

MSc Motorsport Engineering

Transient Vehicle Response Modelling

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Introduction

In this assignment, the general approach to modelling vehicle dynamics will be explored with a focus on the seven degree of freedom vehicle model. This model will then be derived analytically and explained. In Simulink a full seven degree of freedom vehicle model will be created based on variables taken from a Formula Ford racecar. Data from a double lane change manoeuvre and a slalom will then be collected using an rFactor-based motorsport simulator and the same Formula Ford racecar, where the G-Force will be extracted from these runs to be input into the Simulink model. The data output from these models will then be compared and the Simulink response to a variety of suspension setup changes will be analysed for the setup changes' effect on performance, safety, and driver experience.

1: 7 Degree of Freedom Vehicle Response Modelling

Transient vehicle modelling attempts to model the chassis response to an input. This form of modelling is used often in motorsport simulator software and in modelling the difference in vehicle response made by changing suspension characteristics for a given car. This includes testing the response gained from a different set of springs or damping settings, without having to fit them to the car and carry out a real test on the track. In a motorsport context, this modelling mainly allows for engineers to find an approximate suspension setup that can be used as a marker before finetuning. In this sense, 7DoF modelling speeds up the process of getting drivers acquainted with a new track or a new car before they can then work with engineers and mechanics to find maximum pace.

2: Equations for a 7 DoF Spring Mass Damper Model

For this scenario to be modelled, the governing equations must first be derived. For a 7 degree of freedom model, these equations govern the movement of the four unsprung masses, and the heave, pitch, and roll of the sprung mass. Firstly however, the extensions of each component should be individually calculated for use in the main equations, and after the equations are constructed dampers can be simply added.

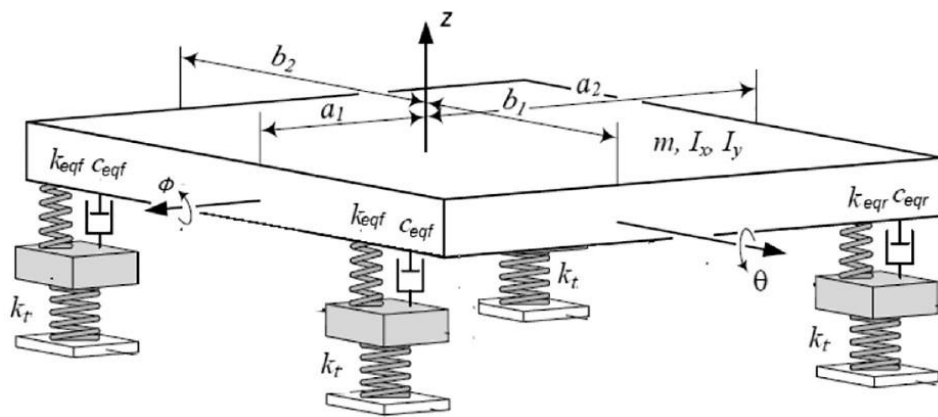


Figure 1 – Example of a 7DoF Full Car Model Diagram

2.1: Extensions

For the extensions, each component is considered with the motion expressed relative to the component, with the upward direction expressed as positive. The definitions of each variable is shown below.

k	Extension of spring
k_t	Extension of tyre from the road surface
y	Vertical position of road surface
z	Vertical of position of sprung mass
z_s	Heave
θ	Pitch
ϕ	Roll
a and b	Distance from CoG to front or rear axle
α and β	Front and rear track width

Figure 2 - Extensions Variables Definitions

Thus, for the front left sprung mass, k_{fl} , heave is known to extend the spring. Conversely, vertical displacement of the corner would compress the spring, therefore making this element a subtraction. Similarly, taking the pitch positive direction to be pitching forward, and the roll positive direction to be rolling left, both these operators become negative as well. This forms the first extension, and this logic can be extended to the other three corners.

$$k_{fl} \rightarrow z_s - z_{fl} - a\theta - \beta\phi$$

$$k_{fr} \rightarrow z_s - z_{fr} - a\theta + \alpha\phi$$

$$k_{rl} \rightarrow z_s - z_{rl} + b\theta - \beta\phi$$

$$k_{rr} \rightarrow z_s - z_{rr} + b\theta - \alpha\phi$$

For the tyre forces, the same logic is applied. When the wheel moves upwards, the tyre extends, and when the road moves upwards, the tyre is compressed.

$$k_{tfl} \rightarrow z_{fl} - y_{fl}$$

$$k_{tfr} \rightarrow z_{fr} - y_{fr}$$

$$k_{trl} \rightarrow z_{rl} - y_{rl}$$

$$k_{trr} \rightarrow z_{rr} - y_{rr}$$

2.2: Unsprung Masses

The equations for unsprung mass again operate on this logic. The tyre force pushes up on the unsprung mass, and the springs act to push downward on the wheels from above.

$$-m_{fl}\ddot{z}_{fl} = k_{tfl}(z_{fl} - y_{fl}) - k_{fl}(z_s - z_{fl} - a\theta - \beta\phi)$$

$$-m_{fr}\ddot{z}_{fr} = k_{tfr}(z_{fr} - y_{fr}) - k_{fr}(z_s - z_{fr} - a\theta + \alpha\phi)$$

$$-m_{rl}\ddot{z}_{rl} = k_{trl}(z_{rl} - y_{rl}) - k_{rl}(z_s - z_{rl} + b\theta - \beta\phi)$$

$$-m_{rr}\ddot{z}_{rr} = k_{trr}(z_{rr} - y_{rr}) - k_{rr}(z_s - z_{rr} + b\theta - \alpha\phi)$$

2.3: Sprung Mass

2.3.1: Heave

The heave equation is simple in that all four corners of the car push up on the sprung mass, meaning all elements are additions.

$$\begin{aligned} -m_s \ddot{z}_s = & k_{fl}(z_s - z_{fl} - a\theta - \beta\phi) + k_{fr}(z_s - z_{fr} - a\theta + \alpha\phi) + k_{rl}(z_s - z_{rl} + b\theta - \beta\phi) \\ & + k_{rr}(z_s - z_{rr} + b\theta - \alpha\phi) \end{aligned}$$

2.3.2: Pitch

The pitch equation multiplies each element by the terms a and b as these values effectively act as lever-arms to the extension of each corner. The final component added to the equation is the longitudinal torque generated by a longitudinal G-force input. This input is converted m/s^2 and again multiplied by the lever arm formed due to the absolute distance between the pitch centre and the centre of gravity to give a torque value.

$$\begin{aligned} -I_x \ddot{\theta} = & bk_{rl}(z_s - z_{rl} + b\theta - \beta\phi) + bk_{rr}(z_s - z_{rr} + b\theta - \alpha\phi) - ak_{fl}(z_s - z_{fl} - a\theta - \beta\phi) \\ & - ak_{fr}(z_s - z_{fr} - a\theta + \alpha\phi) + (G_{Long} \times 9.81 \times abs(h_{CoG} - h_{pc})) \end{aligned}$$

2.3.3: Roll

The roll equation is formed by the same principles as the pitch equation, but of course uses the track width values to calculate the lever arms, considers a latitudinal G-force input, and the G lever is calculated using the height of the roll centre instead of the pitch centre.

$$\begin{aligned} -I_y \ddot{\phi} = & \alpha k_{fr}(z_s - z_{fr} - a\theta + \alpha\phi) + \alpha k_{rr}(z_s - z_{rr} + b\theta - \alpha\phi) - \beta k_{fl}(z_s - z_{fl} - a\theta - \beta\phi) \\ & - \beta k_{rl}(z_s - z_{rl} + b\theta - \beta\phi) + (G_{Lat} \times 9.81 \times abs(h_{CoG} - h_{rc})) \end{aligned}$$

2.4: Adding Damping

To add dampers to these equations, the same components are added again to the equations but this time refer to c , a damping constant, rather than k , a spring constant. Furthermore, the elements within these components refer to the velocity of each element rather than the absolute position. This is due to the nature of dampers acting only to control motion within a system, compared to springs that simply exert a force based on position.

2.4.1: Unsprung Masses:

$$\begin{aligned} -m_{fl} \ddot{z}_{fl} = & k_{tfl}(z_{fl} - y_{fl}) - k_{fl}(z_s - z_{fl} - a\theta - \beta\phi) + c_{tfl}(\dot{z}_{fl} - \dot{y}_{fl}) - c_{fl}(\dot{z}_s - \dot{z}_{fl} - a\dot{\theta} - \beta\dot{\phi}) \\ -m_{fr} \ddot{z}_{fr} = & k_{tfr}(z_{fr} - y_{fr}) - k_{fr}(z_s - z_{fr} - a\theta + \alpha\phi) + c_{tfr}(\dot{z}_{fr} - \dot{y}_{fr}) - c_{fr}(\dot{z}_s - \dot{z}_{fr} - a\dot{\theta} \\ & + \alpha\dot{\phi}) \end{aligned}$$

$$-m_{rl}\ddot{z}_{rl} = k_{trl}(z_{rl} - y_{rl}) - k_{rl}(z_s - z_{rl} + b\theta - \beta\phi) + c_{trl}(\dot{z}_{rl} - \dot{y}_{rl}) - c_{rl}(\dot{z}_s - \dot{z}_{rl} + b\dot{\theta} - \beta\dot{\phi})$$

$$-m_{rr}\ddot{z}_{rr} = k_{trr}(z_{rr} - y_{rr}) - k_{rr}(z_s - z_{rr} + b\theta - \alpha\phi) + c_{trr}(\dot{z}_{rr} - \dot{y}_{rr}) - c_{rr}(\dot{z}_s - \dot{z}_{rr} + b\dot{\theta} - \alpha\dot{\phi})$$

2.4.2: Sprung Mass

Heave

$$\begin{aligned} -m_s\ddot{z}_s = & k_{fl}(z_s - z_{fl} - a\theta - \beta\phi) + k_{fr}(z_s - z_{fr} - a\theta + \alpha\phi) + k_{rl}(z_s - z_{rl} + b\theta - \beta\phi) \\ & + k_{rr}(z_s - z_{rr} + b\theta - \alpha\phi) + c_{fl}(\dot{z}_s - \dot{z}_{fl} - a\dot{\theta} - \beta\dot{\phi}) + c_{fr}(\dot{z}_s - \dot{z}_{fr} - a\dot{\theta} + \alpha\dot{\phi}) \\ & + c_{rl}(\dot{z}_s - \dot{z}_{rl} + b\dot{\theta} - \beta\dot{\phi}) + c_{rr}(\dot{z}_s - \dot{z}_{rr} + b\dot{\theta} - \alpha\dot{\phi}) \end{aligned}$$

Pitch

$$\begin{aligned} -I_x\ddot{\theta} = & bk_{rl}(z_s - z_{rl} + b\theta - \beta\phi) + bk_{rr}(z_s - z_{rr} + b\theta - \alpha\phi) - ak_{fl}(z_s - z_{fl} - a\theta - \beta\phi) \\ & - ak_{fr}(z_s - z_{fr} - a\theta + \alpha\phi) + bc_{rl}(\dot{z}_s - \dot{z}_{rl} + b\dot{\theta} - \beta\dot{\phi}) + bc_{rr}(\dot{z}_s - \dot{z}_{rr} + b\dot{\theta} \\ & - \alpha\dot{\phi}) - ac_{fl}(\dot{z}_s - \dot{z}_{fl} - a\dot{\theta} - \beta\dot{\phi}) - ac_{fr}(\dot{z}_s - \dot{z}_{fr} - a\dot{\theta} + \alpha\dot{\phi}) \\ & + (G_{Long} \times 9.81 \times m_s \times \text{abs}(h_{CoG} - h_{pc})) \end{aligned}$$

Roll

$$\begin{aligned} -I_y\ddot{\phi} = & \alpha k_{fr}(z_s - z_{fr} - a\theta + \alpha\phi) + \alpha k_{rr}(z_s - z_{rr} + b\theta - \alpha\phi) - \beta k_{fl}(z_s - z_{fl} - a\theta - \beta\phi) \\ & - \beta k_{rl}(z_s - z_{rl} + b\theta - \beta\phi) + \alpha c_{fr}(\dot{z}_s - \dot{z}_{fr} - a\dot{\theta} + \alpha\dot{\phi}) + \alpha c_{rr}(\dot{z}_s - \dot{z}_{rr} + b\dot{\theta} \\ & - \alpha\dot{\phi}) - \beta c_{fl}(\dot{z}_s - \dot{z}_{fl} - a\dot{\theta} - \beta\dot{\phi}) - \beta c_{rl}(\dot{z}_s - \dot{z}_{rl} + b\dot{\theta} - \beta\dot{\phi}) \\ & + (G_{Lat} \times 9.81 \times m_s \times \text{abs}(h_{CoG} - h_{rc})) \end{aligned}$$

3: Developing a 7 DoF Simulink Model

3.1: Simulation Parameters

The parameters used for the Simulink model were based on a Formula Ford race car. These values are shown below.

Vehicle Mass (kg)	MT	514
Front Corner Unsprung Mass (kg)	M_{usf}	22
Rear Corner Unsprung Mass (kg)	M_{usr}	22
Vehicle Sprung Mass (kg)	M_s	426
Centre of Gravity Height (m)	CoG	0.277
Weight Split F:R (%)	-	45:55
Wheelbase (m)	W	2.443
Front Track (m)	T_f	1.62
Rear Track (m)	T_r	1.57
Front Tyre Rolling Radius (m)	r_f	0.2667
Rear Tyre Rolling Radius (m)	r_r	0.2794
Front Motion Ratio	MR_f	1
Rear Motion Ratio	MR_r	1
Front Default Spring Rate (N/mm)	K_{sf}	18
Rear Default Spring Rate (N/mm)	K_{sr}	16
Front Roll Centre Height (mm)	h_f	0.9
Rear Roll Centre Height (mm)	h_r	59
Front Slow Bump	N/m/s	1660
Front Fast Bump	N/m/s	511
Front Slow Rebound	N/m/s	1915
Front Fast Rebound	N/m/s	511
Rear Slow Bump	N/m/s	1477
Rear Fast Bump	N/m/s	537
Rear Slow Rebound	N/m/s	1612
Rear Fast Rebound	N/m/s	537

Figure 3 - Formula Ford based Simulink model Parameters

3.2: Assumptions

A number of assumptions have been made in the development of the Simulink model.

- The effect of anti-roll bars have not been modelled, which reduces the real-world accuracy of the response. This will however be mirrored when the Simulink response is compared to simulator data, as the anti-roll bars can be disconnected in the sim setup.
- The damping response is selected based on four given values of bump and rebound in the fast and slow conditions, and so is not based on a look-up table. This reduces the accuracy of the damping simulation of the model, and should be considered when analysing the results.

- The height of the pitch centre was not given in the Formula Ford dataset, so the G-Force term of the pitch equation holds inaccuracy due to this value being estimated to calculate the length of the lever arm.
- The road surface was assumed to be completely flat, so no y input was added to the model. Given the road surface in the simulator data was also devoid of bumps noticeable to the driver, this should cause little inaccuracy but should nevertheless be considered.
- Front and rear roll centres are averaged to calculate an overall roll, and such the roll response is not as accurate.
- The tyre data is approximated based on an Avon tyre for a Formula Ford, and so is a good estimate but may not be exactly accurate.

3.3: Model Construction

The bulk of constructing the model comes from translating the equations laid out in Section 2 into Simulink. Some further details come from adding 'From' and 'Goto' blocks so that values can be pulled into other equations, however other additions are commented on below individually.

3.3.2: Unsprung Masses

One of four unsprung masses equations is shown below, with the other three being identically constructed. The contact patch force of each corner is also logged as a simple sum of the tire forces at play.

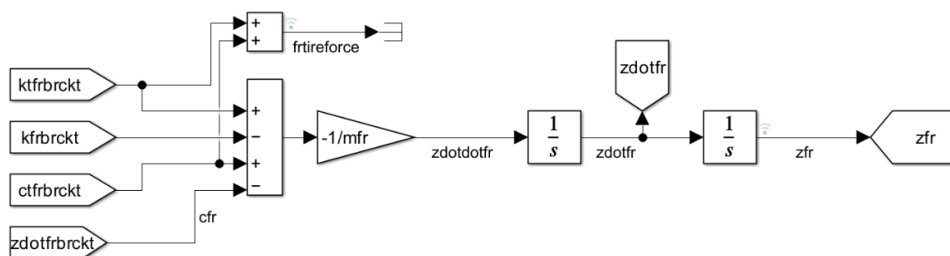


Figure 4 - Unsprung mass subsystem

3.3.3: Sprung Mass

3.3.3.1: Heave

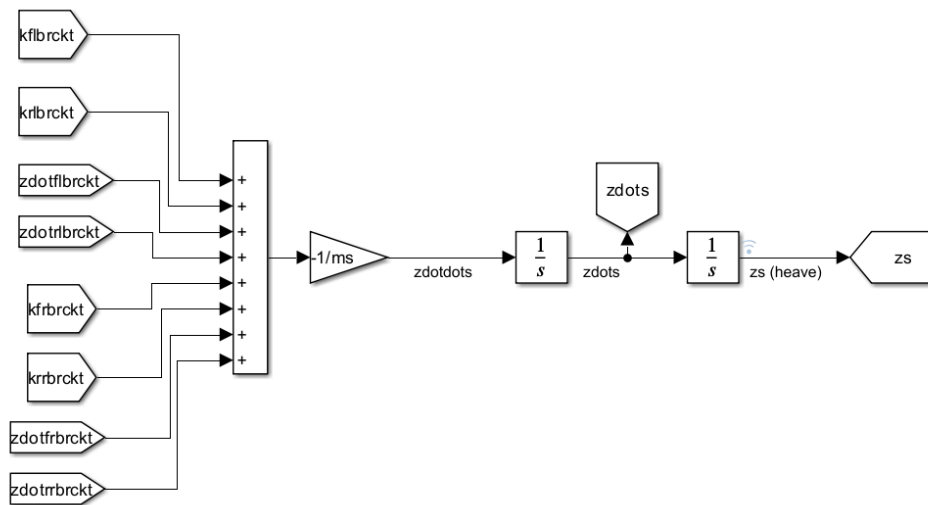


Figure 5 - Heave subsystem

3.3.3.2: Roll

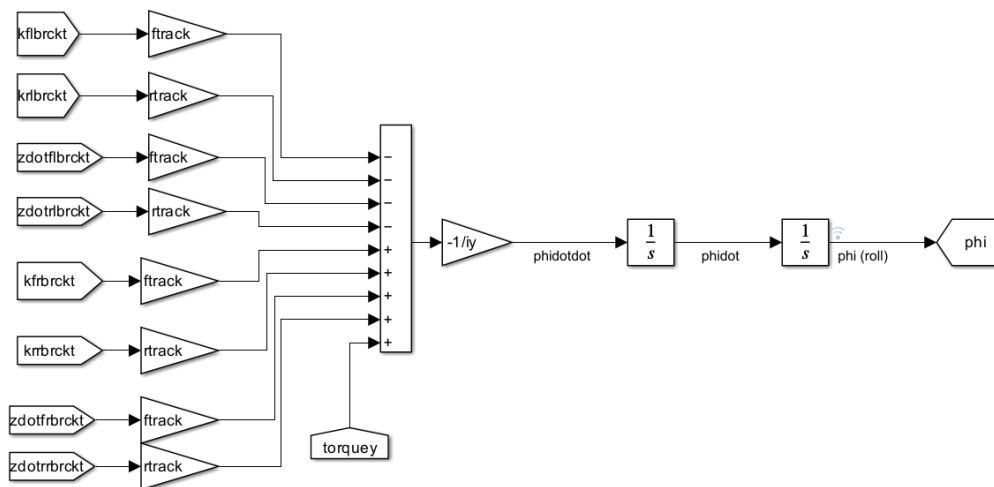


Figure 6 - Roll subsystem

3.3.3.3: Pitch

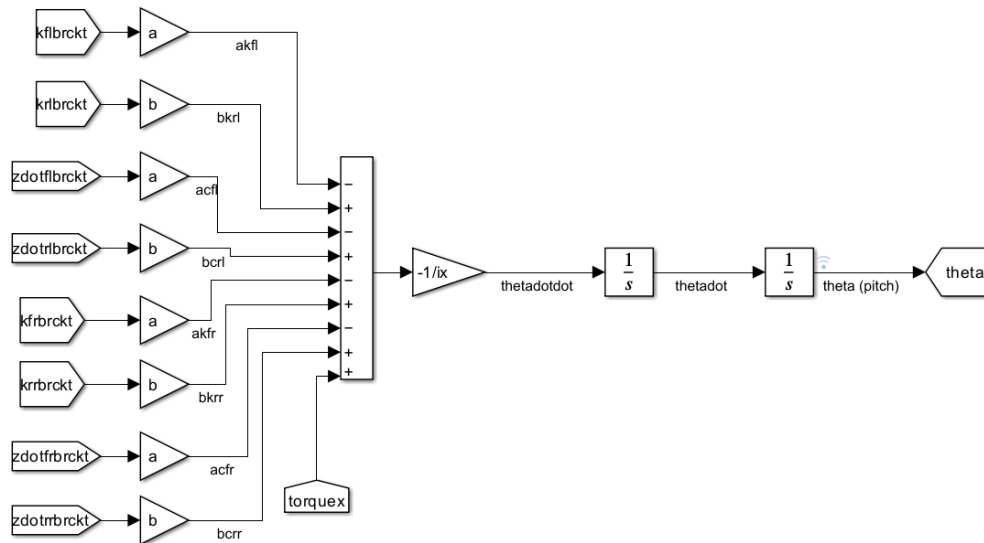


Figure 7 - Pitch subsystem

3.3.4: Auxiliary

3.3.4.1: Damper Logic

The damper subsystems serve to decide which of four damping coefficient to use, depending on the value of the velocity of the damper. The ‘if’ block contains the four possible conditions for if the velocity is above or below zero, determining bump or rebound, and if the magnitude of the velocity is above or below 15m/s (Campbell-Brennan, 2020), determining if the damper gives a fast or slow response. These conditions then activate a smaller subsystem containing a simple gain block that applies the correct damping value, which is then output to the rest of the model.

There are two of these subsystems, each for front and rear, which refer to different variables for front and rear damping, allowing for the tuning of these variables individually.

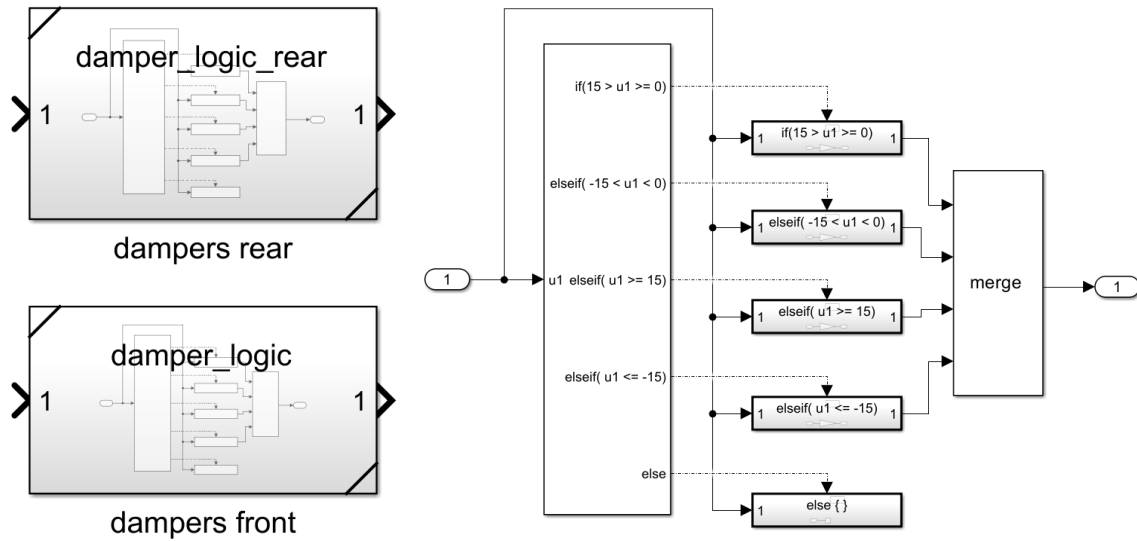


Figure 8 - Damper logic subsystems

3.3.4.2: Extensions

In order to clean up the readability and make troubleshooting the larger sprung mass equations easier, the extensions were calculated in separate subsystems within the model, using ‘Goto’ and ‘From’ blocks to transfer the output to the larger equations. These subsystems are shown below, and are translations of the extensions previously defined analytically.

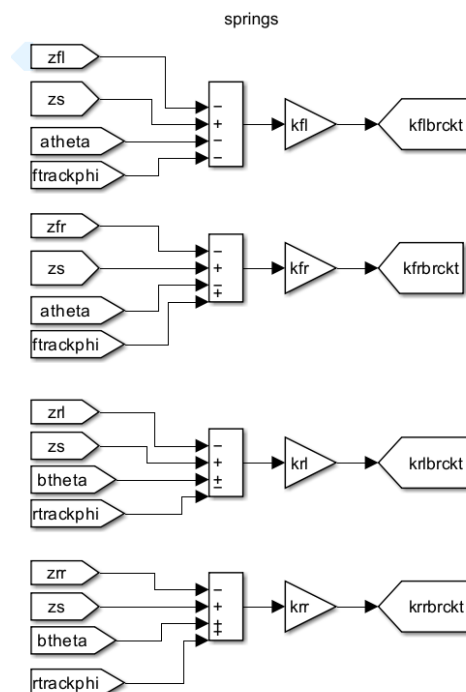


Figure 9 - Ks subsystem

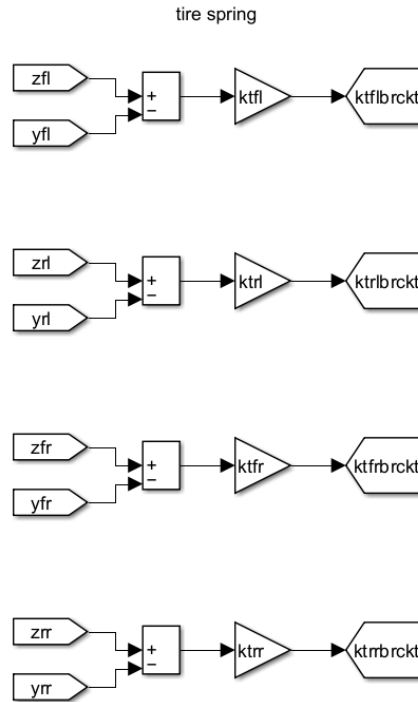


Figure 10 - K_s subsystem

The damping coefficient is chosen via the aforementioned damper logic subsystem, which here is used as a pair of referenced subsystems that simply refer to the same subsystem rather than requiring changes to be made to all identical subsystems.

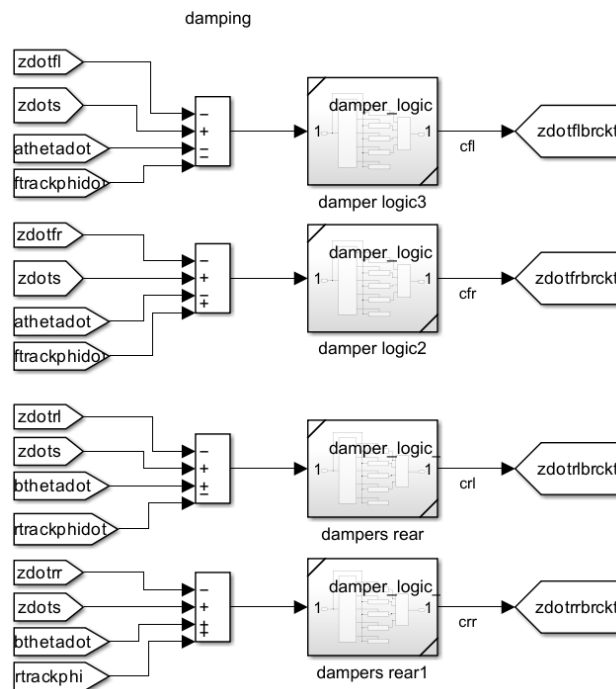


Figure 11 - C_s subsystem

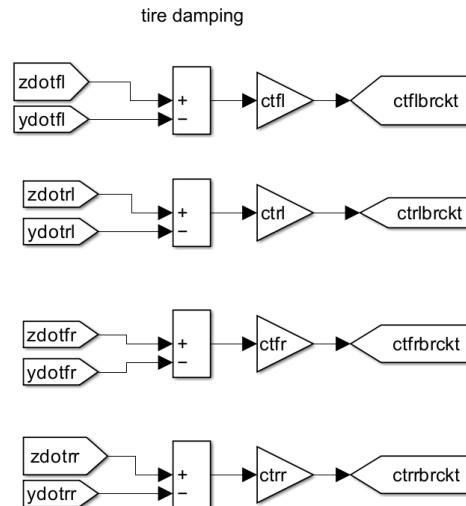


Figure 12 - C_s subsystem

3.3.4.3: G Calculations

The G calculations simply calculate the final term of the pitch and roll equations, by converting the G-force into an acceleration and multiplying that value by the lever created by the difference in position between the roll and pitch centres and the centre of gravity.

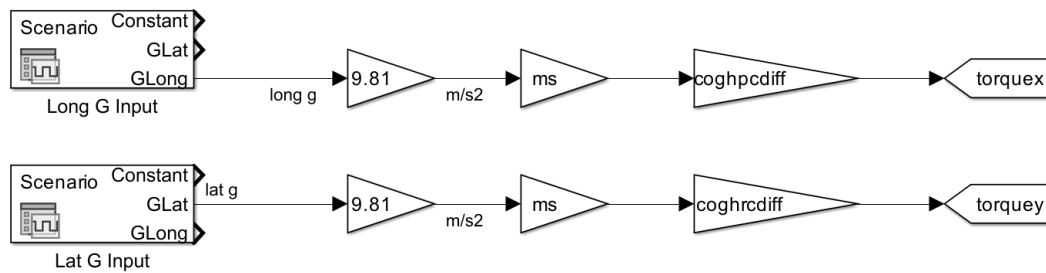


Figure 13 – G-force subsystem

3.3.4.4: Pitch and Roll Calculations

In order to carry out the other calculations, a number of values are required based on the phi and theta values. These are simply calculated below.

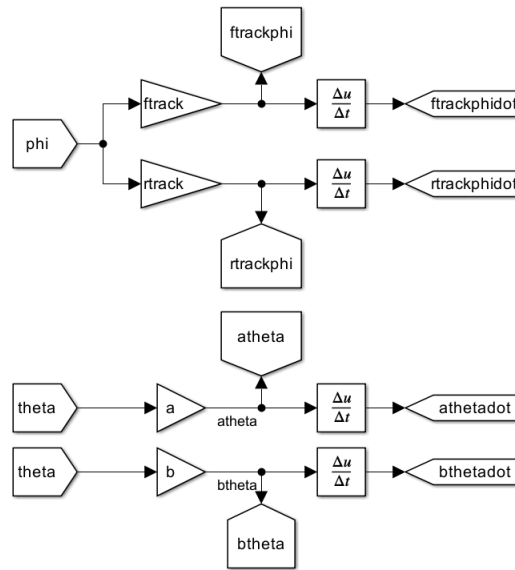


Figure 14 - Pitch and roll subsystem

3.3.4.5: Y Input Calculations

The y input subsystem is simply capable of supplying a y input to each corner of the car, with delay blocks to more finely control this input. A bump can be selected to test the response to see if the model acts as expected, and a flat input can be selected to isolate the G response. Furthermore, the rolling radius of the front and rear tyres is added to the y value, simulating a difference in ride height on the front and rear axle created by this difference. This results in the model pitching forward, which is mirrored in the simulator data. It also, unfortunately, results in a bump artifact at the beginning of the data trace as the model settles, due to the fact that the y values at $time = 0$ are equal to 0. This can be ignored inconsequentially and thus will be.



In order to check the model was responding correctly, a simple bump input was added and the model was troubleshooted until the response showed no errors. This was a long process, as certain systems had to be rebuilt in order to function at all. In particular, the damper logic had to be rebuilt to use an 'If' block rather than using switches, as these did not work, and it was also found that one of the signs in a 'Sum' block in one of the extensions was incorrect, meaning both sides of the front axle was calculating roll in the same direction. However, this was fixed, and it can now be seen that the response of the sprung and unsprung masses act in the expected direction. The results of this troubleshooting are shown below.

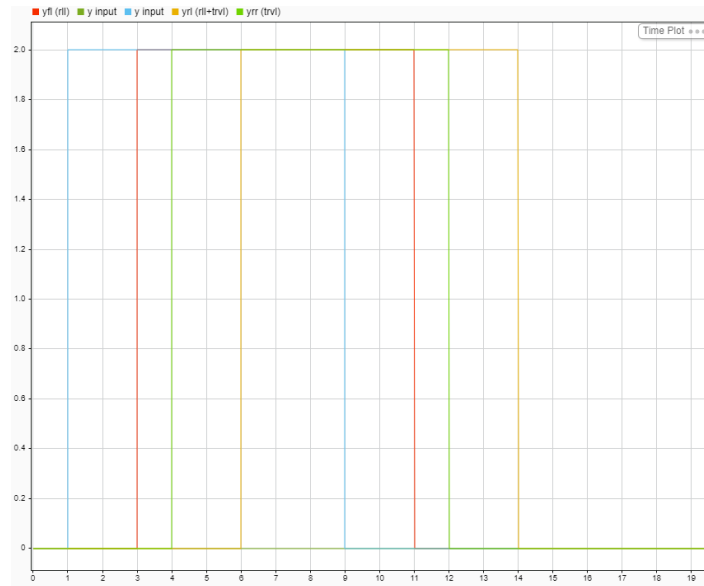


Figure 16 - Y input at each corner of the car for the model check

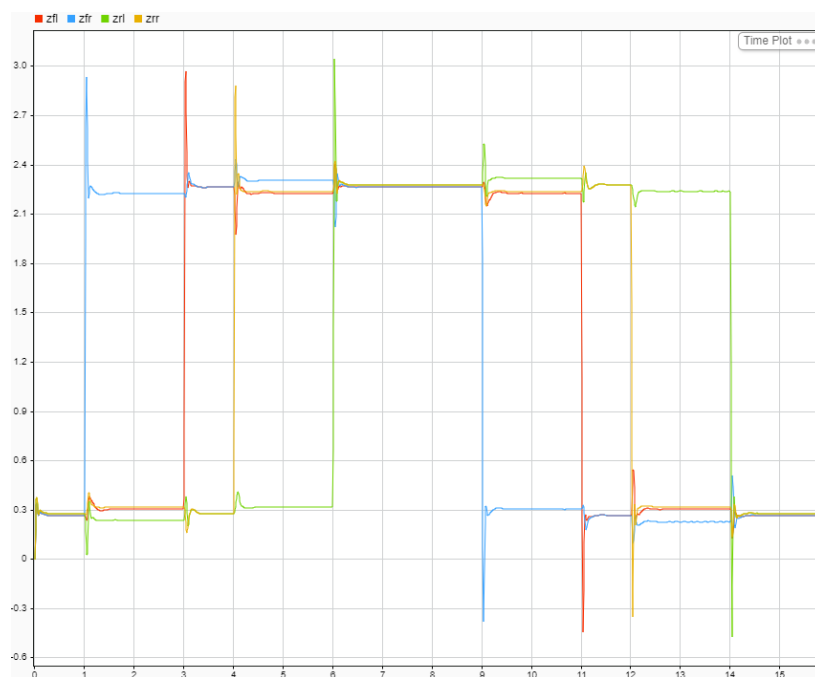


Figure 17 – Z response to y input at each corner of the car

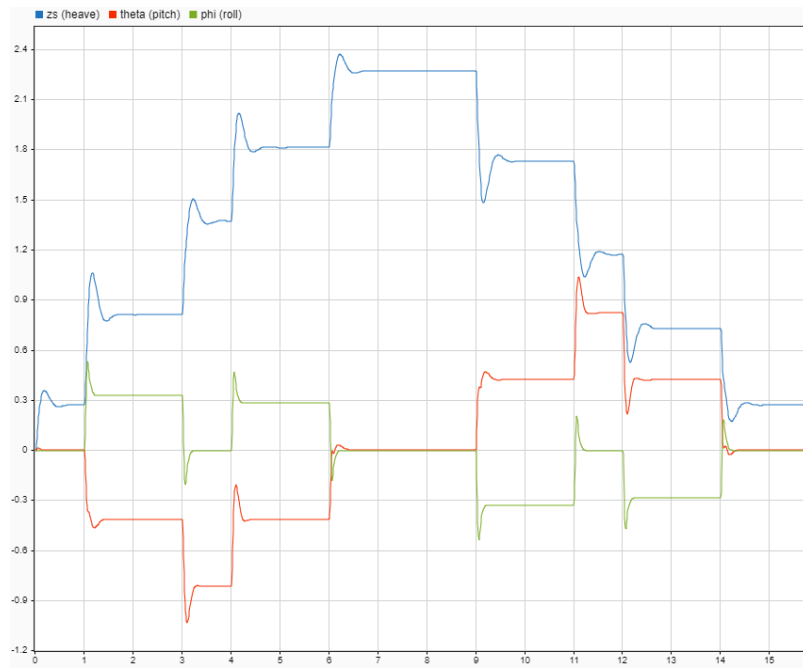


Figure 18 - Heave, pitch, and roll response of the sprung mass to y input

4: Comparing Transient Response of Simulink Model and Simulator Data

The latitudinal and longitudinal G-force data was pulled from the simulator data in MoTec, and input into the Simulink model to compare the chassis response of the car. A series of suspension changes were also made to assess the effect of this on the response.

It is worth noting that the simulator's force feedback settings made it difficult to keep a smooth steering input, as can be seen below. This made the roll response of the vehicle more jittery than it would be in real life, however this of course is mirrored in the Simulink output data given that it affects G-force.

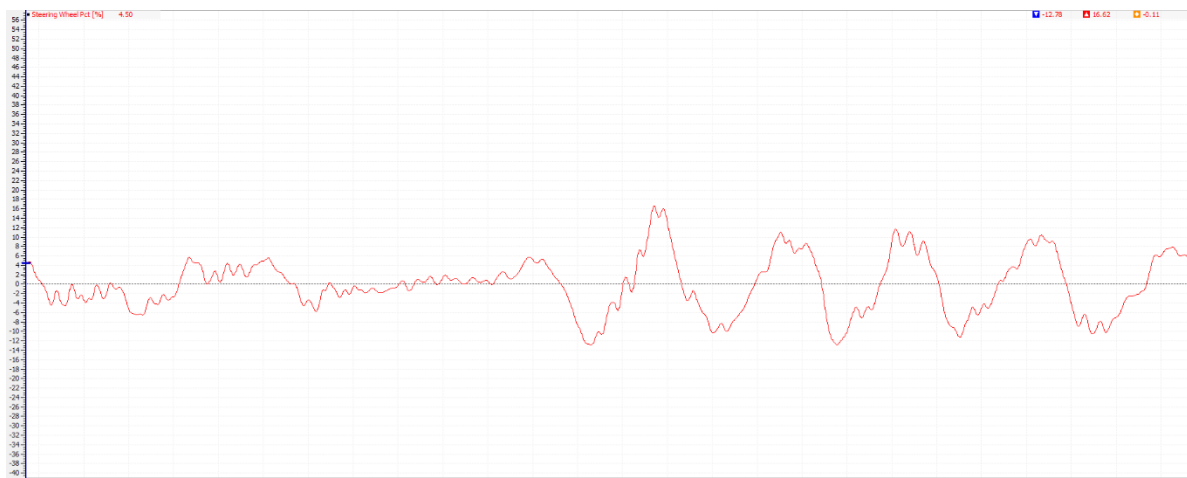


Figure 19 - Jittery simulator steering input

The exact suspension changes made are shown in the appendices.

4.1: Differences in the Model's Approach

The 7 degree of freedom model built in Simulink and the simulation carried out in the RFactor-based simulator approach transient vehicle modelling in different ways. The simulator carries out its physics simulation by a multibody chassis and suspensions simulation polling at 800Hz. This simulation resolves forces to determine the transient state of the vehicle and the force at each contact patch, where the tyres are modelled through a Pacejka-based model to determine the grip condition (rFpro, 2009). This calculation can also be based on the available grip derived from tyre compound and temperature, track surface and temperature, and even variables such as wind condition. Conversely, the 7 degree of freedom model developed in Simulink resolves the system equations to find the transient state of the suspension and chassis. This approach has a generally lower level of fidelity, as it cannot not take into account the plethora of variables available to the simulator. Furthermore however, movements such as yaw or translational movement in the x and y directions are also not accounted for in only a 7DoF model. Moreover, the simulator's method is able to take into account more specific suspension and

chassis geometry of the given vehicle, such as vehicle ride height and weight distribution, in resolving the chassis response of the vehicle.

4.2: Standard response comparison

Firstly, a run with standard settings was carried out, with which the results from a variety of suspension settings could be compared.

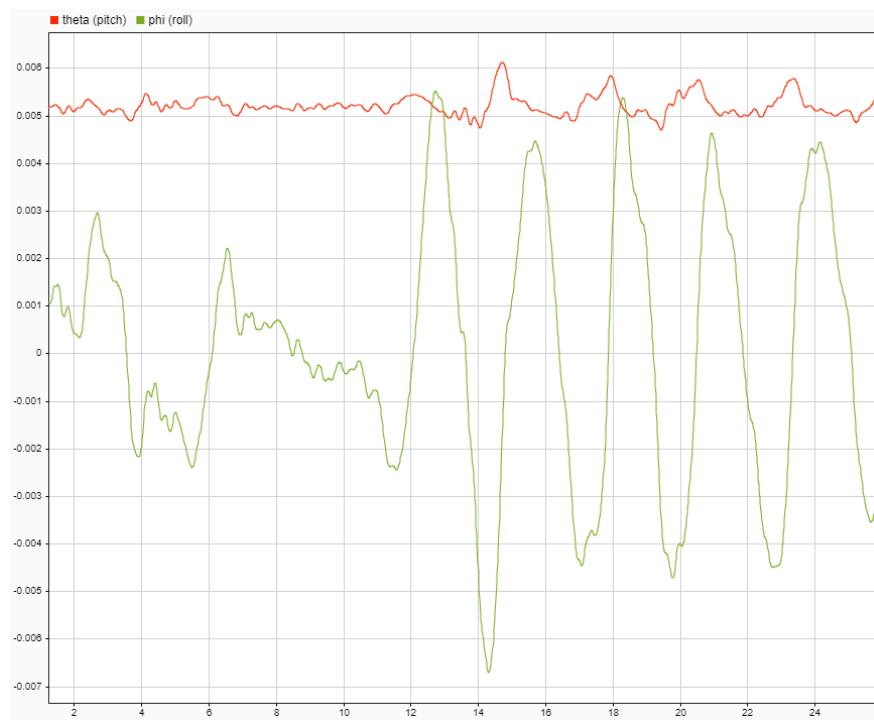


Figure 20 - Pitch and roll Simulink response

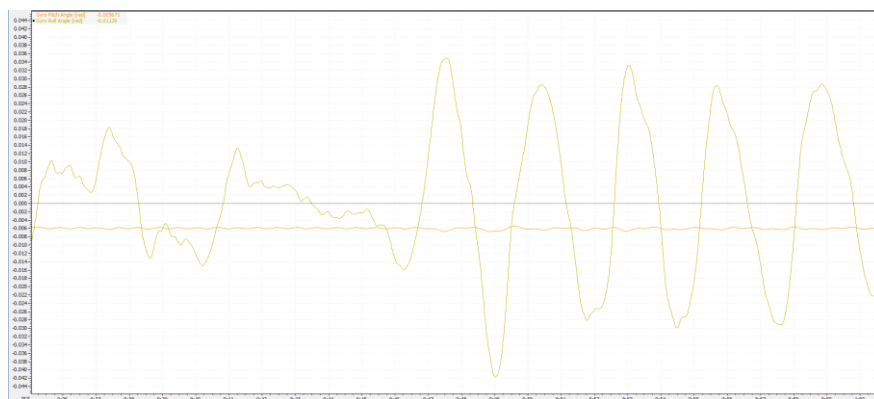


Figure 21 - Pitch and roll Simulator response

The roll response from the Simulink model is similar in shape to the one gathered from the simulator data. One can clearly see the effect of the double lane change manoeuvre until before $t = 12$, and the slalom after this point. The two models do deviate however, most notably in the magnitude of the roll response. The Simulink model's roll max value is -0.0067rad , whereas the max value of the simulator

roll is -0.042rad , a difference of 145%. This is likely due to the differences in the modelling approach outlined previously, as the Simulink model has been tested exhaustively for errors.

4.3: Spring stiffness study

The spring stiffness was changed from the standard 17000N/m , to the minimum 8000N/m , and the maximum 27000N/m . The chassis roll and pitch response is compared below, with the larger amplitude of variation being for the softer spring rates.

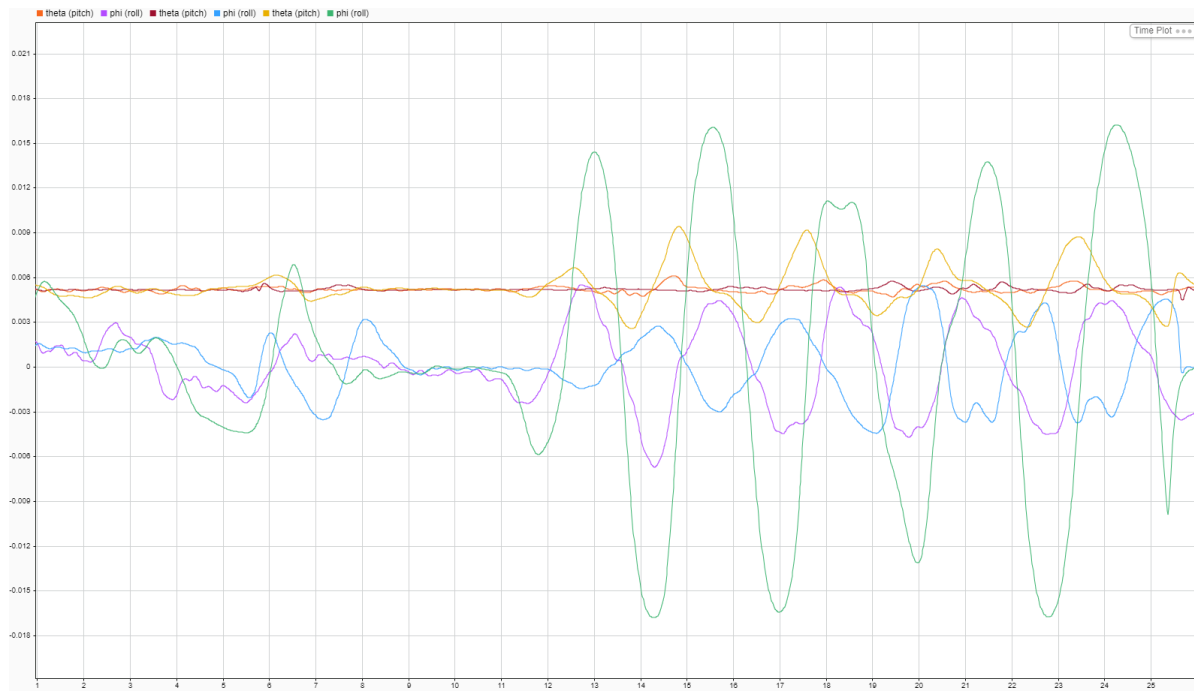


Figure 22 - Roll and Pitch response from spring variance

One can clearly see that as the spring stiffness is lessened, the pitch and roll response is exaggerated. This results in increased weight transfer, which makes the car less agile and less responsive to a change in direction. This change in response also reduces the driver's confidence in the car to change direction, which compounds to make the car slower and less safe. This can be seen in an increase in the pitch response when springs are softer, resulting from variation in speed.

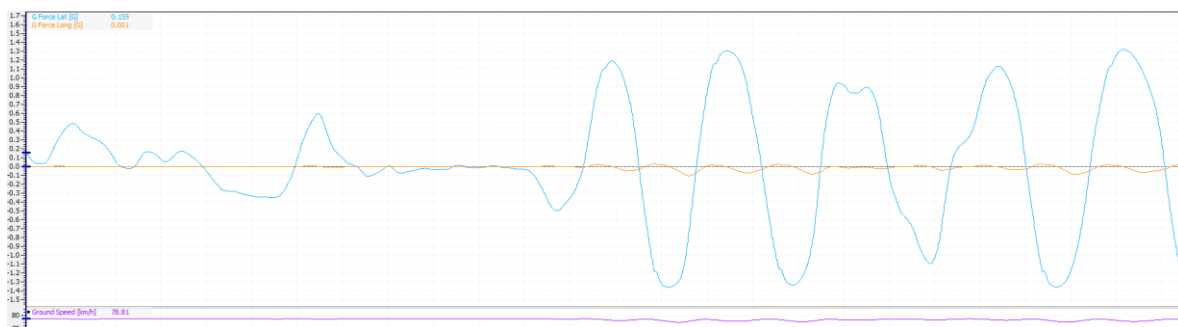


Figure 23 - G response for the soft spring setting, including the ground speed trace

The pink speed trace can be seen dipping slightly where the lateral G is at its highest, creating a small longitudinal G response that produced the pitch variation seen in the first plot. This loss of speed can be likely attributed to the loss of stability caused by a softer setup, thus showing a loss of driver confidence and overall performance.

4.4: Damper stiffness study

The damper settings were changed to a maximum softness position, and compared to a standard and maximum stiffness position.

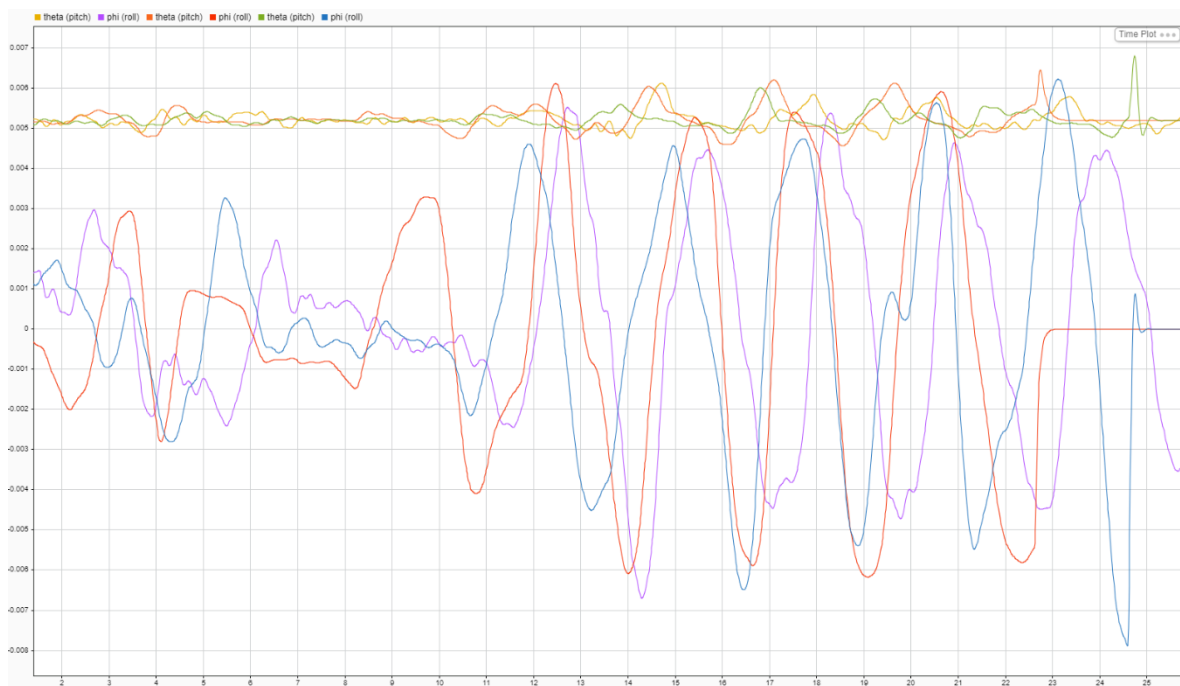


Figure 24 - Roll and Pitch response from damper stiffness variance

The most notable observation that can be made by changing damper settings is that these changes produce far less visible change than changing the spring rates. This is likely due to the breadth of adjustment available for dampers compared to springs in the setup of the Formula Ford. As expressed in Appendix A, the spring rates can be changed by a total of 19000N/m, whereas the damping values may only be changed by 800N/m/s. This is likely to produce a smaller difference in response, and thus it may simply be that the differences are as a result of the differences in how the test course was driven.

Damping has its greatest effect on vehicle dynamics when dealing with bumps and road irregularities, as the stiffness of the dampers directly correlates with the time it takes for the vehicle body to settle. Therefore, the effect of this change would be easier to see in

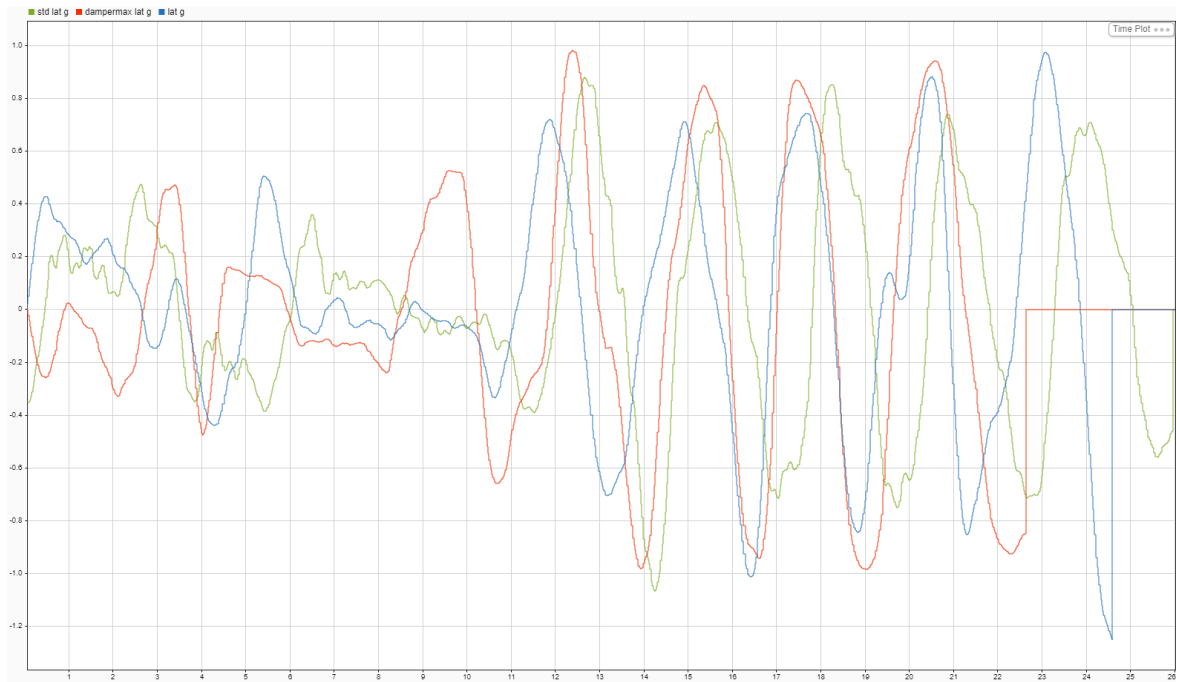


Figure 25 - Lateral G produced in the runs for different damping stiffnesses

Comparing the above lateral G force plot with the previous roll trace, it can be seen that the bulk of the variation between runs can be attributed to the different way in which the runs were driven, in contrast to the different ways in which the chassis responds to the G force input. This could be mitigated in any future tests by having the driving carried out by a driver more used to the simulator who would be able to replicate the manoeuvre more consistently, or by carrying out multiple runs for each setup and taking the best one.

4.5: Front v Rear

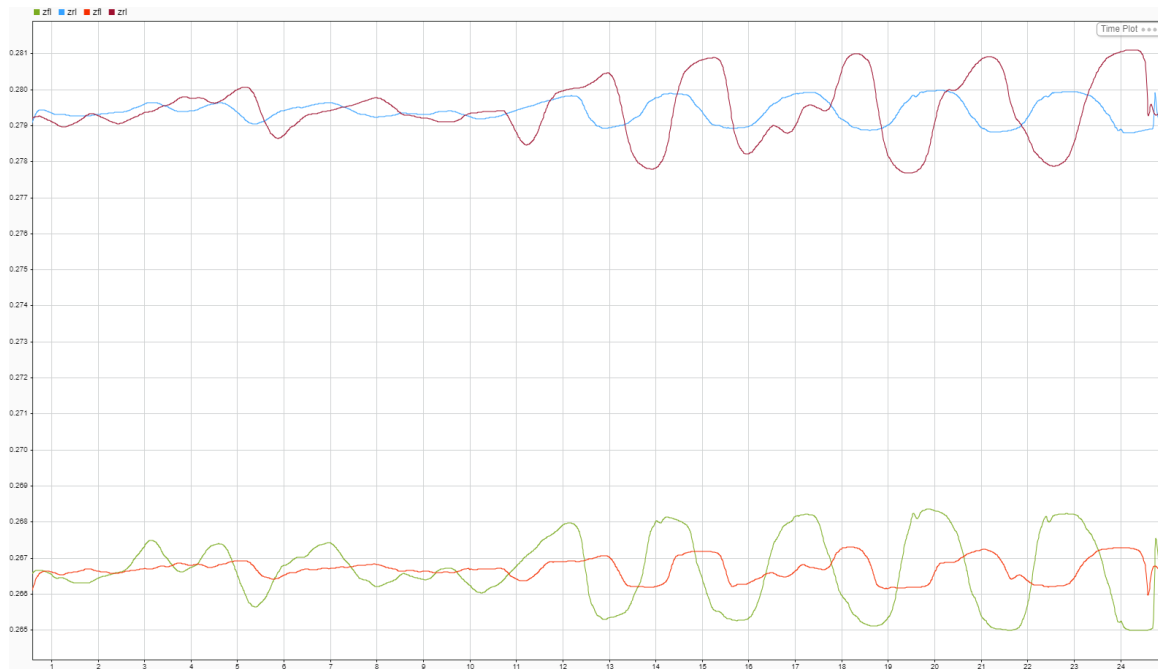


Figure 26 - Front and rear left corner response of the Front v Rear study

In the above Simulink trace, the top traces display the rear left response, and the bottom traces display the front left response. The blue and green denote Run 19, running the front axle at maximum and the rear at minimum, and the orange and red denote Run 20, with the front at minimum and rear at maximum. Here, it can be seen that the z position of the unsprung mass varies again with greater amplitude where spring stiffness is decreased. This type of movement is likely one that would be better found by measuring yaw and slip angle, as an increased weight transfer over the front or rear axle is likely to cause understeer or oversteer respectively, leading to a loss of grip. This reduces performance and safety, especially oversteer in the case of commercial vehicles as the average driver is worse at counteracting oversteer than understeer. In a motorsport application, generally oversteer is favoured, although to what extent usually comes down to individual driver preference.

Conclusion

In conclusion, the general approach to modelling vehicle dynamics was explored with a focus on the seven degree of freedom vehicle model. This model was then derived analytically and created in Simulink, based on variables taken from a Formula Ford racecar. Data from a double lane change manoeuvre and a slalom was then collected using an rFactor-based motorsport simulator and the same Formula Ford racecar, where the G-Force was extracted from these runs to be input into the Simulink model. The data output from these models were then compared and the Simulink response to a variety of suspension setup changes were analysed for the setup changes' effect on performance, safety, and driver experience.

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Appendix A: Available Suspension Settings

Front Suspension:

Component	Minimum Value	Step Size	No. of Steps
Springs	8000N/m	1000N/m	19
Slow Bump	1245N/m/s	100N/m/s	8
Fast Bump	383.25N/m/s	100N/m/s	8
Slow Rebound	1436.25N/m/s	100N/m/s	8
Fast Rebound	383.25N/m/s	100N/m/s	8

Rear Suspension:

Component	Minimum Value	Step Size	No. of Steps
Springs	8000N/m	1000N/m	19
Slow Bump	1107.75N/m/s	100N/m/s	8
Fast Bump	402.65N/m/s	100N/m/s	8
Slow Rebound	1209N/m/s	100N/m/s	8
Fast Rebound	402.75N/m/s	100N/m/s	8