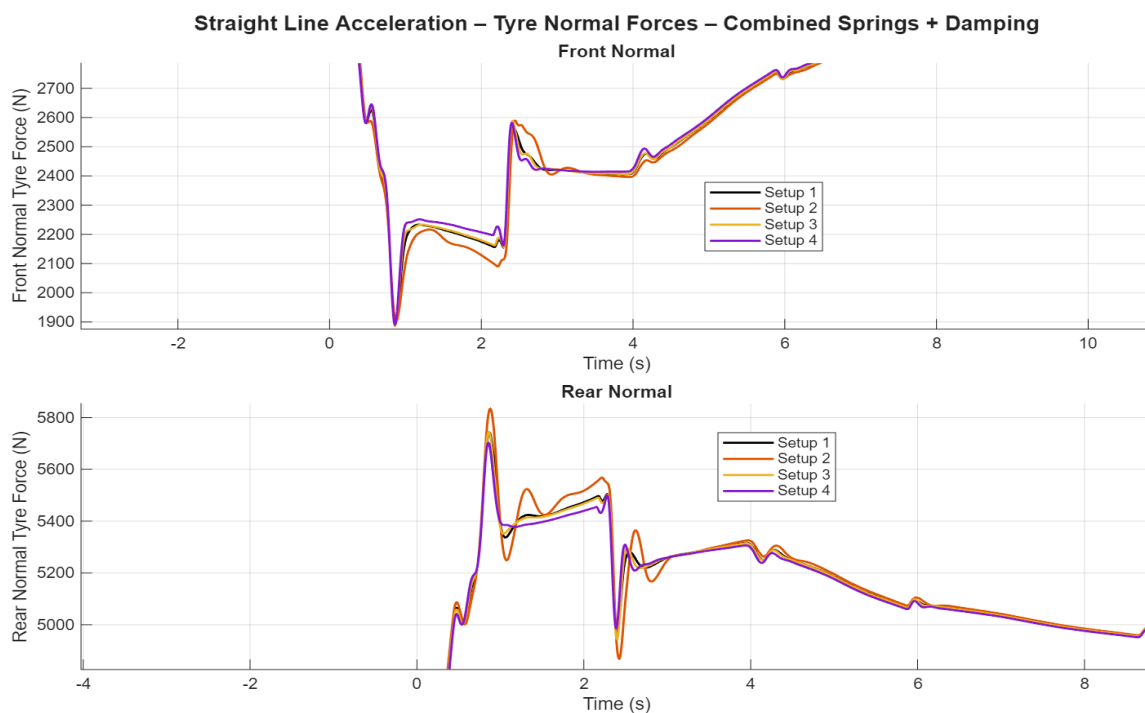
2.1.3 Straight Line Acceleration Test

This acceleration test was carried out in a straight line from a standing start (see Appendix H). I have zoomed in each graph below to capture the hard acceleration during the first 6 seconds, after this the car becomes steadier and there were less obvious differences.



(Graph 9: Damping Setup Pitch vs Test Time)

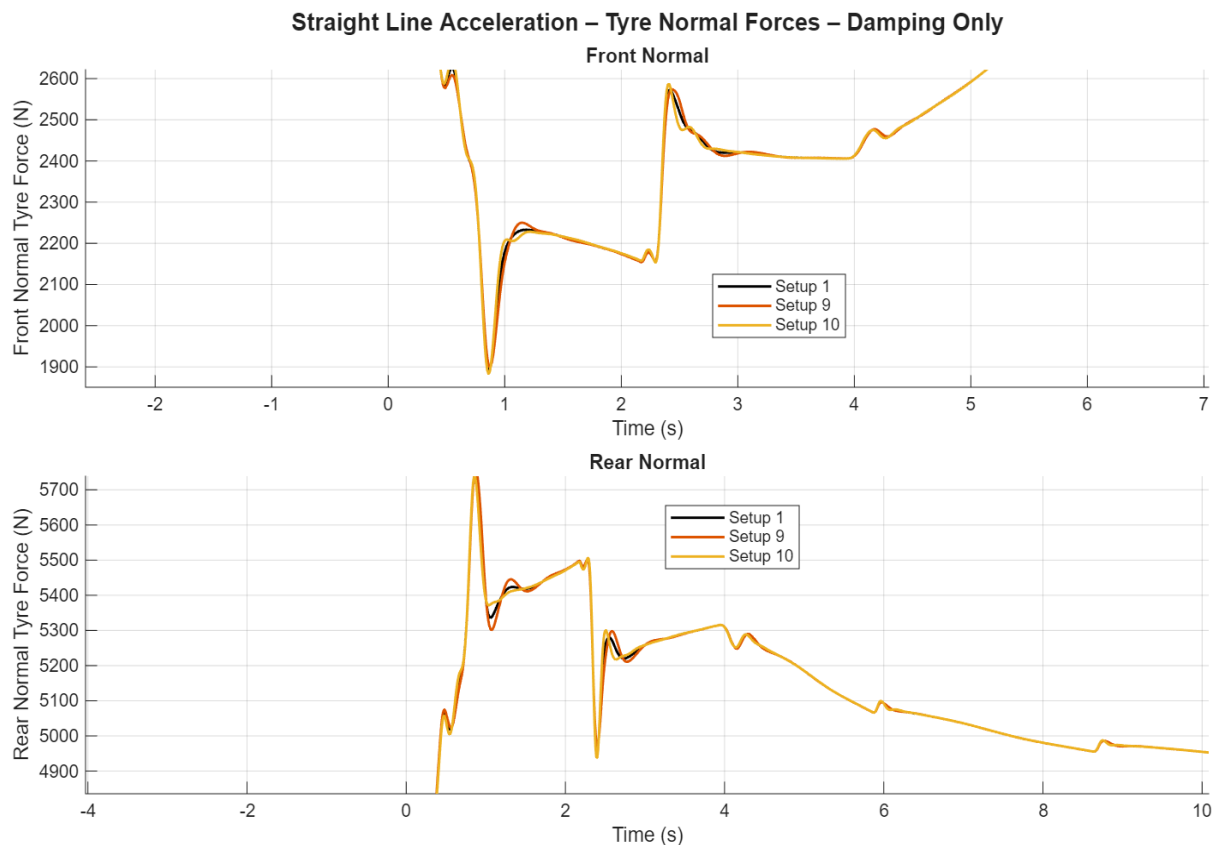
The damper adjustments were much more visible during hard acceleration shown in graph 9. Setup 10, which increased damping by 30%, helped with smoothing out the pitch towards the rear of the car. This setup therefore had better pitch control so could feel more stable during longitudinal weight transfer.



(Graph 10: Combined Setup Tyre Normal Forces vs Test Time)

Damper adjustment was more crucial when adjusting alongside springs as shown by graph 10, this could be due to spring rebound changes with rate adjustment. Stiffening the rear springs and reducing the damper force had a very negative effect on tyre forces (Setup 2) and created largely varying rear forces (Firth, G. R. 1991).

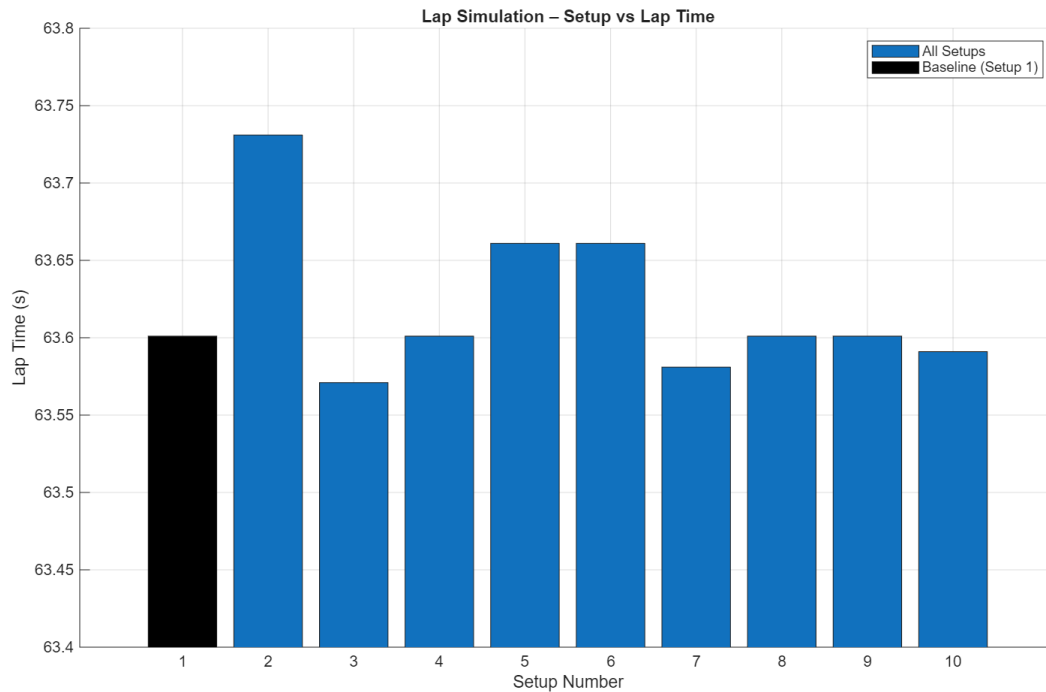
Graph 11 below shows how only the damper changes effected the tyres, slightly more minimal but shows that the forces were more stable with 30% more damping force in setup 10.



(Graph 11: Damper Setup Tyre Normal Forces vs Test Time)

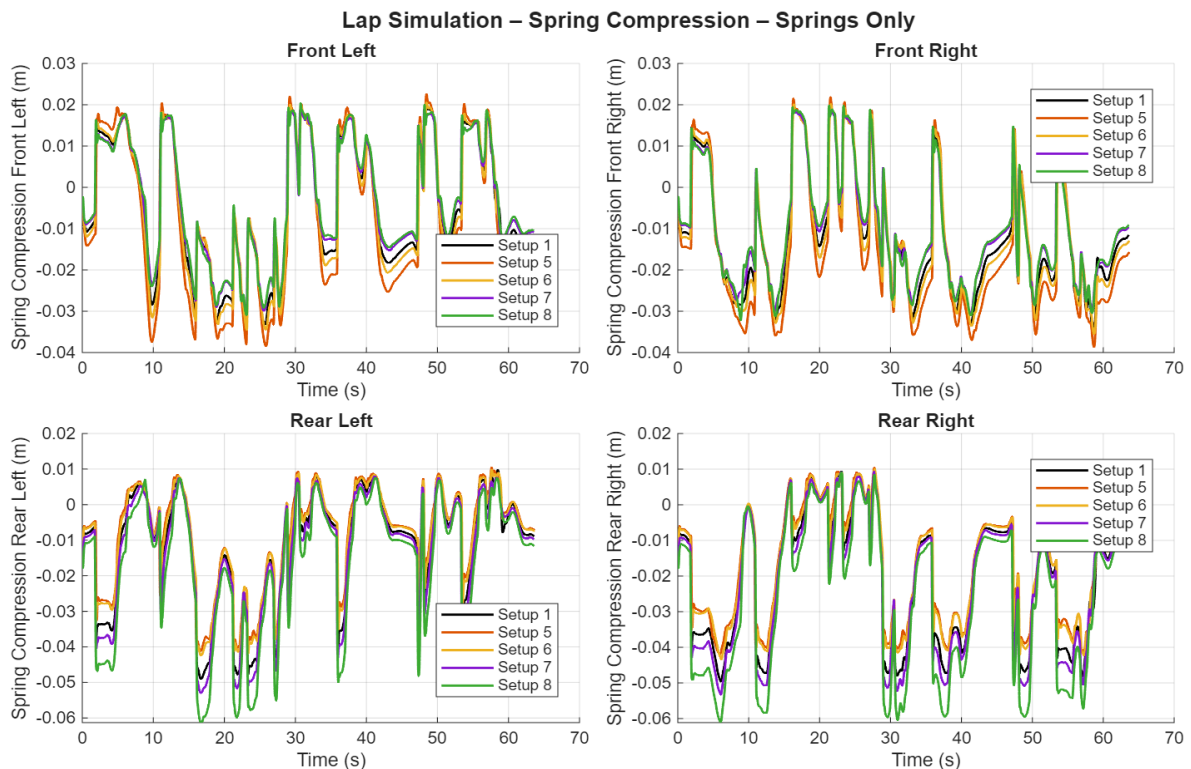
2.2 Lap Simulation

To simulate a lap with the sports car I performed a max performance event in VI-Grade around Pembrey. This will be a flying lap and will find the limit of each of the setups used in the dynamic manoeuvre section (Graph 1 & Appendix A).

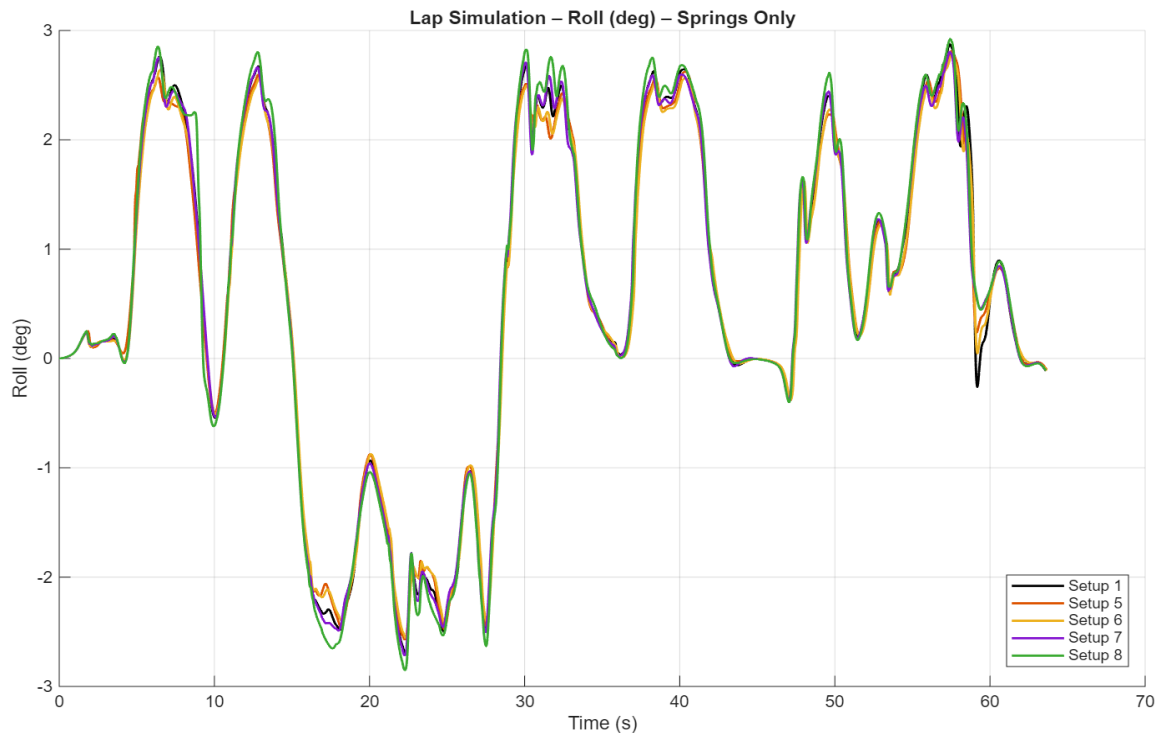


(Graph 12: Lap Time vs Setup Number)

2.2.1 Spring Variation Response

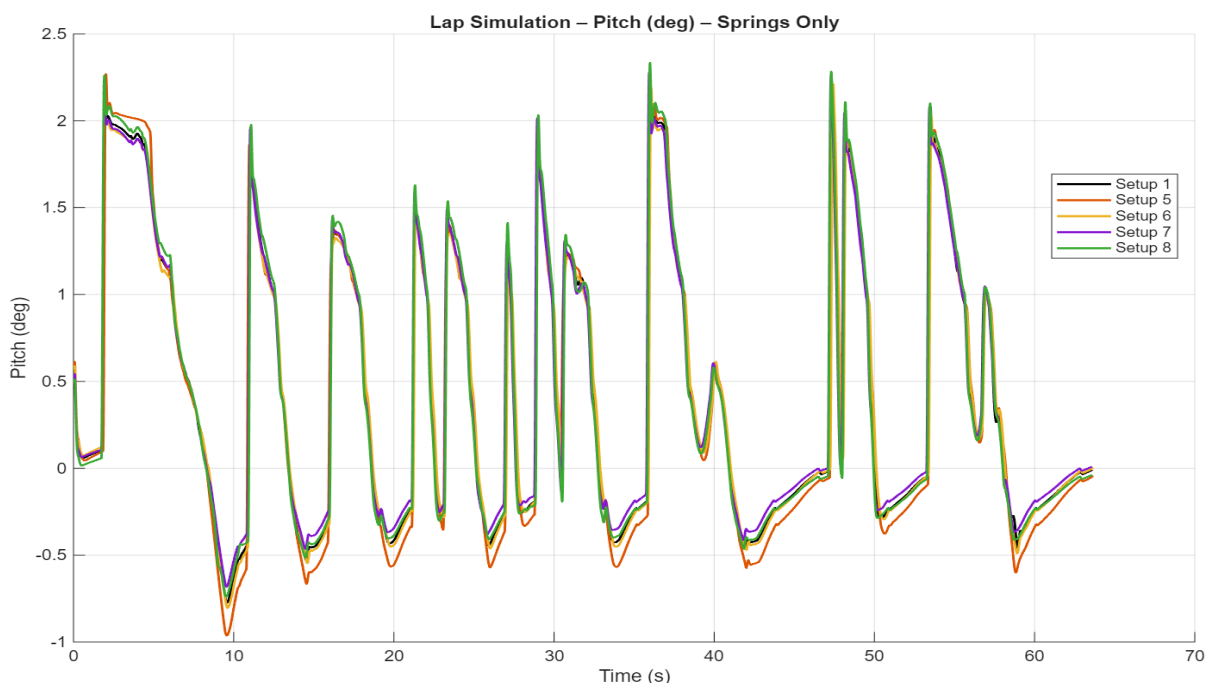


(Graph 13: Corner Specific Spring Compression vs Lap Time)



(Graph 14: Spring Setup Roll vs Lap Time)

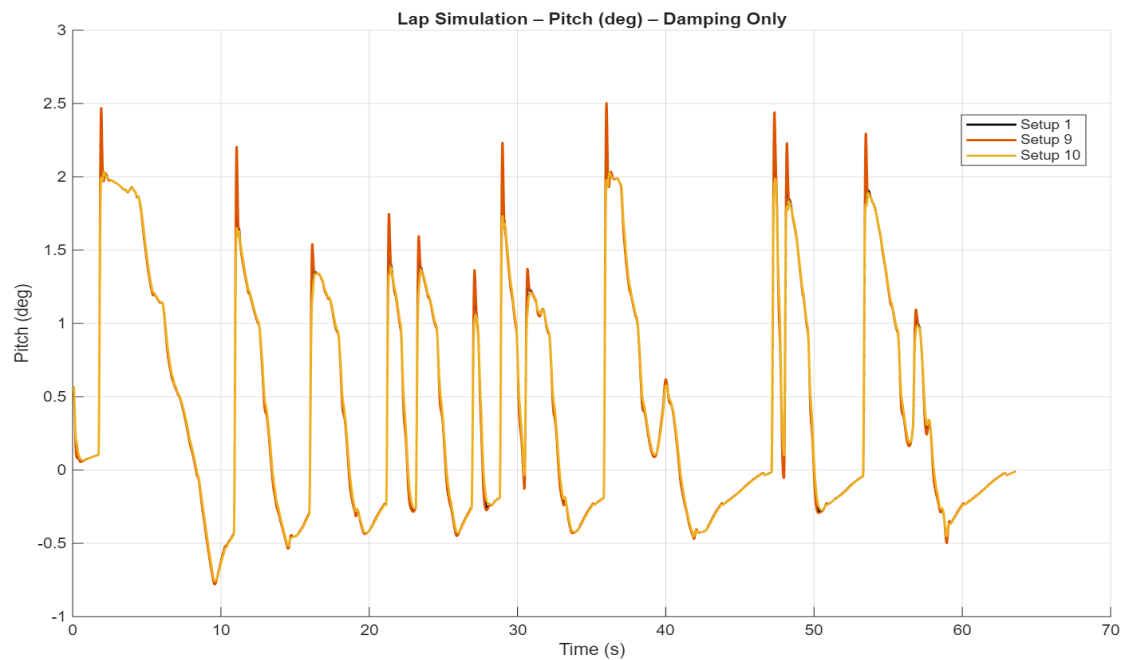
From graph 14 above, it can be observed that there isn't necessarily a correlation between a faster lap time with a reduction in roll. Setup 7 was the quickest out of the spring setups but in most areas has the second largest roll, slightly more than baseline. However, setup 7 has the quickest return to steady roll, meaning potentially higher stability.



(Graph 15: Spring Setup Pitch vs Lap Time)

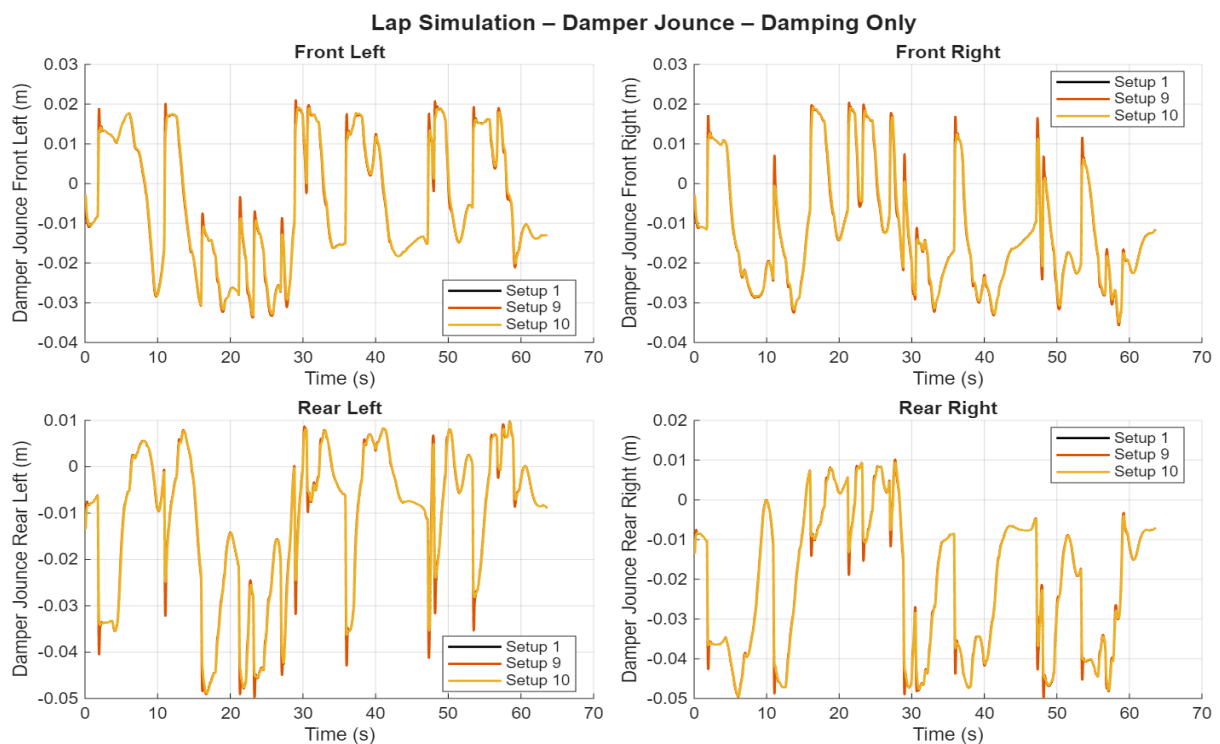
Pitch is reduced for setup 7, shown in graph 15, further explaining the increased stability under weight transfer equalling improved lap time. Setups 5 and 6 had the most forward pitch (dive) meaning the car would have been less stable under braking and initial turn-in.

2.2.2 Damper Variation Response



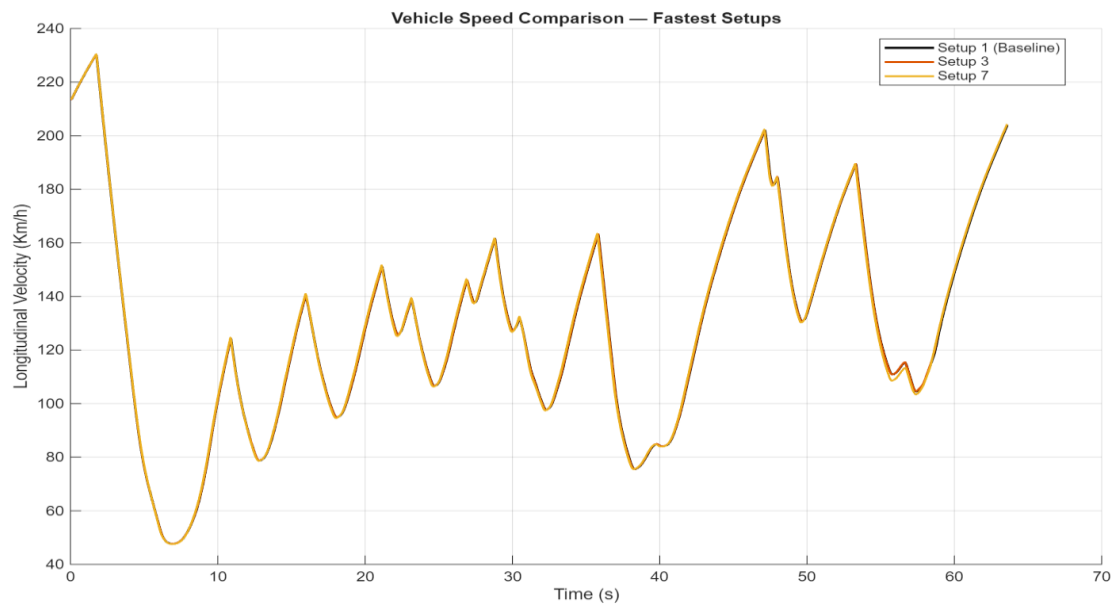
(Graph 16: Damper Setup Pitch vs Lap Time)

Setup 9 reduced overall damper force and affected pitch, shown in graph 16, with large spikes compared to setup 1. Setup 10 provided the most pitch resistance and was faster than the other setups investigated for damper changes. Like the dynamic manoeuvre test, damper alterations made very small differences to roll, therefore it isn't relevant to analyse from this test (see Appendix J). Graph 17 below shows the decrease in damper movement, which was also seen in the previous graph.



(Graph 17: Corner Specific Damper Jounce vs Lap Time)

2.2.3 Performance & Handling Characteristics



(Graph 18: Longitudinal Velocity vs Lap Time)

The velocity trace displayed in graph 18 presents interesting results as the quickest (setup 7) reduces speed more on entry but gets a much better exit often beating the others out of the corners. This is amplified into the last corners of the track around 55 seconds.

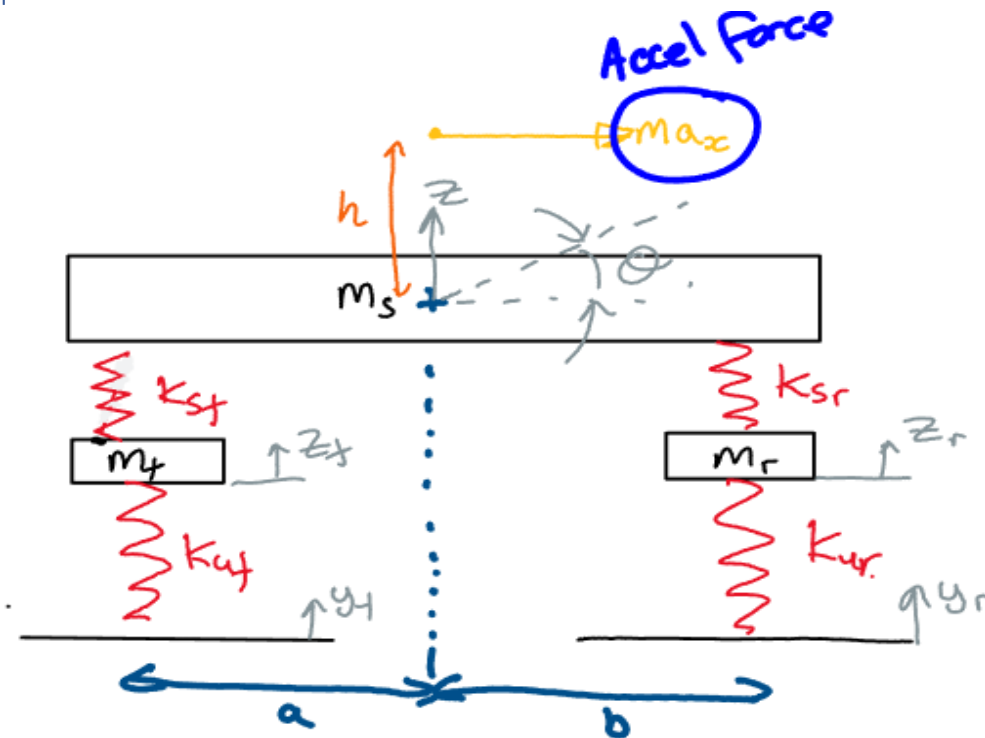
To understand why this is I have gathered a yaw rate graph (below) which maps how readily the car rotates laterally. This graph shows that setup 7 has the highest yaw rate in most areas, meaning it rotated easily, and the driving model slowed more to avoid losing control. However, this finding could be less setup related and instead explain a limitation of having a computed driver model, because the instability from the rear on entry forces the model to get a better exit. The interaction of the operator with the vehicle is a source of both input and disturbance (Blundell, M. and Harty, D. 2015).



(Graph 19: Yaw Rate vs Lap Time)

3 4 Degree of Freedom Simulink Model (Half Car)

3.1 Equations of Motion & Half Car Model



(Figure 2: Half Car Model Diagram (Tudor, T. 2022))

Notation: Z = Vertical Movement (Heave), Theta = Pitch,
Zf = Front Unsprung Movement, Zr = Rear Unsprung Movement

Acceleration x Mass = Spring Force Equation + Damper Force Equation

$$-m_s \ddot{Z} = K_f(Z_s - Z_f - a\theta) + K_r(Z_s - Z_r + b\theta) \\ + C_f(\dot{Z}_s - \dot{Z}_f - a\dot{\theta}) + C_r(\dot{Z}_s - \dot{Z}_r + b\dot{\theta})$$

$$-I\ddot{\theta} = bK_r(Z_s - Z_r + b\theta) + aK_f(Z_s - Z_f - a\theta) \\ + bC_r(\dot{Z}_s - \dot{Z}_r + b\dot{\theta}) + aC_f(\dot{Z}_s - \dot{Z}_f - a\dot{\theta})$$

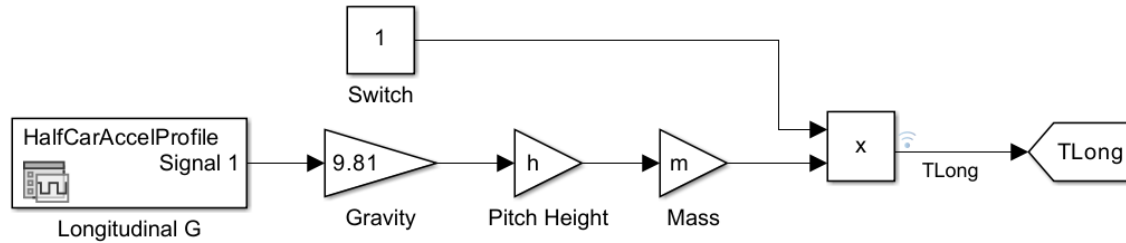
$$-m_f \ddot{Z}_f = K_{tf}(Z_f - y_f) - K_f(Z_s - Z_r - a\theta) \\ + C_{tf}(\dot{Z}_f - \dot{y}_f) - C_f(\dot{Z}_s - \dot{Z}_r - a\dot{\theta})$$

$$-m_r \ddot{Z}_r = K_{tr}(Z_r - y_r) - K_r(Z_s - Z_r + b\theta) \\ + C_{tr}(\dot{Z}_r - \dot{y}_r) - C_r(\dot{Z}_s - \dot{Z}_r + b\dot{\theta})$$

These equations were derived from figure 2 above. To find vertical, pitch, and unsprung movement the spring equation is divided by mass and then integrated twice. Damper forces are added to spring forces and are identical apart from damper rate variables and the equation being derived to find the velocities of each movement signal.

3.2 Model Development

3.2.1 Acceleration Input

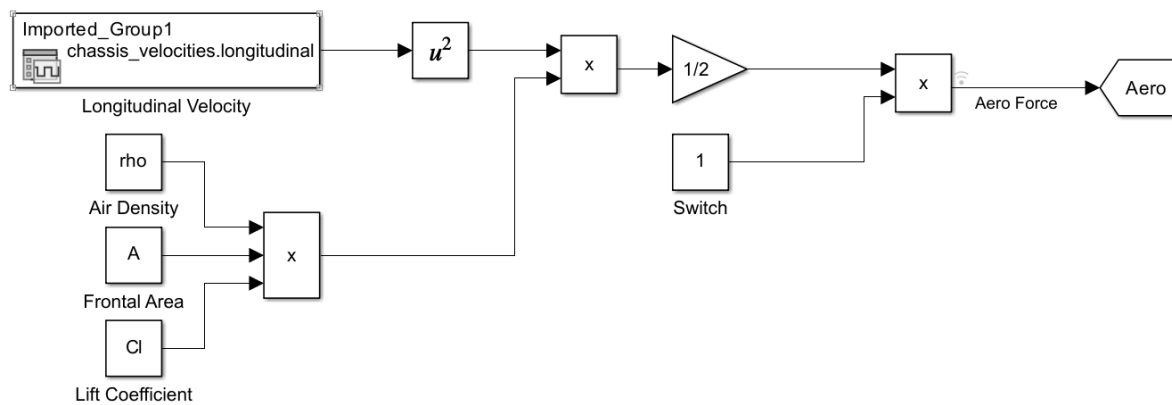


(Figure 3: Simulink Acceleration Input)

Figure 3 shows how I have implemented acceleration within Simulink; the output of this block is TLong which is the total longitudinal acceleration force. This is added to the pitch calculation to make the equation below:

$$-I\ddot{\theta} = bKr(Zs - Zr + b\dot{\theta}) + aKf(Zs - Zf - a\dot{\theta}) + bCr(\dot{Z}s - \dot{Z}r + b\dot{\theta}) + aCf(\dot{Z}s - \dot{Z}f - a\dot{\theta}) + TLong$$

3.2.2 Aerodynamic Input

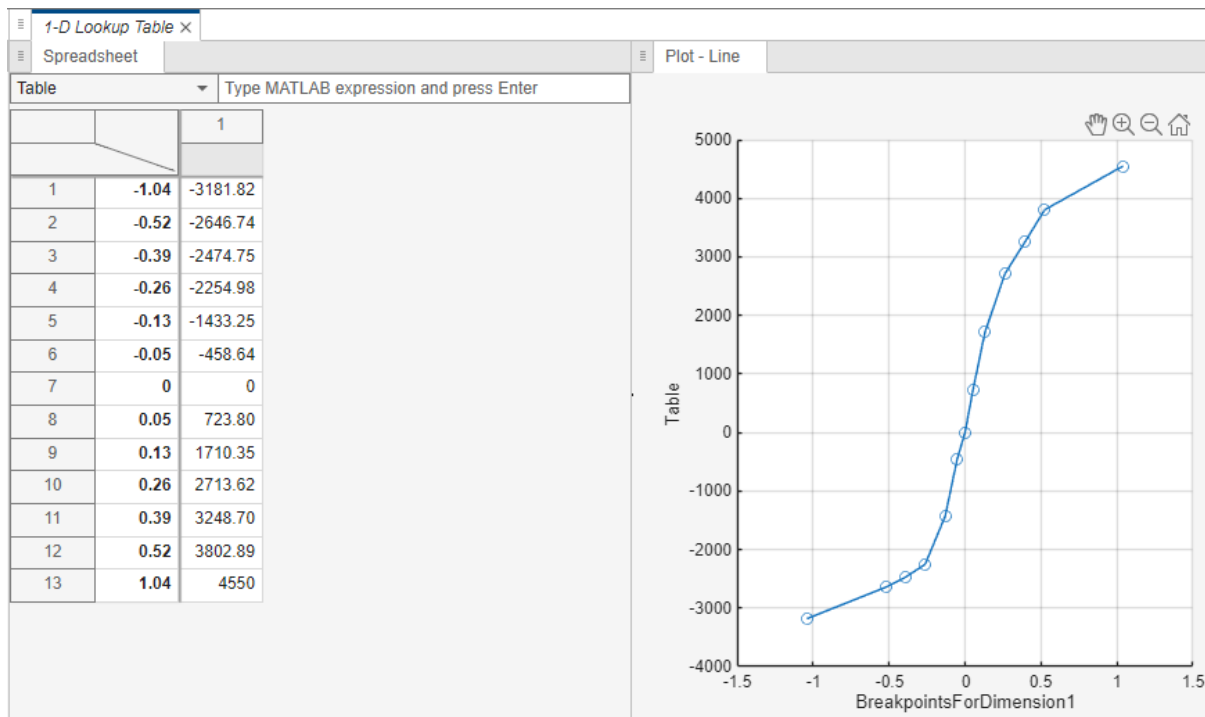


(Figure 4: Simulink Aerodynamic Input)

Figure 4 above is my Simulink block for calculating total aero forces from a longitudinal velocity signal. The total force is taken away from the vertical calculation to act as downforce shown below:

$$-ms\ddot{Z}s = Kf(Zs - Zf - a\dot{\theta}) + Kr(Zs - Zr + b\dot{\theta}) + Cf(\dot{Z}s - \dot{Z}f - a\dot{\theta}) + Cr(\dot{Z}s - \dot{Z}r + b\dot{\theta}) - Aero$$

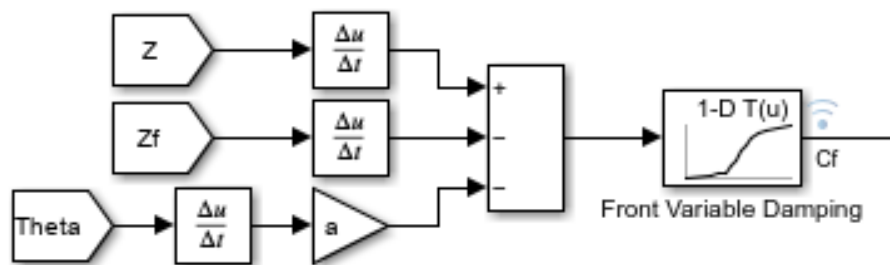
3.2.3 Variable Damping – VI-Grade Sports Car



(Figure 5: Simulink Front Variable Damping Lookup Table)

To add the variable damping curve of the VI-Grade model, I decided to utilise the 1-D Lookup Table block in Simulink (figure 5). This takes damper velocity in and outputs the corresponding damper force according to the damping curve taken from VI-Grade.

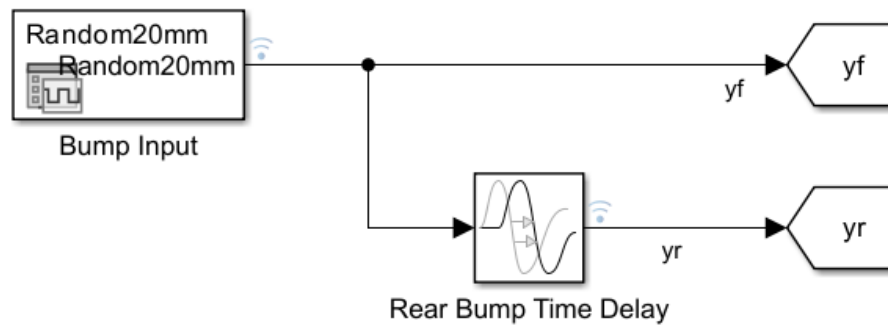
Below (figure 6) is how the C_f (front damping force) is computed within the mass equations.



(Figure 6: Simulink Front Variable Damping Calculation)

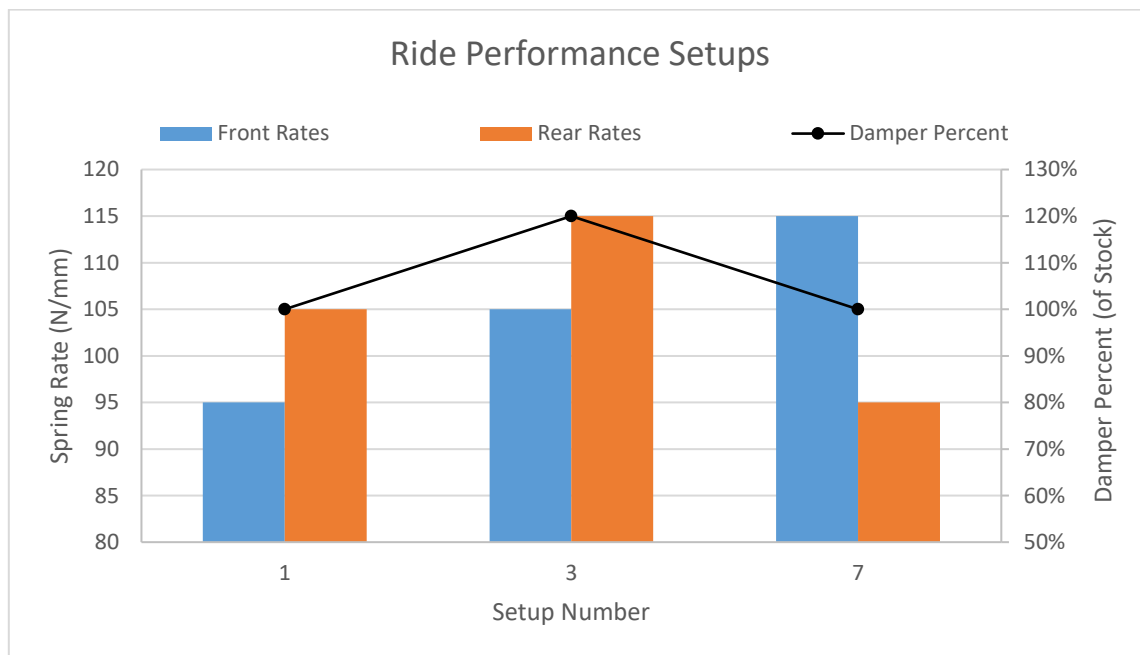
3.3 Vehicle Ride Performance

To evaluate ride performance through my Simulink half car model I will be comparing the quickest lap simulation setups 3 and 7 against baseline, through different bump inputs. This will help understand whether these setups can cope with bumps which is crucial for track performance. Figure 7 below shows the model taking a bump input and outputting direct road movement variables y_f (front) and y_r (rear). There is a 1 second time delay to simulate the time taken for the rear to reach the bump after the front.



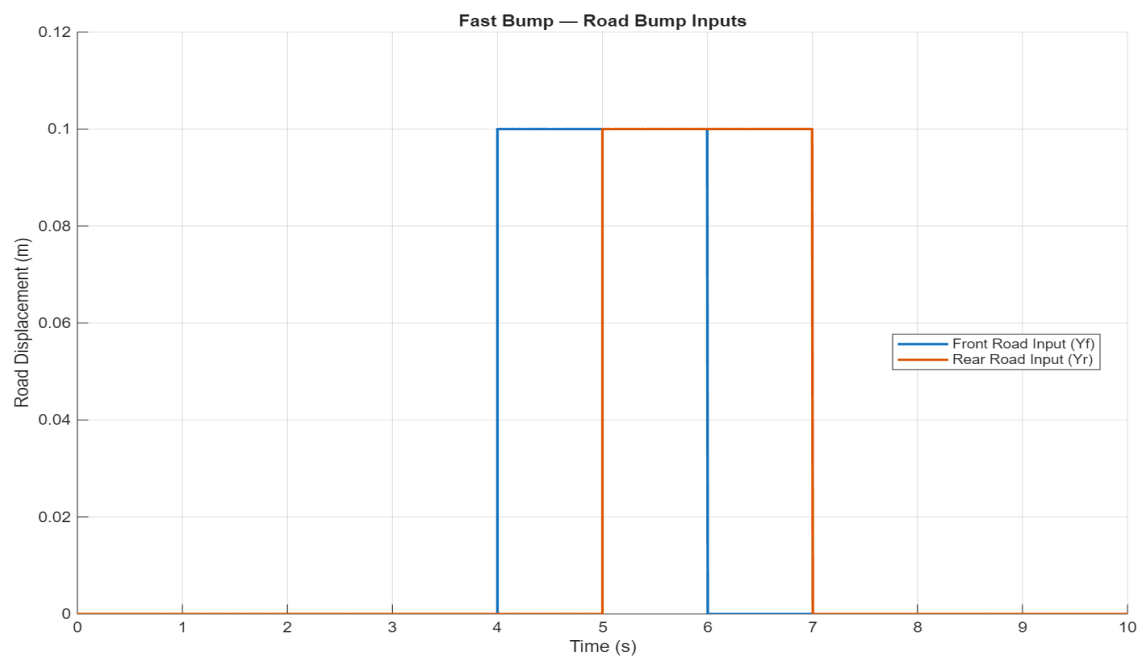
(Figure 7: Simulink Front and Rear Bump Input)

For this test I took the sports car model variables from VI-Grade to input into Simulink (see Appendix K), sprung mass was halved due to it being a half car model.



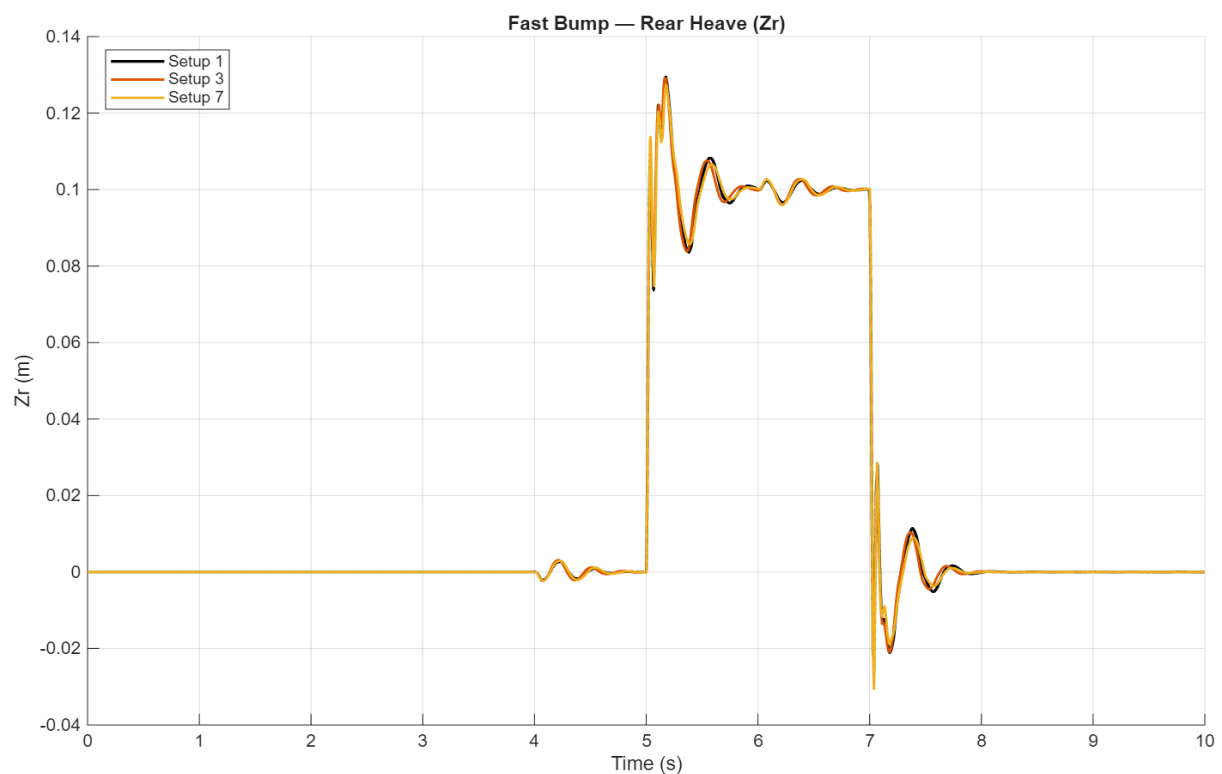
(Graph 20: Ride Performance Test Spring Damper Setups)

3.3.1 Fast Bump Input



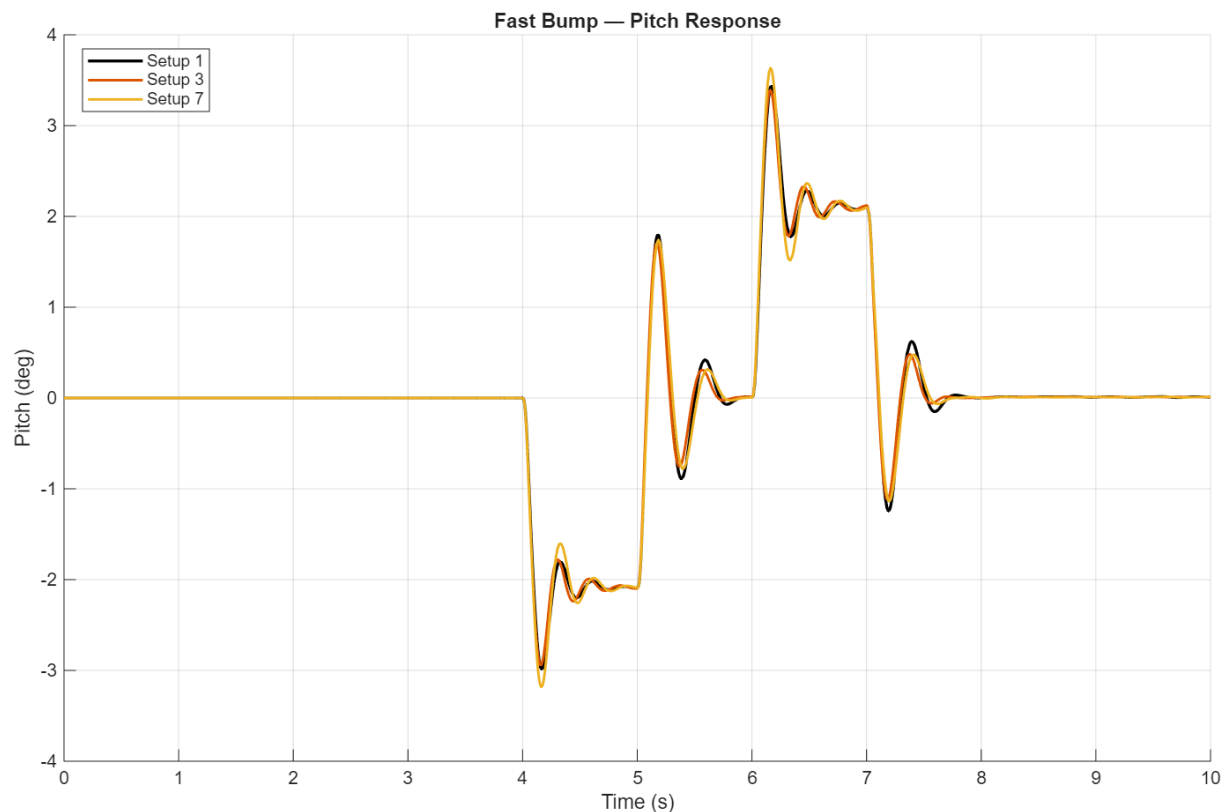
(Graph 21: Fast Bump Road Input)

Graph 21 shows my first bump input tested, which will help understand rapid damper force and heave changes.



(Graph 22: Fast Bump Rear Heave Movement)

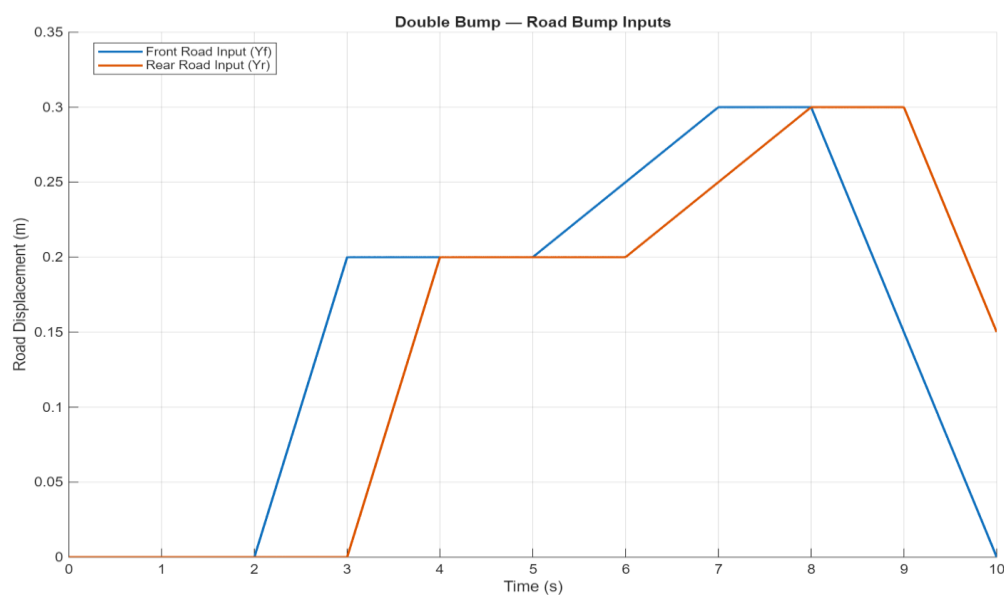
Graph 22 details the overall heave (vertical movement) of the car, and it shows that setup 3 with increased damping settles the rear of the car quicker after bumps. This could allow the car to be more stable at the rear after a harsh bump than the other setups, improving handling.



(Graph 23: Fast Bump Pitch Response)

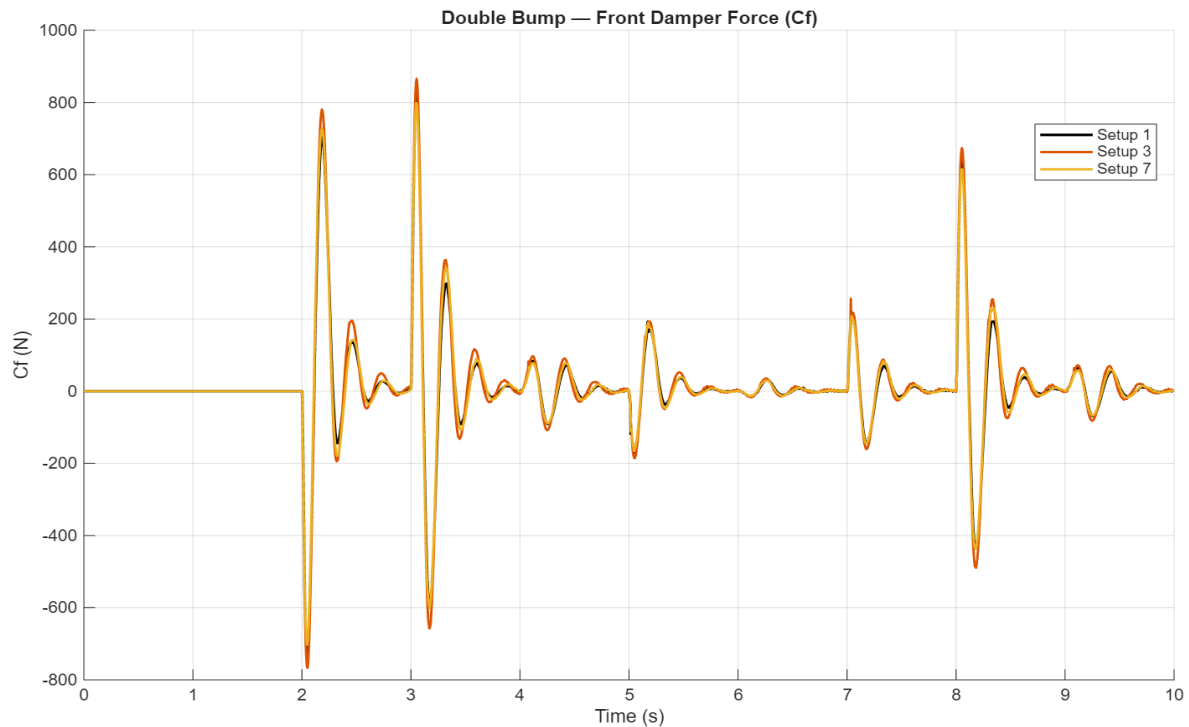
Graph 23 shows that setup 3 controls the pitch better than the other setups however, during rear bump inputs at 5 and 7 seconds the pitch range is very similar to setup 7. Considering this, that could explain why there was differences during the lap simulation in terms of ride performance with setup 7 having better control of rear bump.

3.3.2 Double Bump Input



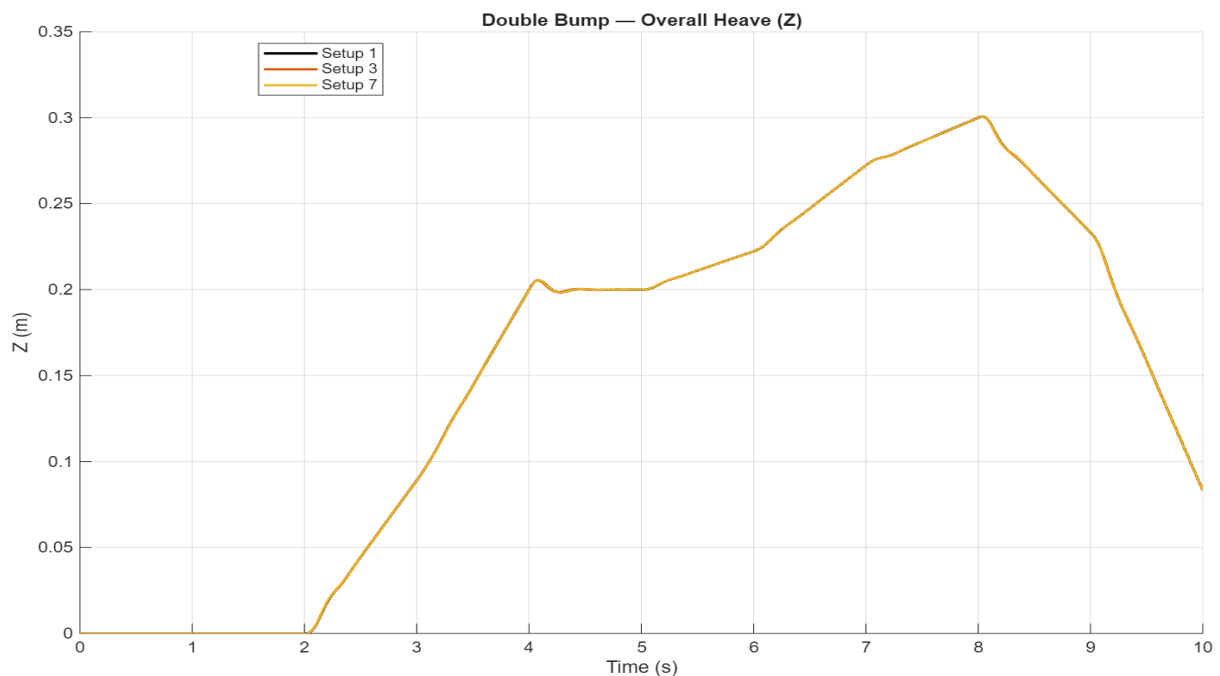
(Graph 24: Double Bump Road Bump Input)

Graph 24 shows my second bump input tested, which analyses multiple bump movements close together to evaluate quick road changes.



(Graph 25: Double Bump Road Bump Input)

The above graph 25 shows how the damper force works to damp the incoming bumps. Setup 3 provides the most force as expected but interestingly setup 7 and setup 1 have different responses with the same damping curve. This could be due to how the spring rate has affected damper force needed to stop rebound.



(Graph 26: Double Bump Road Bump Input)

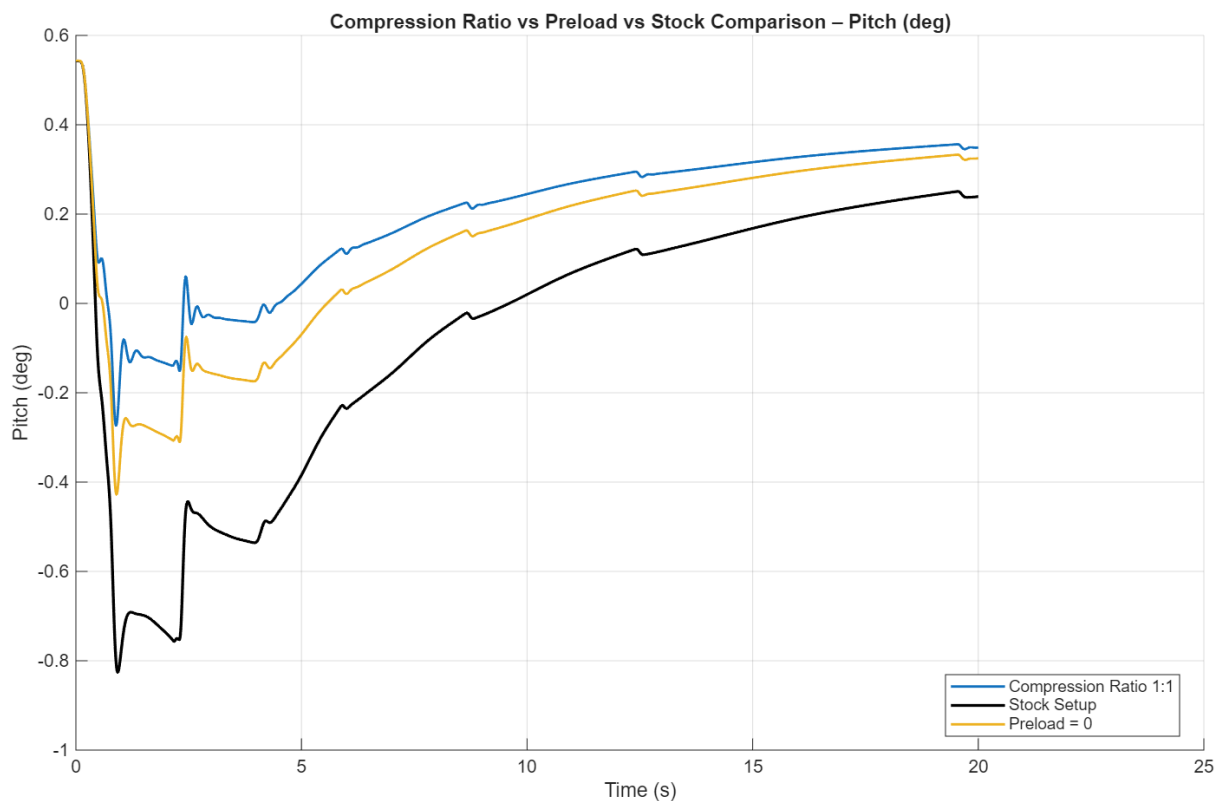
Graph 26 shows how overall vertical movement is affected by the changes and shows a distinct lack of change. This could be due to the setups being relatively close together in terms of balance and overall spring rate, however, this does lead onto a couple of limitations within the Simulink model.

The half car model used doesn't have a compression ratio adjustment meaning that each setup assumes a ratio of 1:1, which could potentially change when adjusting springs on a real car. There is also no ride height adjustment which relates to motion ratios, especially when considering ride during multiple bump inputs. For example, heave may change between spring changes due to ride height, but this isn't modelled meaning the half car model data remains very similar.

3.4 Simulink Half Car VI-Grade Comparison

3.4.1 Compression Ratio & Preload

Before comparing VI-Grade and Simulink, I decided to analyse the effect of changing the compression ratio and preload parameters within VI-Grade on a straight-line acceleration test identical to earlier in this report (see Appendix H).

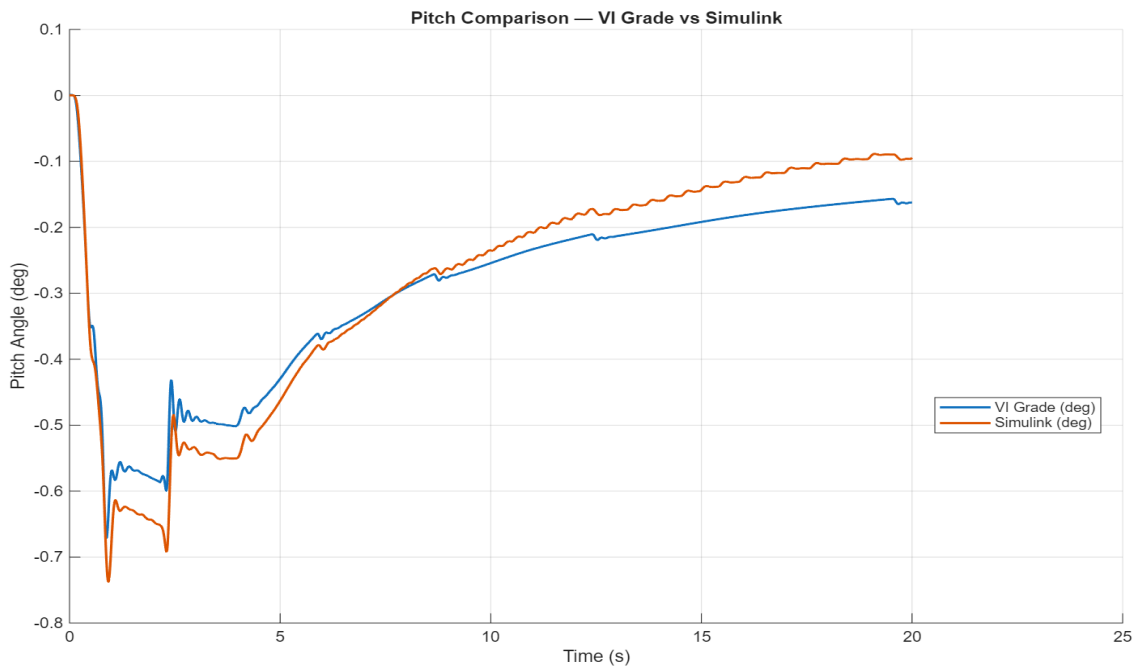


(Graph 27: Compression Ratio & Preload Pitch Comparison)

Graph 27 shown above explains that the preload and compression ratio adjustment have a significant effect on pitch response. Whilst compression ratio has a larger effect, preload also makes drastic changes to pitch, which I didn't anticipate making such a large difference. To accurately compare my half car and VI-Grade moving forward, I will be adjusting preload to 0 and compression ratio to a constant of 1, to match Simulink. Compression effect on each corner from this test can be seen in Appendix L.

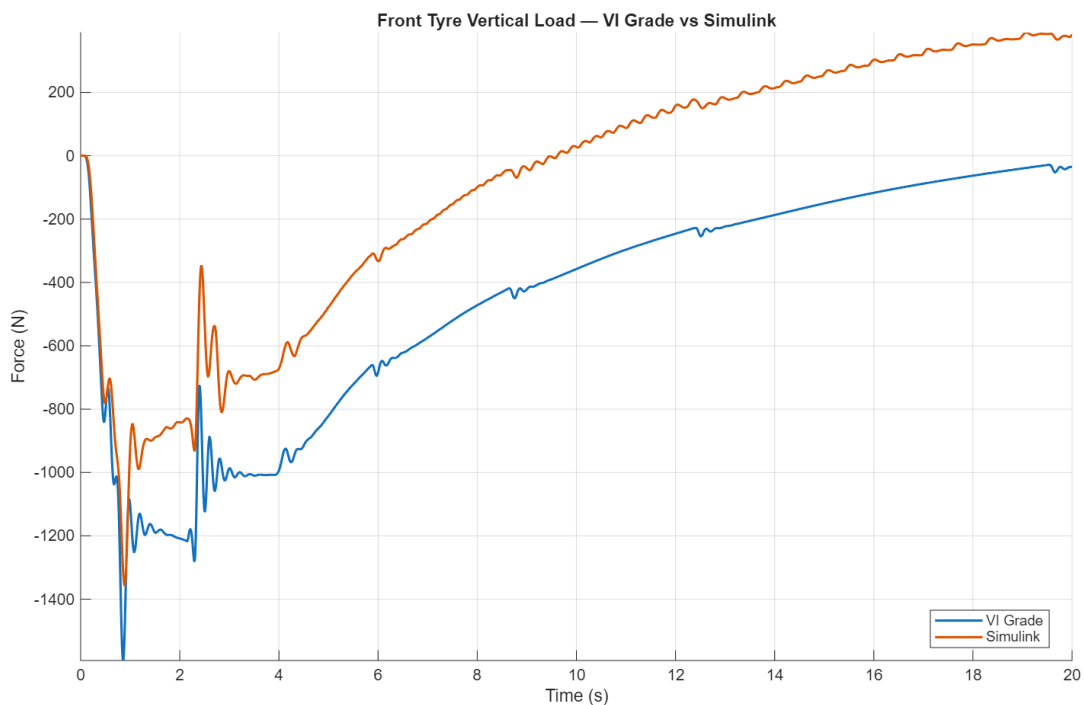
3.4.2 Acceleration Test Comparison

To compare closely I have removed the offset that VI-Grade sets for the simulation as it doesn't assume 0 tyre forces at the start of each test, which is how the Simulink model operates.



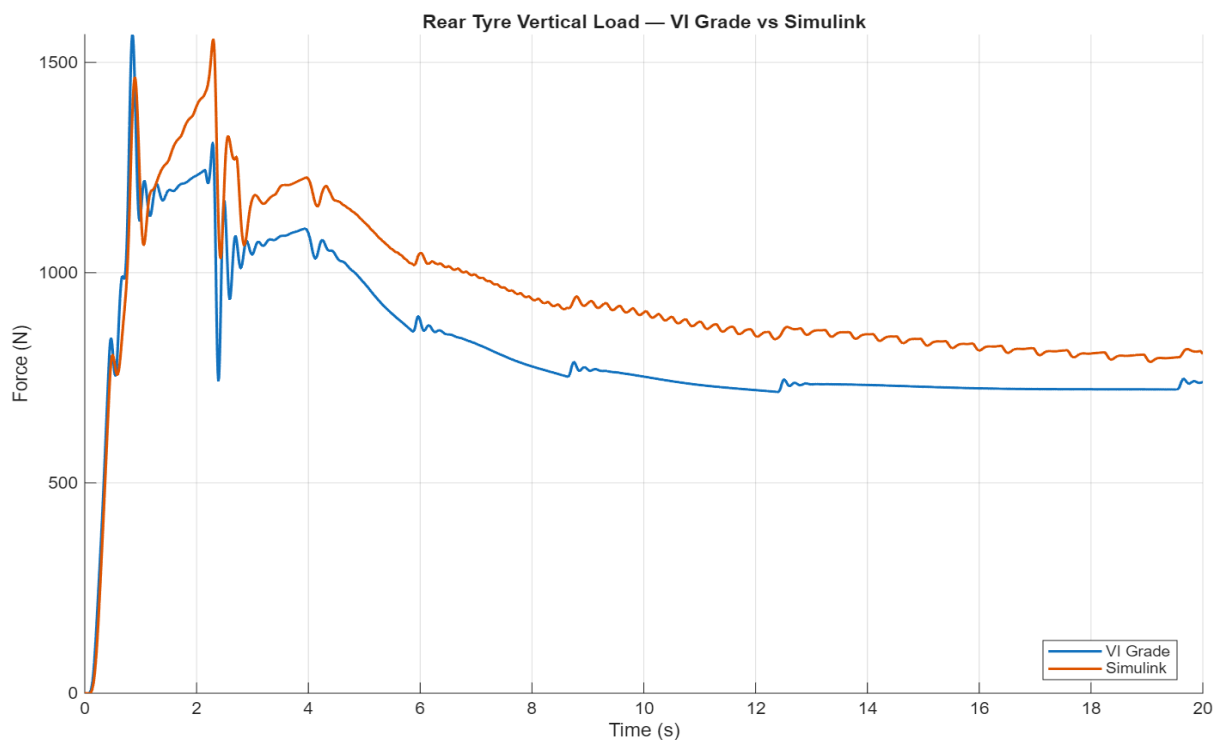
(Graph 28: VI-Grade vs Simulink Half Car Pitch Comparison)

The above graph shows how the pitch trends between Simulink and VI-Grade are similar, but the scale is slightly different. Initially Simulink assumes larger pitch values than VI-Grade until around halfway through the test where the lines cross and settle. This behaviour from VI-Grade could be the effect of the offset I mentioned which models the sports car as having forward pitch naturally, this would be difficult to remove this effect from the data entirely.



(Graph 29: VI-Grade vs Simulink Half Car Front Tyre Load Comparison)

Graph 29 above shows how Simulink predicts less lift on the front of the car for the duration of the acceleration test. The reason for this could be the difference between the aero force input between the models. VI-Grade has a complex system with front and rear aero affecting certain specified areas of the car, whereas Simulink is just a generic whole-car load which just assumes one point of downforce as the whole car. This has affected the front tyre loads because VI-Grade assumes a more rear biased aero map on the sports car which relieves load off the front axle more. The amount of lateral force that can be generated by a tyre is extremely dependent on the vertical force acting upon it (Santos, R. de O. 2014).

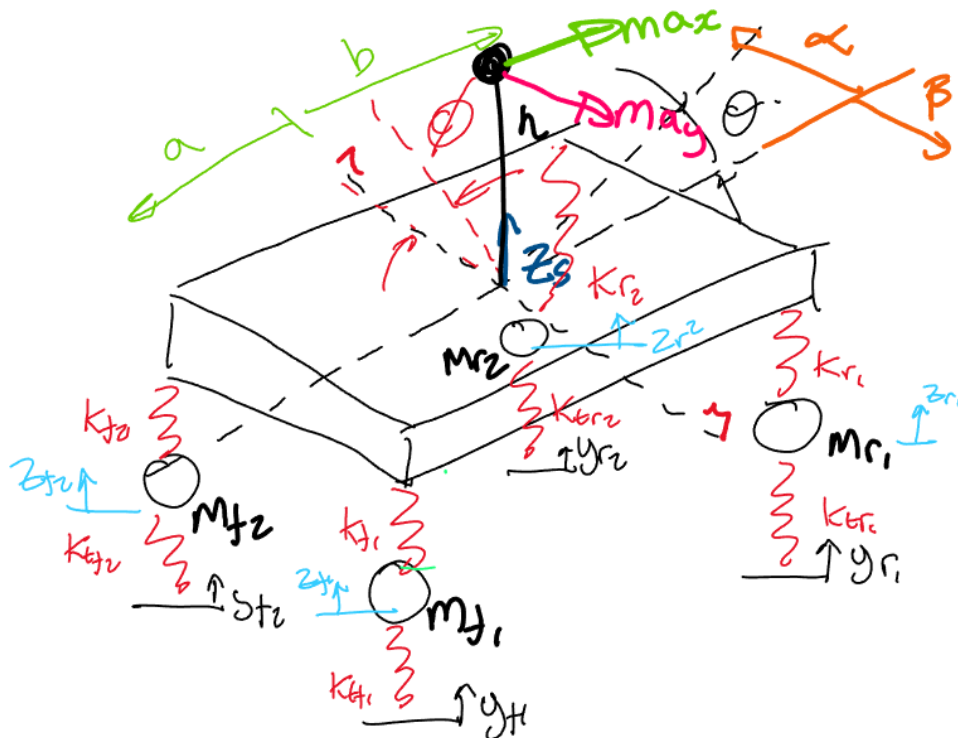


(Graph 30: VI-Grade vs Simulink Half Car Rear Tyre Load Comparison)

Graph 30 shows the correlation between the front and rear vertical load plots is consistent with differences in aerodynamic load distribution and longitudinal load transfer modelling. VI-Grade applies distributed, rear-biased aerodynamic forces and detailed suspension load transfer, resulting in reduced front axle load and redistributed rear load under acceleration. This is because downforce in VI-Grade produces negative values therefore the tyre loads are both less in comparison to Simulink.

4 7 Degree of Freedom Simulink Model (Full Car)

4.1 Equations of Motion & Full Car Model



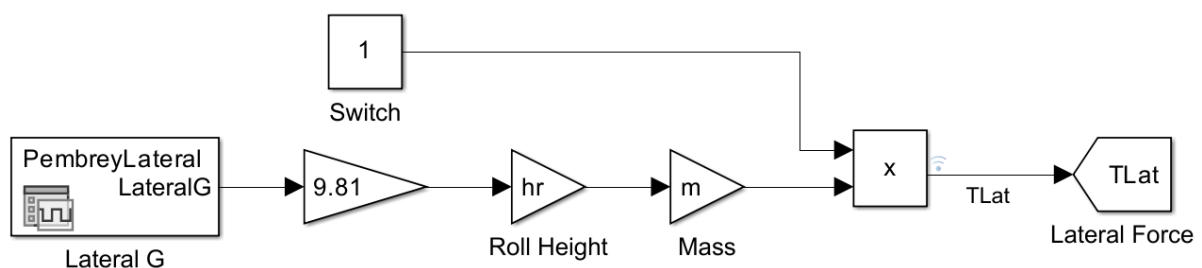
(Figure 8: Full Car Model Diagram (Tudor, T. 2021))

Added notation: Phi = Roll,
Number 1 = Left, Number 2 = Right

Acceleration x Mass = Spring Force Equation + Damper Force Equation

$$\begin{aligned}
 -I_{xx}\ddot{\phi} = & \alpha K_{f2}(Z - Z_{f2} - a\theta - \beta\phi) + \alpha K_{r2}(Z - Z_{r2} + b\theta + \alpha\phi) \\
 & - \beta K_{f1}(Z - Z_{f1} - a\theta - \beta\phi) - \beta K_{r1}(Z - Z_{r1} + b\theta - \beta\phi) \\
 & + \alpha C_{f2}(\dot{Z} - \dot{Z}_{f2} - a\dot{\theta} - \beta\dot{\phi}) + \alpha C_{r2}(\dot{Z} - \dot{Z}_{r2} + b\dot{\theta} + \alpha\dot{\phi}) \\
 & - \beta C_{f1}(\dot{Z} - \dot{Z}_{f1} - a\dot{\theta} - \beta\dot{\phi}) - \beta C_{r1}(\dot{Z} - \dot{Z}_{r1} + b\dot{\theta} - \beta\dot{\phi})
 \end{aligned}$$

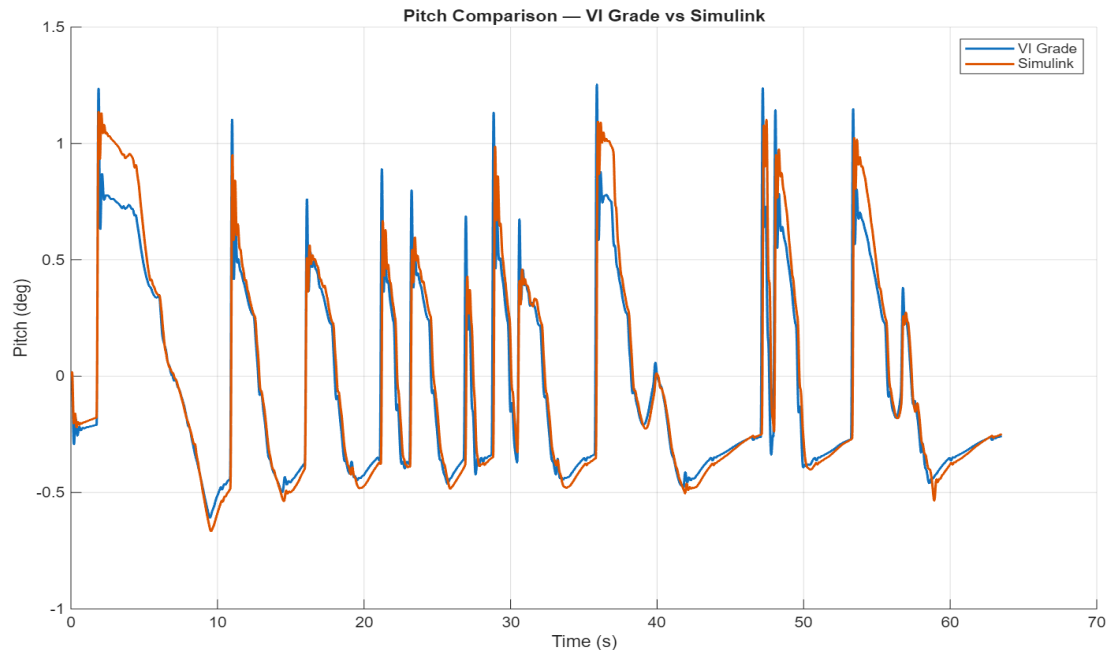
Above is the added lateral equation to calculate roll. Figure 9 below shows how total lateral acceleration (TLat) is calculated from a lateral g input. This is then subtracted from the equation above to calculate roll as Phi. Full 7DOF spring equations are detailed in Appendix M. The displacements were derived in the same way as the 4DOF model to create damper forces.



(Figure 9: Full Car Lateral Force Calculation)

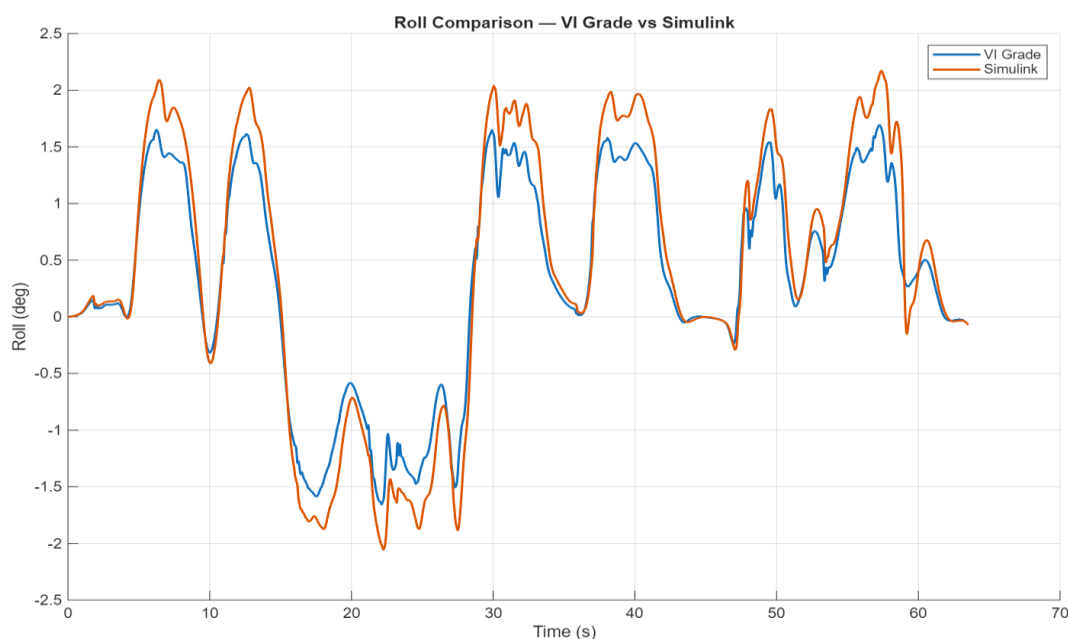
4.2 Lap Simulation Comparison

To complete my comparison of multi-body and analytical modelling, I tested the VI sports car around Pembrey, the same stock setup as before, but with a matching 1:1 compression ratio and 0 preload. I also had to adjust the graphs once again to remove the offset which VI-Grade has.



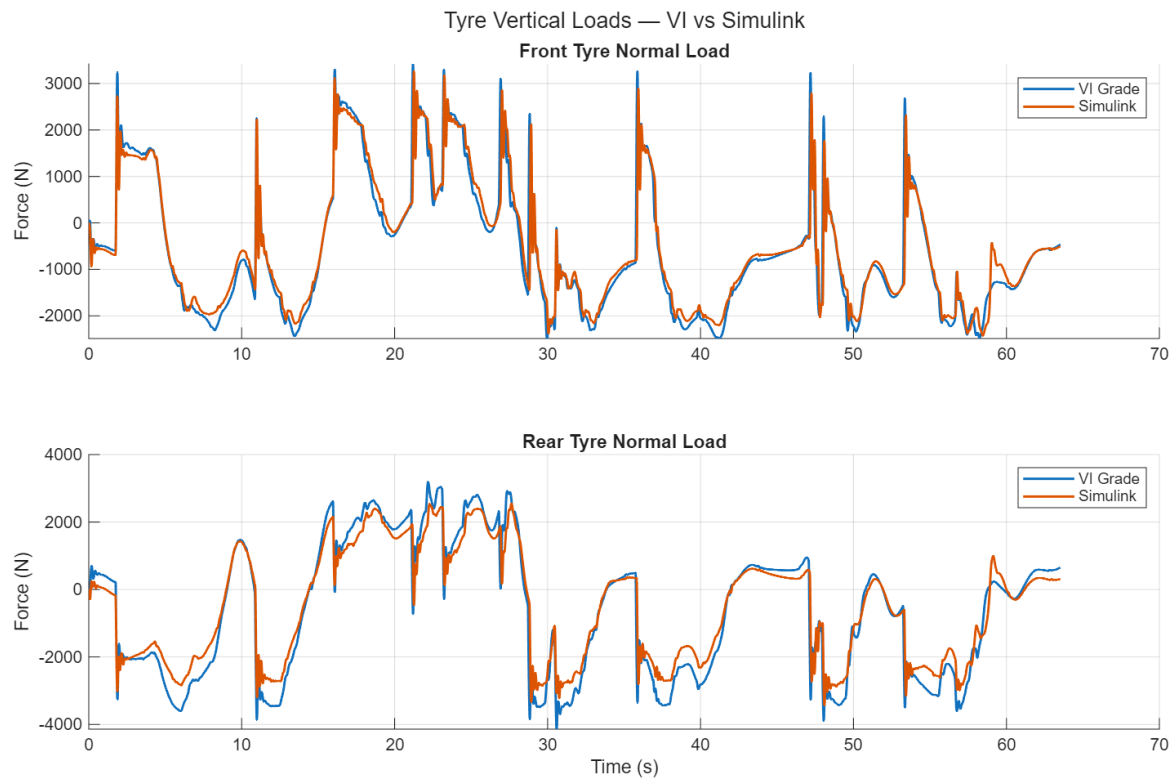
(Graph 31: Full Car Simulink vs VI-Grade Pitch Response)

The above graph shows that the 7DOF Simulink model matches pitch trends exactly with a slight variation in range. VI-Grade CarRealTime was unable to provide pitch and roll height above centre of gravity values, meaning I defaulted to using centre of gravity height for both in Simulink. Ideally there would be values for these available that differ to correctly calculate about pitch and roll centres. I anticipate that the pitch and roll height would be smaller than the value I used therefore, this would reduce the higher Simulink values to be inline, or at least closer in comparison.



(Graph 32: Full Car Simulink vs VI-Grade Roll Response)

Graph 32 above reflects my earlier points about needing the roll centre data, as once again, Simulink has over-predicted the scale of roll around Pembrey. This roll data leads to another limitation of the Simulink model, anti-roll bars (ARBs) aren't modelled, meaning that the VI car would be more resistive to roll. This can be seen in the data as the models are close at smaller roll events but as the car rolls more the separation begins, which I think could be the ARB's distributing load to the opposite side during high lateral load. (Tuan, A.N. and Thang, B.H. 2019).



(Graph 33: Full Car Simulink vs VI-Grade Roll Response)

I then compared the tyre normal forces between the models and from the graph above I am happy that there is good correlation for the majority of the lap. The main differences in tyre force match the correlation seen in the roll and pitch graphs, where larger pitch and roll moments amplify the differences between complex simulation and analytical modelling.

5 Conclusion

This investigation has demonstrated how suspension spring and damping parameters influence vehicle ride, handling and tyre load behaviour using both multi-body simulation (VI-Grade CarRealTime) and analytical modelling (Simulink). The VI-Grade dynamic manoeuvres showed that combined spring and damping changes produced the most significant effects on vehicle response. During sine steer and constant radius tests, stiffer front springs increased understeer whilst softer rear springs promoted oversteer, confirming expected lateral load transfer trends. Damper-only changes produced comparatively small variations in lateral behaviour but were more influential under longitudinal acceleration, where increased damping reduced pitch oscillations and stabilised tyre normal forces.

Lap simulation results around Pembrey demonstrated that reduced roll alone did not guarantee improved performance. The fastest configuration achieved a balance between controlled pitch response and rapid recovery to steady roll, enabling improved corner exit performance despite higher transient yaw rates. This highlights the importance of combined spring–damper optimisation rather than isolated parameter changes.

The 4DOF half-car Simulink model successfully captured key ride and pitch response trends and showed good qualitative agreement with VI-Grade during acceleration testing. Differences in tyre load prediction were attributed primarily to the simplified aerodynamic modelling and the absence of preload and compression ratio effects. Extending the model to a 7DOF full-car configuration improved correlation further by introducing roll dynamics and lateral load transfer. However, limitations remained due to the lack of anti-roll bar modelling, and the assumption of centre-of-gravity roll and pitch heights.

Overall, this study confirms that multi-body simulation provides superior fidelity for full-vehicle behaviour, while analytical models remain valuable for understanding fundamental dynamics and trends. The combination of both approaches offers a powerful framework for suspension development and motorsport vehicle optimisation.

References

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Appendices

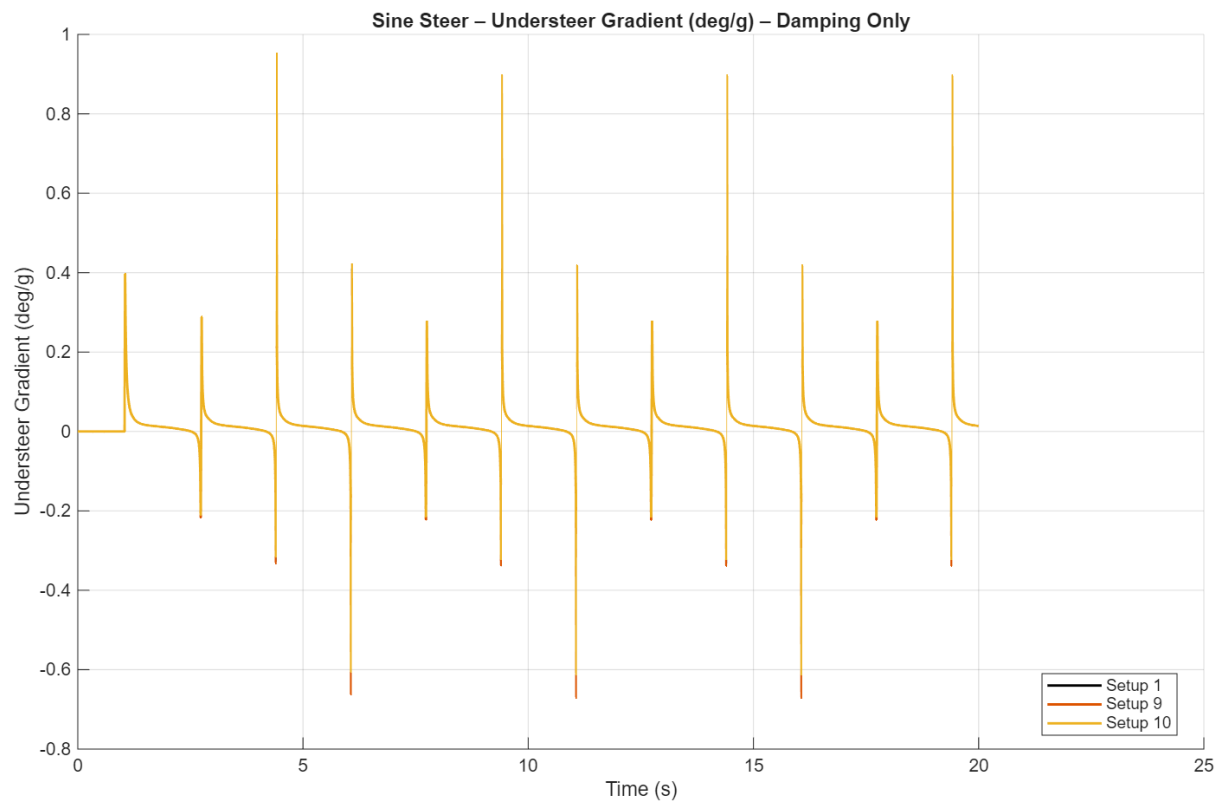
Appendix A – Spring Damper Setup Table

Setup	Damping (%)	Front Spring (N/mm)	Rear Spring (N/mm)
Setup 1 - Stock springs & Stock damping	100%	95	105
Setup 2 - Spring 1 (soft front, stiff rear) & -30% damping	70%	70	130
Setup 3 - Spring 3 (near-stock) & +20% damping	120%	105	115
Setup 4 - Spring 5 (stiff front, soft rear) & +30% damping	130%	120	80
Setup 5 - Spring 1 only (stock damping)	100%	70	130
Setup 6 - Spring 2 only (stock damping)	100%	85	125
Setup 7 - Spring 4 only (stock damping)	100%	115	95
Setup 8 - Spring 5 only (stock damping)	100%	120	80
Setup 9 - Damping -20% only (stock springs)	80%	95	105
Setup 10 - Damping +30% only (stock springs)	130%	95	105

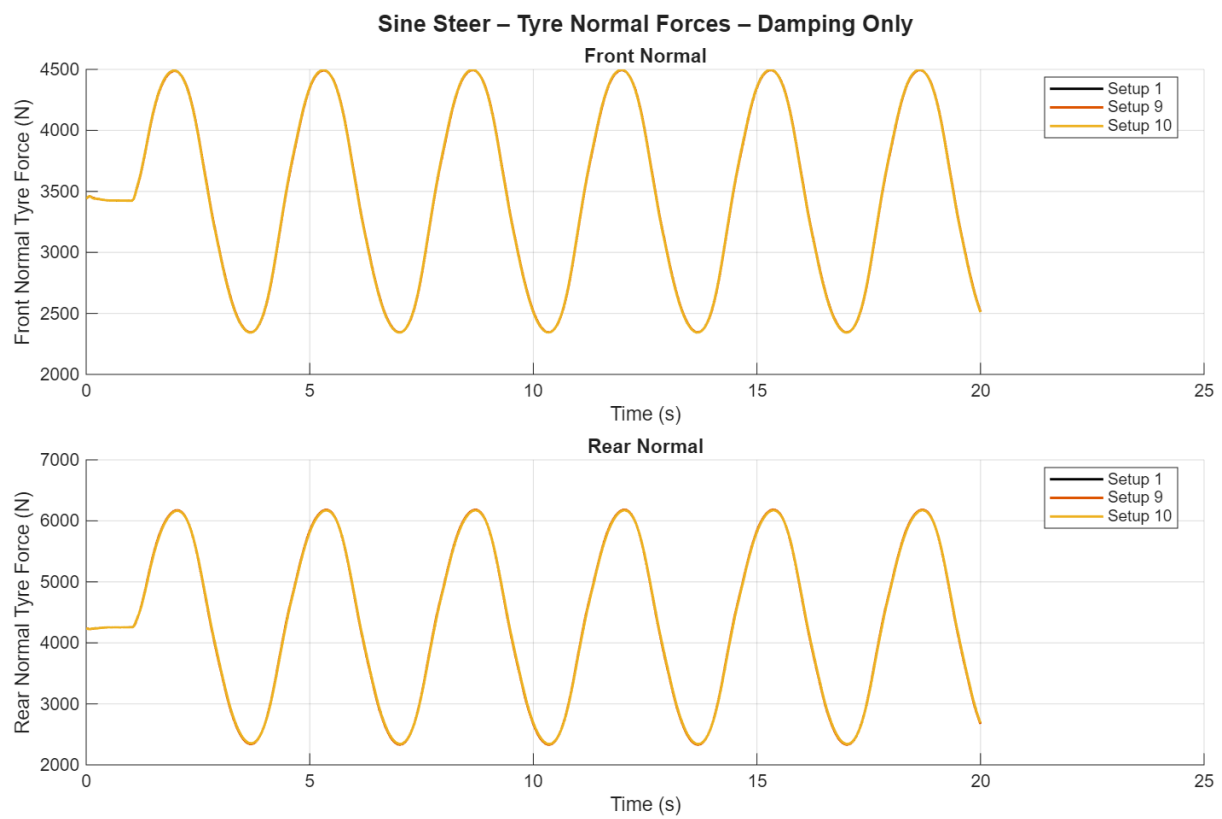
Appendix B – Sine Steer - VI-Grade Test Values

Sine Steer	Value	Units
Test End Time	20	Secs
Initial Velocity	80	Kph
Initial Gear	2	-
Steer Start Time	1	Secs
Initial Steer	0	Deg
Sine Steer Amplitude	50	Deg
Sine Steer Frequency	0.3	Hz
Sine Steer Phase	0	Deg
Cruise Control	Yes at 80	Kph
Steer Release	No	-

Appendix C – Sine Steer Damper Setup - Understeer Gradient Graph



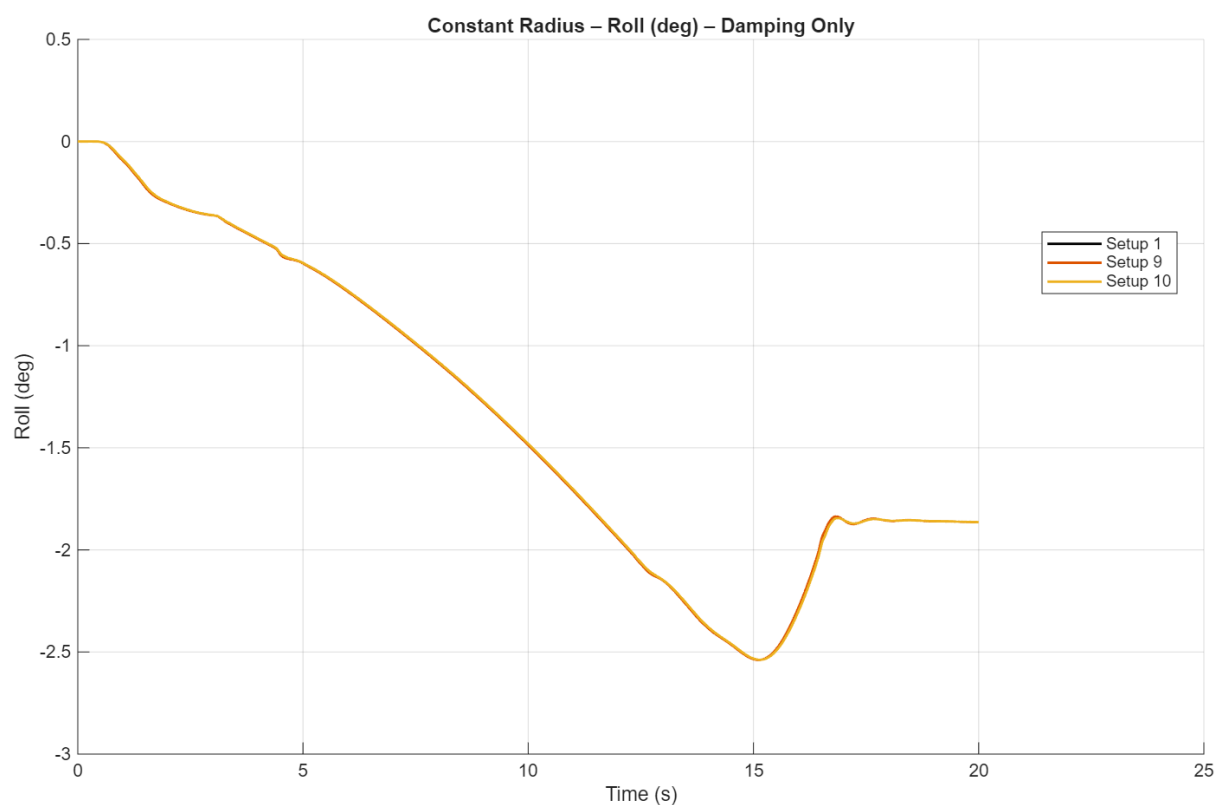
Appendix D – Sine Steer Damper Setup - Tyre Normal Forces Graph



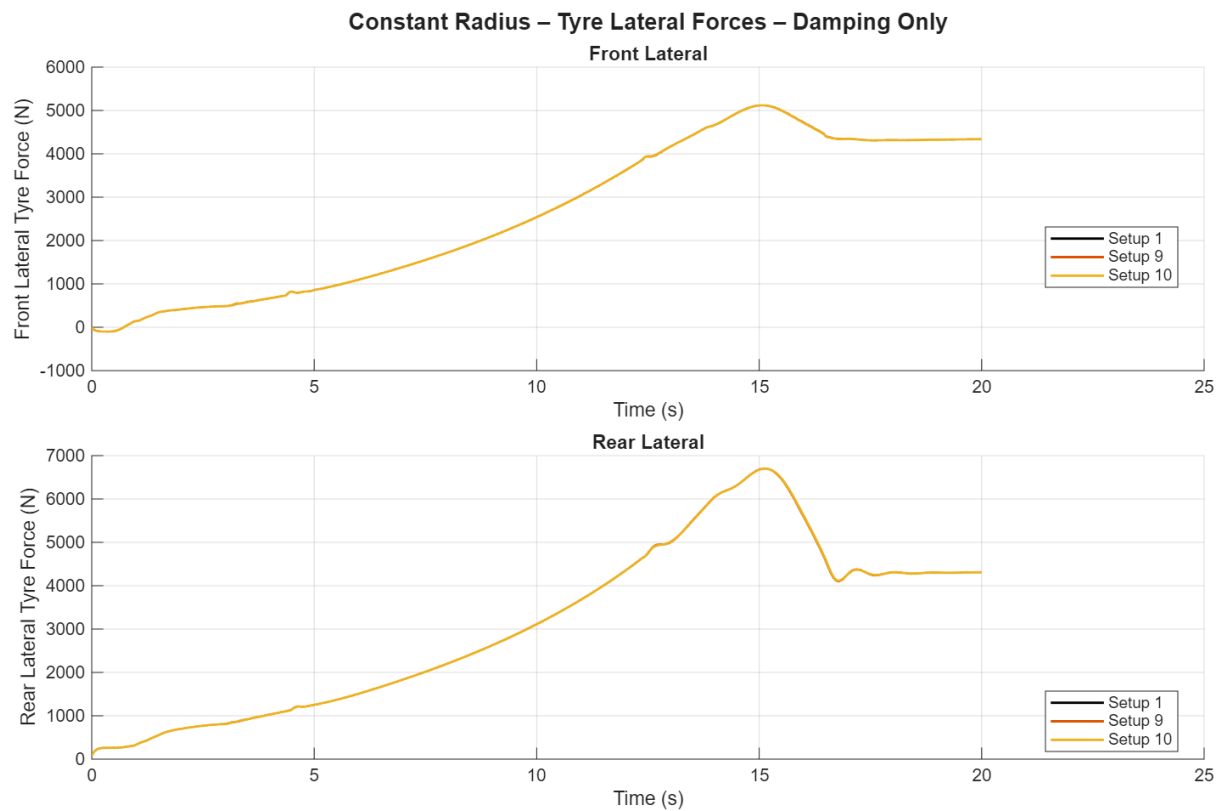
Appendix E – Constant Radius Cornering - VI-Grade Test Values

Constant Radius Cornering	Value	Units
End Time	20	Secs
Initial Velocity	30	Kph
Initial Gear	1	-
Start Time	3	Secs
Steer Duration Time	17	Secs
Entry Distance	5	Metres
Turn Radius	50	Metres
Final Velocity	108	Kph
Turn Direction	Left	-
Shift Gears	Yes	-
Steer Release	No	-

Appendix F – Constant Radius Cornering - Damper Setup Roll Graph



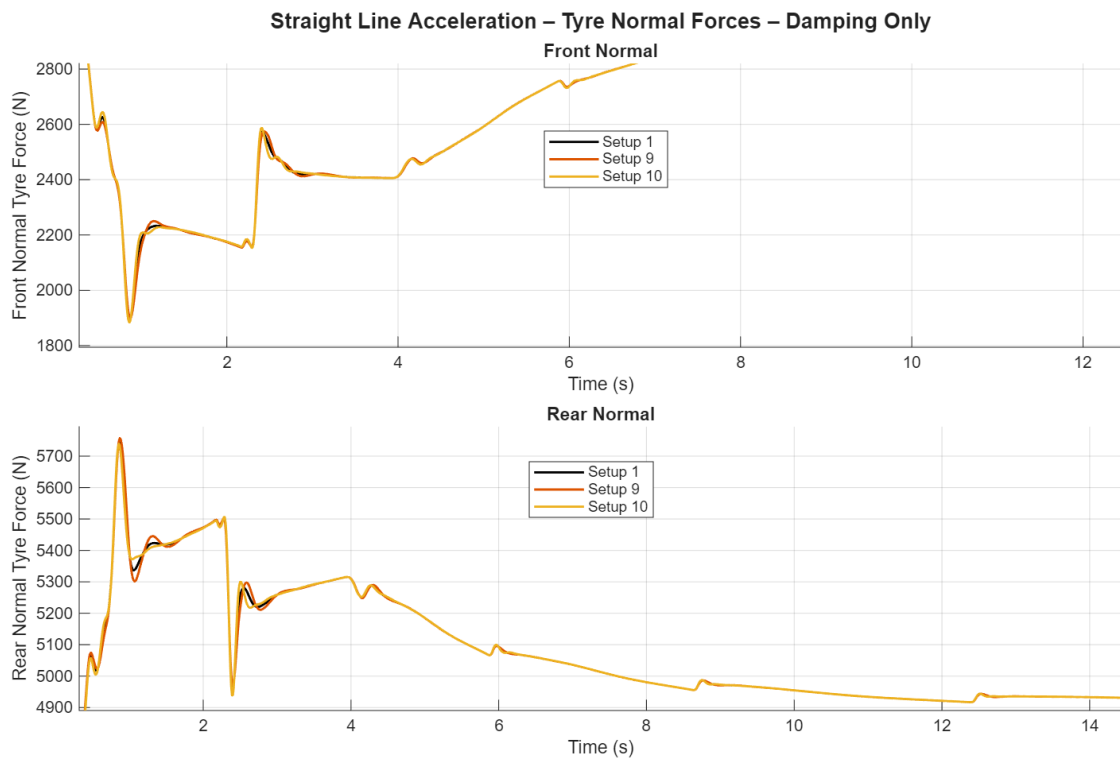
Appendix G – Constant Radius Cornering - Damper Setup Tyre Lateral Forces Graph



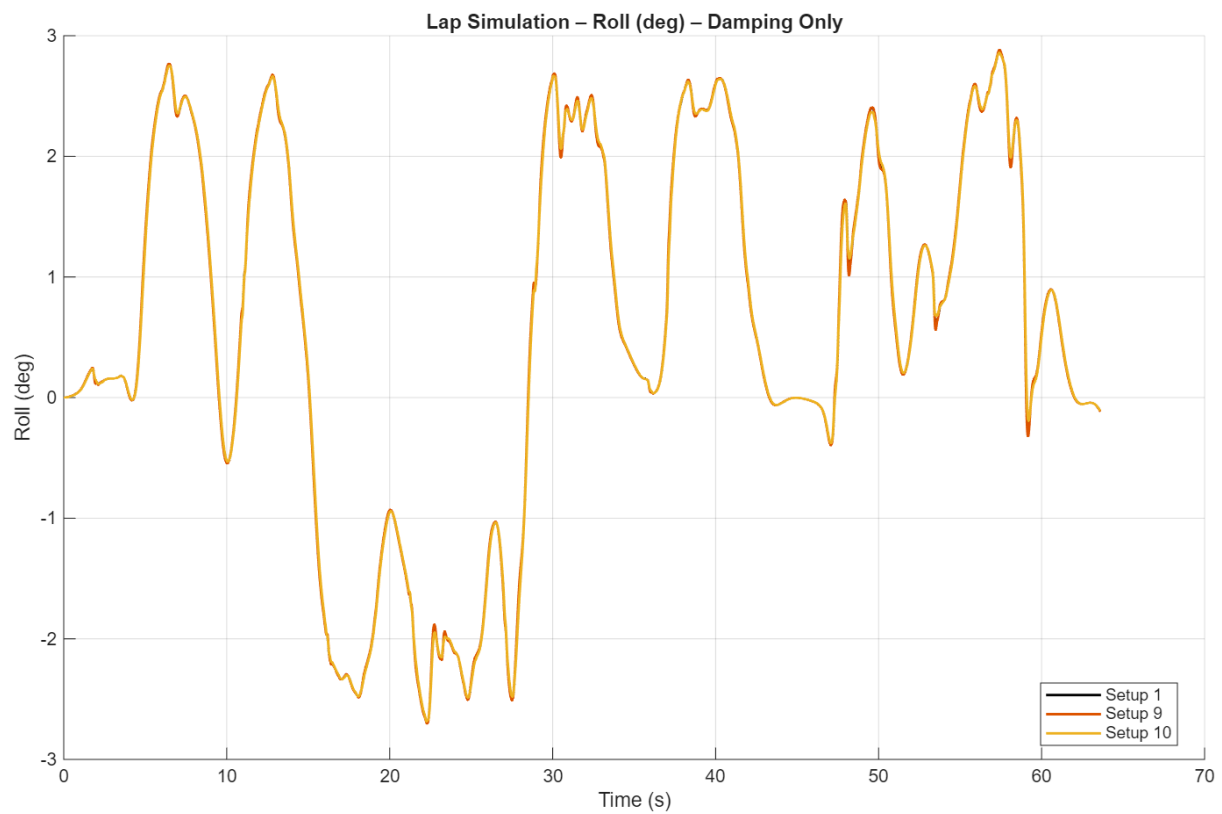
Appendix H – Straight Line Acceleration - VI-Grade Test Values

Straight Acceleration	Value	Units
End Time	20	Secs
Initial Velocity	0	Kph
Initial Gear	1	-
Open Loop Throttle	No	-
Target Acceleration	7.5	m/s^2
Straight Line	Yes	-
Shift Gears	Yes	-

Appendix I – Straight Line Acceleration – Damper Setup Tyre Normal Forces Graph



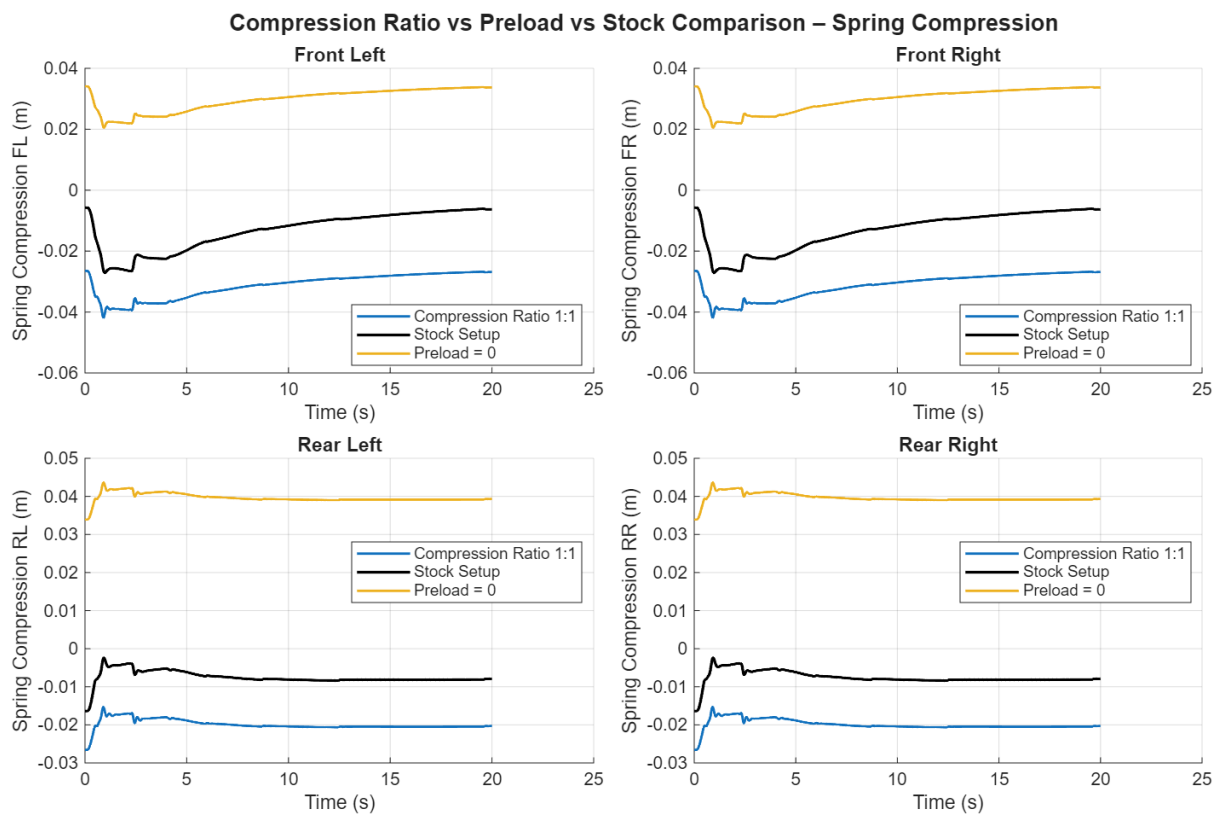
Appendix J – Lap Simulation – Damper Setup Roll Graph



Appendix K – Simulink Half Car – VI Sports Car Input Variables

Variable	VI Half Sports Car	Units
Sprung Mass	$m = 680;$	Kg
Front Unsprung Mass	$m_f = 45.85506;$	Kg
Rear Unsprung Mass	$m_r = 51.3019091;$	Kg
Front Spring Rate	$K_f = 95000;$	N/m
Rear Spring Rate	$K_r = 105000;$	N/m
Front Tyre Sprung Rate	$K_{tf} = 200000;$	N/m
Front Tyre Damping Rate	$C_{tf} = 50;$	N/m
Rear Tyre Sprung Rate	$K_{tr} = 300000;$	N/m
Rear Tyre Damping Rate	$C_{tr} = 50;$	N/m
CofG to Front Wheel Distance	$a = 1.52;$	M
CofG to Rear Wheel Distance	$b = 1.22;$	M
Pitch Inertia	$I = 1040;$	Kg/m ²
Pitch Centre Height	$h = 0.486;$	M

Appendix L – Compression Ratio & Preload Comparison – Spring Compression



Appendix M – All 7 Degree of Freedom Spring Equations

Equations of Motion

$$\begin{aligned}
 & -M_s \ddot{z}_s = k_{f1}(z_s - z_{f1} - a\theta - \beta\phi) + k_{f2}(z_s - z_{f2} - a\theta + \alpha\phi) + k_{r1}(z_s - z_{r1} + b\theta - \beta\phi) + k_{r2}(z_s - z_{r2} + b\theta + \alpha\phi) - \frac{1}{2}\rho v^2 A C_L \\
 & -I_{yf} \ddot{\theta} = b k_{r1}(z_s - z_{r1} + b\theta - \beta\phi) + b k_{r2}(z_s - z_{r2} + b\theta + \alpha\phi) - a k_{f1}(z_s - z_{f1} - a\theta - \beta\phi) - a k_{f2}(z_s - z_{f2} - a\theta + \alpha\phi) + m a x h \\
 & -I_{xf} \ddot{\phi} = \alpha k_{f1}(z_s - z_{f1} - a\theta - \beta\phi) + \alpha k_{f2}(z_s - z_{f2} + b\theta + \alpha\phi) - \beta k_{r1}(z_s - z_{r1} + b\theta - \beta\phi) - \beta k_{r2}(z_s - z_{r2} + b\theta + \alpha\phi) - m a y l \\
 & -m_{f1} \ddot{z}_{f1} = k_{f1}(z_{f1} - y_{f1}) - k_{f1}(z_s - z_{f1} - a\theta - \beta\phi) \\
 & -m_{f2} \ddot{z}_{f2} = k_{f2}(z_{f2} - y_{f2}) - k_{f2}(z_s - z_{f2} - a\theta + \alpha\phi) \\
 & -m_{r1} \ddot{z}_{r1} = k_{r1}(z_{r1} - y_{r1}) - k_{r1}(z_s - z_{r1} + b\theta - \beta\phi) \\
 & -m_{r2} \ddot{z}_{r2} = k_{r2}(z_{r2} - y_{r2}) - k_{r2}(z_s - z_{r2} + b\theta + \alpha\phi)
 \end{aligned}$$

(Figure : 7DOF Spring Equations of Motion (Tudor, T. 2021))

Appendix N – Half Car 4DOF Simulink Model

