



An Investigation into the Characteristics of Dampers and Their Effect on Vehicle Handling



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1) Abstract

Race car performance is dependent on a lot of different elements such as the engine, chassis, tyres, suspension system, the track, aerodynamics and the driver. Due to the complexity of these parameters having evolved throughout the years, today the minor adjustments make a big difference (Katz, 2006). Having a well-balanced suspension system is crucial to achieve success around a racetrack. One of the most important elements of the suspension system are the damping coefficients. Having the perfect damping coefficients is crucial and is achieved with extensive testing to ensure the perfect combination of bump and rebound settings. Having these combinations on a race car allows the suspension system to keep all four tyres in contact with the road, over kerbs, bumps and especially during cornering. Perfecting these combinations will enhance the available grip around the whole track, making the car faster (Kasprzak, 2014). Not only being the fastest around one circuit, the key to a successful race team is that the race engineers can adapt the damping coefficients to the circuit, weather conditions and driver preference. The potential performance of dampers at a wide range of different variables can be unlocked by finding the optimal combination of settings and components for the specific track (DemonTweeks, 2014). Extensive testing is required to achieve the best suspension setup on a race car hence why this project will investigate the effect of changing damping coefficients using a BSI ISO standard dynamic vehicle test along with a circuit test around Silverstone circuit. Development of the damper will continue if motor vehicles continue to evolve. The introduction of electric vehicles with their increased weight and torque values creating different demands will see a new era in damper evolution. Well found science-based knowledge will continue to drive their development for the future of the automotive and motorsport industry.



2) Introduction

The project will examine how different rebound and bump damping coefficients effect the handling characteristics of the vehicle by using different methods of testing and analysis. The vehicles in question include a Sports 2000 Prototype Race Car which is equipped with a pushrod style suspension system along with Penske 8300 series dampers and a GP2 car for the double lane change manoeuvre.

The first phase of the project will include stripping the Sports 2000 car's suspension components, weighing and measuring everything possible and then using the measurements acquired from the workshop and inputting them into mathematical equations to find more information about the characteristics of the car. Once this information is gathered it can be used to conclude statements such as the suspension frequencies and damping ratio. The dampers will also be tested on the damper dynamometer in the workshop to create force vs velocity graphs for all settings available, this will allow for the damping coefficients to be found and will also give an idea of what the changes do to the forces of the damper per increment of velocity.

The next step of the project will be to gather data from the real car from Silverstone and repeat the tests with the same damping coefficients on the simulator, this will give information about both real life data and simulator data to compare and contrast to assess the accuracy of the simulator before moving on to the second test. The two data sets will then be compared to see if the matched damping coefficients have the same effects on the handling characteristics of the car. The channels that will be used to examine the behaviour of both data sets on Motec will be the lateral acceleration, longitudinal acceleration, understeer / oversteer and the lap times.

The last phase of the project will be to test different damping coefficients on the GP2 car using the double lane change manoeuvre built on Bob's Track Builder Pro. The track will be built to specification for the GP2 vehicle and therefore will follow BSI ISO 3888 standards. Each setup will be examined at a speed of 120 Km/h to keep the tests as fair as possible. Each setup will be examined through the analysis of damper speed histograms, weight transfer and understeer / oversteer.



3) Project Aims

The main aim of this project is to critically assess the effect of different damping coefficients on the vehicle's behaviour. This aim is achieved through extensive testing along with scientific clarification and equipment validation.

- Validation of the simulator – Matching both real life damping coefficients with the simulator coefficients while critically assessing the impact of changing the front damping coefficients on the Sports 2000 Prototype Race car.
- Understand and collect informative evidence of the effect that changing the front low and high-speed damping coefficient has on the vehicle's behaviour along with its difference in lap times.
- Investigate the effect of low speed damping coefficients on a vehicle's behaviour through the ISO 3888 double lane change manoeuvre test.



4) Literature Review

4.1) Double Lane Change Manoeuvre

Formerly known as the 'moose test', the lane change manoeuvre was transferred to the international Standard ISO 3888 after the Association of the German Automotive Industry (VDA) revised it. The test is used to evaluate the handling performance of a vehicle and is used as an integral method in conjunction with the design of a new vehicle and also the assessment of vehicles that are being released (Vehico, 2020). In more detail, the test is used to create real life scenarios 'where the driver needs to quickly swerve an object or change lanes at high speeds in the face of an emergency' (Kutluay, 2012).

Kutluay (2012) states that the vehicle might experience understeer or oversteer, most likely oversteer during the counter steer phase of the manoeuvre or the car could completely flip over due to the lateral acceleration involved which sometimes happens to vehicles with a high centre of gravity. He also indicates that the test will demonstrate the agility of the vehicle along with the capabilities of its lateral dynamics hence why the GP2 car will be tested using this ISO and VDA test (Kutluay, 2012).

Winner (2012) conducted a study into the validation of vehicle dynamics using the double lane change manoeuvre. He states that the steering angle of the vehicle differs depending on the skill of the driver and that the steering should return to steady state for a split second while the vehicle is in the left-hand lane, before changing lane back to its original. The author concludes that the measured data is easier to analyse when averaged out as a two interval approach, as can be seen in the figures below (Winner, 2012). Here by, this project will be using the same driver for all tests to conduct each setup as equal as possible.

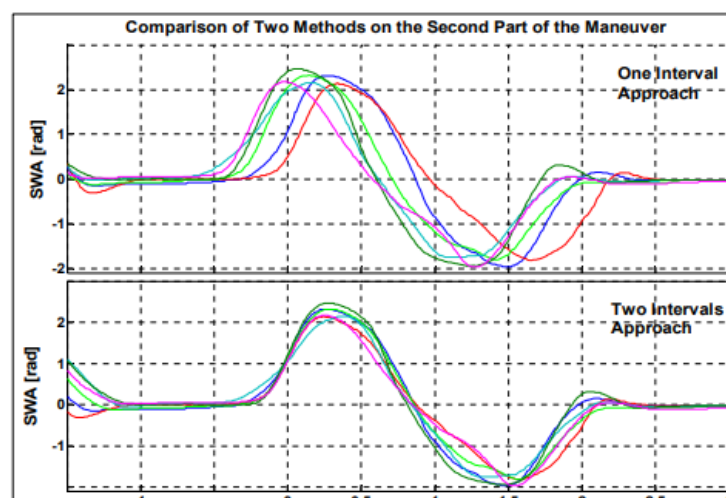
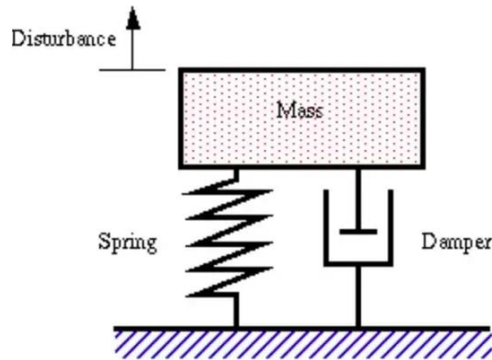


Figure 1 - Steering inputs for a lane change manoeuvre (Winner, 2012)



4.2) Spring mass damper system

Below is a damped spring and mass system which has three forces governing the dynamic behaviour. Although there is a small amount of damping in the tyre (2% approximately), the damper must do the majority of damping in this system (Kasprzak, 2014).



Where Spring = K , Mass = M and Damper = C

Inertial force is created due to accelerating the mass which is a function of the mass and acceleration (Milliken, 1995).

Figure 2 - Spring Mass Damper system (Milliken, 1995)

The damping force is a function of the velocity across the damper including the damper coefficient, this is expressed in terms of force/unit velocity. Finally, the spring force is a function of the spring displacement and the spring rate, which is expressed in terms of force/unit displacement (Milliken, 1995).

4.3) Full Car Spring Mass Damper system

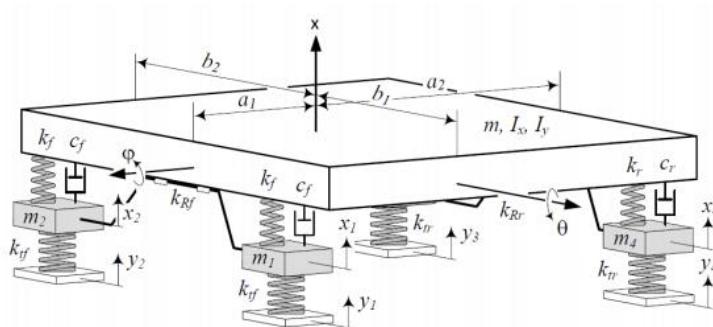


Figure 3 - Full car mass spring damper system (Jazar, 2012)

This full car model can be used as a tool to analyse the ride of a vehicle. This is due to the model having a spring and a damper at each corner. The model describes the roll, pitch and vertical motion as a function of the springs and damper forces. The sprung mass in this model is considered as a rigid body with infinite stiffness in every direction. The model also includes the anti-roll bar stiffness as this will affect the behaviour of the vehicle (Jazar, 2014).



4.4) Data Acquisition

There are many ways to interpret data when looking at Motec and it is used for many different applications. For this project, the different channels and data interpretation will be used to characterize the effect that the dampers have on the handling of the vehicle, therefore weight transfer, lateral and longitudinal acceleration, understeer / oversteer and other channels will be analysed to conclude the effect of different damping coefficients.

Santos (2014) states that there are three main categories when using a data acquisition software, these are powertrain, driver and chassis. The chassis parameters that Santos (2014) recommends are damper displacement, suspension loads, ride height and temperatures (Santos, 2014). Luckily, the simulator is equipped with all these channels whereas the real-life data will need to be interpreted by looking at the G force traction circle and the lateral G force channel. Munkholm (2011) suggests that the G force traction circle enables the analyst to compare different grip levels while changing setups, which is what this project will be analysing around the Silverstone International circuit (Munkholm, 2011).

4.4.1) Damper Speed Histograms

This is a characterization of the damper while it's on the car, giving an indication of how much time the damper is spending in each speed interval. This makes these histograms very useful to tune dampers because they produce evidence of the damper behaviour integrated into the vehicles suspension system (Segers, 2014), as agreed by Trevitt (2019), these are a very useful technique for suspension setup when out on track (Trevitt, 2019). To maximize the performance of a damper, adjustments to the setup should be made to get a symmetrical damper histogram (Segers, 2008).

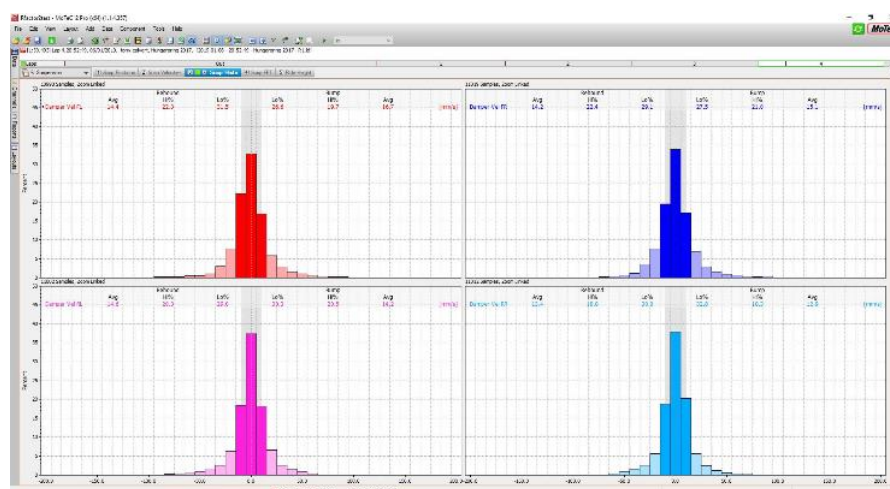


Figure 4 - Damper speed histograms (Motec.com, 2012)



4.5) Understeer and Oversteer

When a vehicle understeers it usually takes an arc 'less than desired' through the corner whereas if the car was oversteering, it would take a 'more than desired' arc through the corner. These occur due to the tyres overcoming their limit of adhesion on the road (Ernst, 2013).

Dampers have an immediate effect on understeer and oversteer due to their capability of controlling how each corner of a vehicle reacts, for instance, when the vehicle travels through a corner the inside dampers will extend and the outside will compress (koni, 2016). This indicates that the dampers can be tuned to reduce the extension (rebound) and compression (bump) depending on the occurrence. Koni (2016) also state that dampers do not have an effect on the amount of weight transfer, but they can control the time it takes to transfer the weight therefore, can increase or reduce the speed of the pitch and roll of the vehicle.

4.5.1) The effect of damper induced understeer/oversteer

A study conducted by De Groote (2002) experiments the dampers effect on each stage of a corner, assuming that the surface is completely flat and smooth. Cornering balance is affected by the distribution of load between the two front tyres and the bigger the load on the tires, the less efficient the front tyres will be. Taking phase 3A (flat out corner entry) into consideration, De Groot (2002) states that when the front dampers are adjusted to increase bump and rebound the load will be bigger at the front end, inducing understeer (Groot, 2002).

Putting this into perspective, if the front dampers are too stiff the car will understeer. When the car is understeering, it takes an arc less than desired which will tend to show on Motec as the driver is forced to input more steering to counter for the understeer. If the car oversteers, the driver will have to counter steer which requires a little less steering or maybe opposite lock.



4.6) Critical Damping

“Amount of damping required to stop movement as quickly as possible” (Tudor, 2019). Using the mathematical terms, K, M and C from the damped spring and mass system, the critical damping coefficient can be explained. This is when the mass is given a step displacement then the damping level that allows that mass to return to steady-state position quickest with no overshoot, is the critical damping. Critical damping coefficient expressed mathematically is;

$$C_{crit} = 2\sqrt{KM} \quad (\text{Milliken, 1995})$$



Figure 5 - Underdamped, Overdamped and Critically damped oscillations (Milliken, 1995)

4.7) Suspension Frequencies

This is how fast the suspension travels up and then back down to the same point when the vehicle drives over a bump. The suspension frequency can be very useful to determine the handling and balance of a vehicle, including how it will react to driver inputs. Before choosing the spring stiffness the first step is to choose the ideal frequency for the front and the rear of the vehicle. The higher the frequency, the stiffer the ride (Routley, 2017). Concluding this statement, Giaraffa (2016) states that lower frequencies produce a softer suspension with more mechanical grip, however drivers will report a ‘lack of support’ due to a slower response, which is opposite to the higher frequency (stiffer suspension) (Giaraffa, 2016).

Here are some examples of ballpark figures for different applications;

- 0.5 - 1.0 Hz – Passenger cars
- 1.5 – 2.0 Hz – Rally cars
- 2.5 – 3.0 Hz – Moderate downforce race cars with up to 50% total weight in max downforce capability.

To calculate the suspension frequency of a corner the mass and spring rate is needed, as shown in the equation below; (Routley, 2017)

$$F = 1/(2\pi)\sqrt{(K/M)}$$



4.8) Motion Ratio

This is defined as the ratio of the damper displacement to the wheel displacement. Simply put, if the wheel travels up an inch and the damper compresses by 0.6 inches then the motion ratio would 0.6 (Performancetrends, 2016).

Therefore, the calculation for the motion ratio is;

$$MR = \frac{\text{Damper Displacement}}{\text{Wheel Displacement}} \text{ (Tudor, 2019)}$$

4.9) Spring Rates

Spring rate is, as expressed by Acxess spring (2019), the constant rate of force expelled by a spring per millimetre of travel once a force is applied to compress it (AcxessSpring, 2019).

4.10) Wheel Rate

From here, the wheel rates can be calculated by using the spring rates and motion ratio for each corner as can be seen in the formula below;

$$WR = SR \times (MR)^2 \text{ (Schermerhorn, 2013)}$$

4.11) Damping Coefficients

This is the slope of a damper's force vs velocity curve which is expressed as a force per unit of velocity (N-s/m). If the damping curve from the dynamometer is linear then the damping coefficient is a constant but when it is no linear then the low speed and high speed have different coefficients (Usually the low speed has a higher coefficient due to the slope vs velocity being at a larger gradient) (Kasprzak, 2020).



5) Methodology

5.1) Data Collection

The first step of data collection will be to use the damper dynamometer with both the Penske and Quantum dampers, testing different combinations of set-ups. The on-board data collected from the vehicle is from the Silverstone International Circuit and was obtained from the data laptop of the MCR racing team.

5.1.1) Real Life Data

The MCR team at the university run the Sports 2000 Prototype race car and have created a test logbook to log every detail of the weekend and the driver's feedback from each test and event. On top of this, the data logger in the car is a Motec C185 model which can log channels that can be analysed on the Motec software, like the simulator data. There are some limitations with using the real-life data, these are the channels that cannot be logged with the real-life vehicle

data such as the steering percentage and tyre loads. Because of this, the real-life data will be approached in the way that Munkholm (2011) suggested, which is to analyse the lateral acceleration channel using methods such as the traction circle and the abnormalities in the full lap channel (Munkholm, 2011).

5.1.2) Damper Dynamometer

All of the damper testing will be done in house in the workshop using the SPA damper dynamometer, this specialised equipment is widely used by many motorsport companies across the world (SPA, 2019). This dynamometer provides different types of data including force vs velocity graphs and force vs displacement graphs, refer to figure 10 for an example (Kasprzak, 2014).

Segers (2014) implies that the damper dynamometer is used to approach a more structured test which is an alternative to on-board data logging limitations, providing more concise answers to the characterisation of the damper (Segers, 2014).

Circuit:	Silverstone International
Date:	04/10/2019
Staff Present:	TT, PD

Testing Log

Climate	T1	T2	T3	T4
Weather	Drying, Overcast	overcast (damp)	rain at end	
Temp	12	13	14	14
Tyres				
Tyre Set	B19	B19	B19	B19
F Pressure	14.5	14.5	14.5	14.5
R Pressure	15	15	15	15
Car Settings				
F Dampers	B: R:	B: R:	see comments	see comments
R Dampers	B: R:	B: R:	see comments	see comments
F ARB	3	see comments	3	3
R ARB	3	see comments	3	3
Rear Wing	7	7	7	8
F Ride Ht	47	47	47	47
R Ride Ht	67	67	67	67
Session Info				
Start Time	09:35	10:45	11:55	14:00
Run Time				
Fastest Lap			1:09.8	1:08.6

T1 Comments	Initial laps for checking car ok. familiarisation with new surface, steering slightly to right, tried to adjust on splines but too far other way, so reverted to previous position. Comments on loose rear end.
T2 Comments	Out for first few laps on a dry line (damp when overtaking). Car felt OK. SETUP TEST SESSION Pit in, full hard front ARB & full soft rear ARB, pit in, full soft front ARB full hard rear ARB, Pit in, full hard bump and
T3 Comments	further data collection exercise. Went out full hard bump and rebound rear, full soft bump and rebound front. out for some laps before coming back in. Driver feedback was positive (best represents current damper settings from combinations tested) rear felt fine, front was
T4 Comments	Following same settings from previous, car suffering from mid corner understeer and snap oversteer. Pit in, -2 front bump (15), +2 rebound rear (3). Completed some laps on this setting and decided to not chase any further, set for qualifying. Tyres are losing a

Figure 6 - Testing log for the MCR race team (UWTSO race team, 2019)



5.1.3) Motec

Motec I2 pro is used by many motorsport teams across the world to organise and chart data in order to identify problems, trends and make improvements. The software system is used to review data from the on-board diagnostics system fitted to the vehicle, by doing this the vehicles damper behaviour can be analysed and improved. Along with improving the balance and behaviour of the vehicle on track, Motec can also be used to develop drivers on different parts of the track by reviewing their performance using different channels such as the throttle and steering positions. It can also be used to compared other drivers that have logged data on the exact same vehicle and circuit (Motec, 2019).

5.1.4) Base Performance Simulator

The simulator in the university has a built-in model of the Sports 2000 Prototype race car, therefore can be used to simulate different damping coefficients using the same vehicle. The positives of this is that all testing can be made in house and that the simulator has many more channels than the actual vehicle, allowing the data to be analysed deeper and in a more detailed manner. The simulator used for the analysis of this project has been used to develop and perfect championship winning teams across the country. The simulator has a very wide range of setup possibilities and parameters that can be changed which makes it a very efficient way of testing (BPSimulators, 2020).

5.1.5) Bob's Track Builder Pro

This product allows the user to create and mesh tracks that can then be downloaded onto different software's to be used (Pywell, 2019). For the double lane change manoeuvre, a circuit was built with cones set out at the right distances, replicating the ISO 3888 diagram that can be seen below (BSI, 2011).

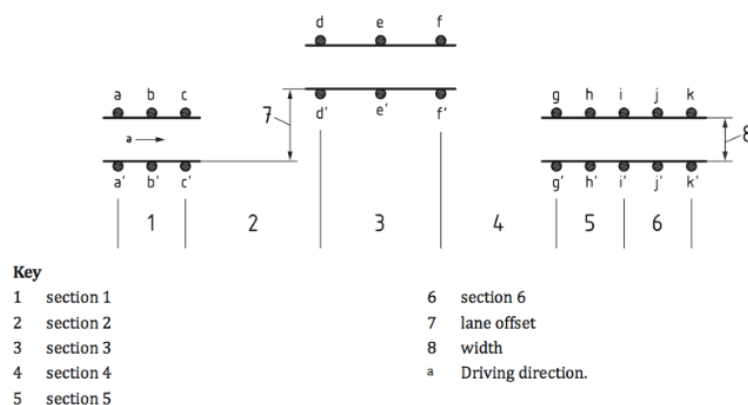


Figure 7 - Lane change manoeuvre dimensioning (ISO 3888, 2011)



5.2) Steady state Analysis

Enabling the project to analyse the understeer / oversteer channel, a steady state cornering test needed to be conducted for both the Sports 2000 and GP2 vehicles. The first step of this phase was to conduct a 360 degree turn on the skid pad built into the simulator. The vehicle is driven around at a constant speed with the same constant steering wheel percentage input where the lateral acceleration should be minimal, suggesting that the vehicle is balanced. Crolla (2000) states that the vehicle needs to 'settle down to its steady state at a certain forward velocity after a steering input has been applied' (Crolla, 2000).

Once the vehicle has been tested on the skid pad the next step is to upload the data onto Motec. As can be seen in the figure below, the math channels created are to predict the understeer and oversteer. The first channel needs to be the corner radius and from here, the Ackerman can be predicted prior to the Ackerman Angle.

Name	Summary
Expressions	
☒ speed [m/s]	('Corr Speed' [km/h] / 3.6)
☒ Corner Radius [m]	('speed ' [m/s] * 'speed ' [m/s]) / ('G Force Lat' [G] * 9.81)
☒ Ackerman [rad]	atan('Wheelbase' [m] / 'Corner Radius ' [m])
☒ WT Lat (front) [N]	'Tire Load - FL' [N]-Tire Load - FR' [N]/2
☒ WT Lat (rear) [N]	'Tire Load - RL' [N]-Tire Load - RR' [N]/2
☒ WT Lat (total) [N]	'WT Lat (front)' [N]+'WT Lat (rear)' [N]
☒ Ackerman Angle [rad]	-('Wheelbase' [m]*'G Force Lat' [G]*'G' [m/s/s])/ (('Corr Speed' [km/h]/3.6)^2)
☒ Theoretical steering pct	'Ackerman Angle' [rad]* (-28.46/0.0994)
☒ Understeer	abs('Steering Wheel Pct' [%]) - abs('Theoretical steering pct')
☒ Rake [deg]	(atan((((('Ride Height RL' [mm]+'Ride Height RR' [mm])/2)-('Ride Height FL' [mm]+'Ride Height FR' [mm])/2)
Constants	
☒ Front Track [m]	1.7
☒ Vehicle Mass [kg]	780
☒ Wheelbase [m]	3.12

Figure 8 - Math channels on Motec (Deion Atkinson, 2020)

Once these parameters are calculated the next step is to analyse the data on Motec and find the ideal steering input and lateral acceleration value, from here the understeer and oversteer can be predicted using the equations below;

$$\text{Theoretical Steering Percentage} = \text{Ackerman Angle (rad)} \times \left(\frac{\text{Steering Percentage}}{\text{Lateral Acceleration}} \right)$$

$$\text{Theoretical Steering Percentage} = \text{Ackerman Angle (rad)} \times \left(\frac{-28.46}{0.0994} \right)$$



$$\textit{Understeer} = \textit{Absolute(Steering Pct)} - \textit{Absolute(Theoretical Steering Pct)}$$

(Tudor, 2019)

Because the understeer equation is the steering percentage take away the theoretical steering percentage, the understeer values will be positive and oversteer will be negative.



5.3) Double Lane Change Manoeuvre – Test planning

The double lane change manoeuvre consists of 11 tests in total, the first 9 being a combination of low speed bump and rebound and then another two tests analysing the different setups from the front and the back. The low speed coefficients are tested with the double lane change manoeuvre for the fact that the low speed damping is influenced by driver inputs and that the track on the simulator is completely flat with no kerbs, meaning that the high speed damping will not be experienced in this scenario (Schroeder, 2018). The coefficients were the available ones for the Gp2 car on the simulator and were split into 3 setups, min, mid and max. This plan was chosen for the vehicle to ensure that all extremes are tested, these are diagonal both ways across the first table, and then assess the minor differences in between the extremes (setups 2,4,6,8).

Table 1 - Lane change manoeuvre test plan (Deion Atkinson, 2020)

		Front		
		min	mid	max
Rear	min	1	4	7
	mid	2	5	8
	max	3	6	9

Table 2 - Damping coefficients for the lane change manoeuvre (Deion Atkinson, 2020)

Test No	Front Bump	Front Rebound	Rear Bump	Rear Rebound
1	268	-268	268	-268
2	268	-268	521	-521
3	268	-268	803	-803
4	521	-521	268	-268
5	521	-521	521	-521
6	521	-521	803	-803
7	803	-803	268	-268
8	803	-803	521	-521
9	803	-803	803	-803

To ensure the test results were consistent and all tests were made fairly and accurately the same driver was used to complete all setup changes and the driver had repeated each test 5 times to erase any inaccuracies. The test was completed at 120 Km/h due to the handling of the vehicle being more advanced than regular road vehicles which normally complete the test at around 80 km/h (Kutluay, 2012).



6) Vehicle Parameters and Characteristics

Table 3 - Vehicles un-sprung parts masses (Deion Atkinson, 2020)

Part	Front/Rear	Mass (kg)
Brake disc	F	4.2
	R	4.2
Push rod	F	0.6
	R	0.65
Tie rod	F	0.35
	R	0.45
Stub Axle	F	0.9
	R	0.9
Upright	F	3.65
	R	6.75
Rim & Tyre	F	9.25
	R	11.05
Lower Wbone	F	2.05
	R	1.6
Upper Wbone	F	0.85
	R	1.3
Rocker	F	0.55
	R	0.7
Steering Rack	F	3.6
Calliper	F	2.05
	R	2.05
Damper	F	1.45
	R	1.55

Table 4 - Sprung and un-sprung masses of the vehicle (Deion Atkinson)

Front sprung: 187.1833333 Kg	Front Un-sprung: 43.31666667 Kg
Rear sprung: 309.9666667 Kg	Rear Un-sprung: 53.03333333 Kg
Total Ms: 497.15 Kg	Total Mus: 96.35 Kg

Table 5 - Corner masses of the vehicle (Deion Atkinson, 2020)

FL MASS 115.5		FR MASS 115
	TOTAL 593.5	
RL MASS 176.5		RR MASS 186.5

Motion Ratio: $MR = \frac{\text{Damper Displacement}}{\text{Wheel Displacement}}$

Front: $MR = \frac{55}{66} = 0.83$

Rear: $MR = \frac{90}{96} = 0.93$

Wheel Rate: $WR = SR \times (MR^2)$

Front: $WR = 39053 \times (0.83^2) = 26903.6$

Rear: $WR = 67598 \times (0.93^2) = 58463.5$



Suspension Frequencies - Sprung:

$$F = \frac{1}{2\pi} \sqrt{\frac{\text{Spring rate}}{\text{Sprung mass}}}$$

$$\text{Front} = F = \frac{1}{2\pi} \sqrt{\frac{39053}{93}} = 3.2 \text{ Hz (for each corner)}$$

$$\text{Rear} = F = \frac{1}{2\pi} \sqrt{\frac{67598}{154}} = 3.3 \text{ Hz (for each corner)}$$

Referring to Routley (2017), race cars with up to 50% of their weight in downforce capability should be between 2.5 – 3.0 indicating that the vehicle has a stiff suspension system considering that it is a moderate downforce race car (Routley, 2017).

Suspension Frequencies – Un-sprung:

$$F (\text{us}) = \frac{1}{2\pi} \sqrt{\frac{\text{Wheel rate} + \text{Tire spring rate}}{\text{Un-sprung mass}}}$$

$$\text{Front} = F (\text{us}) = \frac{1}{2\pi} \sqrt{\frac{26903.6 + 250000}{21.65}} = 17.9 \text{ Hz (each corner)}$$

$$\text{Rear} = F (\text{us}) = \frac{1}{2\pi} \sqrt{\frac{58463.5 + 250000}{26.5}} = 17.2 \text{ Hz (each corner)}$$

$$\text{Critical Damping: } C_{\text{critical}} = 2 \sqrt{\text{Spring rate} \times \text{Mass}}$$

$$C_{\text{crit}} (\text{front right}) = 2 \sqrt{39053 \times 115} = 4238.4 \text{ N-s/m}$$

$$\text{Front left} = 4247.6 \text{ N-s/m}$$

$$\text{Rear right} = 7101.3 \text{ N-s/m}$$

$$\text{Rear left} = 6908.3 \text{ N-s/m}$$



Damping Ratio – Bump and Rebound at midway settings

Rear damper rates =

Bump low speed - 5763.533 N-s/m

Bump high speed – 2649.723 N-s/m

Rebound low speed – 13302.140 N-s/m

Rebound high speed – 5127.831 N-s/m

$$\varepsilon = \frac{\text{Damping Rates}}{\text{Critical Damping}}$$

Damper ratios for rear right:

Low speed bump: $\varepsilon = \frac{5763.533}{7101.3} = 0.81$

High speed bump: $\varepsilon = 0.37$

Low speed rebound: $\varepsilon = 1.81$

High speed rebound: $\varepsilon = 0.72$ Average = 0.92

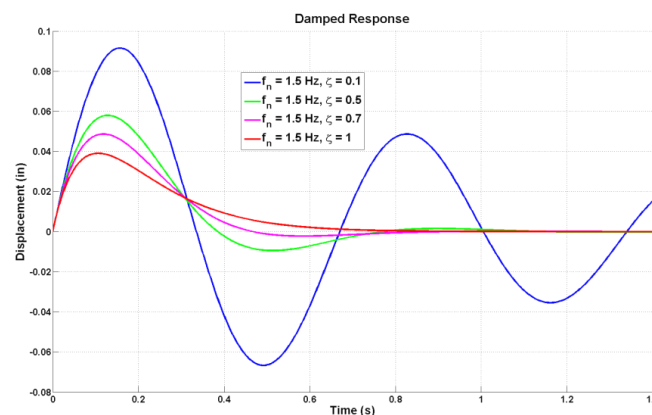


Figure 9 - Damping displacements of different frequencies (Kasprzak, 2014)

From looking at the damping ratios above, it is certain that the Sports 2000 prototype race car is nearly critically damped as the average results are 0.92 and critically damped is 1 (as can be seen in the figure above). Kasprzak (2016) states that non-aero race cars should be between 0.5 – 0.7 and that moderate aerodynamic race cars should be between 0.7 – 1, therefore 0.92 critical damping value indicates that the Sports 2000 race car is moderately aerodynamically equipped (Kasprzak, 2014).



7) Results Analysis and Discussion

7.1) Force vs Velocity Graphs – Penske vs Quantum

The figure to the right was obtained from a damper dynamometer test on the Penske 8300 series damper at full soft bump and rebound, these are the dampers fitted onto the Sports 2000 race car. Circled in black are the low speed forces, these forces are experienced by driver induced actions such as cornering, pitch and roll. The high speed (not circled) damping is experienced when hitting a kerb or a hole which are surface irregularities (Schroeder, 2018).

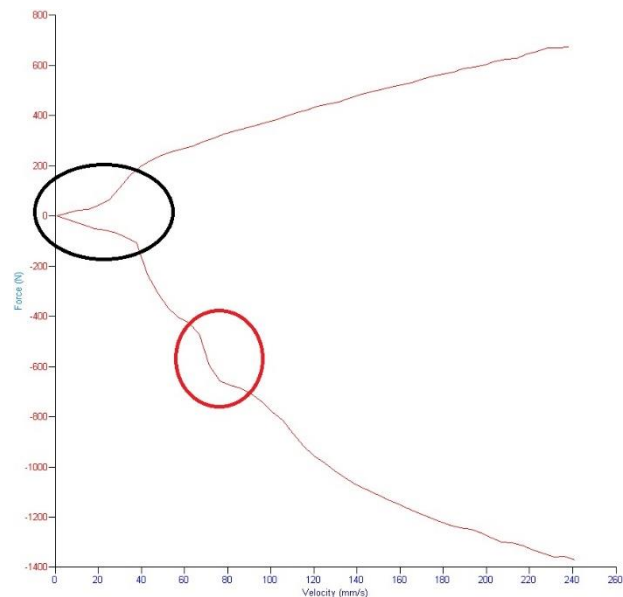


Figure 10 - Force v Velocity for Penske 8300 series damper (Deion Atkinson, 2020)

Circled in red is the fast range of rebound which show some irregularities in the graph line. This loss in force could be an issue with valving or a bent rod as Kasprzak (2014) has previously experienced in his study of dampers (Kasprzak, 2014). The graph indicates that the forces of rebound are much higher per increment of velocity, this is because the rebound function of the damper controls the motion of the body which is the sprung mass (Erjavec, 2016). Taking into consideration that the sprung mass is 5 times the amount of the un-sprung mass in Kg's explains why the rebound has a higher force value.

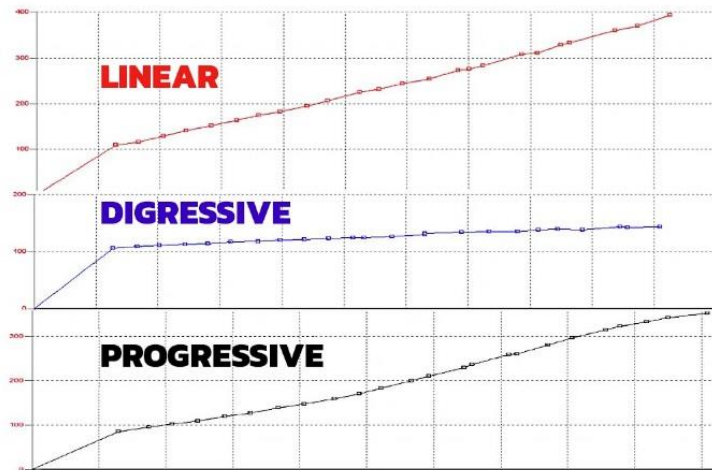


Figure 11 - Different piston effects on force v velocity graphs (Shocksurplus.com, 2020)

A study made by Shock surplus (2020) defines the difference between pistons inside the damper. Looking at the results of figure 10, the Penske damper is fitted with a linear piston that causes the high speed forces to rise gradually along with the velocity (Shocksurplus, 2020)

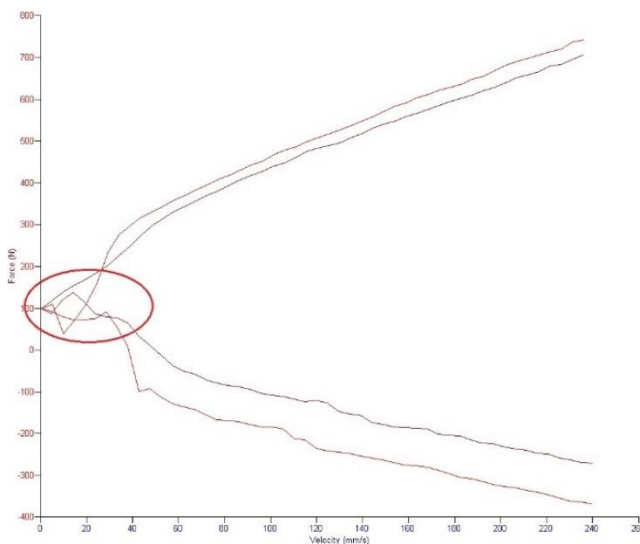


Figure 12 - Quantum force v velocity graph with both extreme settings (Deion Atkinson)

Looking at a different damper, the Quantum single-adjustable Force v Velocity graph doesn't look too dis-similar to the Penske graph, but it is only a single adjustable damper therefore, changing both bump and rebound together. Figure 12 is the force v velocity graph for full soft in purple and full hard in red for the Quantum damper. Circled in red is the low speed for both setups and by looking at the graph there is an indication of faults in the damper (Kasprzak, 2014).

Analysing the difference between compression and rebound there is a bigger difference between both setups in rebound, but the compression shows a higher force by 380 Newtons at 240 mm/s, giving a ratio of 1.9 bump to 1 rebound. Unlike the Penske damper, this is a compression biased damper, but the adjustment has a larger effect on rebound throughout each sweep.



7.2) Real-life On-board Data – Silverstone International Circuit

The damping coefficients for both damper setups can be seen in the tables below, these were then matched on the simulator for comparison and validation;

Table 6 - Damping coefficients for test 1 at Silverstone (Deion Atkinson, 2020)

Test 1 – Lap time = 1.11.170 s		
Front	Bump High	2400
	Rebound High	-4947
	Bump Low	12000
	Rebound Low	-4000
Rear	Bump High	6000
	Rebound High	-4957
	Bump Low	12000
	Rebound Low	-9474

Table 7 - Damping coefficients for test 2 at Silverstone circuit (Deion Atkinson, 2020)

Test 2 – Lap Time = 1.09.149 s		
Front	Bump High	3200
	Rebound High	-5131
	Bump Low	12000
	Rebound Low	-6105
Rear	Bump High	6000
	Rebound High	-4957
	Bump Low	12000
	Rebound Low	-9474



6.2.1) Damper Histograms

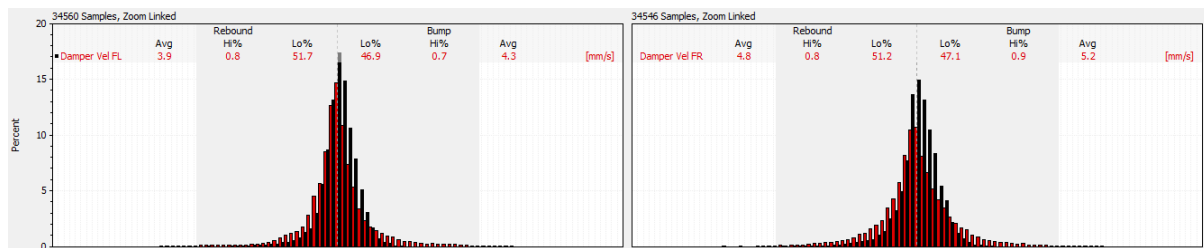


Figure 13 - Damper speed histograms for real-life data sets (Deion Atkinson, 2020)

By looking at the front damper speed histograms setup 1 (black) spends more of its time in low speed and that the front left damper is compression biased and the front right is rebound biased. This suggests that both dampers need to be tuned separately for this circuit where the front left could do with a bit less compression or a little more rebound (Rijk, 2010). Looking at the spread of both setups, setup 2 (red) seems to have a wider spread, therefore spends more time at different velocities. Rijk (2010) states that this is an indication of the body not being overdamped at low frequencies suggesting that the body returns to steady state equilibrium at a critically damped pace (Rijk, 2010) (Madnick, 2020) (Rouelle, 2019).



7.2.2) Understeer / Oversteer Analysis

The figure below represents both setups overlaid on Motec. The black channel is setup 1 and the red channel is setup 2 and to clarify, the black channel is the softer setup of both tests as can be seen in the tables 6 and 7. The times for both setups are very close but setup 2 is faster by two seconds per lap. Circled in black is the exit of turn 2, with both setups at a 150 Km/h there is a drastic drop in acceleration but it is clear that setup 2 (red) experiences higher values of G force and that they are both oversteering due to the instant loss in G force value (Fey, 1993). Similar is circled in red where both setups are experiencing oversteer, this time setup 2 experiences the most loss suggesting that it has slightly more oversteer than setup 1.

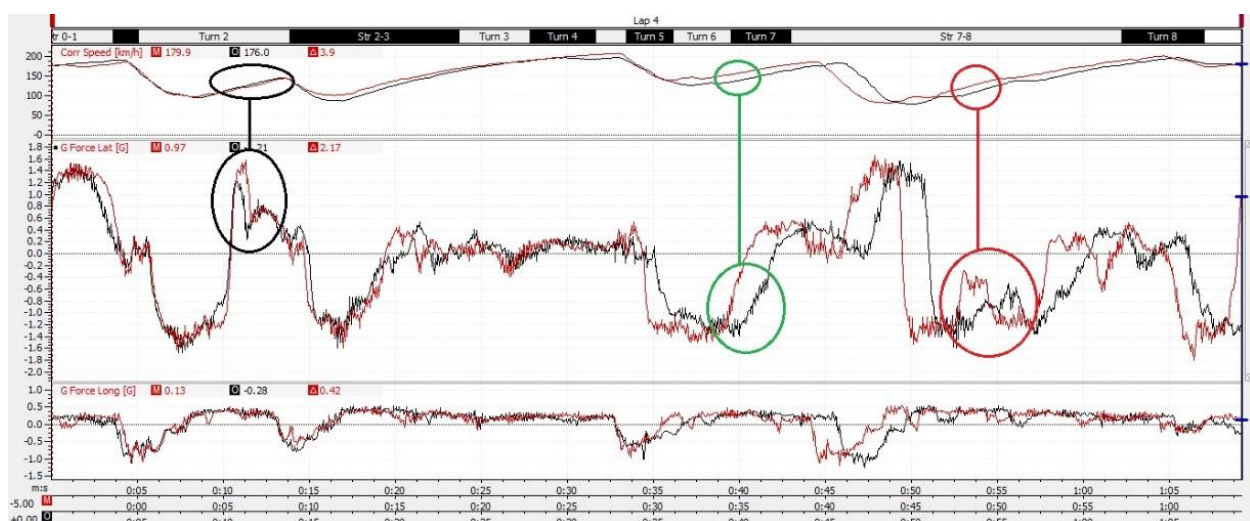


Figure 14 - Motec channels for analysis of the real-life data sets (Deion Atkinson, 2020)

Taking the green circled area into consideration, at the exit of turn 6 it shows that setup 2 returns to steady state at a faster pace than setup 1. This suggests that setup 2 is understeering due to the irregularities in the channel as it returns to the zero position (Fey, 1993). Post analysis of the two setups there aren't many differences between them, but the understeer or oversteer of both could be driver induced as this can't be ruled out due to the car not having a steering position sensor. Because of this, drivers' feedback will be taken into consideration to assess the handling and for setup 1, the feedback was that the car was suffering from snap oversteer as can be seen on the data sheet. Setup 2 for the driver felt fine and that the rear felt good, but the front was too stiff, inducing understeer as the lateral acceleration channel suggests (Groot, 2002).



7.2.3) G-G Scatter plot – The Traction Circle

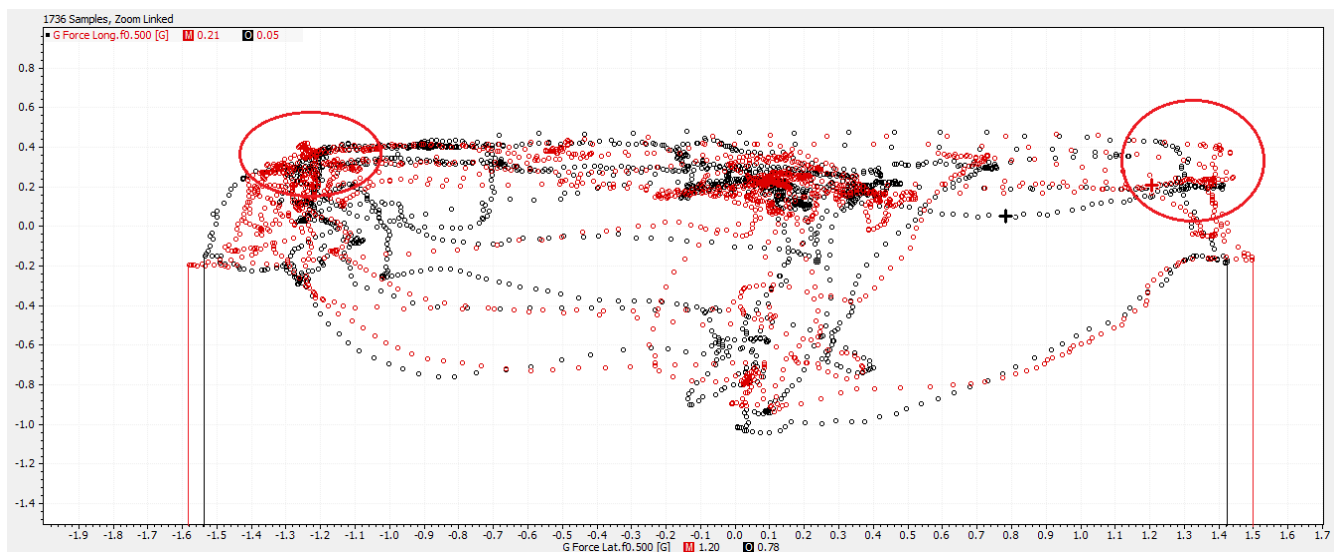


Figure 15 - G-G Scatter plot for the real-life data sets (Deion Atkinson, 2020)

Above is the traction circle for both setups, using the same colour reference as before where setup 1 is black and 2 is red. Both red and black lines indicate that setup 2 (stiffer) can generate more lateral acceleration by 0.08 G. The more lateral acceleration the better the cornering forces for that setup and overlaying two sets of data can indicate which setup has the most grip when cornering (Munkholm, 2011). Fey (1993) states that when braking, the higher the longitudinal G values are the harder the car is braking which is an indication that setup 1 is using the most of its traction longitudinally (Fey, 1993). This could be a consequence of stiffening the front of the car because the stiffer the front, the faster the weight is transferred and this will force the car to push into the corner (Groot, 2002). The circled areas on the top right and top left show the values of grip available when the car is cornering and accelerating at the same time. The higher the positive value of G's the stronger the car is accelerating therefore, setup 2 seems to be utilising more tyre grip when exiting corners (Rouelle, 2019). The results of the G-G diagram indicate that when the front of the car is stiffer, the better it is at accelerating out of the corner. In argument, De Groot (2002) and Mansell (2019) state that when the front of the car is stiffer it is more likely to push through the corner, inducing understeer (Groot, 2002) (Mansell, 2019).



7.3) Silverstone International Simulator Test

This test was conducted with the same damping coefficients as the above, therefore data should be comparable between both data sets taking into consideration the overlaying of the data not being accurately matched indicating each corner as they should be and the inaccuracies of the simulator.

6.3.1) Damper Speed Histograms

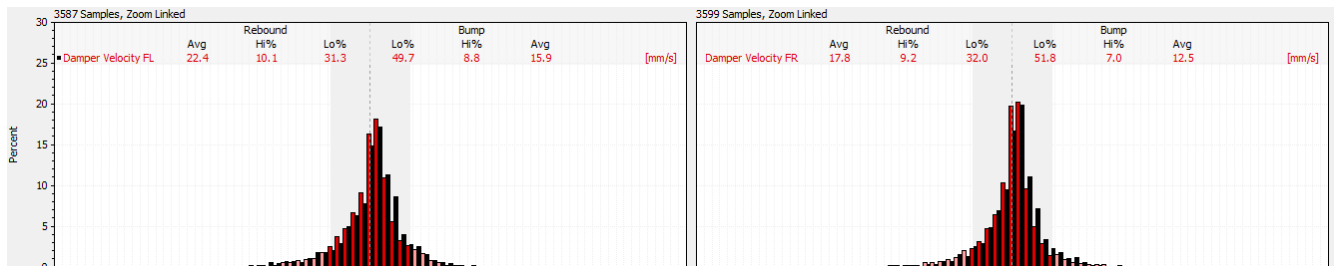


Figure 16 - Damper speed histograms for simulator test (Deion Atkinson, 2020)

On the simulator test both damper settings seem to be bump biased around the circuit. Setup 1 (black) is a lot more un-even and spends more time at higher speeds than setup 2. Setup 2 looks to be a lot more symmetrical which is the same as the real-life data, indicating that this setup works better than setup 1 does. By looking at these histograms it can be said that setup 1 (black) needs to be slightly stiffer in rebound to compensate for the amount of bump, as explained by (Rijk, 2010) and (Rouelle, 2019).



7.3.2) Understeer / Oversteer Analysis

Due to the simulator having specific channels that the real car doesn't have, it is possible to create an understeer/oversteer channel. Like the real-life data, setup 2 (red) encounters more understeer than setup one when travelling through turn 1 but there is a 15 Km/h difference in speed. Circled in red is the low speed corner of the circuit, this shows that setup 1 has more oversteer and setup 2 has understeer which can also be seen in the steering percentage channel where setup 2 turns for longer than setup 1. Ernst (2013) states that when a car is understeering it takes an arc 'less than desired' through the corner therefore, more steering input is needed (Ernst, 2013).



Figure 17 - Motec channels for analysis of simulator test (Deion Atkinson, 2020)

In the green circled areas both setups are travelling at 150 Km/h into the corner where they both experience 'lift off oversteer' when the rear of the car loses grip due to the weight shifting forwards to the front of the car (Mansell, 2019). Mid corner it's clear that setup 2 has more understeer than setup 1 as it stays at full lock for longer to counter and, as De Groot (2002) explained if the front bump and rebound settings are increased then understeer is more likely to happen (Groot, 2002).



7.3.3) G-G Scatter Plot – The Traction Circle

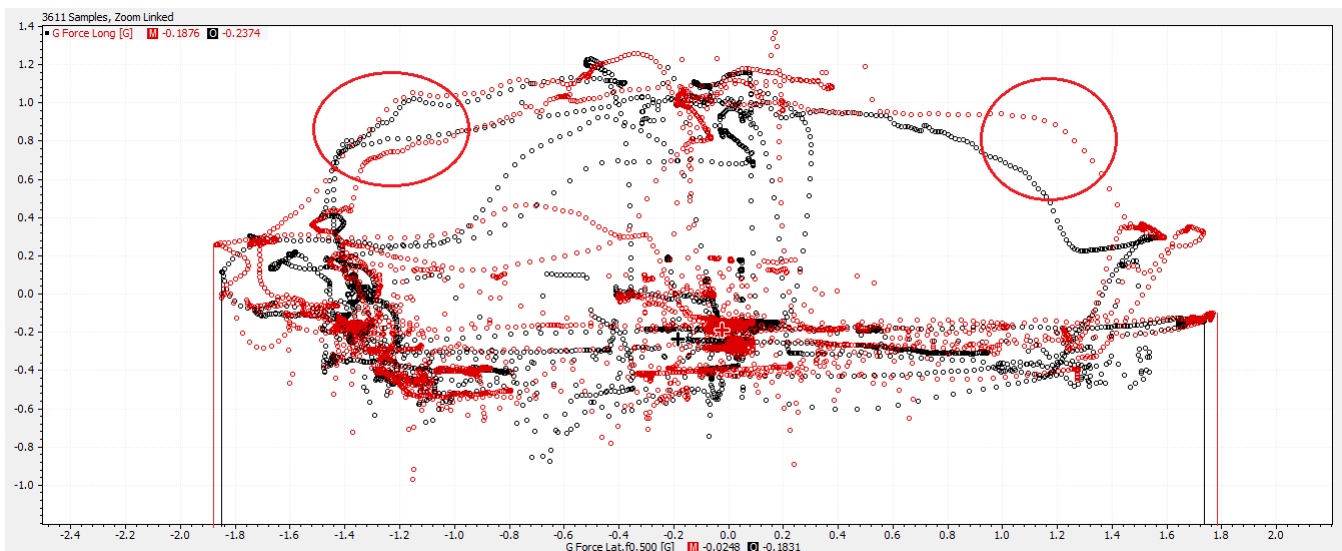


Figure 18 - G-G Scatter plot for the simulator data sets (Deion Atkinson, 2020)

The scatter plot above shows that setup 2 (red) experiences most lateral G force values when cornering at 0.3 G of longitudinal acceleration. This is very similar to the real-life data scatter plot as the most G force values are at 0.2 longitudinal G, giving a difference of 0.1 G longitudinally. The results are the same as they were in the real car where setup 1 doesn't experience as much G force as setup 2. Another point is that at around 1.0 G on the right-hand side setup 2 seems to have a perfect circular shape which shows it is gaining higher cornering forces that suggests it has higher grip levels than setup 1 (Fey, 1993).

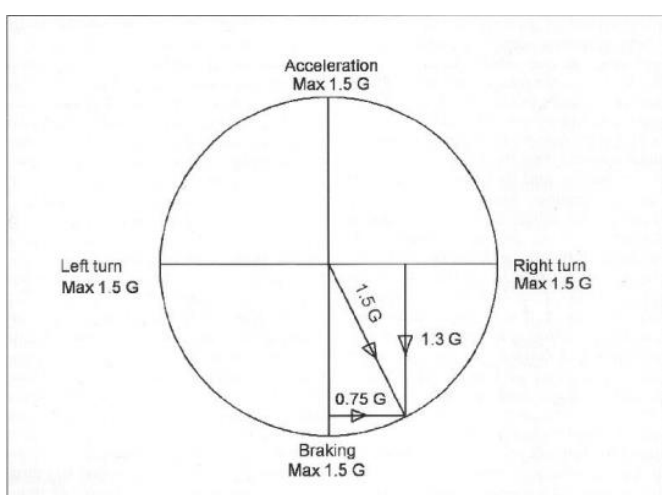


Figure 19 - Ideal G-G diagram of a vehicle with maximum tyre capability of 1.5G (Rijk, 2010)

This is an example of the perfect traction circle by Rijk (2010) which states that this is based on a maximum tyre grip of 1.5 G (Rijk, 2010). From the log sheet created by the race team, tyres were known to be 'scrubbed' and in good condition, but not new. This could be the difference in G force values between the simulator and the real-life data where the simulator shows a higher value by 0.3 G.



7.3.4) Simulator Validation

Looking at both data sets it's clear that both setups replicate the same behaviour when used on the sim and in real-life. Setup 1 seems to be the slower setup of both as it has softer damping at the front, this sacrifices the traction ability of the car as can be seen in the G-G diagram. Both data sets show a similar trend in understeer and oversteer as the setups change, this being that setup 2 has slightly more push but can travel through the corners at an average of 10 Km/h faster. The damper speed histograms also show a similar trend where setup 1 spends more time at low speed whereas setup 2 is evenly spread at both front corners. The irregularities in the histograms is that the real-life data seems to be biased to both bump and rebound but the simulator is bump biased. The reason for this difference could be that the track on the simulator hasn't got dips or bumps or that the driver could be inputting different low speed changes to the car which would manipulate the histograms symmetrical shape.



7.4) Double Lane Change Manoeuvre

6.4.1) Setup 1, 2 and 3

The first setups to be analysed doing the double lane change manoeuvre will be setup 1, 2 and 3.

The first three setups are based on the front being at minimum bump and rebound, therefore only changing the rear from min, mid to max. The details of each coefficient can be seen in table 2. Each setup has a Force v Velocity graph example from Excel which can be seen in appendix 1.A.

7.4.2) Damper Speed Histograms



Figure 20 - Damper speed histograms for lane change manoeuvre setups 1, 2 and 3 (Deion Atkinson, 2020)

The histograms above show that Setup 2 is the only setup that uses both bump and rebound the same amount whereas setup 3 is very much bump biased and only spends time at lower speeds. Setup 1 seems to be spending more time at higher velocities than the other two setups. This is an indication that both bump and rebound would ideally need stiffening to compensate for the lack of low speed damping, this would then result in a more symmetrical shape (Rijk, 2010).



7.4.3) Understeer and Oversteer

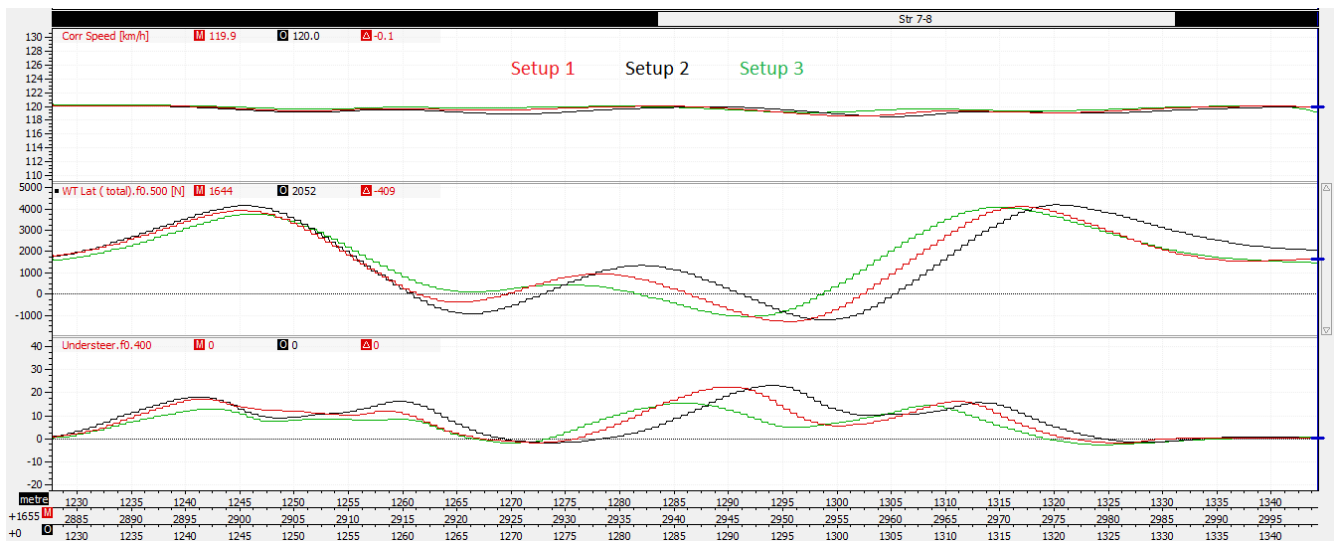


Figure 21 - Understeer channels for lane change manoeuvre setups 1, 2 and 3 (Deion Atkinson, 2020)

Above are the channels to compare weight transfer along with understeer and oversteer. Setup 1 is the one that experiences most understeer over the whole course of the test except for the first transition where setup 2 seems to have a little more. Setup 3 has a lot less understeer throughout the whole test but, at the last transition when the vehicle is returning to steady state it experiences slight oversteer unlike the other two setups. This is a consequence of stiffening the rear dampers because, as stated in Suspension secrets (2019), the method to remove oversteer is to reduce the stiffness at the rear of the car or increase the stiffness at the front (Suspensionsecrets, 2019).

Comparing the weight transfer of each setup it's clear that setup 3 transfers its weight before the other setups do when transitioning back into the original lane. This indicates that the stiffer the damper settings the quicker the weight transfers which is the general effect of dampers on vehicle handling (Alexander, 2015). De Groot (2002) states that when the weight is transferred quicker the car will respond faster when cornering and should be able to withstand higher lateral acceleration values. This explains why setup 3 has less G force than the other two setups in the G-G diagram (Figure 22) indicating that it has slightly more potential tyre grip (Groot, 2002).



7.4.4) G-G Scatter Plot

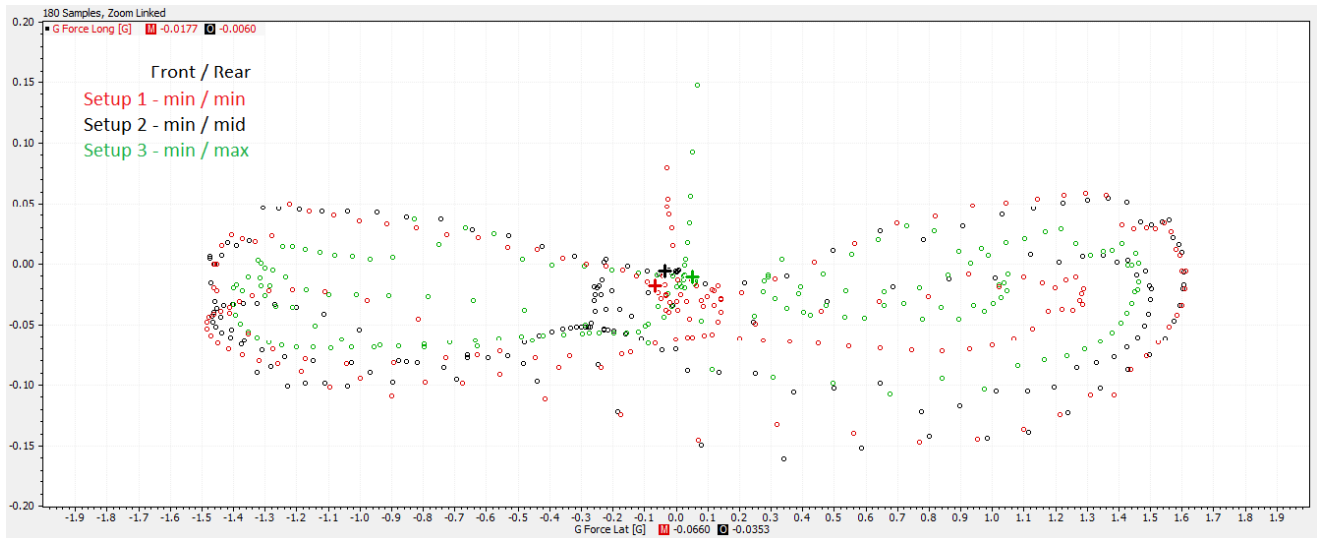


Figure 22 - G-G scatter plot for lane change manoeuvre setups 1, 2 and 3 (Deion Atkinson, 2020)

This G-G diagram indicates that setup 1 is accelerating at 0.1 G more than setup 3 which suggests that setup 3, in theory, could go faster through the manoeuvre as it has smaller values. Looking at both setup 1 and 2 they seem to be very similar and when turning left (positive lateral G force value) the car experiences higher acceleration values which matches up to the understeer graph where the value is higher at the third transition of the test (Munkholm, 2011).



7.5) Setups 4, 5 and 6

7.5.1) Damper Speed Histograms



Figure 23 - Damper speed histograms for lane change manoeuvre setups 4, 5 and 6 (Deion Atkinson, 2020)

The damper speed histograms for these setups indicate that each setup is spending time at both low and high speed during the test. Setup 4 and 6 on the left-hand side of the car are compression biased whereas setup 5 is nearly symmetrical suggesting that this setup doesn't require much change. When looking at the right-hand side of the car, setup 5 is showing a rebound biased histogram whereas setup 6 is most symmetrical but could still do with a decrease in rebound at the rear or an increase in rebound at the front to compensate for the lack of bump (Rijk, 2010) (Rouelle, 2019). Increasing the coefficients of both bump and rebound at the front has increased the speed variations of all four dampers. This has caused all three setups to be symmetrically better, especially the previously known setup 1 which is now setup 4, as it now spends a little more time at lower speeds.



7.5.2) Understeer and Oversteer

Analysing the understeer graph, it shows that as the damping coefficient increases at the rear the understeer decreases. Looking at the understeer of each setup, there is a higher rate of decrease when the rear is stiffened from mid to max unlike the difference between min and mid. At the second transition which would be counter steering back onto the off-set lane, setup 6 encounters a longer duration of understeer and setup 4 has slight oversteer. This concludes that setup 5 spends a longer amount of time at steady state (being balanced) before turning back onto the original lane.



Figure 24 - Understeer channels for lane change manoeuvre setups 4, 5 and 6 (Deion Atkinson, 2020)

At the end of the test setup 6 clearly has a lot more understeer because of the rear being at its stiffest between the three setups. De Groot (2002) and Suspension Secrets (2019) state that as the dampers are adjusted to increase bump and rebound at the rear, the car will oversteer and to overcome this the front rebound will need to be increased (Suspensionsecrets, 2019) (Groot, 2002). The effect that the front damping coefficients has on the vehicle is that, as the front gets stiffer from min to mid setting, the understeer for setup 1 from figure 20 is less than setup 4. The reason for this is that when the front of the car is stiffer than the rear, the weight will transfer faster towards the front and induce understeer (Groot, 2002). As the front and rear get stiffer together the car seems to have less understeer because of both front and rear having the same coefficient suggesting that the weight transfer isn't biased towards one another (Ernst, 2013).



7.5.3) G-G Scatter Plot

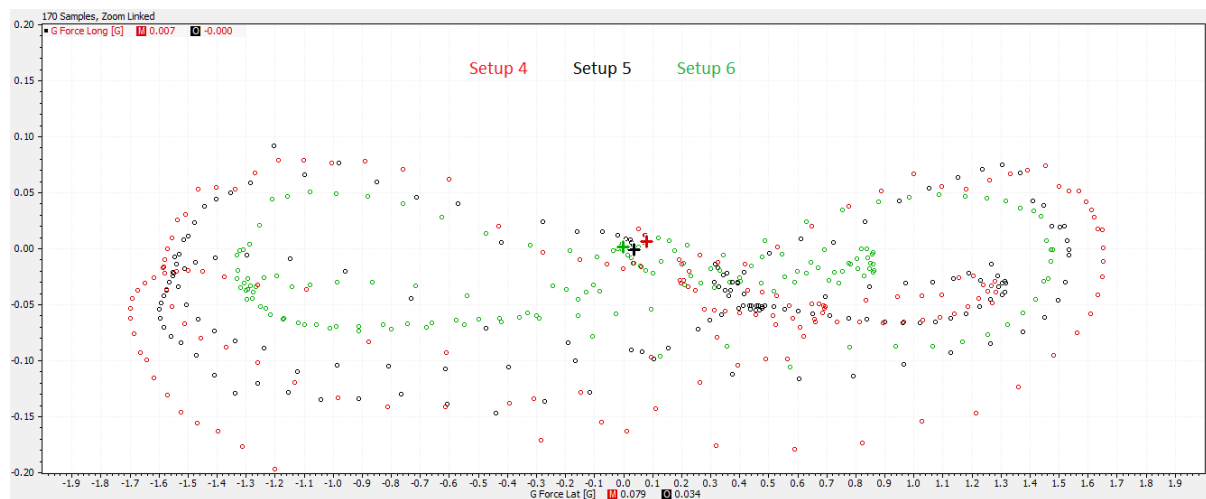


Figure 25 - G-G Scatter plot for lane change manoeuvre setups 4, 5 and 6 (Deion Atkinson, 2020)

Following the same trend as figure 22, the lateral acceleration decreases as the rear of the car gets stiffer. Setup 4 experiences 0.1 G more than setup 1 does, this is an indication that increasing bump and rebound at the front of the car has more of a negative effect than increasing them at the rear. In contrast, setup 5 has 0.06 G less lateral acceleration than setup 2, which means that when the front bump and rebound is increased along with the rear, the vehicle responds better to the test by not utilising the maximum ability of the tyres (Williams, 2016). Setup 6 has the same amount of lateral acceleration as setup 3 and also the same amount of understeer, this suggests that the coefficient changes at the front of the car don't have much effect when the rear suspension is at maximum bump and compression.



7.6) Setups 7, 8 and 9

7.6.1) Damper Speed Histograms



Figure 26 - Damper speed histograms for lane change manoeuvre setups 7, 8 and 9 (Deion Atkinson, 2020)

A further increase in damping at the front of the vehicle has caused the histograms of setup 8 to lose the symmetrical shape it achieved in setup 5. Looking at the front left and rear right, setup 8 is symmetrical but on the front right it is bump biased and rear left is rebound biased, therefore for the test in question, a reduction in bump or increase in rebound at the front right and the opposite for the rear left would achieve a more symmetrical shape for these corners of the car. With both front and rear at maximum bump and rebound, the histograms show that there is slightly too much bump. Considering that both bump and rebound coefficients are 803, Kasprzak (2014) mentions that the rebound coefficient of dampers are usually higher because they control the motion of the sprung mass, suggesting that this could be a solution as to why setup 9 spends a lot more time in bump (Kasprzak, 2014).



7.6.2) Understeer and Oversteer

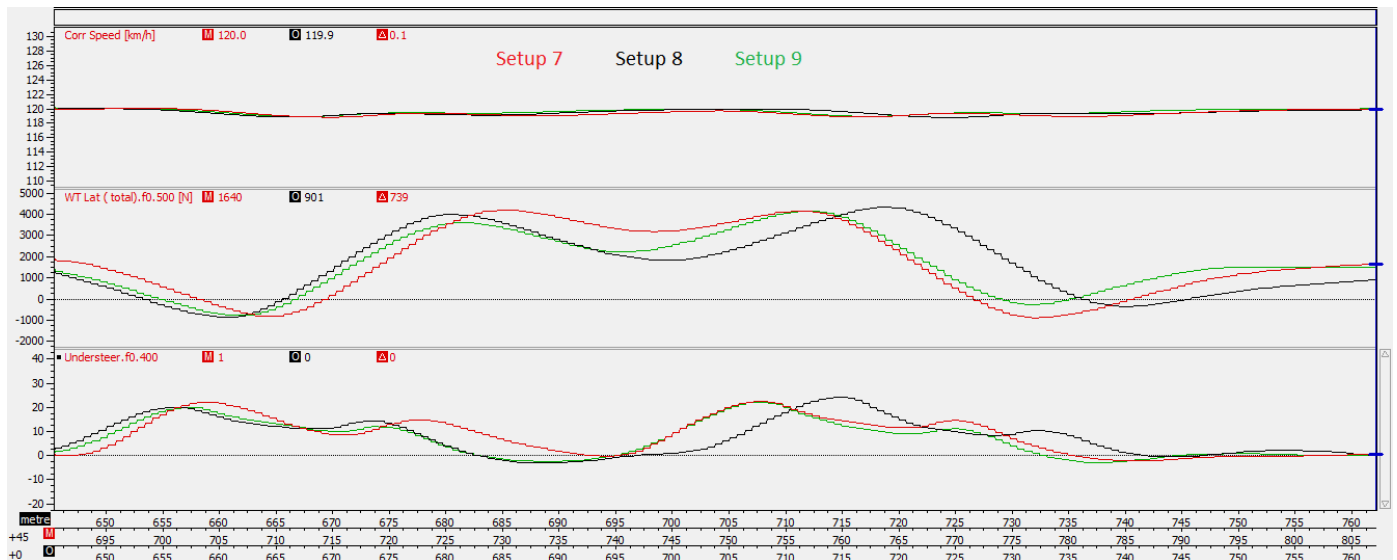


Figure 27 - Understeer channels for lane change manoeuvre setups 7, 8 and 9 (Deion Atkinson, 2020)

Opposite to the other two data sets, the understeer value doesn't decrease as the coefficient increases. The highest amount of understeer in figure 27 is setup 7, this is due to the front dampers being at maximum and the rear being at minimum, this will increase the speed of weight transfer towards the front and see an increase in understeer (koni, 2016). As the rear of the car gets stiffer, the understeer doesn't decrease at the same rate of the other two data sets. This is an indication that the front of the vehicle is too stiff and causing the car to understeer nearly twice as much in comparison to setup 6. It can also be said that both setups 8 and 9 oversteer during the transition of lanes, this is a compromise of stiffening the rear dampers as stated by Suspension Secrets (2019), the way to reduce oversteer is to reduce the rear damping stiffness (Suspensionsecrets, 2019).



7.6.3) G-G Scatter Plot

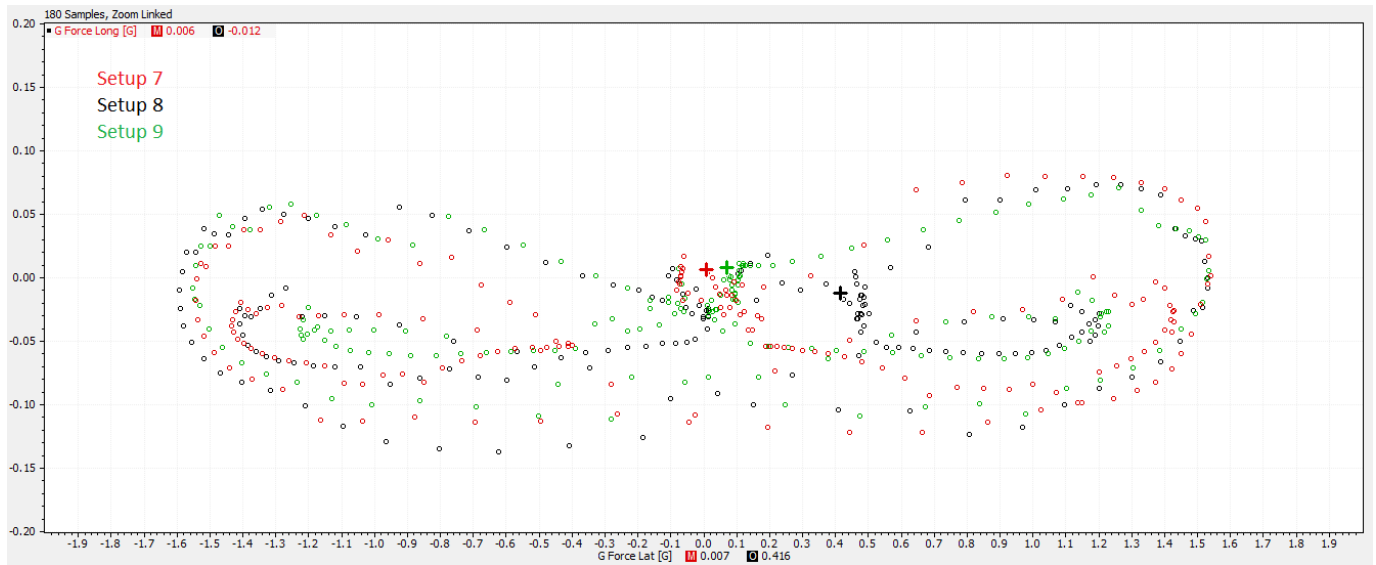


Figure 28 - G-G Scatter plot for lane change manoeuvre setups 7, 8 and 9 (Deion Atkinson, 2020)

All three setups have 1.55 G of lateral acceleration so from this perspective, there are no differences between them. Each of these setups show 0.5 G more acceleration than setup 3 and a further 0.5 G more than setup 6. This is an indication that setup 6 shows the least value indicating that it uses the least potential of the tyre during the test. Setup 1 and 2 have a maximum of 1.6 G whereas setup 4 increases to 1.65 G which makes these three setups (7,8,9) slightly better, but not the best. Slight differences in the G-G diagram also support the findings in figure 26 where the understeer differences are only minute. Koni (2016) suggests that if the bump coefficient is decreased then the weight transfer would slow down and decrease the understeer of the car (koni, 2016). Linking the statement by Koni (2016) and the results of the histograms, if the bump coefficient was decreased to achieve a symmetrical shape, the understeer would decrease along with the lateral acceleration values (Ernst, 2013) (koni, 2016).



7.7) Setups 10 and 11

This part of the test was conducted to assess the effect of bump and rebound separately where rebound will be at full hard and bump will be at full soft and vice versa. The coefficients for each test can be seen in the table below;

Table 8 - Damping coefficients for lane change manoeuvre setups 10 and 11 (Deion Atkinson, 2020)

Setup	Front Bump	Front Rebound	Rear Bump	Rear Rebound
10	268	-803	268	-803
11	803	-268	803	-268

6.7.1) Damper Speed Histograms



Figure 29 - Damper speed histograms for lane change manoeuvre setups 10 and 11 (Deion Atkinson, 2020)

It can be seen in figure 29 that setup 10 spends most of its time in rebound and setup 11 is the opposite, spending most of its time in bump. Both bump and rebound coefficients are the same at max and min therefore, when rebound is stiffer than bump, the histograms seem to be spending an average of 40 percent of their time at very low speeds. Setup 11 seems to be slightly better shaped, but the front left damper spends a lot more time at lower velocity's than the other corners, this is a combination of too much bump or not enough rebound to settle the histogram. As quoted by Motec (2019), 'To spend more time in a region, stiffen that region. To spend less time in a region, soften that region', therefore reducing the bump in setup 11 would compensate for the lack of rebound experienced (Motec, 2019).



7.7.2) Understeer and Oversteer

Between both setups in figure 30 there isn't much difference in understeer or weight transfer. The minor difference in weight transfer is that at the final transition back onto the original lane, setup 10 encounters slightly longer weight transfer which also induces more understeer at the same point. This suggests that rebound has more of an effect on the vehicles handling, even though they are minor differences.

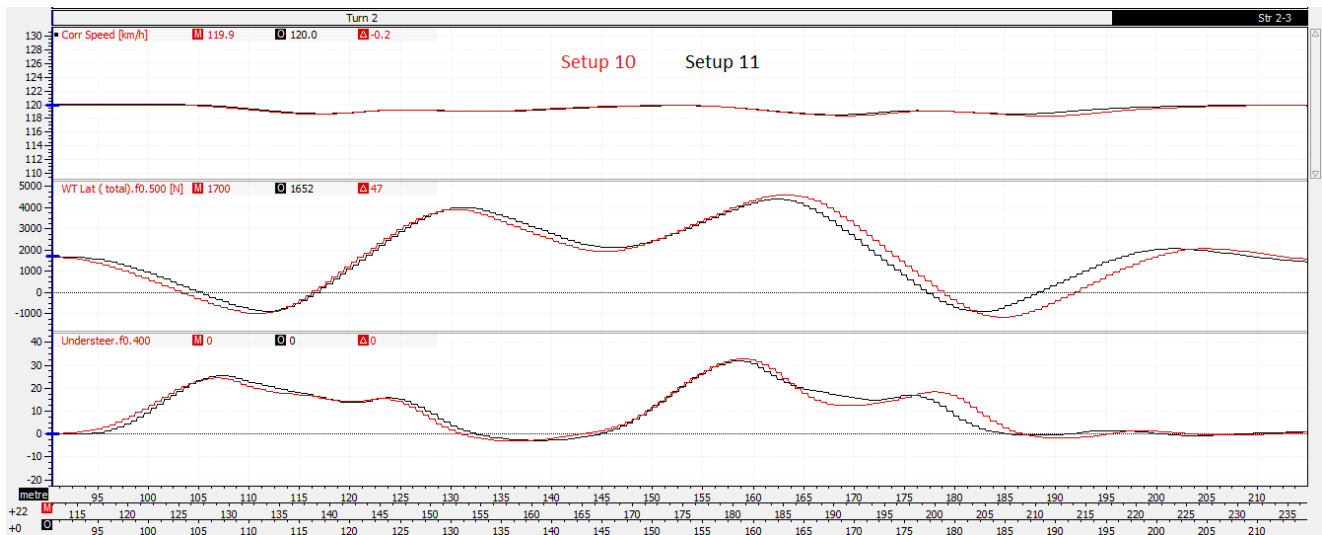


Figure 30 - Understeer channels for lane change manoeuvre setups 10 and 11 (Deion Atkinson, 2020)

Setup 11 seems to have a longer lasting amount of understeer at all points in the test and this is, according to Rapid-racer (2016), the effect of too little rebound. This occurs when the suspension of the car will not settle as fast and will oscillate for longer (Rapid-racer, 2016). Rapid-racer (2016) and Suspension secrets (2019) also suggest that if there is too much bump and too little rebound, understeer will occur due to the weight transfer velocity. An increase in rebound at the rear of the car along with a low damping coefficient of bump will cause the vehicle to roll slightly more and cause oversteer at corner exits, which is clear at the last transition of lanes in the test (Rapid-racer, 2016) (Suspensionsecrets, 2019).



7.7.3) G-G Scatter plot

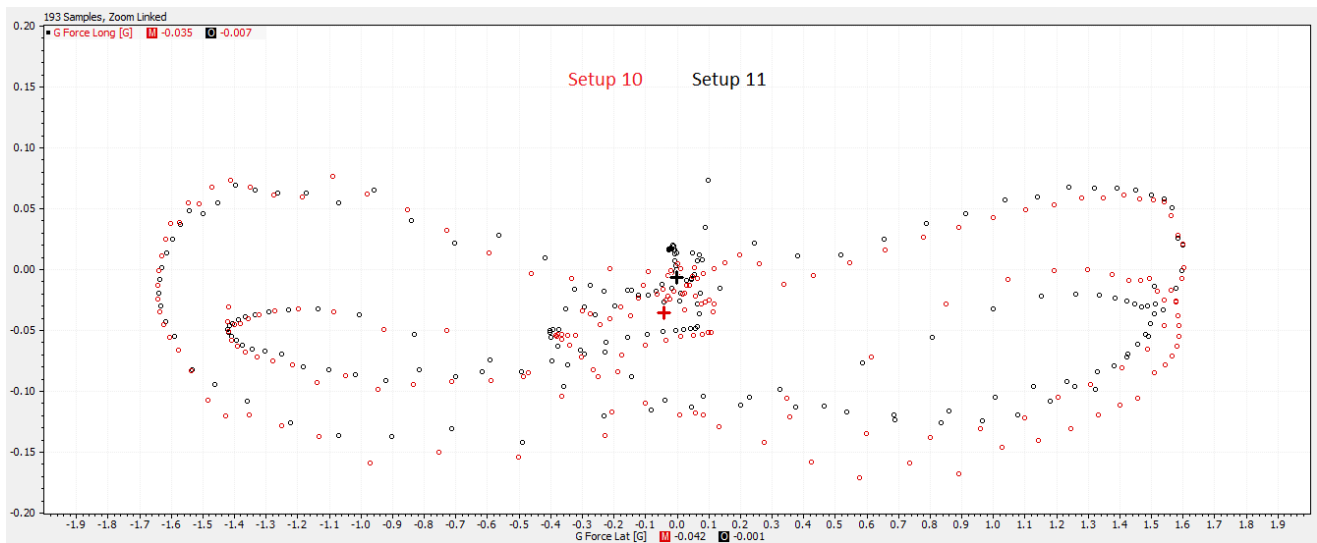


Figure 31 - G-G Scatter plot for lane change manoeuvre setups 10 and 11 (Deion Atkinson, 2020)

Like the understeer values of both setups, there are only minor differences between the two on the G-G diagram. Setup 10 seems to experience 0.01 G more than setup 11 which relates to figure 30 as it indicates that it is using more of the tyre's ability. Setup 10 shows the same values as setup 4 which also had a high understeer value of 32. Looking at the shape of the traction circle, setup 10 has more of a circular shape to it. This means it has larger longitudinal acceleration values when bump is stiffer than rebound. This is a result of underdamping (not enough bump) where the suspension system will repeat the movement until the car returns to its original ride height. This then increases the loads on the tyres and uses more of the available grip than setup 11 (Rapid-racer, 2016).



7.8) Results Comparison

7.8.1) Understeer

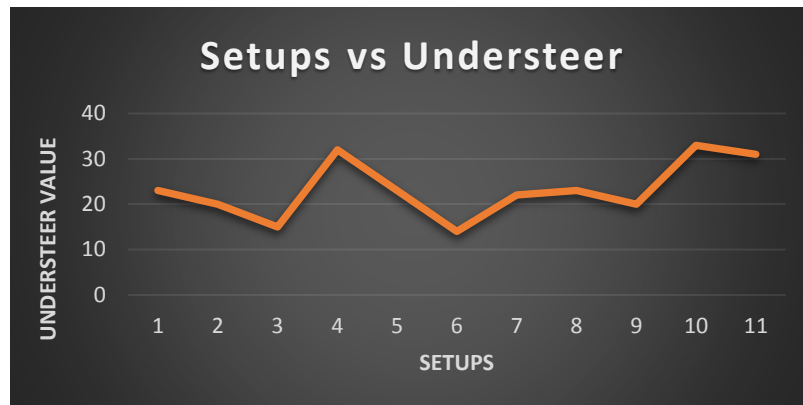


Figure 32 - Understeer graph for all setups on the lane change manoeuvre (Deion Atkinson, 2020)

As the rear of the car gets stiffer towards setup 3 the understeer value decreases but setup 4 sees the second highest value of understeer, this is when the front of the car is at mid and the rear is at soft. From here, the value decreases again following the same trend as the first three setups. Setup 6 has slightly less understeer than setup 3, indicating that the least understeer is experienced when the front is at mid and the rear is at full hard. There are no other trends to be seen after this point due to the front of the car being too stiff, raising the weight transfer velocity along with the understeer (Rapid-racer, 2016). Comparing bump and rebound setups, when bump is stiffer the understeer value is slightly less than when the rebound is stiffer, but these two setups are of the highest understeering along with setup 4.

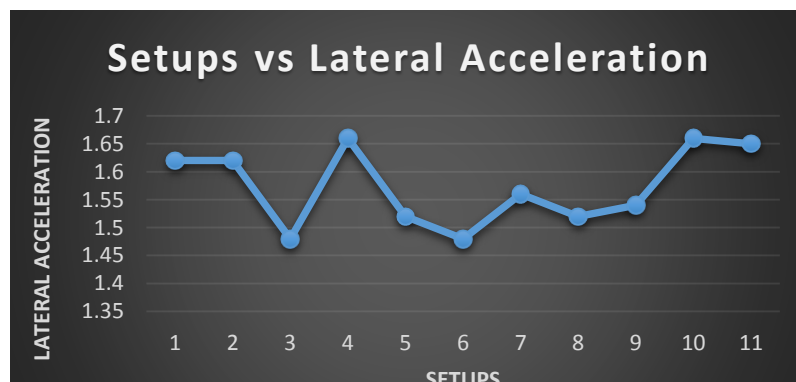


Figure 33 - Lateral acceleration graph for all setups on the lane change manoeuvre (Deion Atkinson, 2020)

Looking at figure 33, the lateral acceleration follows the same trends as the understeer where the value decreases as the rear gets stiffer. The lower the value the better the setup as there is more tyre grip available, therefore the car has the ability to travel a lot faster through the test.



7.8.2) Oversteer

The oversteer of each setup was taken from the same point of each manoeuvre. This is at the last transition where the vehicle moves back onto the original lane.

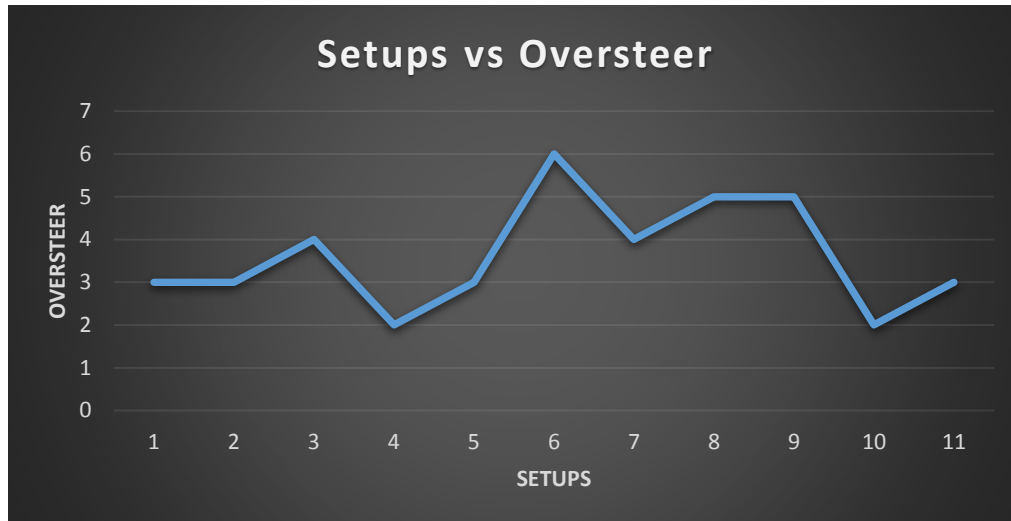


Figure 34 - Oversteer graph of each setup on the lane change manoeuvre (Deion Atkinson, 2020)

Figure 34 follows a different trend between setup 1 and 9 where the oversteer increases as the front of the car is softened. Setup 6 has the most oversteer as the rear of the car is stiffer, this is a consequence of having less understeer. Also, setup 10 and 11 have the least oversteer as they have the most understeer along with setup 4 that has a low amount of oversteer. A compromise between figure 32 and 33 would be setup 5 where it has moderate understeer and oversteer in comparison to the other setups which, in theory, means that this is the most balanced setup. De Groot (2002) mentions that when understeer is a problem, the rear of the car needs to be stiffer and when oversteer is the issue, the front of the car needs to be stiffer (Groot, 2002). This indicates that a compromise between the two needs to be found as an optimum setup and in a race car application, along with these results, the car needs to be set up with the driver's preference as one of the main guidelines to follow.



8) Conclusion

Looking at the overall results of the project, changing the damping coefficients of a vehicle has a drastic effect on how the vehicle responds to driver inputs and handles around a circuit. The real-life data set at Silverstone Circuit shows a positive effect of increasing the damping coefficients on the front of the car where the stiffer setup could carry more speed through some corners and produce a quicker lap time by 2 seconds. When increasing the front of the car around the track it's clear that the understeer increases, with the increase in understeer it returns to steady state quicker than setup 1. Like these results, the stiffer setup on the simulator was also quicker. The simulator G-G diagram has slightly higher values of lateral acceleration over the real-life data where the possible difference could be the tyres adhesion limit. Overall, both the simulator data and the real-life data are very similar in the results of changing the front coefficients, therefore validating the simulator. Along with simulator validation, the similarities in results prove that the steady state analysis equations and experiment was successful as the simulator data's understeer and oversteer characteristics match the real-life data.

Clarifying the double lane change manoeuvre test results, the setup with the least amount of understeer is setup 6. This means that when the front of the car is in mid setting, the understeer values are low. A sacrifice for this is oversteer, where setup 6 has the most of it. The graph of understeer and oversteer indicates a trend that when the understeer decreases the oversteer increases, and vice versa. In collaboration with the understeer and oversteer, the G-G scatter plot of each setup indicates that setup 3 and 6 have the least amount of lateral acceleration which means that the tyres have slightly more potential grip with these two setups. Taking understeer, oversteer and lateral G force values into consideration, setup 5 doesn't have drastic differences and doesn't sacrifice understeer for oversteer as much as the other setups. This is an indication that setup 5 could arguably be the setup that is most balanced during the double lane change manoeuvre.



9) Further Work

Due to limitations and time constraints on the project, deeper investigation was not possible and limited the project from clarifying any further effects of changing damping coefficients. The problems faced to expand the project included limited software usage on personal equipment and also the inaccessibility of the simulator. Delving further into the project could have been done by implementing other BSI ISO tests such as the Power-off reaction of a vehicle in a turn, Braking in a turn or the Steady state circular driving behaviour (BSI, 2016). Changing the low speed damping coefficients within these test methods would have given a further, more detailed analysis of the effects the dampers have on vehicle handling. These three test methods along with the Double lane change manoeuvre illustrate the dynamic behaviour of a vehicle in such situations that happen on a daily basis on public roads (BSI, 2016). These various tests would further validate the effect of changing low speed damping coefficients by looking at different aspects of vehicle handling such as the throttle application and brake application while cornering.

The real-life data of Silverstone could have also been taken into further detail by examining other damping coefficients on the front and also the rear of the car. This would have validated the simulator even better with further evidence of similarities between the two data sets. Real-life data cannot be collected in house and was therefore, unfeasible due to the team not doing any winter testing and the remainder of the 2020 testing and race days being cancelled.



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11) Appendices

1.A) Force v Velocity graphs – Setups 1, 2 and 3

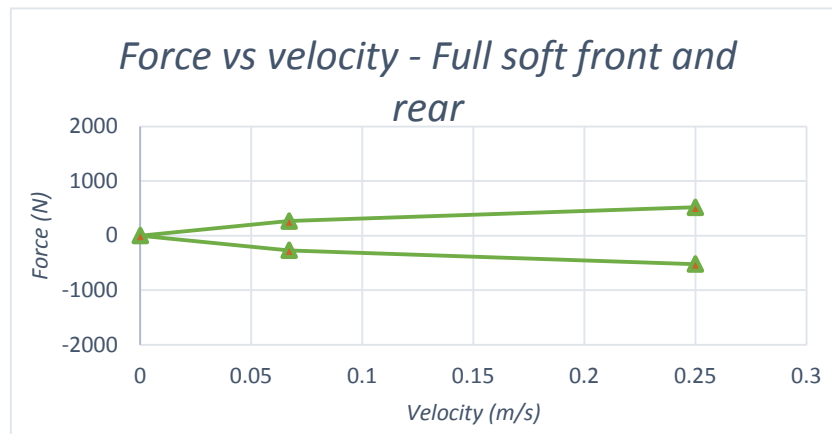


Figure 35 - Force v Velocity graph setup 1 (Deion Atkinson, 2020)

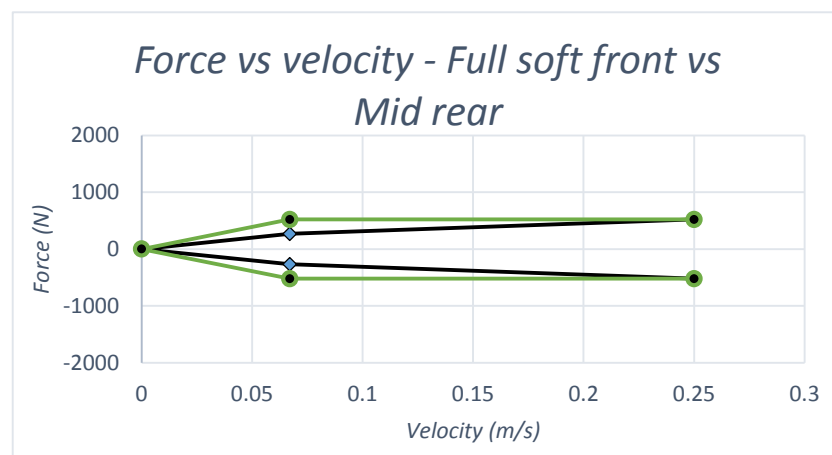


Figure 36 - Force v Velocity graph setup 2 (Deion Atkinson, 2020)

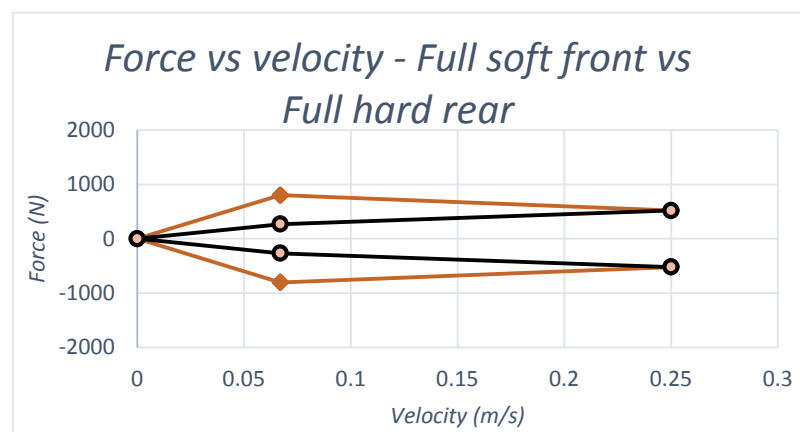


Figure 37 - Force v Velocity graph setup 3 (Deion Atkinson, 2020)



1.B) Force v Velocity graphs – Setups 4, 5 and 6

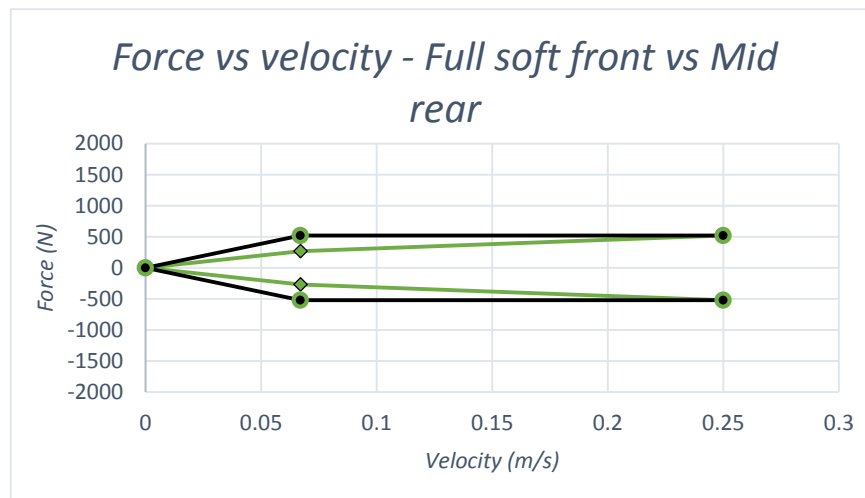


Figure 38 - Force v Velocity graph setup 4 (Deion Atkinson, 2020)

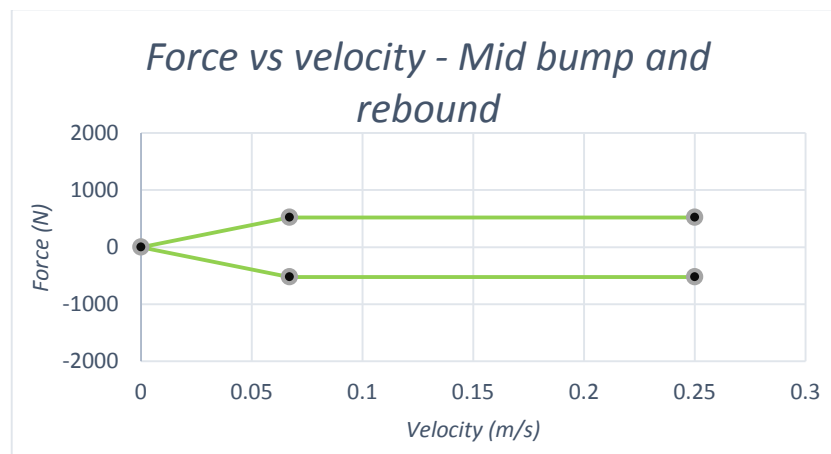


Figure 39 - Force v Velocity graph setup 5 (Deion Atkinson, 2020)

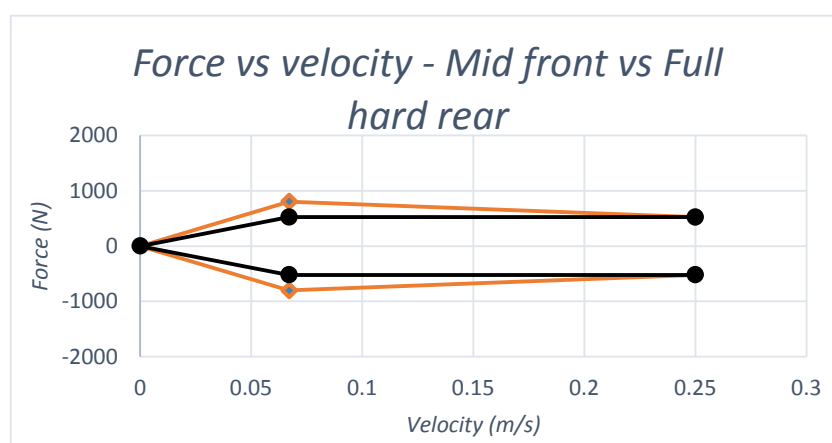


Figure 40 - Force v Velocity graph setup 6 (Deion Atkinson, 2020)



1.C) Force v Velocity graphs – Setups 7, 8 and 9

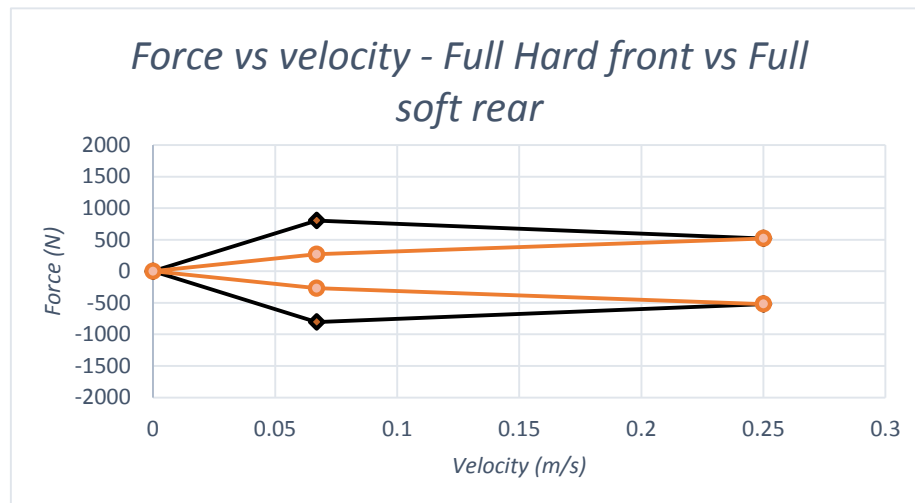


Figure 41 - Force v Velocity graph setup 7 (Deion Atkinson, 2020)

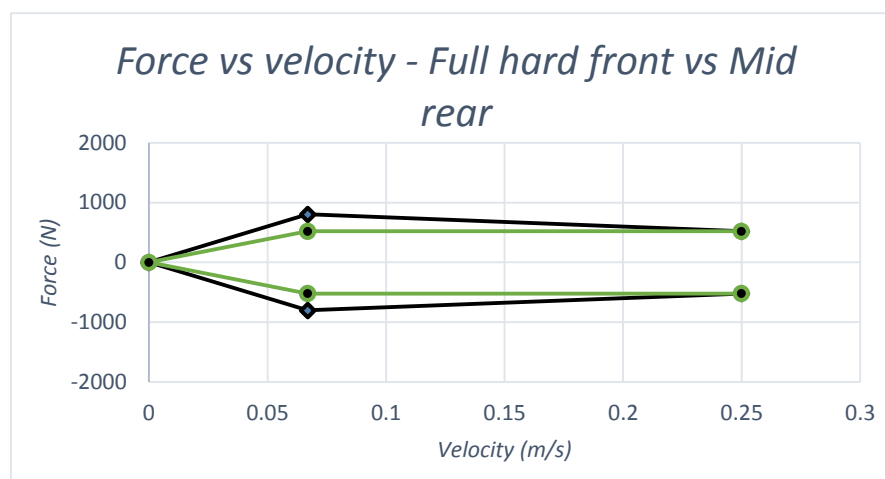


Figure 42 - Force v Velocity graph setup 8 (Deion Atkinson, 2020)

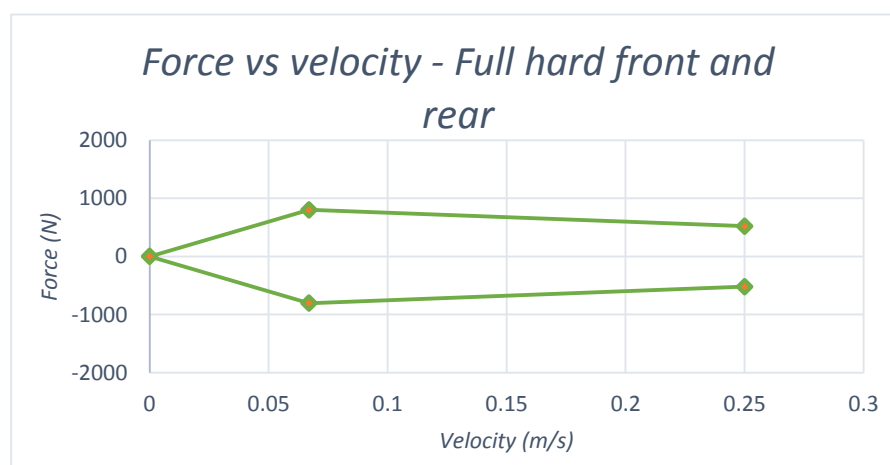


Figure 43 - Force v Velocity graph for setup 9 (Deion Atkinson, 2020)



1.D) Force v Velocity graphs – Setups 10 and 11

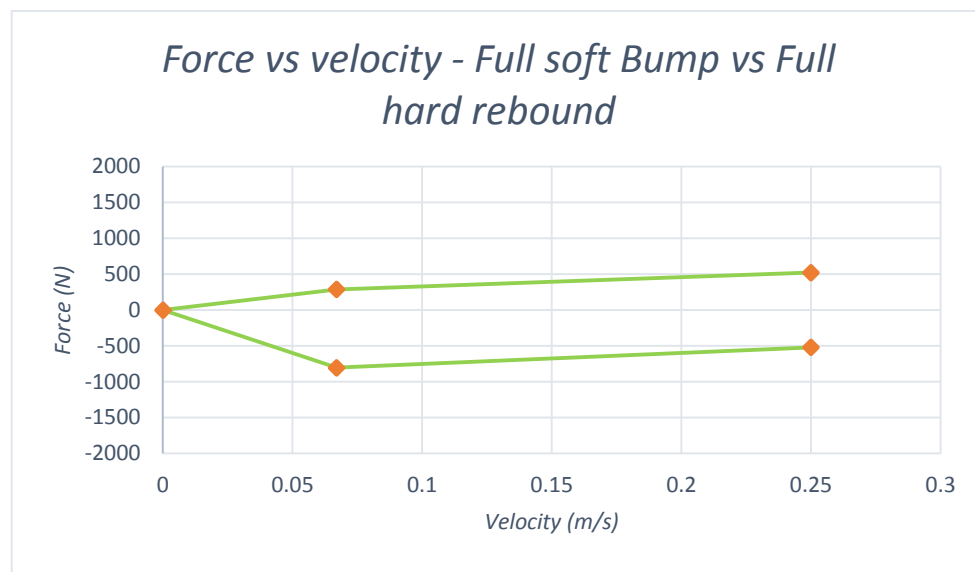


Figure 44 - Force v Velocity graph for setup 10 (Deion Atkinson, 2020)

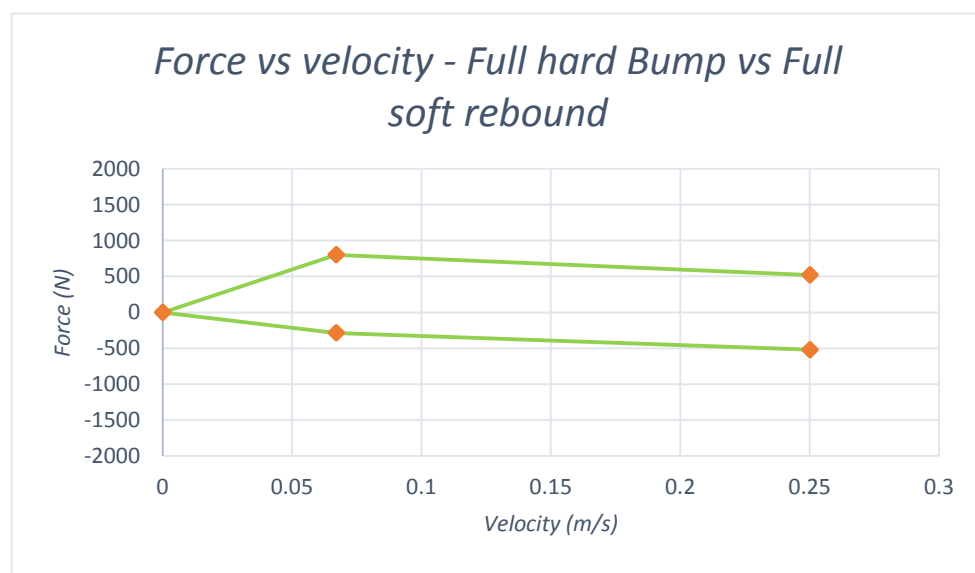


Figure 45 - Force v Velocity graph for setup 11 (Deion Atkinson, 2020)