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Understanding spring rates and damper settings of sports 2000 car using driver in the loop simulator

1 Abstract

Damper and spring rates have always been a passion since studying engineering from working on different settings from racing mountain bikes as well working with racing drivers on damper settings for a Radical sr8 for RAW motorsports. Having previously worked with driver in the loop simulator it seemed important adjusting damper and spring rates in the simulator was an obvious choice for the project.

1 Contents

1	Abstract.....	2
2	Introduction	5
3	Aims and objectives	6
4	Timing plan.....	7
4.1	How could project be delayed	7
4.2	Pandemic impact.....	7
5	Background research	8
5.1	Dampers.....	8
5.1.1	Characteristics.....	8
5.1.2	Damper characteristics effect on handling.....	10
5.1.3	Damping ratio	10
5.1.4	Damper construction	11
5.1.5	Damper internals	13
5.1.6	Damper curves	15
5.1.7	Histogram.....	17
5.1.8	Adjusting compression displayed through histograms.....	18
5.1.9	Adjusting rebound displayed through histogram	19
5.2	suspension kinematics	19
5.2.1	motion ratio	19
5.2.2	suspension configuration.....	20
6	Methodology.....	22
6.1	Automobilista simulator model	22
6.2	Damper dyno	22
6.3	Real life data from motec	23
6.4	Real damper forces inputted into BPS simulator model	23
6.4.1	Low speed testing	24
6.4.2	High speed testing.....	25
7	Results and analysis	27
7.1	Inputting real damper forces into the BPS simulator model	27
7.1.1	Histograms	27
7.2	Analysing the handling effect of the real damper forces in the simulator model.....	28
7.2.1	Low speed compression.....	28
7.2.2	Low speed rebound.....	32
7.2.3	High speed compression	36
7.2.4	High speed rebound.....	40
7.3	Analysing the damper internals	44
7.3.1	Main piston	44

7.3.2	Shim stacks.....	45
7.3.3	Needle valve.....	46
7.4	Analysing the handling effect of the new spring rate settings	46
7.4.1	Spring rate effect on handling.....	47
7.4.2	Spring rate over bumps.....	50
7.5	Analysing the handling effect of the new damper coefficients	51
7.5.1	Low speed compression.....	51
7.5.2	Low speed rebound.....	54
7.5.3	High speed compression	58
8	Discussion.....	66
8.1	Spring set up guide.....	66
8.2	Low speed set up guide.....	66
8.3	High speed set up guide.....	67
9	Conclusions	69
10	Appendices.....	70
10.1	Comparison between automobilista and real life	70
10.2	Force vs velocity graphs	70
10.3	Histograms	70
10.4	Sprung and unsprung frequencies of the MCR.....	71
10.5	New coefficients low speed testing	71
10.6	New coefficients high speed testing	71
10.7	Maths channels	72
11	References	73

2 Introduction

This project is a study into evaluating the current spring rate and damper settings of the MCR from track data and damper dynamometer testing which will then be imported into the simulator model to show the effects on handling. The damper settings that will be investigated will be low/high speed compression/rebound and what affect they have on the force vs velocity graph, histograms and handling and what is the relationship between the three. Once evaluated the existing spring and damper rates will be compared against new spring and damper rates that would be considered “ideal” and to see how that effects the handling of the MCR with a suggested set up handling guide at the end to summarise the handling effects of spring and damper rates.

3 Aims and objectives

- What affect does spring rate have on handling and how does spring rate relate damper coefficients?
- What affect does low speed compression have on histograms and handling?
- What affect does low speed rebound have on histograms and the effect handling?
- What affect does high speed compression have on histograms and handling?
- What affect does high speed rebound have on histograms and handling?
- Produce a set up guide for the sports 2000 for the dampers rate and spring rates depending on the desirable handling balance and provide data to back up the validity of the findings.

4 Timing plan

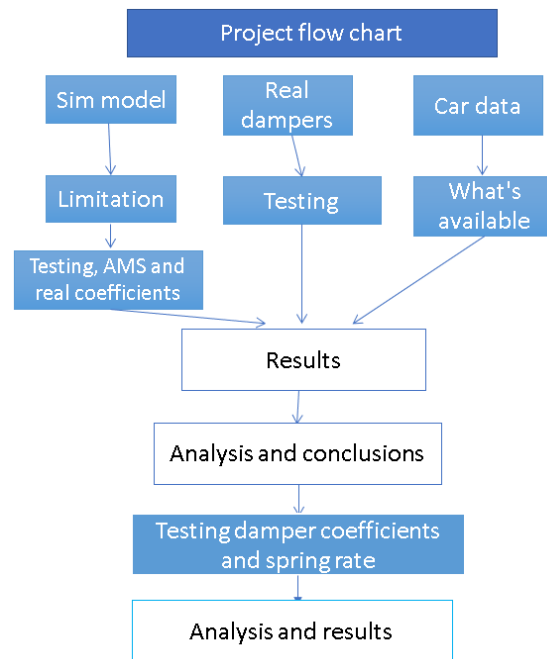


Figure 3-1 Flow chart of the project

4.1 How could project be delayed

In the early stages of the project there could be little to delay the process as all of the data is been done generated in the simulator which is available anytime as long as the simulator model works, otherwise there could be correlation errors between real life and the simulator. The delays could come from the damper dyno testing as the dampers need to be taken off the car which was only available for a period of 3 weeks which started from the end of October but the main delays would be using the damper dynamometer later in the project modifying the dampers and the dynamometer broke which means examining the damper internals were speculative and no data to back up findings.

4.2 Pandemic impact

The only area the pandemic impact had on the project was using the damper dynamometer in university, but all of the simulator testing was conducted beforehand. This meant that modified force vs velocity graphs couldn't be compared against modified simulator data. If the project were to continue this would be the area to focus on.

5 Background research

5.1 Dampers

Dampers are designed to reduce and control oscillations from impacts acting on the wheel. For example, a wheel would hit a bump, the damper will compress and absorb the wheel movement reducing the impact on the chassis. The damper will then extend controlling the oscillation. The main purpose of dampers are to control the unsprung and sprung mass as well absorb any harsh impacts from the road. They can also control the pitch, roll and squat which can affect the characteristics of the car handling. (Giraffa, n.d.) (Bastow, 2004) (Mcbeath, 2002)

5.1.1 Characteristics

5.1.1.1 Compression

Compression is the force reacting against the damper decreasing displacement, for example hitting a bump. Compression controls the unsprung mass by controlling the movement of the wishbones, upright and wheel. More compression force restricts the movement of the unsprung mass causing tyre force variation, not enough compression force allows for too much movement of the unsprung mass and can overload tyres. The lower the compression the less resistance against the damper movement resulting in softer force. (Kasprzak, n.d.) (J.Rowley, 2003)

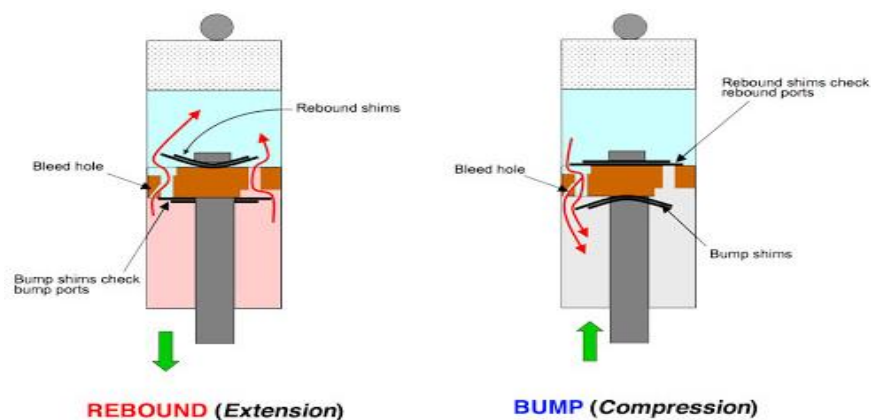


Figure 5-1 Diagram showing damper internals through compression and rebound

5.1.1.2 Rebound

Rebound is the speed of damper returning to full stroke from displacement under compression.

Firm rebound force - slower the damper recovers displacement

Soft rebound force - faster the damper recovers displacement

Rebound force controls the sprung mass which has a larger effect on handling as the sprung mass weighs more than the unsprung mass (controlled by compression). This is why there tends to be more rebound force than compression to keep the body of the vehicle stable. (pitch, roll and squat)

5.1.1.3 Damper speed ranges

Dampers can provide different forces based on the velocity, this would be low and high speed.

- 0-0.1 m/s- low speed- roll, pitch and squat.
- 0.1-0.250 m/s- high speed – big impacts, kerbs and bumps

5.1.1.4 Low speed

Controlled by the bleed holes in the main piston and needle valve design. (Kasprzak, n.d.)
For circuit racing this is where the majority of adjustment will make a difference as dampers velocities only go into high speed over large bumps and kerbs.

5.1.1.5 High speed

High speed is controlled by the quantity of shim stacks and the thickness of the shims. Once the bleed holes in the piston start to choke (no more oil can pass through) the shim stack opens and the amount of oil passing through will control the coefficient.

Figure 5-2 is an open diagram of the difference in low and high speed adjustment of a piggyback on a monotube damper.

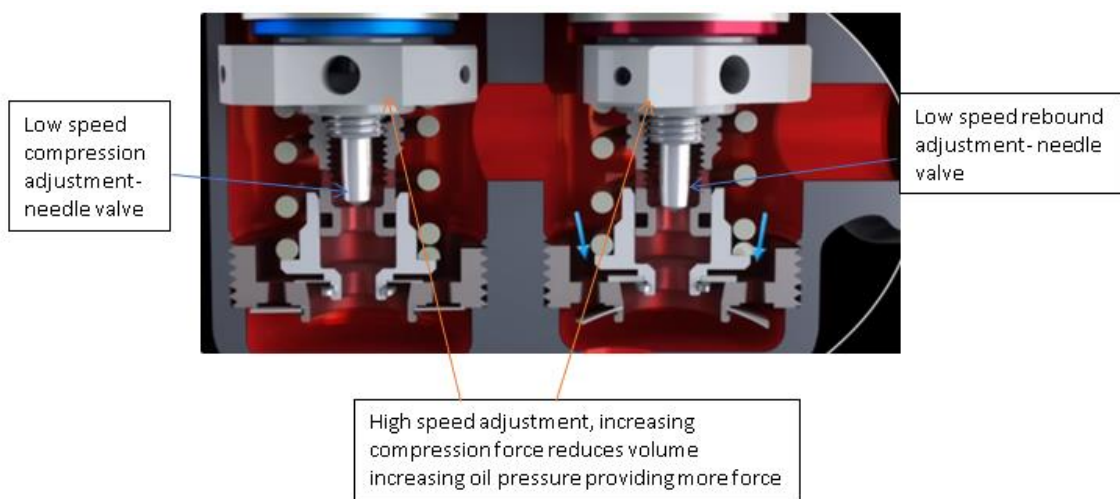


Figure 5-2 Internals of fox dhx2 damper showing low and high speed adjust

5.1.2 Damper characteristics effect on handling

The table below summarises the effects of damper forces on handling. (Smith, 1978)

Compression		
Low speed	Too much	Chassis roll slow to develop Reduces understeer on initial turn in
	Too little	Dives and squats excessively Roll velocity increases
High speed	Too much	Doesn't absorb impacts from kerbs and bumps Instability over bumps
	Too little	Plush feeling Car could ground over bumps and compressions
Rebound		
Low speed	Too much	Wheels don't return to ground quickly after displacement. Overload outside tyre and underload inside tyre
	Too little	Difficult to gain traction at the rear Faster roll velocity
High speed	Too much	Damper packs, doesn't return to full stroke to absorb next impact Firm ride
	Too little	Excessive oscillation over bumps and kerbs Instability over bumps on corner entry, mid corner.

Figure 5-3 Handling guide of the effects of damper settings based on Carroll Smiths findings

As can be seen from the table the majority of the handling effects are controlled by the low speed for circuit racing and the high speed is only for bumps and kerbs which may not be used, for example very smooth race circuits. This is why most dampers at low to mid end cost the damper only focuses on low speed through the needle valve whereas more expensive dampers are 4 way adjustable to adjust the high speed.

5.1.3 Damping ratio

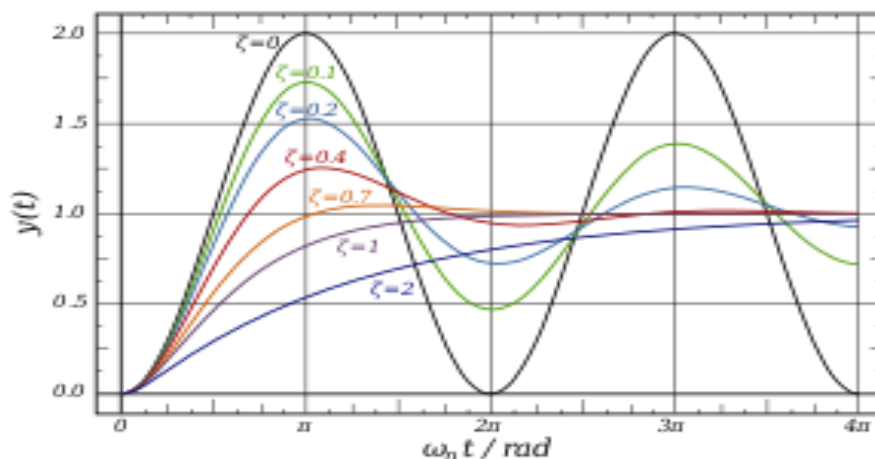


Figure 5-4 Damping ratio graph

The firmness of the damping can be defined by the damping ratio which shows if the suspension is under, critically or over damped. Critically damped is =1 and anything below is under and anything above is over damped. Passengers cars are under damped with a damping ratio around 0.2 with race cars around 0.6-0.7 to provide a stiffer ride. The damping ratio is useful guide to find out how the dampers are performing with a reference to different types of vehicles. To find the damping ratio the level of critical damping needs to be calculated.

Damping Ratio

$$\xi = \frac{C}{C_{cr}}$$

ξ = Damping Ratio (N-s/m)
 C = Damping coefficient (N-s/m)
 C_{cr} = Critical damping coefficient (N-s/m)

Critical Damping Coefficient

$$C_{cr} = 2\sqrt{K_S * M}$$

C_{cr} = Critical Damping Coefficient (N-s/m)
 K_S = System Spring Rate (N/m)
 M = System Mass (kg)

5.1.4 Damper construction

5.1.4.1 Comparison between twin tube to a monotube

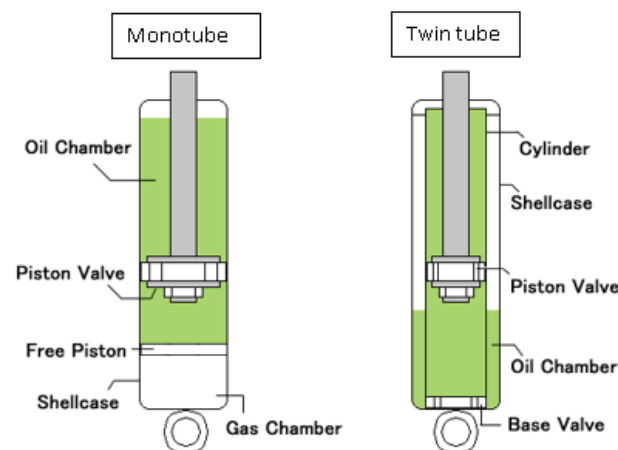


Figure 5-5 Comparison of a twin tube to mono tube internals

In a twin tube damper there is no piston or barrier between the oil chamber and gas chamber which means that they need to be mounted upright to avoid cavitation, so the nitrogen gas is at the top of the outer tube diameter pressurising the oil, otherwise air escapes into the main chamber depressurizing the damper causing bubbles in the main chamber.

A base valve is designed to let oil through into the outer chamber, this can cause oil shear as there is a lot of oil passing between the outer chamber to the main chamber through a tight orifice increasing oil temperature. (F. Milliken), This more likely to overheat because of the reduced inner tube size which can decrease compression and rebound force as the oil viscosity decreases. Also the main tube isn't exposed to ambient compared to a monotube which also increases oil temperature. (Anon., n.d.) Below is a table summarises the difference between twin tube to a monotube.

Twin tube		Monotube	
Advantages	Disadvantages	Advantages	Disadvantages
Cheap / inexpensive	Damper needs to be mounted up right, otherwise air leaks into the main tube causing cavitation – 35% less performance	Oil separated from the gas for a gas separator piston, no cavitation	The damper length increases as it's harder to mount the damper as the main tube is longer
Compression piston adjustment	Gas and oil mixing reduce damping forces which leads to aeration	Can be mounted in any direction	Piggyback can avoid increasing damper length but can also be more difficult to mount and can lead to cross torque with certain pipe lengths
External damage to the outer tube does not affect the internals of the damper- less prone to damage	Smaller main tubes, resulting in smaller piston size causing higher temperatures resulting in more fade (less compression and faster rebound)	Main working tube is larger resulting in less damper fade compared to a twin tube damper	Can run higher gas pressure increasing the pressure of the oil in the damper increasing the damping forces, ideal for circuit racing.
Lower gas pressure because of the valve at the base of the damper, less compression providing more grip	Generally less damper forces, more ideal for off road rallying compared to circuit racing which require more damper force	The compression needle valve is larger increasing the compression coefficient and low speed force	The main tube is exposed to ambient temperature which means that there's less damper fade.
Less stress on the seals as there is less gas pressure compared to a monotube			More expensive

Figure 5-6 Table comparing monotube to a twin tube

5.1.4.2 Monotube with piggyback

The diagram below shows a variation of different approaches to the “piggyback” design.

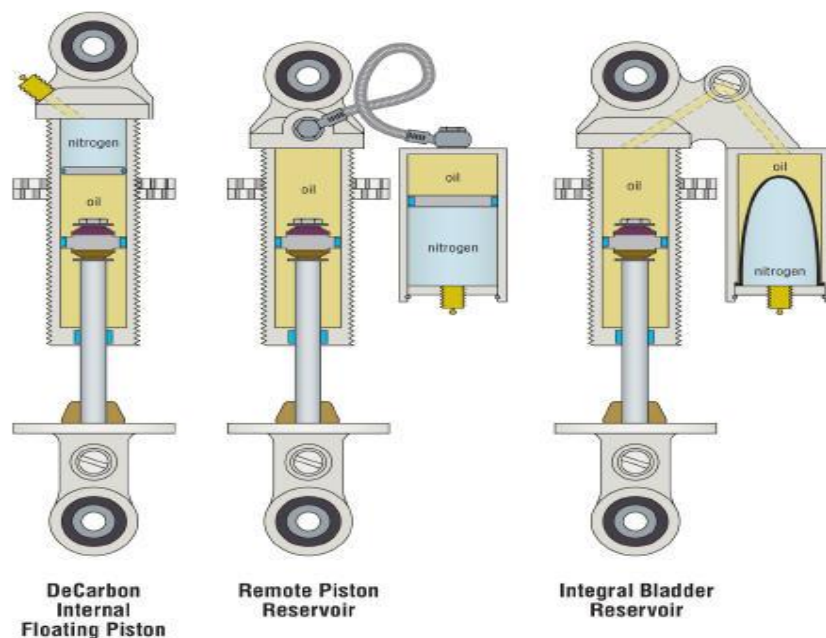


Figure 5-7 Comparing a standard monotube to a monotube with a piggyback

5.1.4.2.1 Advantages of a piggyback compared to a monotube

- Higher quality construction
- Reduces the damper length so can be mounted easier than a standard monotube.
- The suspension system can be tighter packaged as the shock length is smaller, reduces unsprung mass.

5.1.4.2.2 Disadvantages

- With a piggyback design oil has to pass through a tube before it can reach the piggyback. This takes longer to react the piston displacement which can cause cross torque, for example increasing compression will restrict the oil passing into the piggyback, once the damper goes into rebound there is still a high volume of oil in the hose which is trying to go back into the main chamber which increases rebound force.
- Some of the damping is in the piggyback instead of the main chamber-faster for the oil to overheat.
- Increased pressure on the hose between the piggyback and main chamber- increases wear.

5.1.5 Damper internals

5.1.5.1 Main piston tube

The main tube of the damper controls the amount of oil in the damper which will control the default damper force, for example large main tube and less oil, softer damper force.

5.1.5.2 Reservoir tube

Outer tube on a twin tube damper and creates the area for extra oil to through a base valve, this increase oil shear and damper fade as the oil is restricted by the valve increasing damper temperature. The gas pressure is in the reservoir tube to pressurize the oil but needs to be mounted upright as there is no gas separator piston like in a monotube. (Kasprzak, n.d.)

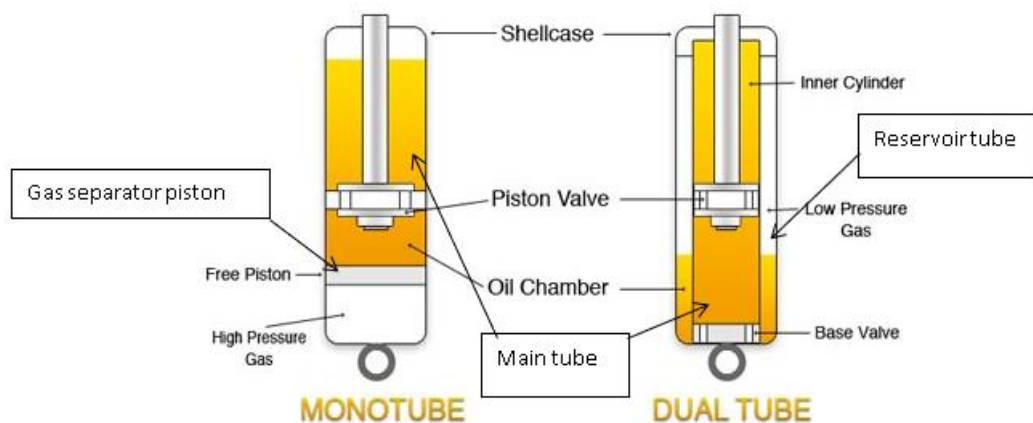


Figure 5-8 Diagram showing the difference between a monotube and twin tube internals

5.1.5.3 Gas separator piston

Keeps the gas separated from the oil in a monotube tube damper which stops cavitation and also pressurizes the oil in the main chamber. (Kasprzak, n.d.)

5.1.5.4 Main piston

Produces the majority of the damping force as the hole size controls the oil movement in compression and rebound.



Figure 5-9 Showing the difference between rebound and compression side of the main piston

The smaller the diameter the less oil can travel through the hole which increases the low speed coefficient producing more damper force. This also affects the knee point speed (switch from low to high speed) as the smaller the hole size the quicker it takes for the holes to choke and open the shim stacks for high speed damping. Therefore the smaller the holes the increased coefficient and a lower velocity knee point and larger the holes the lower the coefficient and the higher the knee point velocity.

As can be seen from figure 5-9 there are differences in the quantity and diameter of the holes between the compression and rebound face. The rebound face has less holes and smaller diameter as the rebound force is greater than the compression force to control body movement of the vehicle. (Kasprzak, n.d.)

5.1.5.5 Needle valve

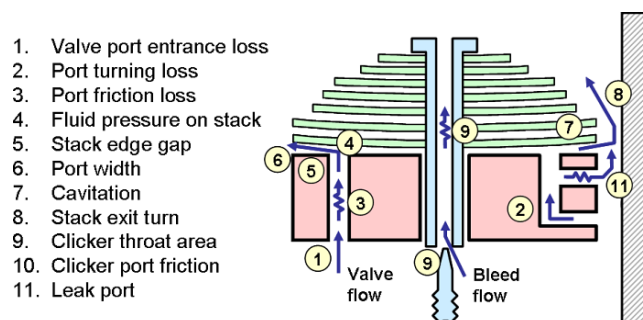


Figure 5-10 Diagram of the needle valve design

Controls the low speed coefficient adjustment and controls the amount of oil that passes through the tapered valve. To increase the damper force the needle valve winds in to restrict the oil passing through the tube which can be seen in figure 5-10 from the adjuster. By restricting the oil flow passed the valve this increases the damper coefficient. The valve design can also affect the damper coefficient because a needle with a short tip that is wide can provide a larger increment when adjusted, but also produce high damper force as a wider taper restricts more oil compared to long thin needle.

5.1.5.6 Shim stacks

Controls the amount of oil to pass through with more shims requiring more force from the oil to open them to cause damper movement. This means more shims increase the high speed coefficient. There will be more shim's in rebound compared to compression as the rebound force needs to be greater to increase stability.

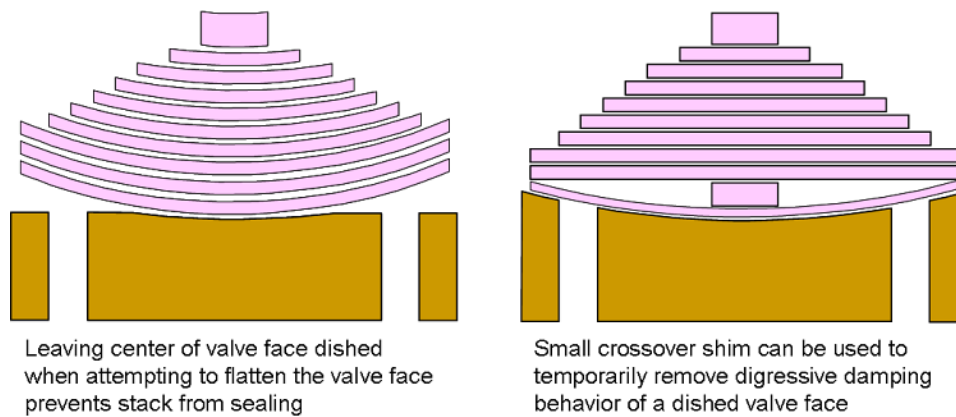


Figure 5-11 Diagram of shim stacks above the main piston

5.1.6 Damper curves

5.1.6.1 Force velocity graph

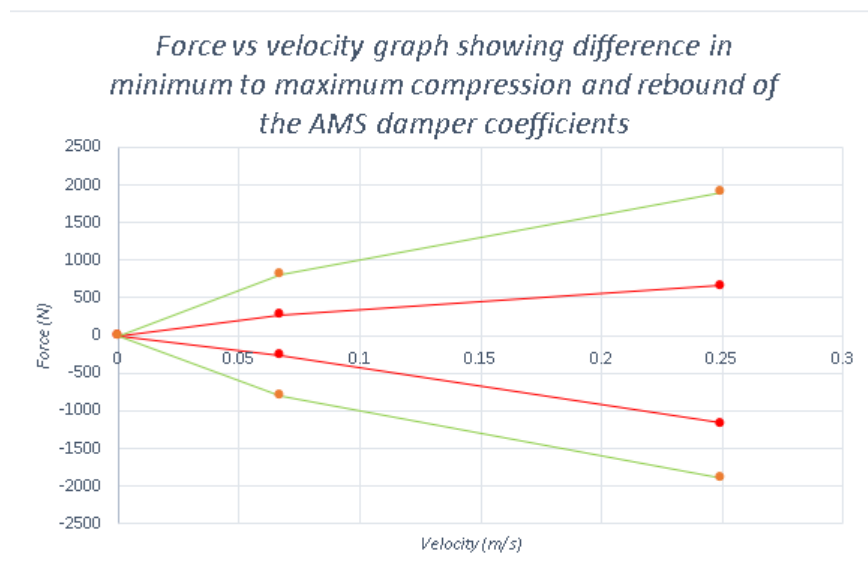


Figure 5-12 Force vs velocity graph of the AMS damper coefficients

Force vs velocity graphs show the force produced at a certain velocities; the gradient of the lines are the damper coefficients. The steeper the gradient the higher the damper coefficient (stiffer) which will result in more damper force.

5.1.6.1.1 low speed

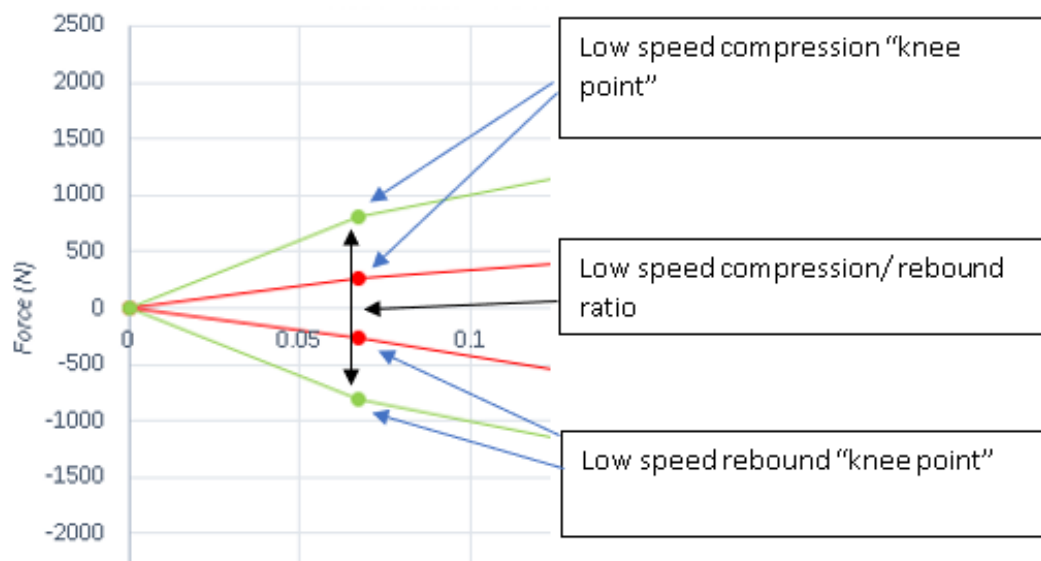


Figure 5-13 Force vs velocity graph focusing on low speed

The “knee point” of a force vs velocity graph is the velocity at which the dampers move from low speed to high speed. As can be seen in figure 5-13 the low speed coefficients (gradient of the line) are steeper than the high speed gradient. This is because you want the low speed damper coefficient to be firmer because it provides greater body control, especially in rebound. Whereas high speed you want to be softer, so the damper absorbs the bumps in compression and returns to full damper stroke in rebound after a bump, the balance between compression and rebound can be seen as the compression to rebound ratio.

The knee point speed can affect the low speed coefficient as by having the knee point at a lower velocity the low speed coefficient increases as there is more low speed damper force at a lower velocity, this increases the stiffness of the low speed coefficient. As most low speed damper velocity happens up to 35mm/s maximum by bringing the low speed knee point to 40mm/s compared to 70mm/s it makes it easier to adjust the low speed damping to control body movement.

5.1.6.1.2 High speed

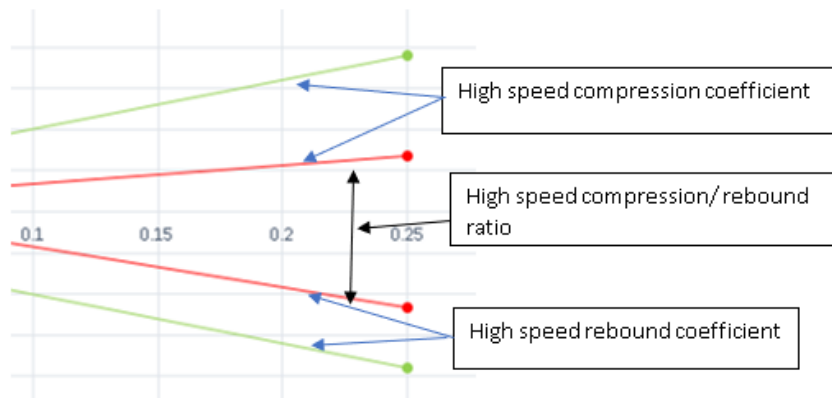


Figure 5-14 Force vs velocity graph focusing on high speed

The high speed section of the force velocity focuses on the damper coefficient and the knee point speed is irrelevant. Just like at low speed the steeper the gradient the stiffer the damper coefficient, as mentioned previously the high speed would need to be softer to absorb impacts such as bumps and kerbs, which results in lower coefficients.

5.1.7 Histogram

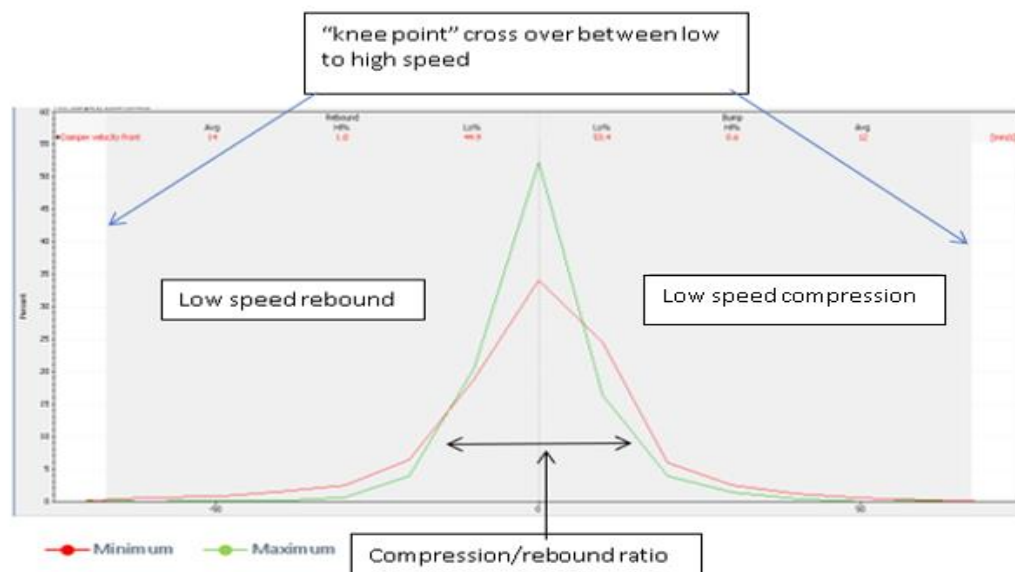


Figure 5-15 Histogram showing the difference from maximum to minimum in compression and rebound

Histograms are used to display what velocities the dampers are operating at; this will depend on the damper settings. A thinner narrower histogram would be a higher damper force/coefficient whereas a wider shorter histogram would be lower damper force/coefficient. The force vs velocity graphs can correlate to the histograms as they should show similar trends as to how the dampers are working. The force vs velocity graph shows how the dampers react on a damper dyno and histograms show how the damper react on track. If the low speed coefficient increase you will see a steeper gradient on a force vs velocity graph and a larger output force, on the histogram the trend would be the thinner and would be spending less time at low speed velocities because the low speed velocities have increased.

5.1.8 Adjusting compression displayed through histograms

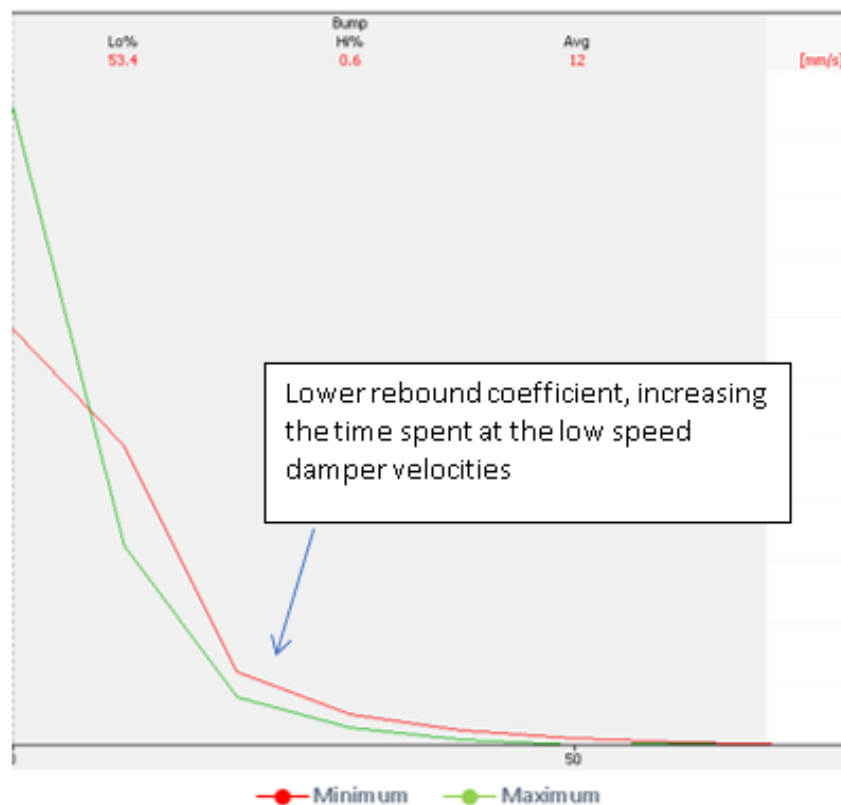


Figure 5-16 Histogram showing the compression adjustment

Figure 5-16 shows the compression side of the histogram and what can be seen is that at minimum compression the damper spends more time at low speed compared to maximum, this will result in a softer ride, but could increase body roll depending on the coefficients, the force vs velocity graph will show a flatter gradient which will produce less damper force.

5.1.9 Adjusting rebound displayed through histogram

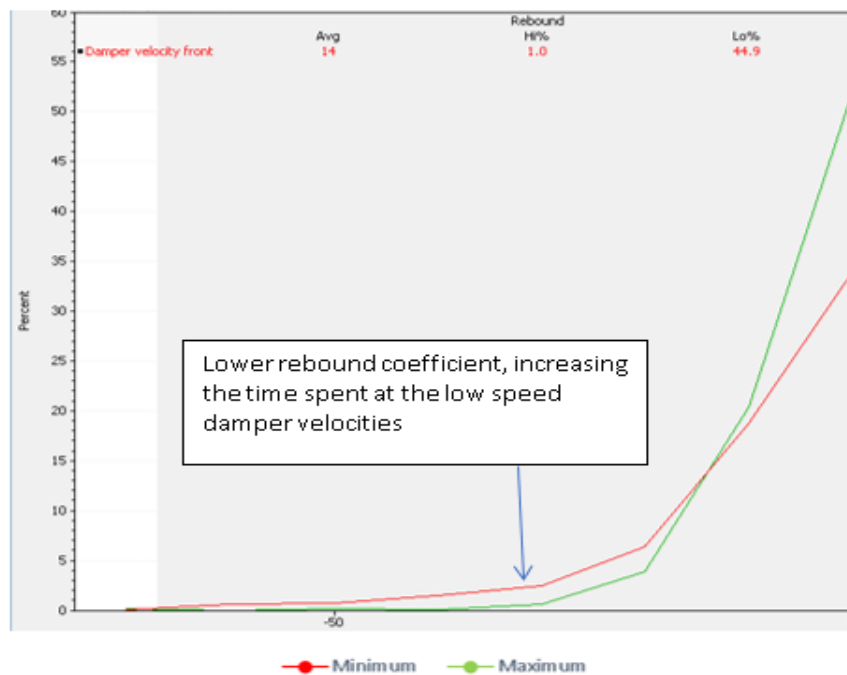


Figure 5-17 Histogram showing the rebound adjustment

Figure 5-17 shows the rebound side of the histogram when adjusting rebound, the trends of the histogram are the same as compression as by making the rebound softer this means the damper spends more time at those velocities, this will provide a flatter gradient in the force vs velocity graph.

5.2 suspension kinematics

5.2.1 motion ratio

Motion ratio is the relationship between wheel travel to damper stroke. In a suspension system the rocker and wishbone length contribute to the motion ratio. (Campbell, 1981)



Figure 5-19 MCR front rocker



Figure 5-18 MCR rear rocker

5.2.1.1 Effect of the motion ratio of suspension design

A higher motion ratio will make the damper stroke similar to the amount wheel travel. This means that the wheel travel will move similarly to the stroke length of the shock. This will make the suspension feel a lot firmer. A higher motion ratio would be very common in track circuit race cars such as single seater F3 cars. (F. Milliken, 1995) The motion ratio can affect the damper coefficients as a high motion ratio means that shock is more likely to bottom out because the wheel travel is similar to damper stroke, this means that there will need to be more damper force to reduce this.

5.2.2 suspension configuration

5.2.2.1 Springs

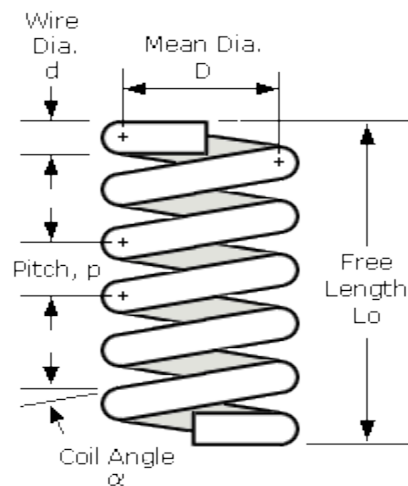


Figure 5-20 Diagram of a coil spring

Springs produce force based on displacement which means the more they compress the stiffer the spring becomes, the spring is designed to support the sprung mass, heavier the mass the firmer the spring. What determines the spring stiffness is the pitch in between the coils, if the coils are closer together the spring is further wound in which produces more force. A firmer spring also increases the friction on the bushing and bearings of the wishbones because of the helical effect.

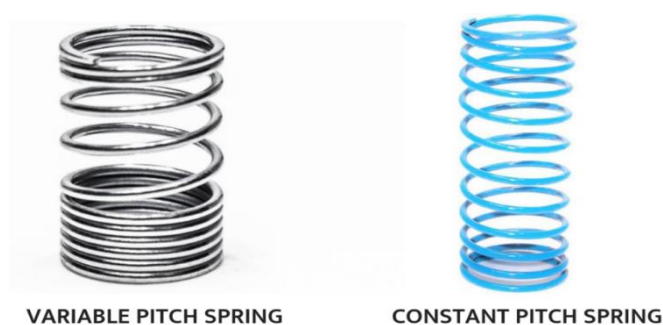


Figure 5-21 Comparison between a linear and progressive spring

Springs act linearly if the pitch is equal of every coil which means that they can compress through damper travel more and can hit the bump stops. Progressive springs have a higher spring rate towards to the bottom of the travel to stop the dampers from bottoming out and

hitting the bump stops, it also reduces the chances of the car grounding whilst still being soft in cornering which increases grip.

5.2.2.2 Sprung, unsprung frequency

Sprung frequency is the frequency of the sprung mass and main indication value how stiff the vehicle is. The firmer the spring, the higher the frequency. For high aero cars the aero force increases the force acting on the tyres which would need stiffer springs even if the mass was around 600kg, the same principles apply for unsprung frequency with harsher the ride the higher the unsprung frequency. Passenger vehicles have a sprung frequency of around 1-2 hz and unsprung of 10-12hz

Sprung Mass

$$\omega_n(s) = \frac{1}{2\pi} \sqrt{\frac{(K_W + K_T)}{M_S}}$$

$\omega_n(s)$ = Sprung mass Natural Frequency (Hz)
 K_W = Wheel Rate
 K_T = Tire Spring Rate
 M_S = Sprung Mass

Unsprung Mass

$$\omega_n(us) = \frac{1}{2\pi} \sqrt{\frac{K_W + K_T}{M_{US}}}$$

$\omega_n(us)$ = Unsprung mass Natural Frequency (Hz)
 K_W = Wheel Rate
 K_T = Tire Spring Rate
 M_{US} = Unsprung Mass

The rear ride frequency will need to be higher at the rear (firmer spring) so over a bump the rear frequency will catch up to the front which stops the body of the vehicle oscillating compromising handling. The MCR has 60% of the mass over the rear axle which would need a stiffer spring to support the extra mass.

Different Front and Rear Ride Frequencies

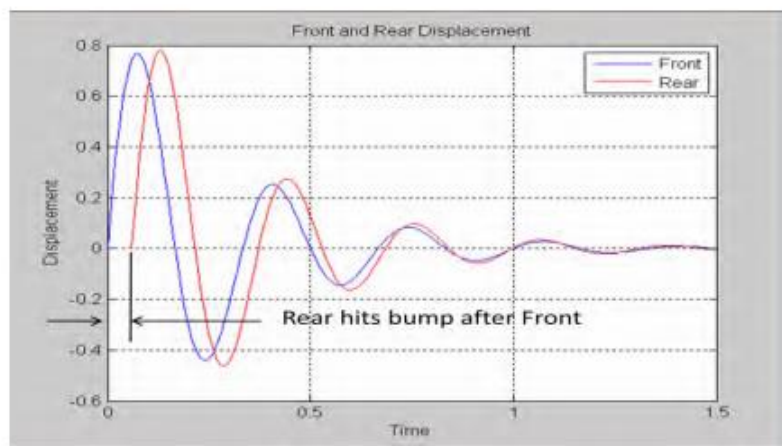


Figure 5-22 Comparison of front to rear spring rate frequencies

6 Methodology

6.1 Automobilista simulator model

It was important to start with the automobilista simulator model. Automobilista is an updated version of BPS (base performance simulator) which already has a sports 2000 car model on it which was developed using data provided from the university. The damper coefficients were very generic damper forces for a low-mid aero race car (force vs velocity graphs and histograms can be seen in research section) which would be a good starting point to just to get an overview on how adjusting damper settings can affect handling starting with quite basic damper coefficients.

Although using automobilista to conduct the simulator testing to start with there were a lot of limitations going forward:

- Unable to adjust the HDV file (would not be able to input the real damper coefficients and new coefficients)
- Limitations with amount of circuits developing new circuits

Therefore going forward using BPS would be the best software to use as there was a sports 2000 car, this model needed to be updated but there would be freedom to do this. There was also a variety of more circuits available to use which means that data could be compared to real life at Silverstone international (automobilista does not have Silverstone international).

6.2 Damper dyno



Figure 6-1 Damper dynamometer

Adjusting compression		
Run	Compression	Rebound
1	0 (min)	15 (mid)
2	4	15
3	8	15
4	12	15
5	16	15
6	20 (max)	15

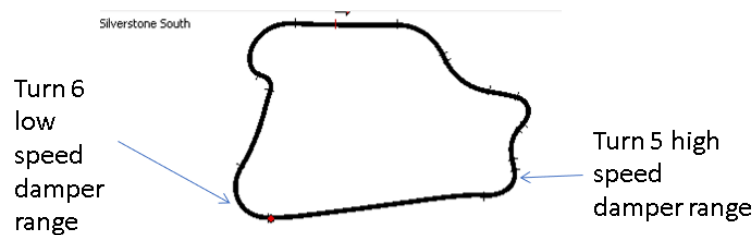
Adjusting rebound		
Run	Compression	Rebound
7	10 (mid)	0 (mid)
8	10	5
9	10	10
10	10	14
11	10	18
12	10	22
13	10	26
14	10	30 (max)

As the damper dynamometer doesn't take long to get force readings the dampers would be adjusted at 4 clicks at a time to get a sweep through the settings, ideally it would be 2 clicks at a time to get a more accurate representation of what happens with a sweep through the settings but this would take too long. The duration of this test for a sweep through compression and rebound was 30 minutes so the session was not too long. The rear dampers had the same amount of adjustment, so the test plan was exactly the same. The damper test plan could

also display cross torque (only increasing compression but also increases rebound) from the minimum and maximum values, these forces at the knee point could be implemented into the simulator model to represent cross torque.

6.3 Real life data from motec

setting	Clicks			
	Front compression	Rear compression	Front rebound	Rear rebound
Minimum	1	1	1	1
Maximum	20	20	30	30



There was also data available from real life which could be compared to the simulator to see if the simulator correlates to real life. There would differences in simulator model compared to real life apart from the damper coefficients and also the accuracy of the track in the simulator model as well as drivers so analysing the handling differences wasn't the priority. The purpose of comparing the data from real life to simulator was to show that the histograms can correlate and display the same trends.

6.4 Real damper forces inputted into BPS simulator model

Once the data was collected from the damper dyno the forces could be converted into the coefficients which could then be implemented into the simulator. This would be able to correlate the simulator to real life (histograms) to show the simulator was accurate enough to replicate the damper behaviour.

The limitations of inputting the damper forces into the simulator would be the knee point speed. As mentioned in the research section because of the main piston holes size and oil flow the knee point speed changes on different settings. Higher coefficient, more force, lower velocity knee point speed. This would change every click slightly but make a larger difference comparing minimum to maximum.

6.4.1 Low speed testing

real damper simulator testing plan				
Run	Front low speed compression (click)	Rear low speed compression (clicks)	Front low speed rebound (clicks)	Rear Low speed rebound (clicks)
1	1	1	15	15
2	10	10	15	15
3	20	20	15	15
4	10	10	1	1
5	10	10	15	15
6	10	10	30	30

compression
minimum = 1 clicks
middle = 10 clicks
maximum = 20 clicks

Rebound	
minimum = 1 clicks	change
middle = 15 clicks	
maximum = 30 clicks	

The testing would be split into low speed and high speed, this would reduce duration of the entire testing and break down the analysis separating low to high speed. As like the automobilista testing using min, mid and max settings would still be in use as this was effective enough during the testing in automobilista to notice a difference and the test session not to be too long.

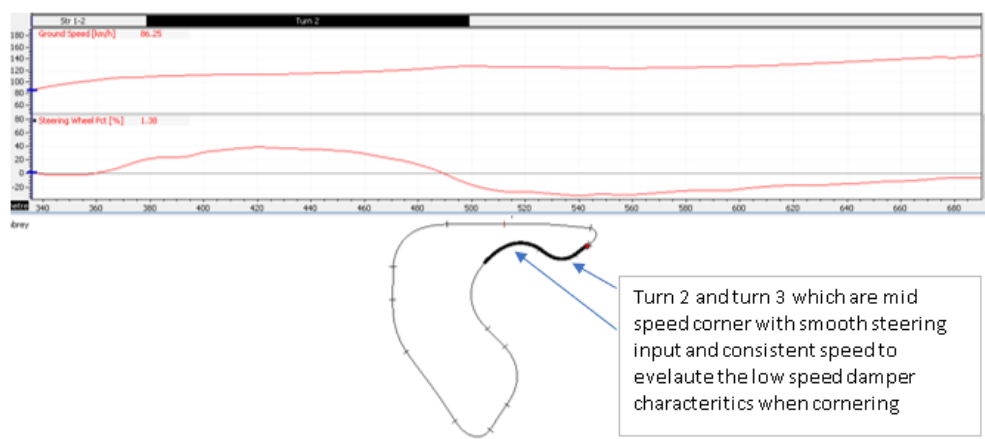


Figure 6-2 Showing the two corners that would be evaluated for all of the low speed testing

The section that would be analysed at Pembrey for the low speed testing, this was because the speed and steering input was consistent and smooth which meant the driver variable was been reduced and also because there was a change in direction in mid speed corners. This means that you could see each damper going from fully loaded up to unloaded showing full compression and rebound stroke on the outside and inside damper. The reason for just looking at 2 corners would be that if data were looked at for the whole lap then there could be so much data that it would take a long time to analyse and limit understanding of the damper settings. All of the low speed testing for the real dampers and updated damper setting in cornering would be looking at turn 2 which is right hander and turn 3 and left hander.

To show understanding of damper behaviour it was important not just to look at cornering because this would be useless for car set up as the understanding would be too limited. Therefore it seemed important to look at the braking zone of turn 1 and accelerating out of turn 1 to look at dive and squat, this would be very important especially in spring rate (which has a greater affect than compression)

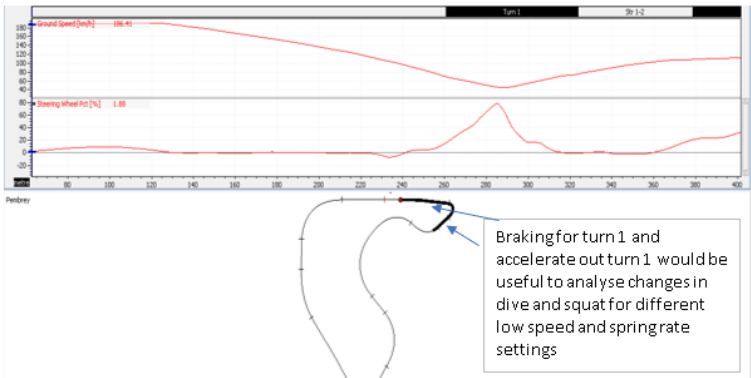


Figure 6-3 Section of track at Pembrey to evaluate squat and dive

6.4.2 High speed testing

real damper simulator testing plan					
Run	Front high speed compression (clicks)		Rear high speed compression (clicks)		Front high speed rebound (clicks)
1	1		1		15
2	10		10		15
3	20		20		15
4	10		10		1
5	10		10		15
6	10		10		30

compression

minimum = 1 clicks

middle = 10 clicks

maximum = 20 clicks

Rebound

minimum = 1 clicks

middle = 15 clicks

maximum = 30 clicks

change

The high speed testing would follow the same structure as the low speed testing as this was an effective testing plan with the min, mid and max. However the high speed testing plan isn't as time consuming because only 1 corner from the track would be analysed as this was the only time dampers go into the high speed velocity so any data that wasn't in high velocities wasn't relevant.

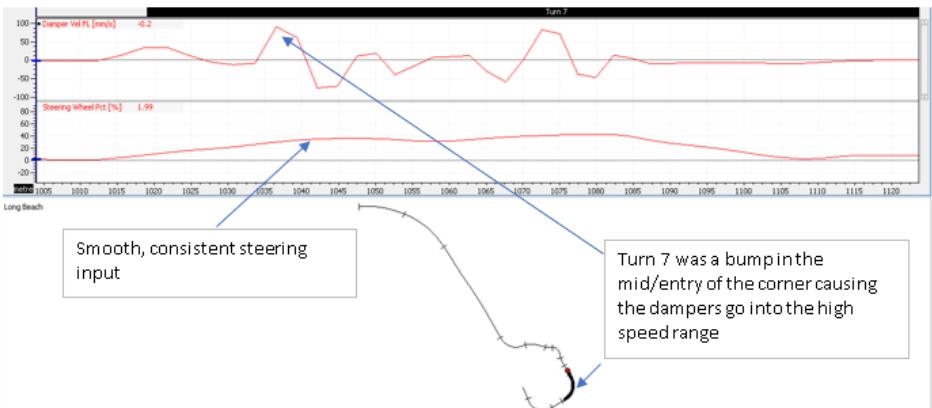


Figure 6-4 High speed input at long beach

Figure 6-4 shows the corner at long beach which show high speed damper characteristics which would be analysed, the steering input was smooth as well as ground speed consistent to reduce driver input discrepancies.

At Silverstone there is a dip/compression in the circuit which was different to long beach where there was a bump in the corner. This means that there could be a comparison between a dip and bump to see if the car and dampers react in the same way and if not why? Therefore the test session at an extra circuit would not be more time consuming but would be important to gain and understanding of the high speed set up.

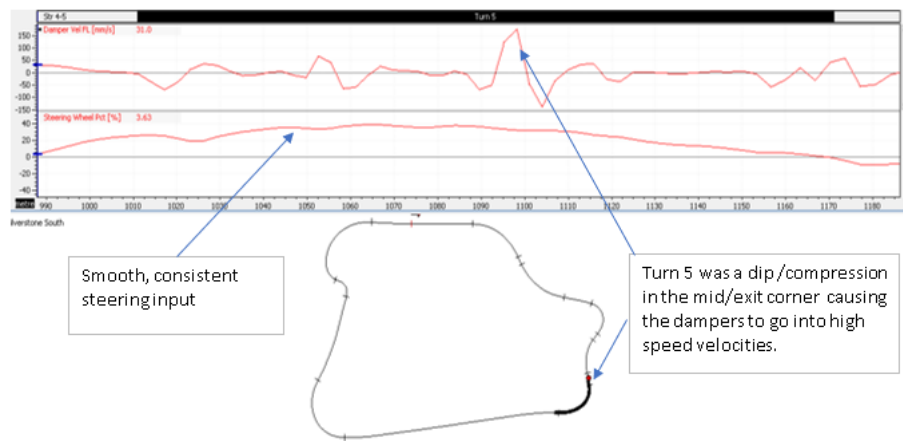


Figure 6-5 High speed input at Silverstone

7 Results and analysis

7.1 Inputting real damper forces into the BPS simulator model

Correlating the histograms could provide useful data to correlate the simulator to real life as the histogram shows how the dampers are behaving at different settings, there would be a comparison from front to rear to understand the effect on handling balance.

7.1.1 Histograms

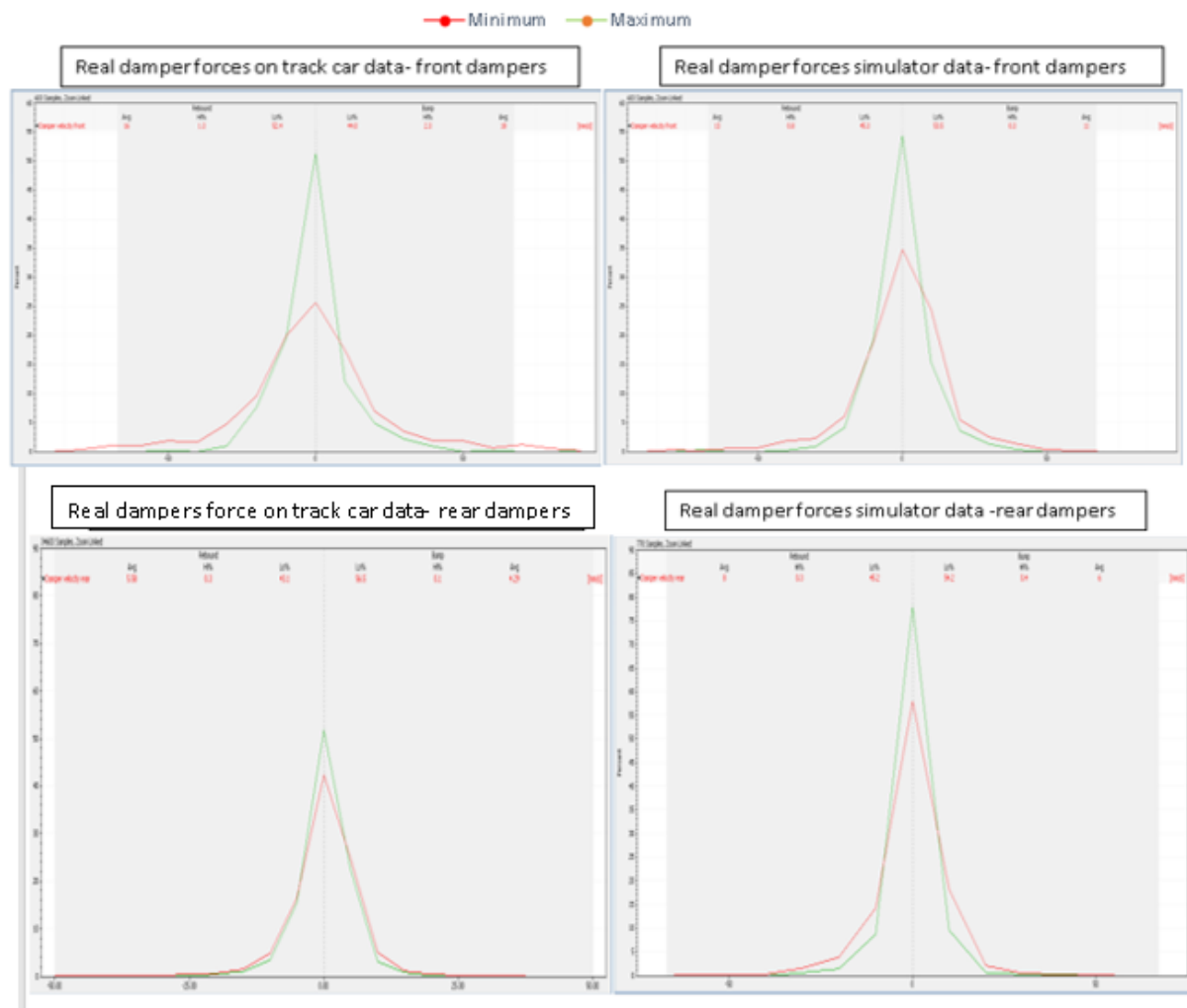


Figure 7-1 Histograms comparing real life to simulator data

7.1.1.1 Analysis of histograms

It can be seen that when comparing the front dampers that the histograms have the same trend when adjusting compression and rebound from maximum to minimum with the on track data showing a larger difference with more high speed input. This could be that the driver used more kerb or more likely that Silverstone is bumpier in real life compared to simulator model. The histograms also show roughly the same increment of adjustment from minimum to maximum which show that the dampers behave the same, this show that the simulator is

accurate enough that the handling affects are relevant to real life as the simulator model is responding correctly.

7.2 Analysing the handling effect of the real damper forces in the simulator model

7.2.1 Low speed compression

front				rear			
low speed compression				low speed compression			
setting	coefficient (n/m/s)	force (N)	damping ratio	setting	coefficient (n/m/s)	Force (N)	damping ratio
min	13134.32836	880	1.5	min	9656.716418	647	0.8
max	15256.71642	1030.1	1.7	max	14548.50746	974.75	1.3

Figure 7-2 Tables comparing front to rear damper forces

The table above shows the rear dampers produce less force at the knee point but a larger range of adjustment on the rear than the front as comparing maximum to minimum on the rear is 327N compared to 150N even though they have the same amount of clicks of adjustment. The reason for this would be in the needle design as the rear would have a larger, flatter and less pointy needle providing a larger adjustment in clicks compared to long, thin needle. What affect this has is that there is less adjustment which means there's less complication in set up as the rear dampers have not much difference from minimum to maximum.

7.2.1.1 Force vs velocity graph

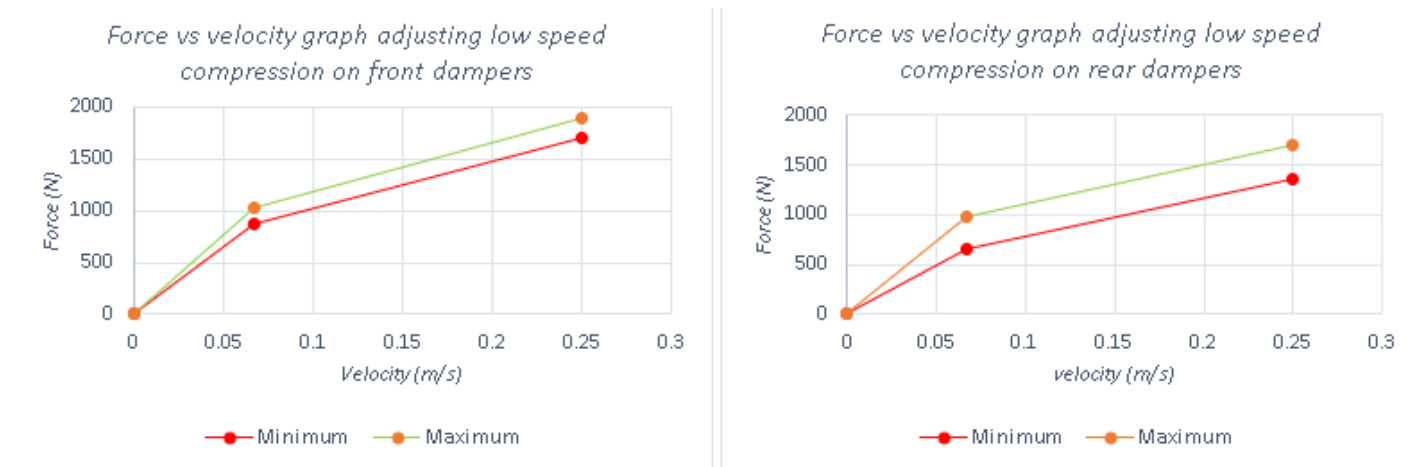


Figure 7-3 Comparison of front to rear force vs velocity graphs

7.2.1.1.1 Analysis of force vs velocity graph

The low speed compression stiffness could be so high is because of the knee point speed as the knee point is at 0.067 m/s whereas common knee point speed ranges between 0.09-0.1 m/s. The will reduce the coefficient as the dampers produce more force at a faster velocity and reducing the gradient of the low speed coefficient.

7.2.1.2 Histogram

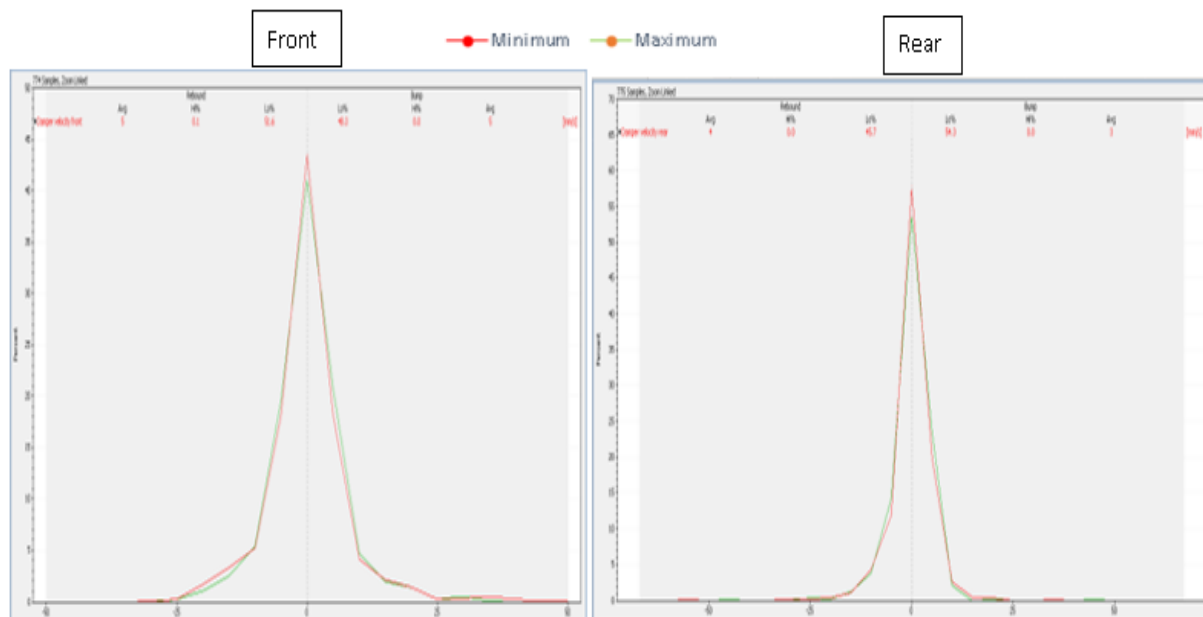


Figure 7-4 Comparison of front to rear histograms

7.2.1.2.1 Analysis of histogram

As can be seen from the force vs velocity graphs and damper forces tables both front and rear dampers are overdamped, this shows in the histograms as the trends are thin and tall as the dampers are stiffer, they operate at lower velocities as there is more resistance to damper movement reducing velocities. The reason why the front appears to be softer is because of the spring rate as the front spring is 39000 N/m compared to 67000 N/m and that causes the front dampers to spend more time at velocities between 10-50 m/s as the softer spring is providing less force to displacement allowing the dampers to move more.

7.2.1.3 Effects on handling

To understand the damper characteristics there will need to be more analysis of the data, this would be done by comparing specific data to what Carroll Smith states. Carroll Smith is a respected motorsport engineer who generated a step by step handling guide of the effect's dampers have on handling. By comparing the data and the statements it could validate the simulator model and further understanding of the damper settings (sprung frequency and damping ratio values comparing to handling).

The table below summarises some of the statements made from Carroll Smith comparing maximum to minimum low speed compression. (Smith, 1978)

Compression		
Low speed	Too much	Chassis roll slow to develop Reduces understeer on initial turn in
	Too little	Dives and squats excessively Roll velocity increases

7.2.1.3.1 Handling effect of firmer compression

Firmer compression “*chassis roll is slow to develop*” (Smith, 1978)

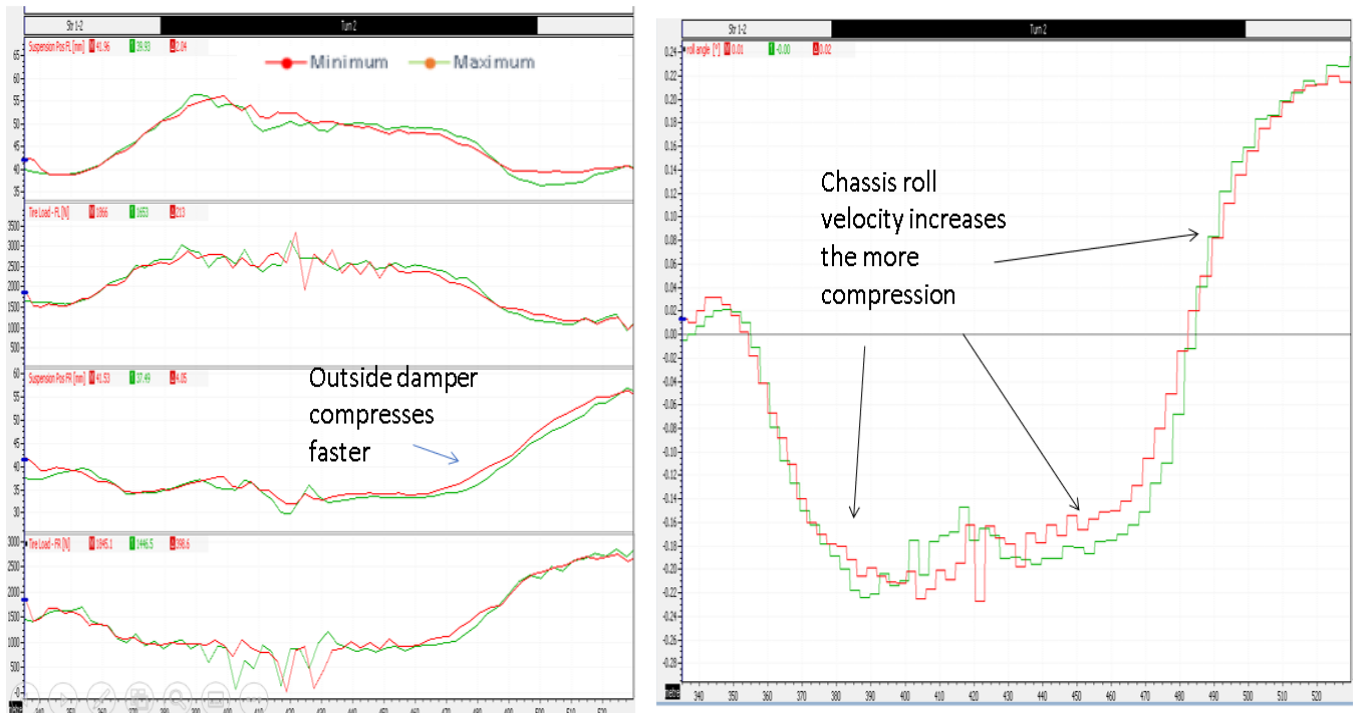


Figure 7-5 Right to left hand corner at Pembrey

Firmer compression “*reduces initial understeer on turn in*” (Smith, 1978)

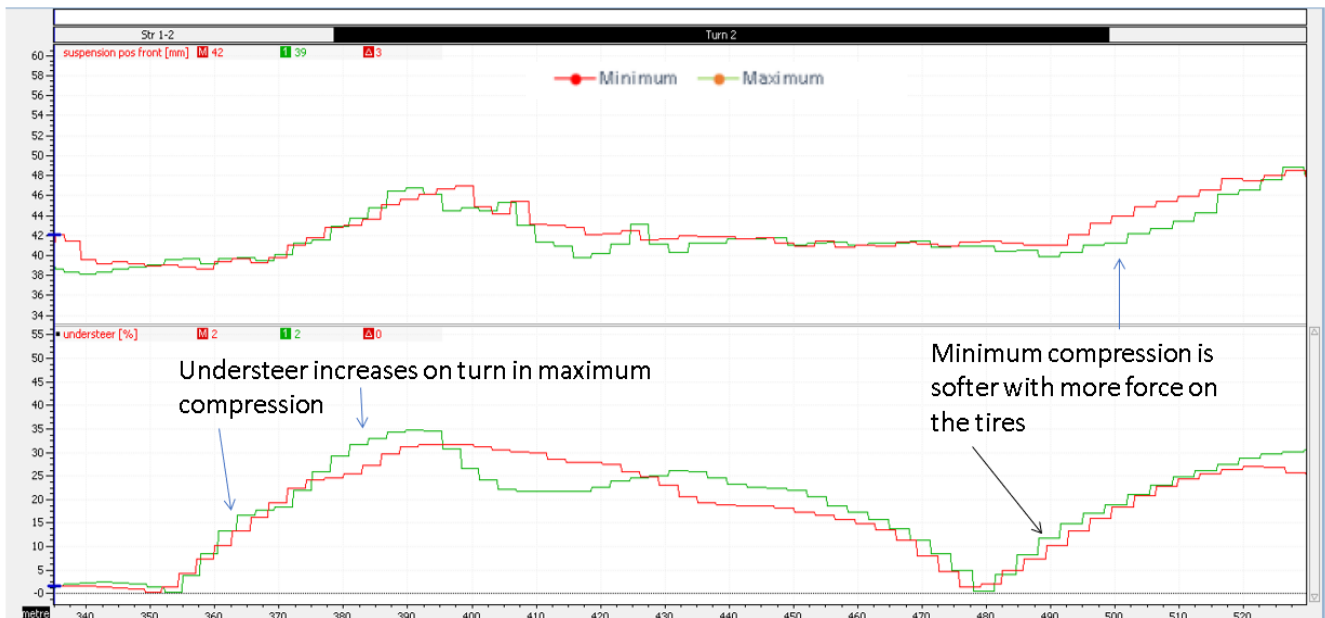


Figure 7-6 Right to left hand corner at Pembrey

7.2.1.3.2 Analysis of handling effect with firmer compression

Figure 7-5 shows that on maximum compression chassis roll is slow to develop, this can be seen when looking at the FL suspension position for a right hand corner as on minimum compression the outside damper loads up faster and further into the damper travel which should increase roll angle, however when looking at the roll angle on maximum compression there is more roll. This could be because the dampers are very stiff (1.5-1.7 damping ratio at the front) that the minimum setting is reducing WT allowing more force on the tyres and keeping the body more stable. This keeps the COG height lower which in turn decreases roll angle. When Carroll Smith states “*roll is slow to develop*” this could be that if the damping ratio ranges from 0.2-0.8 as the damper settings vary from very soft to very hard instead of just 2 very firm damper settings.

Softer compression causes the car to “*squat and dive excessively*” (Smith, 1978)

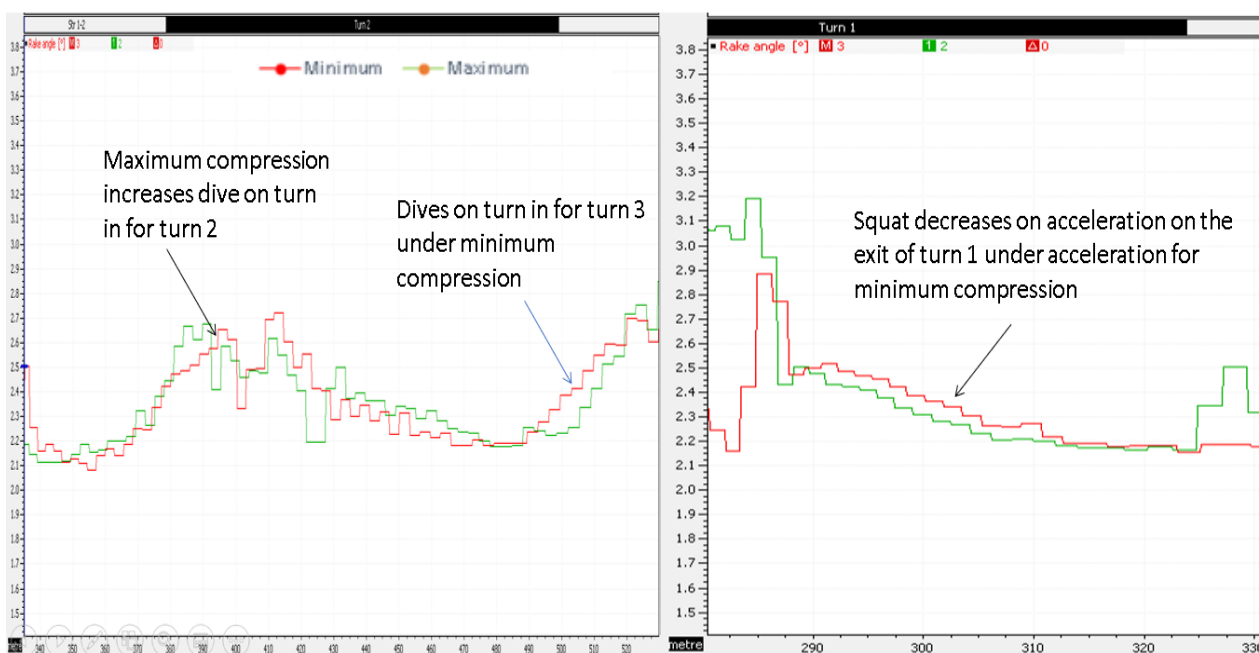


Figure 7-7 Data from Pembrey through turns 1 to 3

7.2.1.3.3 Analysis of handling effect with softer compression

Figure 7-7 shows that softer compression increases dive and squat, this can be seen in the data for turn 3 that dive increases as the minimum compression causes the rake angle to increase on entry, this can increase grip on the front tyres as there is more force on the front tyres compared to the rear which decreases WT on the front axle and increases WT on the rear axle increasing yaw and reducing understeer. However on entry to turn 2 dive increases on firmer compression. As the difference in compression is very minimal this could be that the driver turned in more aggressive or dabbed the brakes slightly firmer causing there to be dive on the front axle, especially since the firmer compression spikes more than the gradual dive on softer compression.

The rake angle on the exit of turn 1 shows that the softer compression decreases squat which contradicts what Carroll Smith states, this again could be that the compression is so stiff that the softer compression setting reduces WT and there is only a small difference between

maximum to minimum whereas Carroll Smiths statements are based on excessively soft to excessively firm which means that it's difficult to compare against the real life damper settings.

7.2.2 Low speed rebound

front	Low speed rebound		
setting	coefficient (n/m/s)	force (N)	damping ratio
min	2319.402985	155.4	0.3
max	11489.46269	769.794	1.3

rear	Low speed rebound		
setting	coefficient (n/m/s)	Force (N)	damping ratio
min	1402.985075	94	0.2
max	9412.626866	630.646	1.1

Figure 7-8 Comparison of front to rear damper forces

The table above shows that the balance front to rear from maximum to minimum is very similar with the difference in the front at around 610 N and at the rear 530N with a damping ratio ranging from 0.2-0.3 to 1.1-1.1. This means the damper produces a similar force front to rear which should give the MCR a balanced handling characteristic with the fronts slightly stiffer which will provide more stability at the front end.

7.2.2.1 Force vs velocity graph

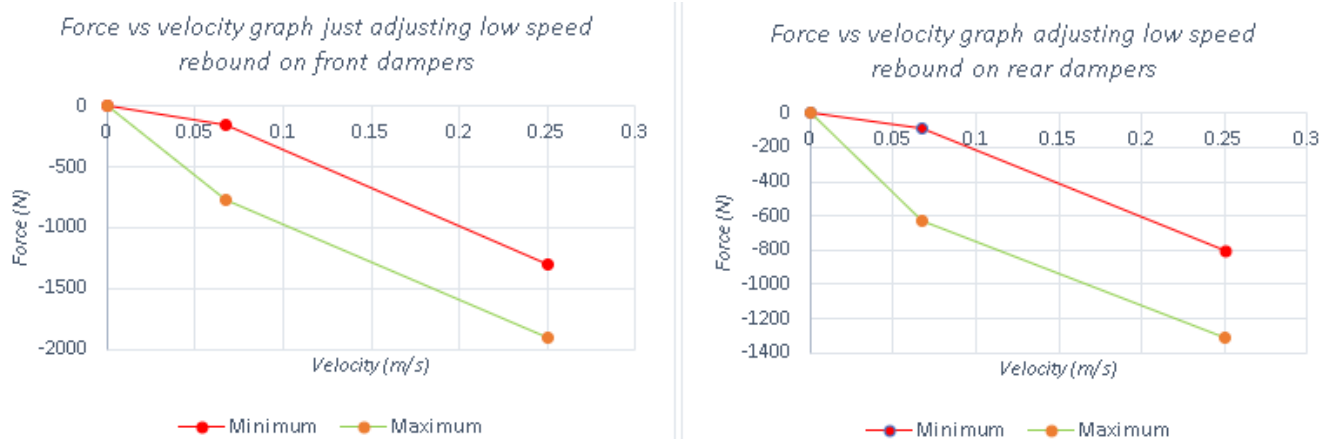


Figure 7-9 Comparison of front to rear force vs velocity graphs

7.2.2.1.1 Analysis of force vs velocity graph

The force vs velocity graphs show the distribution front to rear is very similar in damper force at the knee point as the damping ratio ranges from 0.2-0.3 on minimum to 1.3-1.1 maximum, this will provide a large range of adjustment from under damped and over damped which should provide accurate data to be able to analyse the effect on handling. This shows that the damper design is intended to be very stiff (damping ratio 1.5-1.7) in compression to reduce roll and a larger difference in rebound with a softer minimum setting (damping ratio 0.2-1.3).

7.2.2.2 Histograms

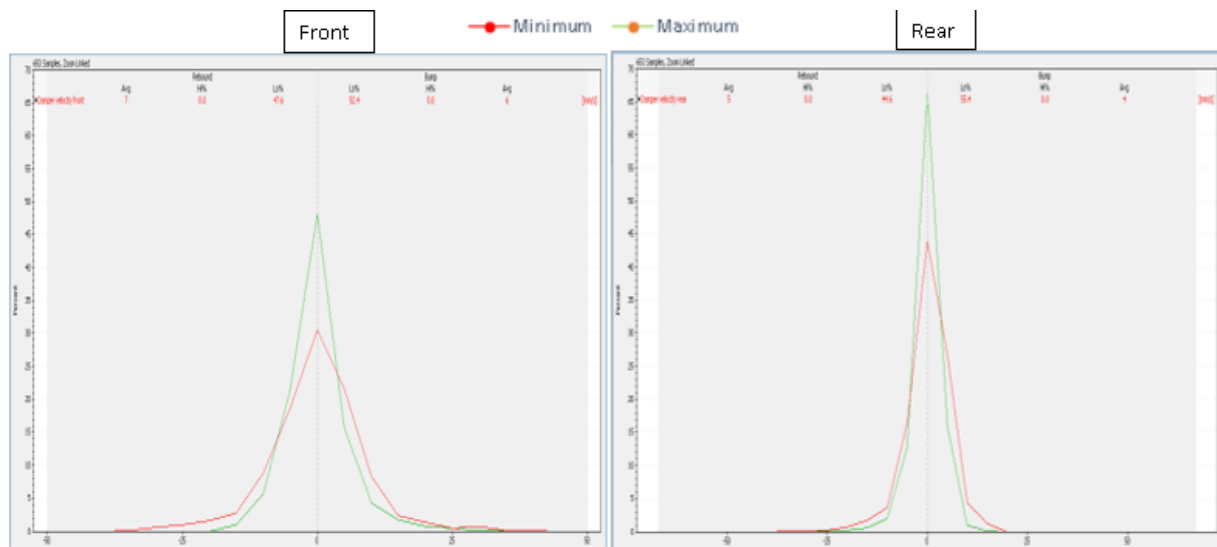


Figure 7-10 Comparison of front to rear histograms

7.2.2.2.1 Analysis of histograms

The histogram shows similar trends front to rear with the rear showing a skinnier trend spending less time at low speed velocities even though the rear dampers are softer. This is like with compression because of the spring rate difference front to rear. As the rear spring rate is 67000 N/m compared to 39000 N/m at the front the rear spring is resisting damper movement at the rear which stops the dampers spending more time at low speed velocities despite the lower coefficients.

7.2.2.3 Effects on handling

Just like in compression comparing the handling data to Carroll Smiths statements ensure the rebound damping in the simulator model is responding correctly. Below is table with Carroll Smiths statements comparing maximum to minimum rebound damping. (Smith, 1978)

Rebound		
Low speed	Too much	Wheels don't return to ground quickly after displacement. Overload outside tyre and underload inside tyre
	Too little	Difficult to gain traction at the rear Faster roll velocity

Slower/firmer rebound “overloads the outside tyre and underloads the inside tyre on corner exit” (Smith, 1978)



Figure 7-11 Right to left hand turn at Pembrey

Slower/firmer rebound cause the “inside wheel returns to the ground slowly” (Smith, 1978)

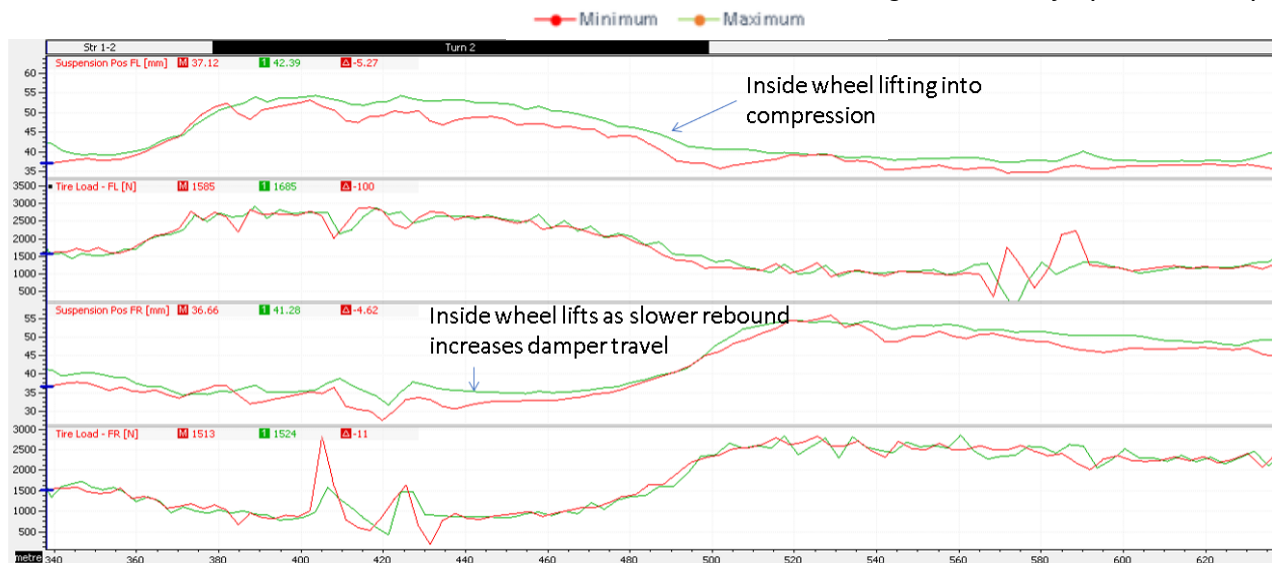


Figure 7-12 Right to left hand turn at Pembrey

7.2.2.3.1 Analysis of handing effect on maximum rebound

Figure 7-11 shows this as the suspension position corresponds to the tyre load force, (more compression more tyre force). This matches up with Carroll Smiths statements as the damping ratio shows a wider range in force with a damping ratio of from 0.2-1.1 which goes from excessively soft to over damped whereas compression was much stiffer with a damping ratio ranging from 1.5-1.7 which isn't a significant enough difference to cause a large enough handling effect.

Figure 7-12 shows that when cornering the outside damper goes further into compression the firmer the rebound setting, this could be because the outside damper is producing more force holding the damper in compression whereas as faster rebound wants to return to full damper stroke faster, this means that according to the data a firmer rebound decreases compression. This doesn't correlate to the histogram as by only adjusting low speed rebound the minimum setting softens rebound and compression which doesn't correlate with the data as the damper is loaded up the maximum setting reduces compression. The reason why the histogram shows this trend is because that on the faster/softer rebound the roll velocity increases, this means that outside damper loads up faster on a softer setting, even if it isn't softer mid corner.

Softer rebound “increases roll velocity” (Smith, 1978)

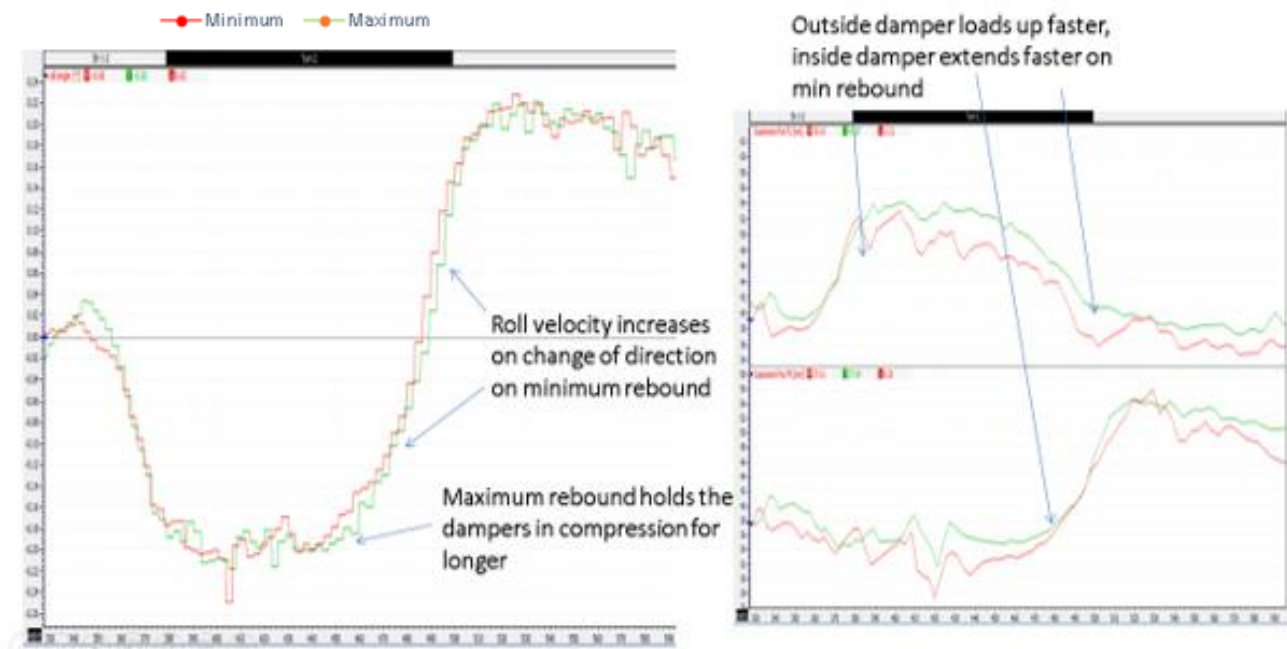


Figure 7-13 Right to left hand turn at Pembrey

Softer rebound is “difficult to gain traction at the rear” (Smith, 1978)

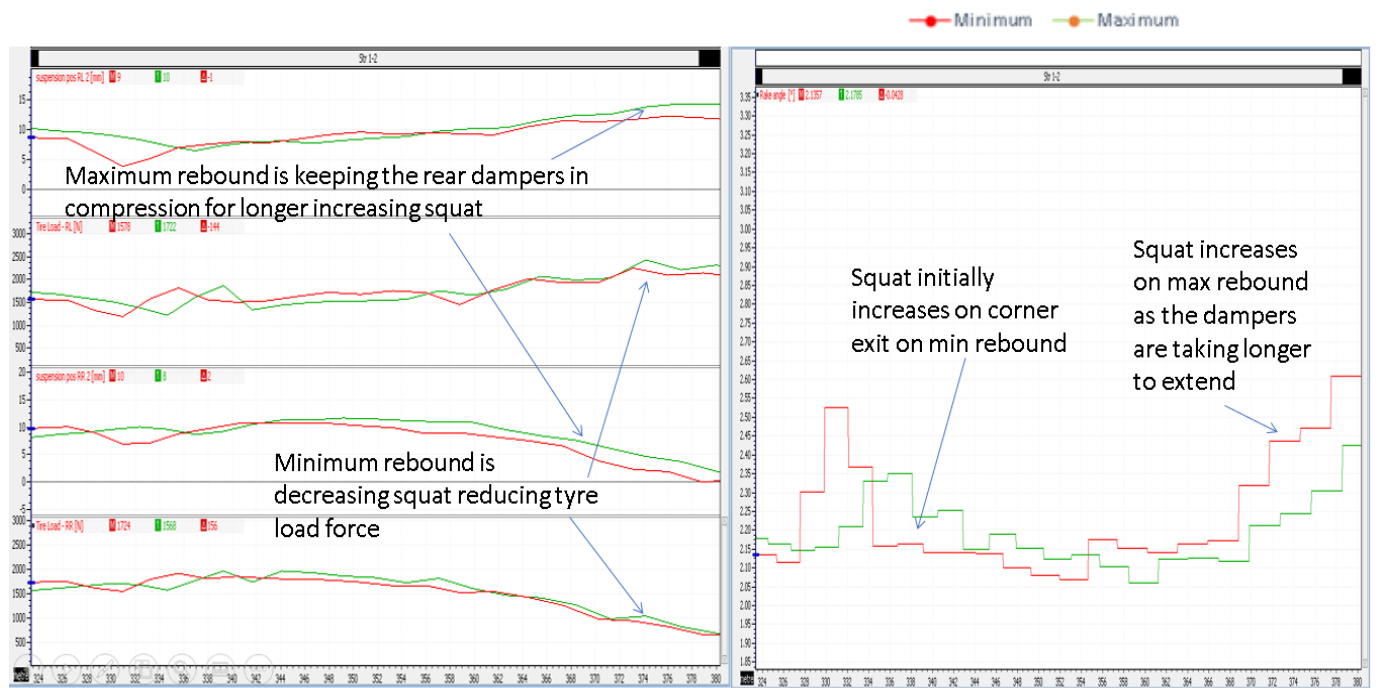


Figure 7-14 Exit of turn 1 accelerating in a straight line

7.2.2.3.2 Analysis of handling effect minimum rebound

Figure 7-13 shows that on softer rebound the roll velocity increases on the minimum rebound setting when changing direction, this will be caused by the inside damper extending faster as there is less force restricting oil through the needle valve. This unloads the inside corner faster and increases load on the outside tyre faster reducing understeer. The roll angle also shows that the firmer rebound setting holds the car in roll for longer as the outside damper takes longer to extend, this can overload the outside tyre and increases understeer depending on the coefficients. As the damping ratio varies by 0.2-1.1 the coefficients provide a clear difference in handling that correlates with Carroll Smiths statements.

Carroll Smiths statement that soft rebound “is difficult to gain traction at the rear” will be referring to tyre force variation on the rear tyres causing oversteer. This can be seen in figure 7-14 when looking at the rear suspension position and tyre loads as the maximum rebound setting is in compression for longer when accelerating out of turn 1. This is because when accelerating the WT shifts towards the rear which loads up the dampers and pushes them into compression (squat).

7.2.3 High speed compression

Front	high speed compression			Rear	high speed compression		
setting	coefficient (n/m/s)	force (N)	damping ratio	setting	coefficient (n/m/s)	force (N)	damping ratio
min	2366.120219	1313	0.3	min	2622.95082	1127	0.2
max	3425.136612	1656.9	0.4	max	5945.355191	2062.75	0.5

Figure 7-15 Tables comparing front to rear damper forces

When adjusting the compression on the damper dyno from minimum to maximum the low speed adjust is the only variable, as from the force vs velocity graphs at low speed increasing the low speed coefficient in turn increases the high speed force which is why the high speed increases from 1313 N to 1657 N even though there is no difference in the high speed coefficient. When converting the forces to coefficients the tables show there is a difference in coefficients when in reality there is no difference from the damper dyno. The affect this has is that the low speed adjusts the high speed force not the coefficient.

7.2.3.1 Force vs velocity graph

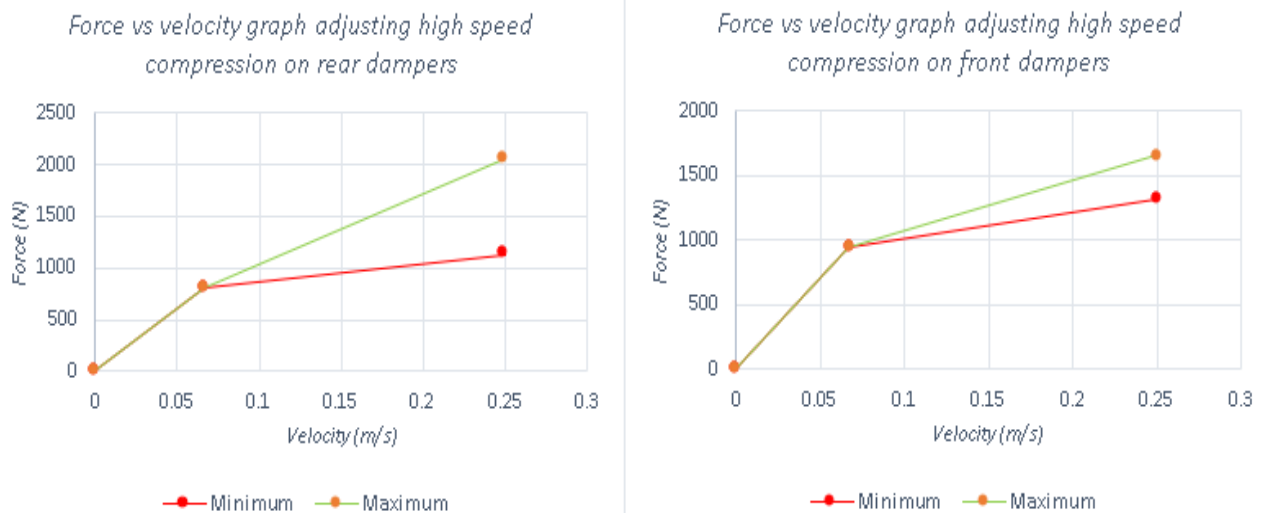


Figure 7-16 Comparing front to rear force vs velocity graphs

7.2.3.1.1 Analysis of force vs velocity graphs

The force vs velocity graph shows the rear dampers have a larger range from minimum to maximum with 950 N at the rear compared to the front which has 300N. This is because the dampers are 2 way adjustable which means the low speed adjust controls the high speed force.

7.2.3.2 Histograms

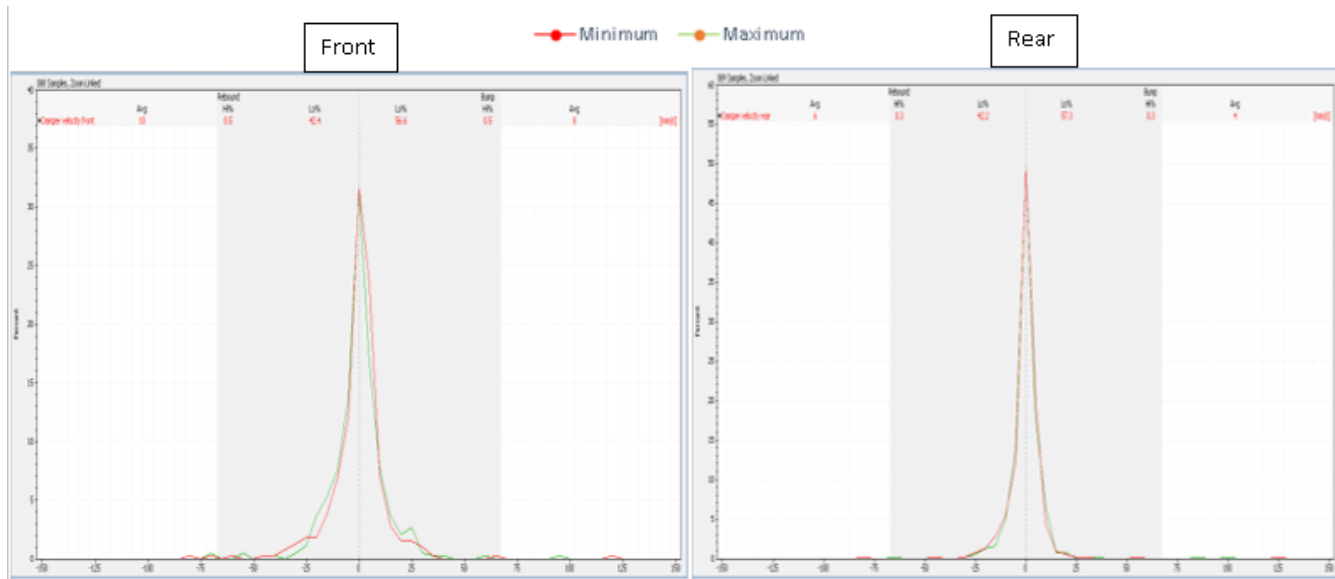


Figure 7-17 Comparing front to rear histograms

7.2.3.2.1 Analysis of histograms

The histograms show differences between minimum to maximum but is difficult to spot as dampers spend majority of time at low speed which means the histograms are difficult to determine high speed differences. What can be seen is that on the front dampers the minimum setting spends more time at high speed velocities as the dampers are softer, this then causes the minimum setting to spend less time at low speed velocities because the dampers are spending more time at high speed velocities. This can be seen in figure 7-17 for front and rear dampers.

7.2.3.3 Effects on handling

Firm compression “doesn’t absorb impacts from kerbs and bumps” (Smith, 1978)

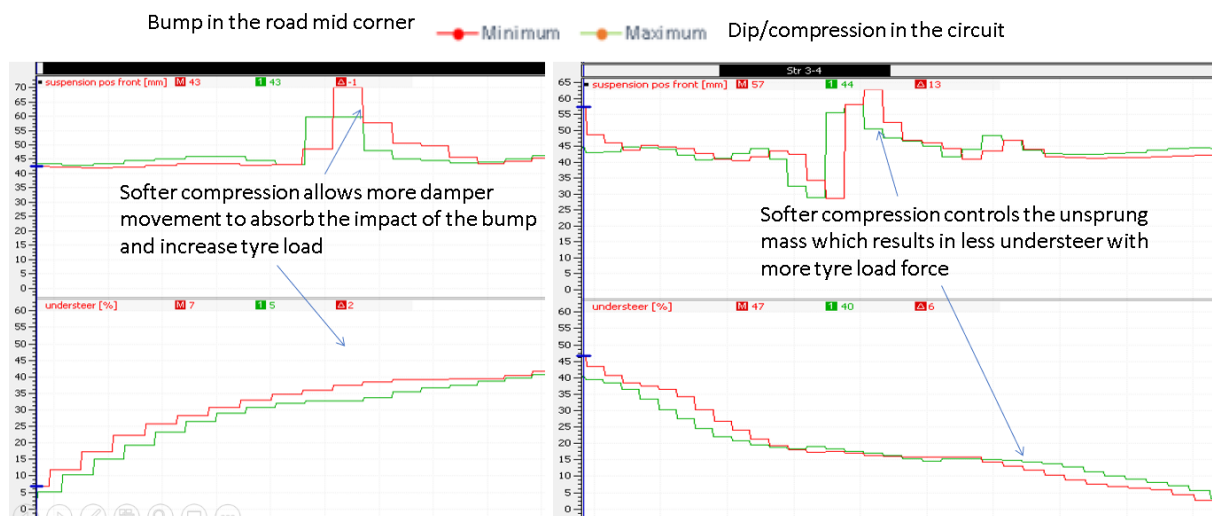


Figure 7-18 Showing the difference between long beach and Silverstone high speed input

Firm compression causes “*instability over bumps, skips over bumps*” (Smith, 1978)

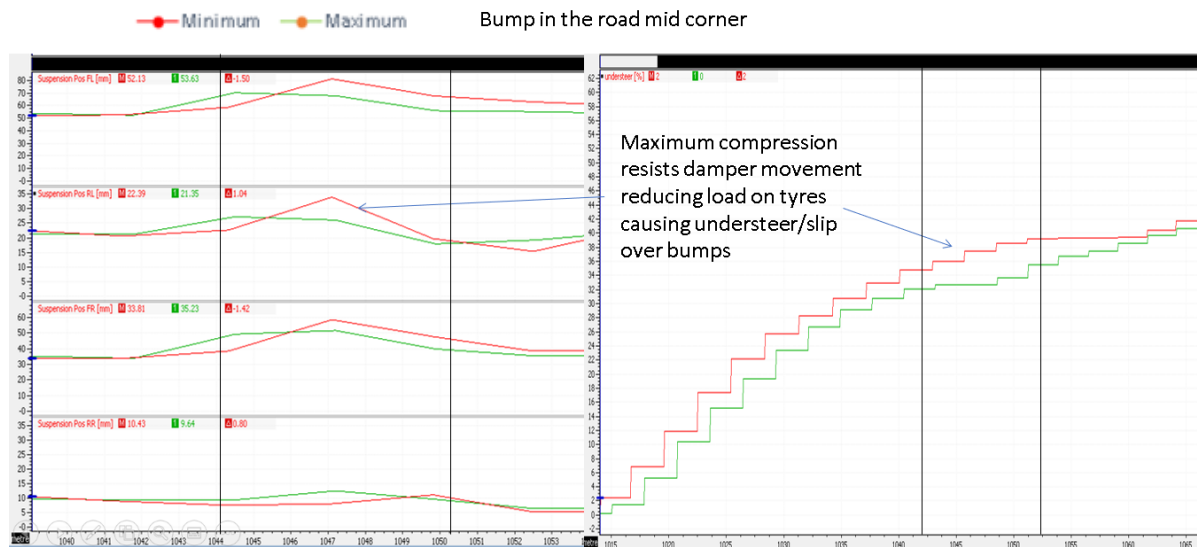


Figure 7-19 Showing high speed input at long beach

7.2.3.3.1 Analysis of the handling effect on maximum compression

Comparing the bump in the corner to the dip in the circuit display the same trends that maximum compression increases understeer as there is less damper movement to absorb impacts. The reason why firm compression can cause instability is when looking at figure 7-17 you can see that firmer compression restricts damper movement which reduces tyre force and cause more movement of the sprung mass as the damper is not absorbing the bumps. Over bumps this can cause the car to skip as when the car hits the bumps the dampers resist movement over the bumps reducing tyre force causing slip to increase, this can be seen in the data over the bump with the suspension position traces to the understeer trace.

Softer compression could cause the “*car to ground over bumps and compressions*” (Smith, 1978)

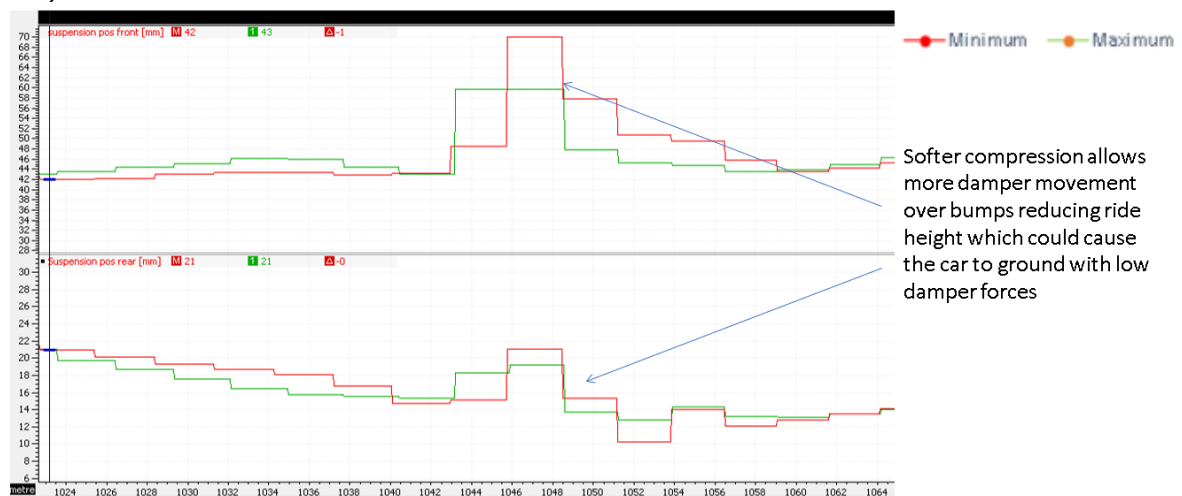


Figure 7-20 High speed input from long beach

Softer compression could cause the car to have a “floaty feeling over bumps” (Smith, 1978)



Figure 7-21 High speed input from long beach

7.2.3.3.2 Handling analysis of minimum compression

Figure 7-20 shows that the softer compression increases damper movement which reduces ride height, this means that going through corners with a large compression/dip and bumps (corner like eau rouge) the car could ground out if there is insufficient damper force. The softer the damper coefficient the increase in suspension travel.

Whether the car could ground will be down to spring rate and not just the damper coefficients, this is because the spring provides force based on displacement and even if the damper fails the spring will still provide force to keep the sprung mass above the ground. This shows in figure 7-21 as there is less damper movement in the rear dampers than the front despite the rear having a larger range of adjustment and the dampers are softer at the rear. The reason why the front dampers use more travel is because the spring rate is 38000 N/m (softer) at the front which has a larger effect over the bumps compared to high speed adjustment.

7.2.4 High speed rebound

Front				Rear			
high speed rebound				high speed rebound			
setting	coefficient (n/m/s)	force (N)	damping ratio	setting	coefficient (n/m/s)	force (N)	damping ratio
min	-2315	-1157	0.3	min	-3590.82	-957	0.3
max	-5368.7	-1193.424	0.6	max	-4678.32	-1311.67	0.4

Figure 7-22 Table comparing front to rear forces

The high speed rebound dampers show the same trends as the high speed compression as the low speed controls the high speed as they are 2 way adjustable dampers from the damper dyno. This can be seen when looking at the front dampers as the range of damper force is 40N and a coefficient difference on 3000 N/m/s. This coefficient difference is much larger because the difference in low speed was 12000 N/m/s which controls the high speed adjust.

7.2.4.1 Force vs velocity graph

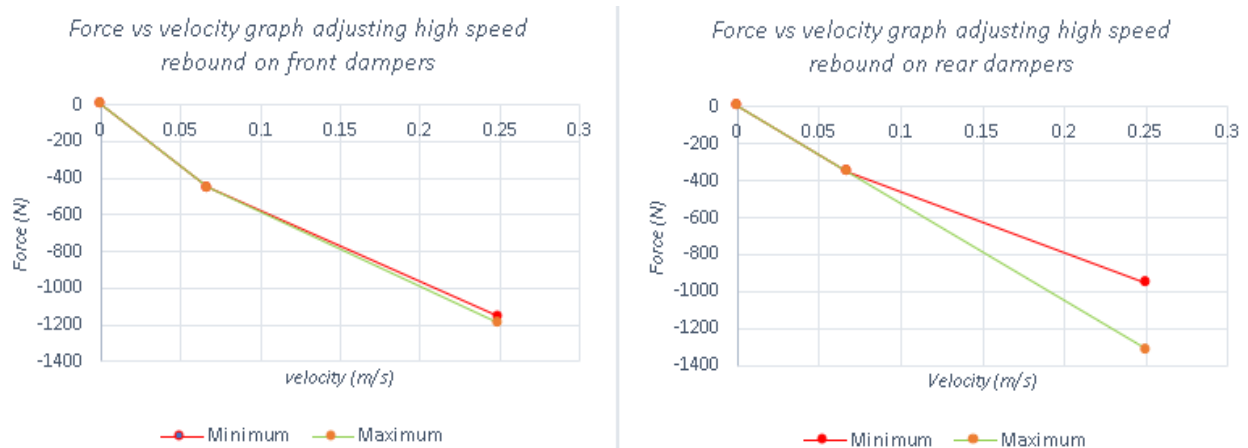


Figure 7-23 Comparing front to rear force vs velocity graphs

7.2.4.1.1 Analysis of the force vs velocity graph

The force vs velocity graph show the high speed rebound effect of the simulator model as only high speed is being adjusted whereas in real life the low speed affects the high speed. This is why the front force vs velocity graph shows inaccuracies with the coefficients as the graph shows there is minimal difference between minimum to maximum whereas there is actually a difference of 3000 N/m/s. This is because the low speed adjust is on the middle setting and varying high speed which doesn't correlate with the damper dyno graphs in real life as low and high speed adjust at the same time.

7.2.4.2 Histograms

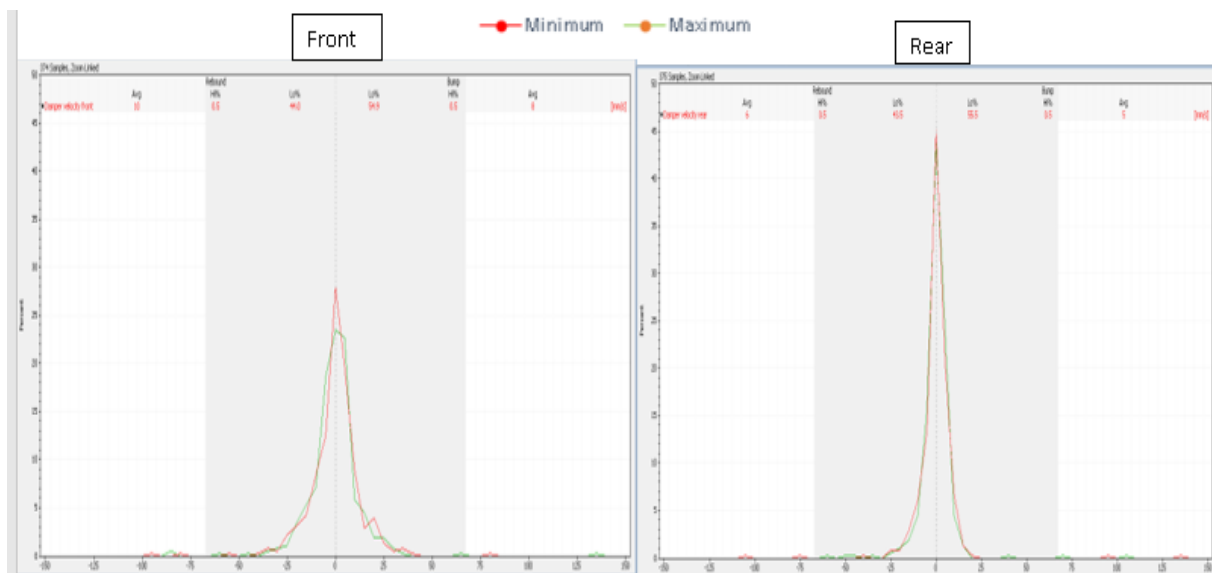


Figure 7-24 Comparing front to rear histograms

7.2.4.2.1 Analysis of histograms

The difference in high speed rebound from the histograms is difficult to see as the dampers rarely reach the high speed range but what be seen is that on the front dampers the velocities

between 40-70 that on the minimum rebound setting spends less time at these velocities compared to the maximum setting, this is because the high speed coefficients are softer which spend more time at higher velocities compared to lower.

7.2.4.3 Effect on handling

Firmer/slower rebound causes the “damper not to return to full stroke to absorb the next impact” (Smith, 1978)



Figure 7-25 High speed input from long beach

Firmer/slower rebound causes the dampers to “slide over bumps” (Smith, 1978)

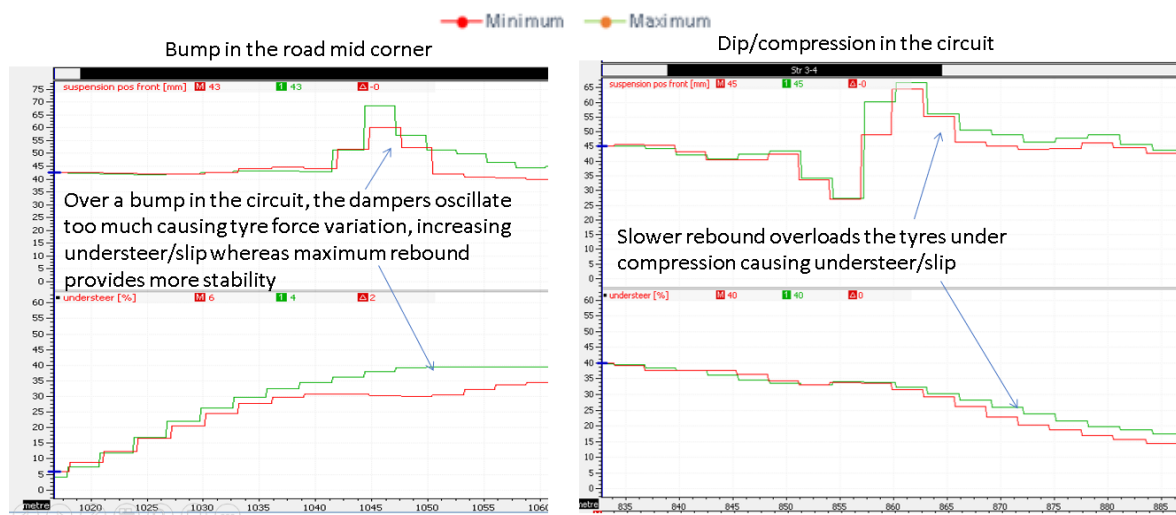


Figure 7-26 Comparing high speed input from long beach to Silverstone

7.2.4.3.1 Handling analysis of maximum rebound

Figure 7-25 shows that the maximum rebound setting holds the dampers in compression for longer after impact from a bump, what this does is increase the tyre force for longer as there is less damper oscillation. This controls the sprung mass as the rebound force acting on the vehicle body is more stable and consistent which can make the car more predictable over bumps. What this also means for repetitive bumps is that the dampers cannot react faster

enough to recover to full displacement in time to absorb the next bump (going over a rumble strip), this will make the dampers feel very firm as the damper stroke is reduced.

Faster/softer rebound causes “excessive oscillation over bumps and kerbs” (Smith, 1978)

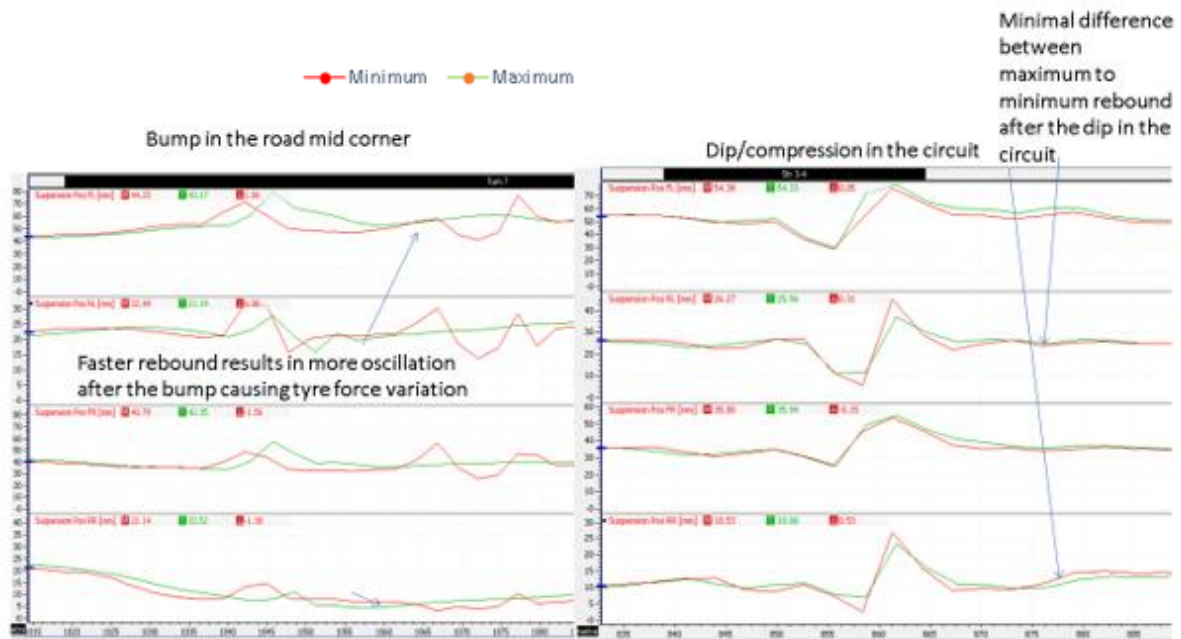


Figure 7-27 Comparison between long beach to Silverstone high speed input

Faster/softer rebound increases “instability over bumps and corner entry” (Smith, 1978)



Figure 7-28 Long beach on the left and Silverstone on the right difference in high speed input

7.2.4.3.2 Handling analysis of minimum rebound

Figure 7-27 shows the dampers return to full displacement faster on the minimum setting compared to maximum setting which can reduce stability as the dampers aren't controlling the

sprung mass as well as a slower setting. This is shown in figure 7-28 as the understeer increases more on the faster setting than the slower setting. This could be that there is insufficient rebound force as the damping ratio ranges from 0.3-0.6 which shows that 0.3 could be too soft for rebound force.

When looking at figure 7-28 the data shows that with the bump min corner than understeer increases the softer the rebound setting whereas with a dip/compression mid corner that more rebound force increases understeer. The reason why there are differences between the corners is because when there is mid corner this can unsettle the car and a fast rebound setting will not be able to control the impact.

7.3 Analysing the damper internals

Looking and modifying the damper internals would be important to gaining a further understanding of dampers but there were some issues:

- Unable to modify the real damper internals as they needed to be used on the MCR
- The dampers that could be modified were one way adjustable and could only modify the amount of shim stacks, the modifiable main piston would not fit.
- Most importantly the damper dynamometer was broken and the pandemic causes university to close.

7.3.1 Main piston

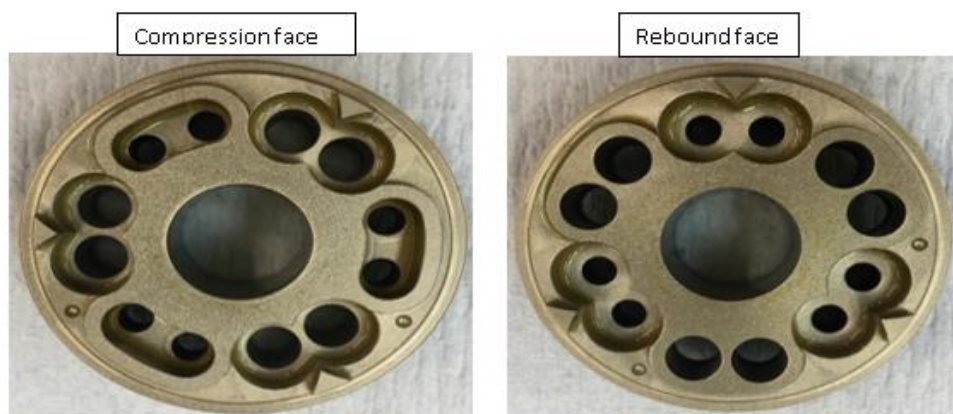


Figure 7-29 Showing the compression and rebound face of a Penske damper

When comparing the main piston faces you can see that the surface area of the compression face is less than rebound allowing more oil to pass through the damper decreasing damper force. The surface area is lower as all the holes are in ports which means there's less metal on the piston up against the shim stacks which means it will take longer for the holes to choke (block with oil) and open the shim stacks. This means the knee point will happen at a higher velocity which will decrease low speed damper force. This will decrease damper fade as the initial diameter of the hole is larger with another port around the hole allowing more oil to pass through without getting stopped on the main piston face.

From the damper dynamometer testing compression the low speed was overly critically damped for the spring rate, therefore a way to decrease this would be to increase the size of the port holes to allow more oil to pass through and decrease damper force, this will also increase the knee point speed as it will take more oil to block up the ports and cause the shim stacks to open.

7.3.2 Shim stacks

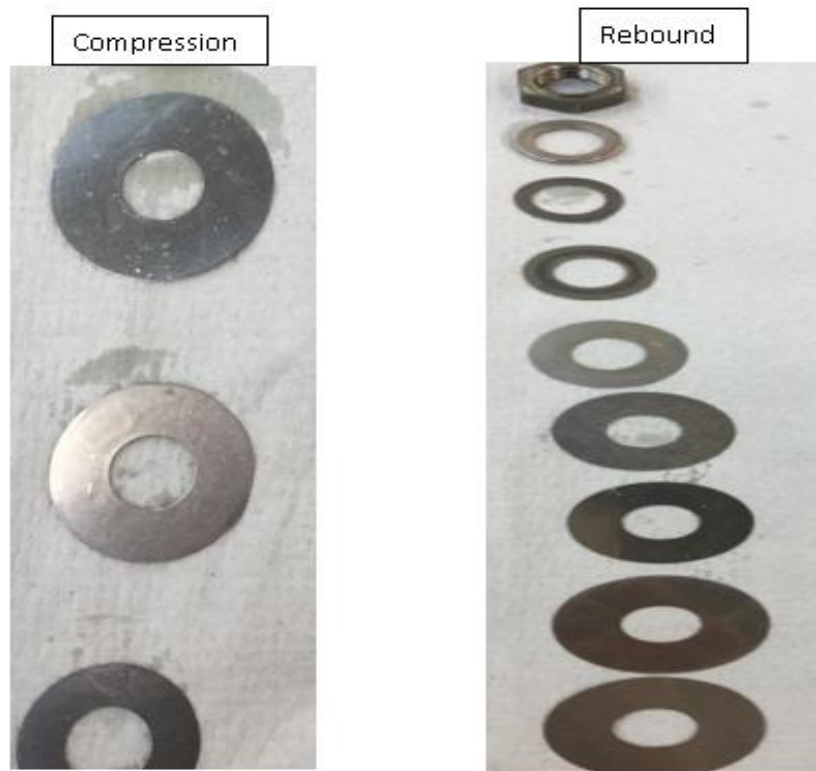


Figure 7-30 Differences in rebound to compression shim stacks

It can be seen the high speed rebound force is greater than the compression as there are 5 more shim stacks, this will require more oil to open the shim stacks. More high speed rebound force will increase stability and keep the damper movement more consistent which is why there are more shim stacks.

7.3.3 Needle valve



Figure 7-31 Needle valve of the Penske

The spring in figure 7-31 controls the movement of the needle valve wound in which can cause issues as the spring is very small and has little control. This means that it can provide inconsistent damper forces which show up on a damper dyno. It also means there is less effect adjusting the damper from one click to another as the spring is not able to control needle valve accurately which struggles to replicate accurate damping force, this can also be down to the needle valve shape.

A wider shorter needle can increase oil shear and damper fade as the oil goes from the main tube through a tight orifice where the needle is which compresses lots of oil increasing viscosity and causes oil shear when it passes through the needle valve. This will cause damper fade as the oil becomes hotter from being compressed which increases, roll and dive in compression and reduces stability in rebound. A longer thinner needle will decrease this effect as it allows more oil to pass through the orifice but will decrease damper force, this is the main disadvantage of needle valves which try to produce high damper force for a long time.

7.4 Analysing the handling effect of the new spring rate settings

As the sprung frequency at 2Hz would be lower than expected for a mid-aero race car it seemed important to test with different spring rates. The spring rate range during testing was between 3-2 Hz as 4 Hz from the table seemed excessively stiff and the MCR weighs 568 kg which would be a lightweight race car. The rear spring rate was always going to be stiffer than the front because of the weight distribution of 60% over the rear which will reduce understeer as a softer spring rate reduces WT as long as it's not excessive. This means the stiffer spring rate at the rear will need to support the extra 90 kg of mass which could increase slip but will support the mass over bumps and compressions.

The front spring rate is also much softer because the front frequency needs to be lower than the rear so the car will not continually oscillate, and the rear will catch up with front when going over bumps.

setting	front sprung rate (n/m)	Rear sprung rate (n/m)	front sprung frequency (hz)	rear sprung frequency (hz)	average frequency (hz)
1	130000	260000	3.8	4.2	4.0
2	100000	200000	3.3	3.7	3.5
3	75000	150000	2.9	3.2	3.0
4	50000	100000	2.4	2.6	2.5
5	35000	70000	2.0	2.2	2.1

car	sprung frequency (hz)
passenger car	0.5-1
modified springs	1-1.5
rally car	1.5-2
low downforce car	1.5-2.5
FSAE car	3.2-3.5
mid downforce car	2.5-3.5
high downforce car	3.5-5.0

total mass	568	Kg
mass on front axle	228	Kg
mass on rear axle	340	Kg
mass % on front axle	40.14085	
mass % on rear axle	59.85915	
sprung mass weight distribution	60.06494	

Figure 7-32 Tables showing typical sprung frequency values, effect of spring rate on sprung frequency

7.4.1 Spring rate effect on handling

Setting	front sprung rate (n/m)	rear sprung rate (n/m)	front critical (n/m/s)	rear critical (n/m/s)	front sprung frequency (hz)	rear sprung frequency (hz)	average sprung frequency (hz)
Maximum	75000	150000	12210.65109	17268.46837	2.9	3.2	3.0
Middle	50000	100000	9969.954864	14099.64539	2.4	2.6	2.5
Minimum	35000	70000	8341.462701	11796.60968	2.0	2.2	2.1

Figure 7-33 The maximum and minimum settings for spring rate testing

For the testing plan spring rates that were not excessive but large enough to see a difference in handling would result in a spring rate between 2-3 hz. The reason for the spring rate testing going above the current spring rate was because 2.1 hz seemed too soft for a mid-aero race car based on other car sprung frequencies.

Hard spring rate causes “unprovoked sliding and a harsh ride” (Smith, 1978)

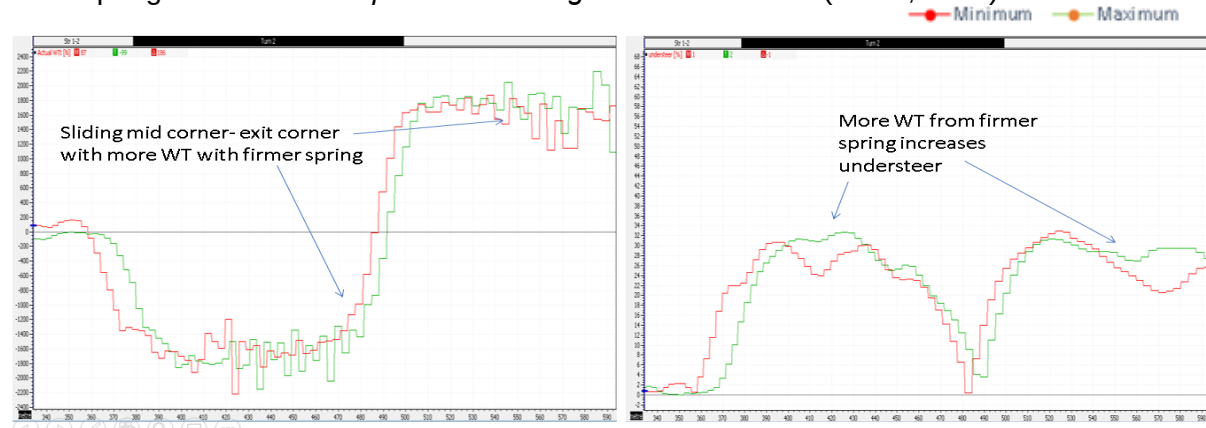


Figure 7-34 Right to left hand corner at Pembrey

Hard spring rate causes the “car to point in on turn but mid corner understeer” (Smith, 1978)

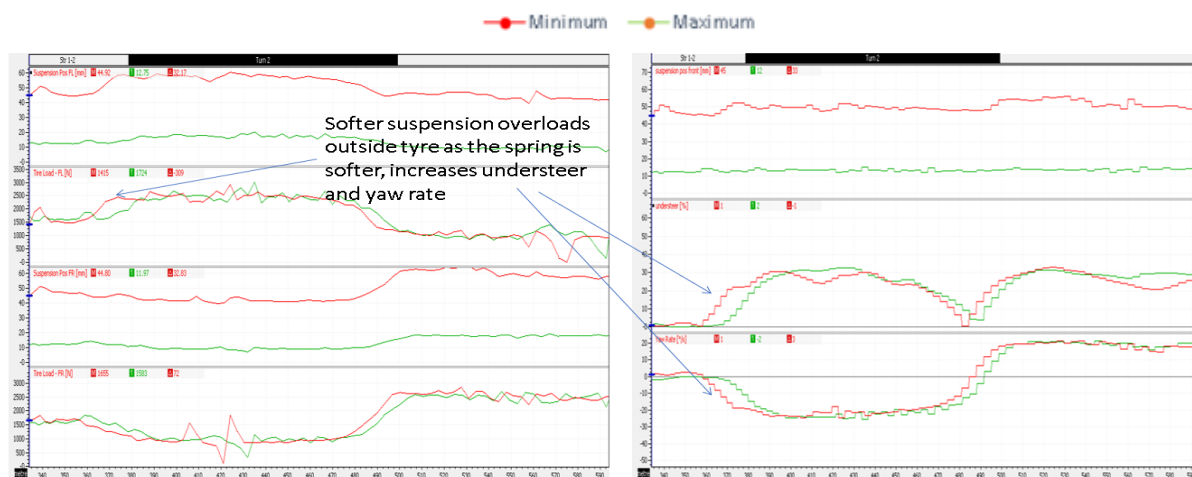


Figure 7-35 Right to left hand corner at Pembrey

7.4.1.1 Analysis of handing effect with the maximum spring rate

The handling data matches with the Carroll Smiths statements because the spring rate goes from soft to hard that is excessive enough to notice, also the spring rate change has bigger difference on handling compared to dampers as they spring supports the sprung mass, therefore there doesn't need to be as large change in spring rate as dampers because the spring has a bigger effect on handling. Therefore a spring rate change from 1-4 hz would be too excessive and provide irrelevant data as although a difference could be shown, those spring rates would not be used in real life.

The data shows during change of direction that there is more WT mid corner leading to more slip which is shown in the understeer channel. Slip increases because the outside corner resists displacement during cornering and reduces load on the outside tyre as there is less damper movement. This reduces tyre force on the outside tyre and causes “unprovoked sliding”.

Softer spring causes a “floating ride, sloppy response” (Smith, 1978)

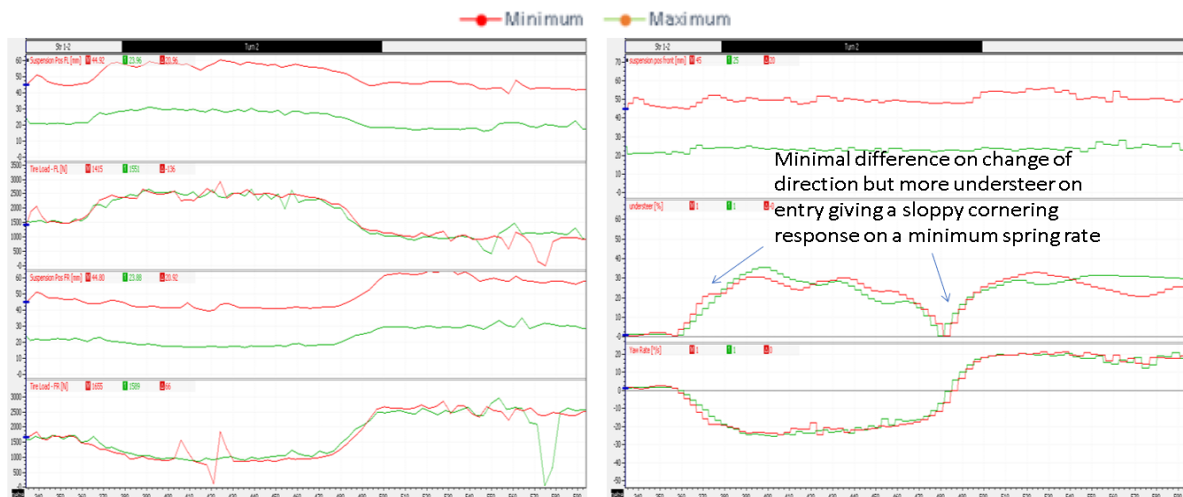


Figure 7-36 Right to left hand corner at Pembrey

Soft spring rate causes “excessive roll, dive and squat” (Smith, 1978)



Figure 7-37 Right to left hand corner at Pembrey as well as squat and dive at turn 1

7.4.1.2 Handling analysis of the minimum spring rate

Figure 7-36 shows that a softer spring increases initial understeer, this is because when the car initially turns in the lateral acceleration shifts WT to the outside wheel and loads up the outside tyre, the softer spring reduces this force because the damper loads up and the spring resists displacement. The softer spring overloads the outside tyre causing slip and also increases roll angle, this can be seen in figure 7-37.

When looking at braking and acceleration it can be seen that the softer spring rate increases dive and squat because of the WT shifting forwards and rear wards with a softer spring unable to support the extra force causing a bigger change in rake angle. This can cause a “sloppy

response” because there is more body movement which reduces handling response as roll angle is increased for longer as well during cornering seen in figure 7-37.

7.4.2 Spring rate over bumps

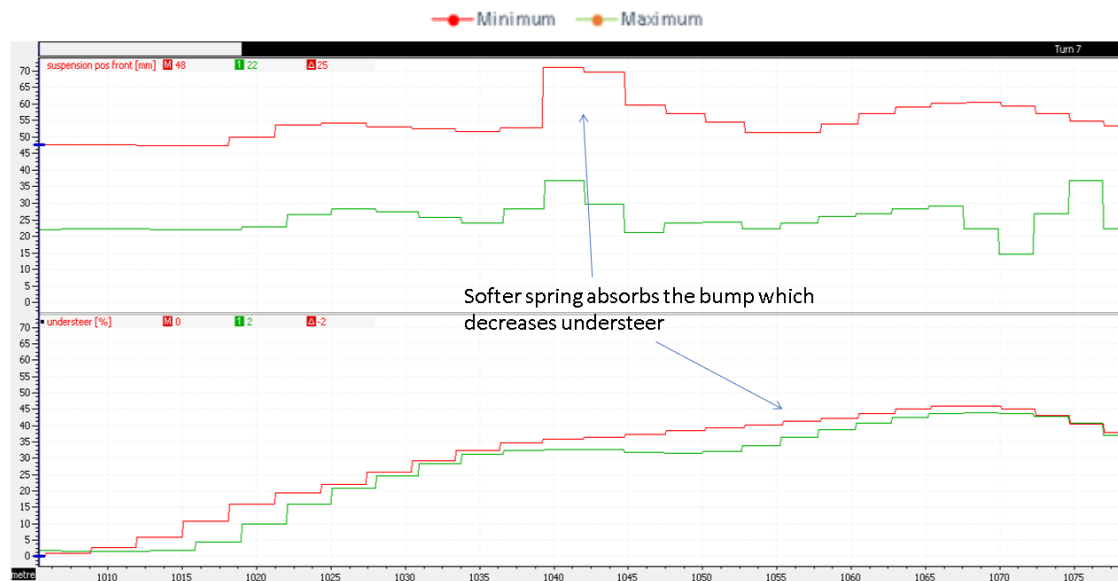


Figure 7-38 Mid corner bump at Long beach

7.4.2.1 Handling analysis of spring rate over bumps

Figure 7-38 shows that the softer spring setting reduces the ride height which can decrease WT as the COG is lower which reduces roll angle and slip. Over the bump the softer spring increases damper travel compared to the maximum setting which reduces tyre load on the firmer spring rate, this increases understeer as can be seen in the understeer trace as the softer spring allows more movement of the unsprung mass controlling the sprung mass. However a softer spring over the bumps can cause excessive damper movement which could cause a “floaty feeling” and the car could ground out over bumps, the reason why you can’t see this in the data is because 2hz is not soft enough to show this whereas 3hz is firm enough to show the understeer increase. Below is a table of the recommended front and rear default spring rate based on the investigation.

setting	front spring rate (n/m)	Rear spring rate (n/m)	front sprung frequency (hz)	rear sprung frequency (hz)	average frequency (hz)
Default spring rate	50000	100000	2.4	2.6	2.5

Figure 7-39 Recommended default spring rate

7.5 Analysing the handling effect of the new damper coefficients

7.5.1 Low speed compression

7.5.1.1 Force vs velocity graph

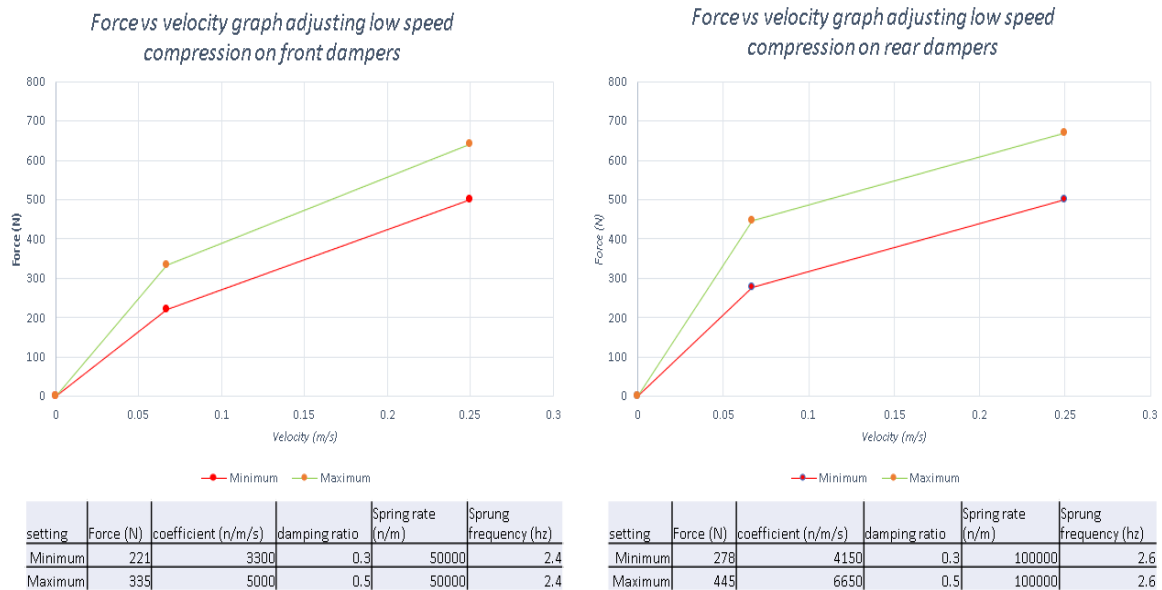


Figure 7-40 Comparison of front to rear new damper coefficients

7.5.1.1.1 Analysis of the force vs velocity graph

The force vs velocity graph shows that difference from maximum to minimum was a damping ratio of 0.3-0.5, the reason for this is that a lower compression would increase tyre load, reduce damper fade, and match the damping ratio of recommended compression coefficients. As the compression is softer a stiffer spring rate would be used to counteract the softer dampers.

Figure 7-40 shows the rear coefficients are stiffer than the front, the reasons for this will be because 60% weight distribution over the rear and a stiffer spring rate of 50000 n/m increases the critical damping rate. This means the coefficients need to be stiffer at the rear to get the damping ratio to 0.3-0.5 to match the front. This means that there will be more WT over the rear causing oversteer but the 60/40 weight distribution decreases understeer to balance the handling affect.

7.5.1.2 Histograms

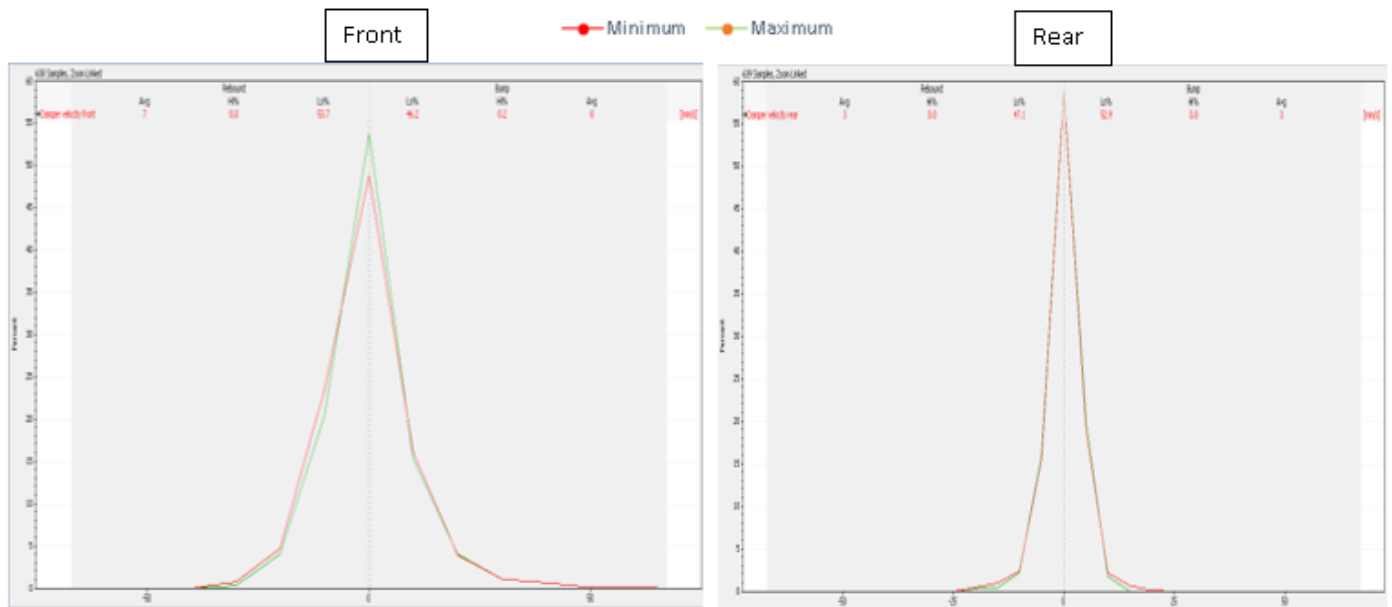


Figure 7-41 Comparison of front to rear histograms of the new coefficients

7.5.1.2.1 Analysis of the histograms

The histograms show a minimal difference between minimum to maximum as the difference is 2500 n/m/s for a damping ratio between 0.3-0.5. This is a lot smaller incremental difference compared to real life as for the new damper coefficients will be looking in more detail compared to looking at excessive values.

7.5.1.3 Effect on handling

Firm compression causes “chassis roll slow to develop” (Smith, 1978)



Figure 7-42 Right to left hand corner at Pembrey

Firm compression “reduces understeer on initial turn in” (Smith, 1978)

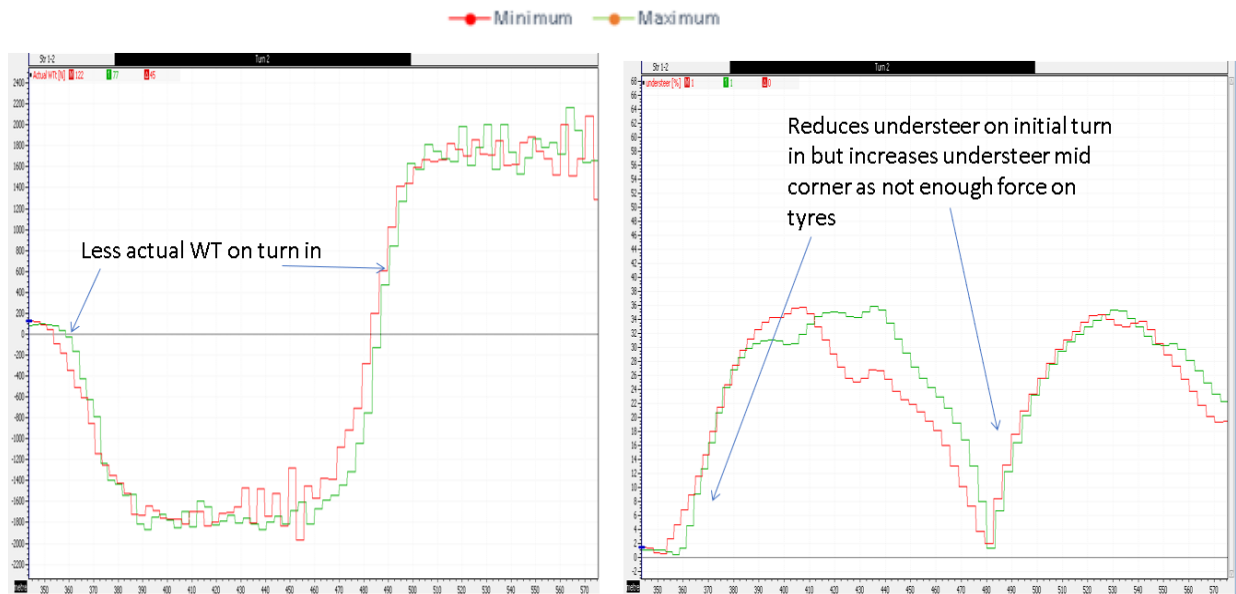


Figure 7-43 Right to left hand corner at Pembrey

7.5.1.3.1 Handling analysis of maximum compression

The data correlates to Carroll Smiths' statements more than the real life damper coefficients as the damping ratios are set to the recommended values for circuit racing which are underdamped compared to over damped. This can be seen in figure 7-42 where “*chassis roll is slow to develop*” on the firmest setting because the outside damper reduces damper travel and reduces roll angle, this reduces WT and decreases understeer as can be seen in figure 7-43. But only on turn in as the firmest setting reduces tyre load force. Therefore a damping ratio between 0.3-0.5 is wide enough to correlate to Carroll Smiths statement but also accurate enough to real life as the difference in coefficients from maximum to minimum is 2500 n/m/s.

What is shown in figure 7-43 is that mid corner to corner exit understeer increases significantly on the firmest setting, this is because there isn't enough force on the outside tyre which shows that 0.5 is too firm for low speed compression.

Soft compression and “causes excessive squat and dive” (Smith, 1978)

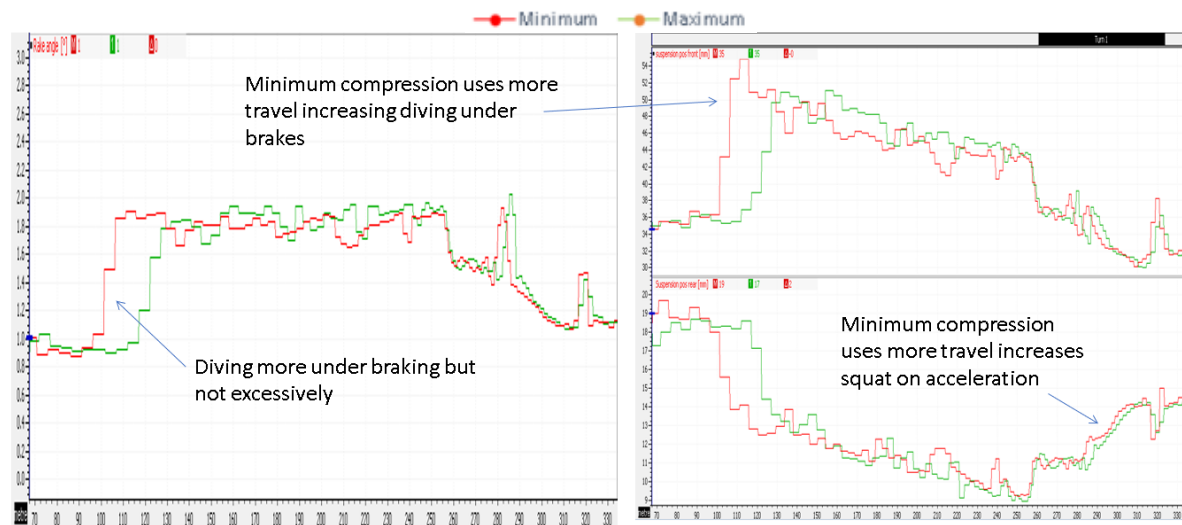


Figure 7-44 Right to left hand corner Pembrey

7.5.1.3.2 Handling analysis of minimum compression

A softer compression damping ratio (0.3-0.4) provides more grip and less WT in cornering than a damping ratio of 0.5 but the softer setting reduces stability in accelerating and braking.

7.5.2 Low speed rebound

7.5.2.1 Force vs velocity graph

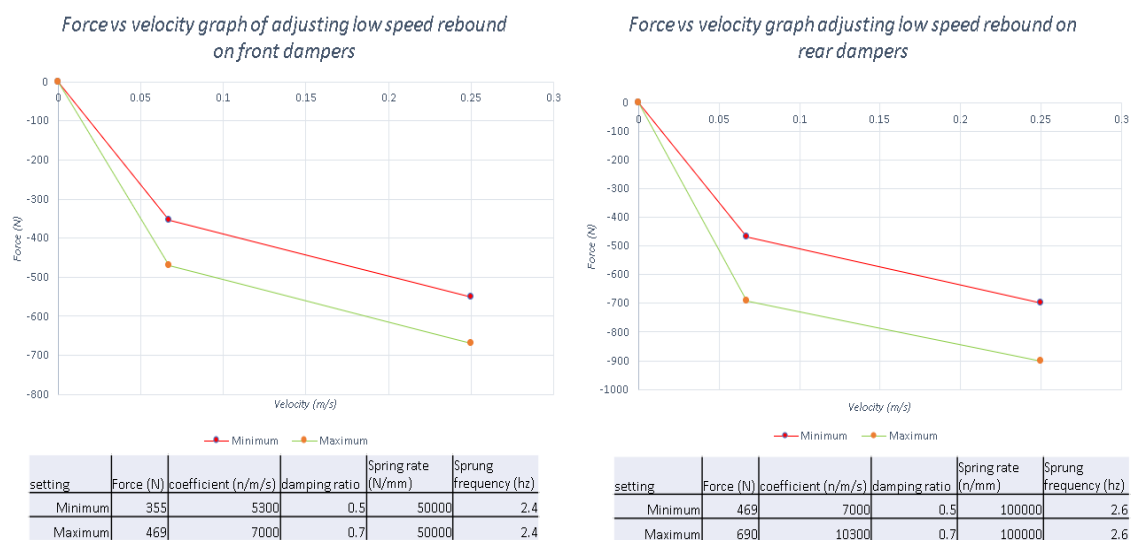


Figure 7-45 Comparing front to rear force vs velocity graph

7.5.2.1.1 Analysis of the force vs velocity graph

The force vs velocity graph show similar trends to compression force vs velocity graph as ideal damping ratio for rebound is between 0.5-0.7 which is stiffer than compression, so the extra rebound force provides more stability by controlling the unsprung mass.

7.5.2.2 Histograms

7.5.2.2.1 Analysis of the histograms

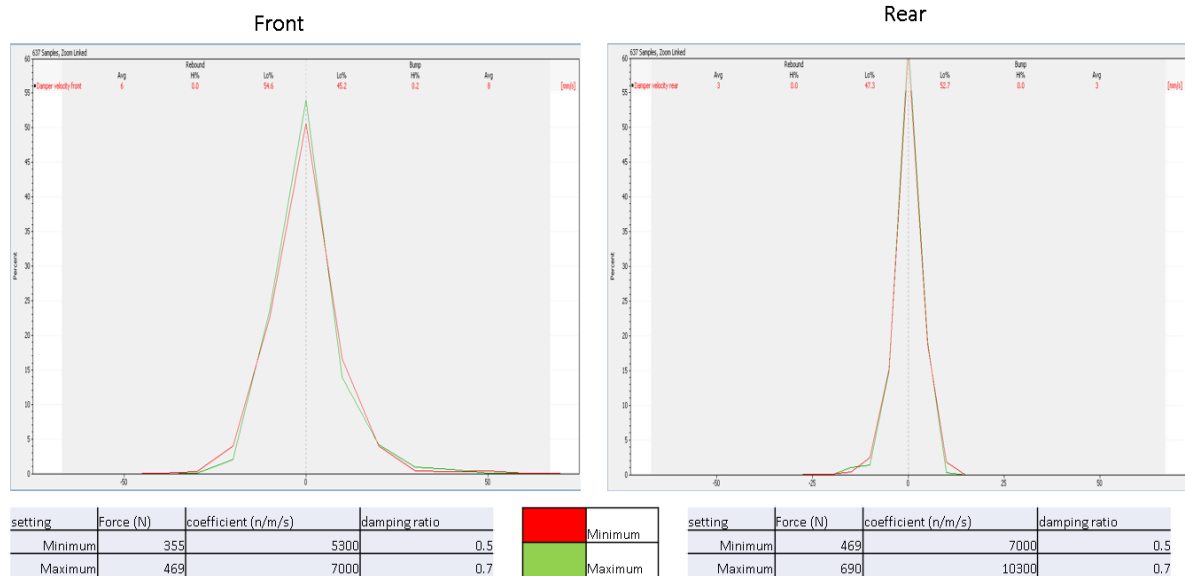


Figure 7-46 Comparing front to rear histograms

The trends from the histograms correlate to the new compression coefficients as there is minimal difference between maximum and minimum because the damping ratio range from 0.5-0.7 is only 3000 n/m/s. The reason for having a small damping ratio range was to go into more detail on the handling affects instead of excessive value. This means that the trend of the histogram isn't too important as long as it can be seen that the softer setting spends more time at lower velocities. This can be seen in rebound from velocities 20-45 mm/s that the red softest setting spends more time at these velocities.

7.5.2.3 Effect on handling

Slower/firmer rebound can cause “the inside wheel to return to the ground slowly” (Smith, 1978)

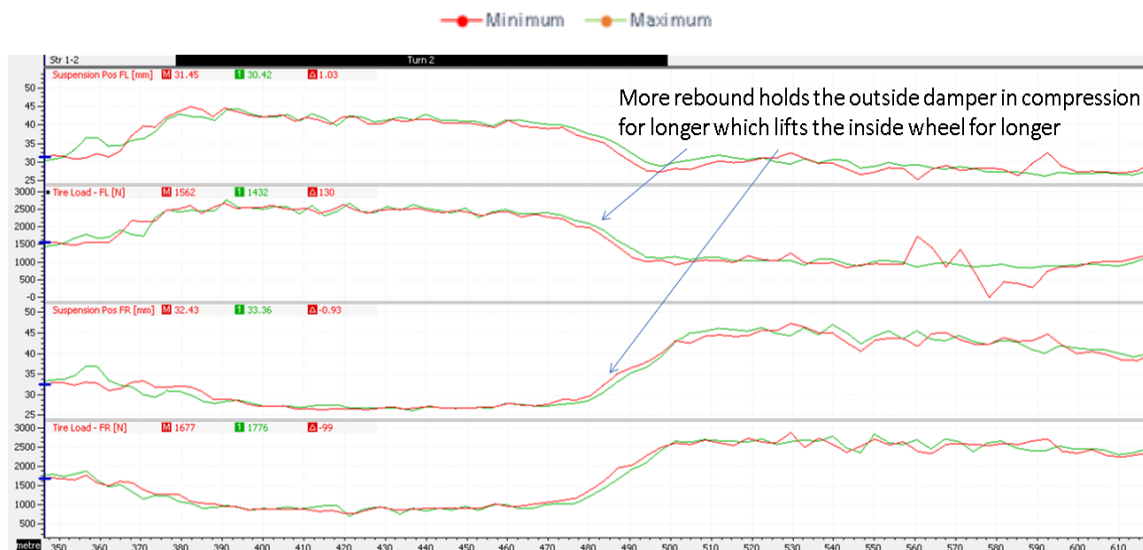


Figure 7-47 Right to left hand corner at Pembrey

Slower/firmer rebound can “overload the outside tyre and underload the inside tyre” (Smith, 1978)

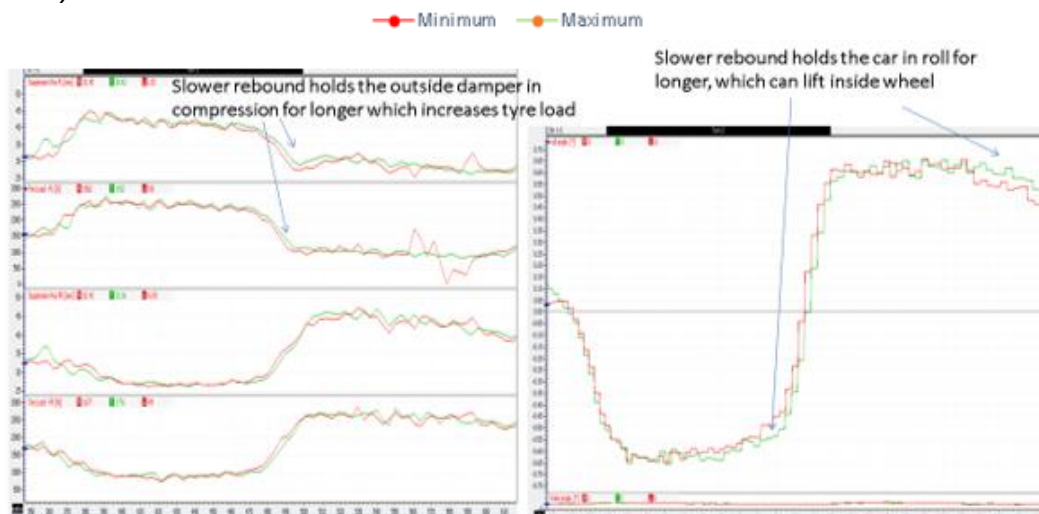


Figure 7-48 Right to left hand corner at Pembrey

7.5.2.3.1 Handling analysis of maximum rebound

What can be seen in figure 7-47 is the slower/firmer rebound setting is holding the outside damper in compression for longer but not further into compression like the real life damper forces, this will be down to the range from maximum to minimum as the real life damping ratio values go from 0.3-1.3 which an excessive difference in damper behaviour.

Faster/softer rebound can be “difficult to gain traction at the rear” (Smith, 1978)

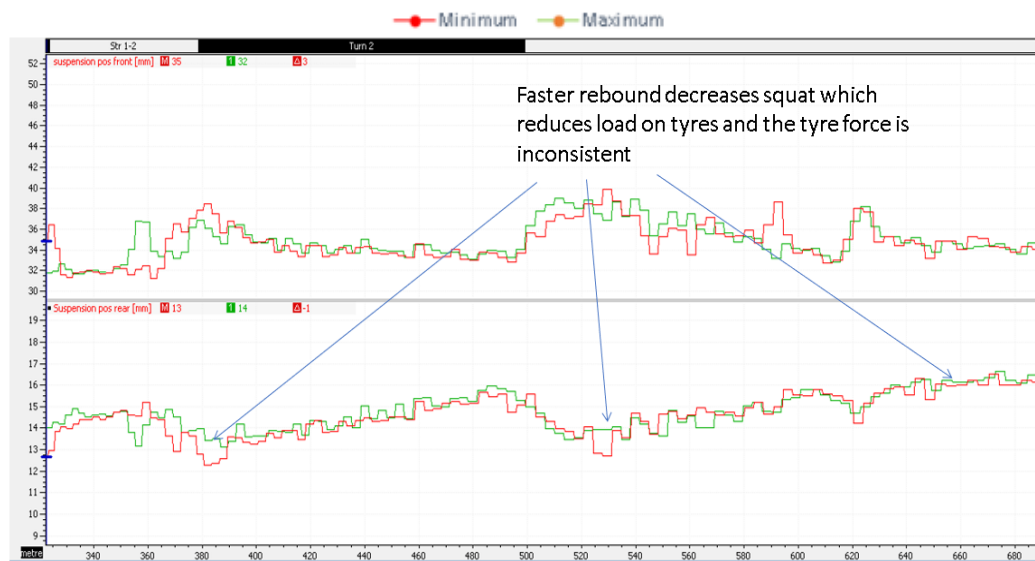


Figure 7-49 Acceleration of turn 1 at Pembrey

Faster/softer rebound can “increase roll velocity” (Smith, 1978)



Figure 7-50 Right to left hand corner at Pembrey

7.5.2.3.2 Handling analysis of minimum rebound

A softer/faster rebound setting can be “difficult to gain traction at the rear” because once the dampers are compressed from the weight transfer accelerating that the rebound can’t control the damper extension causing oscillating, this is emphasized on the minimum setting and can cause tyre force contact variation which decreases grip load whereas a firmer setting can increase slip but provides more consistent tyre force meaning it’s easier to gain traction..

Figure 7-50 shows the roll velocity increases the softer/faster the setting, this can decrease stability as the body movement of the sprung mass is more unstable because the dampers

are oscillating more, this can increase turn in response but also make the car twitchier on a change of direction (chicane). This means that although a firmer setting can increase understeer and roll for longer the sprung mass is controlled more and can increase stability, this can make the car easier to drive and give the driver more confidence as the sprung mass is more predictable.

7.5.3 High speed compression

7.5.3.1 Force vs velocity graph

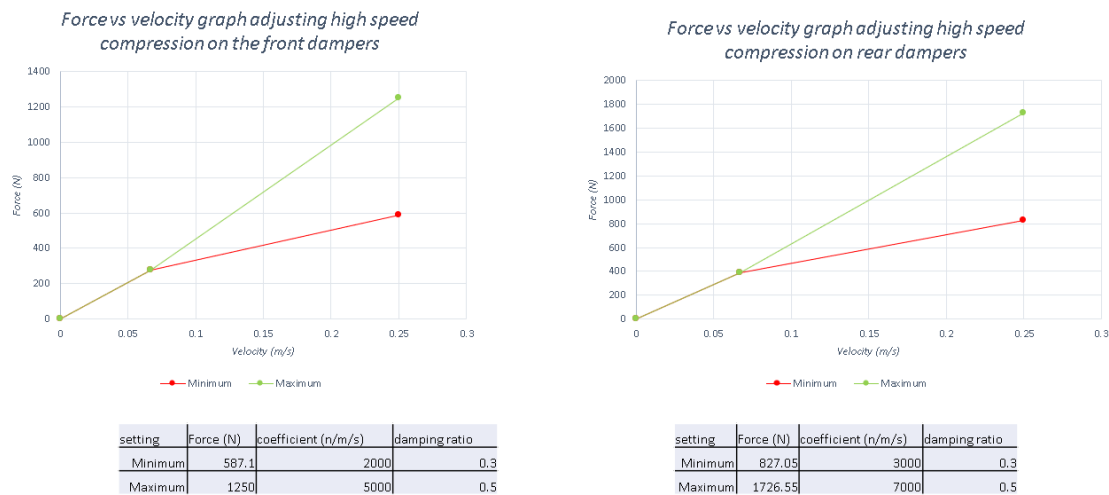


Figure 7-51 Comparing front to rear force vs velocity graphs

7.5.3.1.1 Analysis of force vs velocity graph

What was shown when looking at the real dampers was that the high speed compression would need to be softer than the low speed to absorb the high speed impacts. Therefore the damping ratio range was from 0.2-0.5 to be large enough to show a difference between maximum and minimum.

Figure 7-51 shows that a damping ratio of 0.5 is very stiff as the gradient of the high speed trend is very steep compared to 0.2 which looks very different to real life damper forces. Therefore the force vs velocity graph does not correspond to real life damper forces so before analysing the handling 0.5 would be too stiff.

7.5.3.2 Histograms

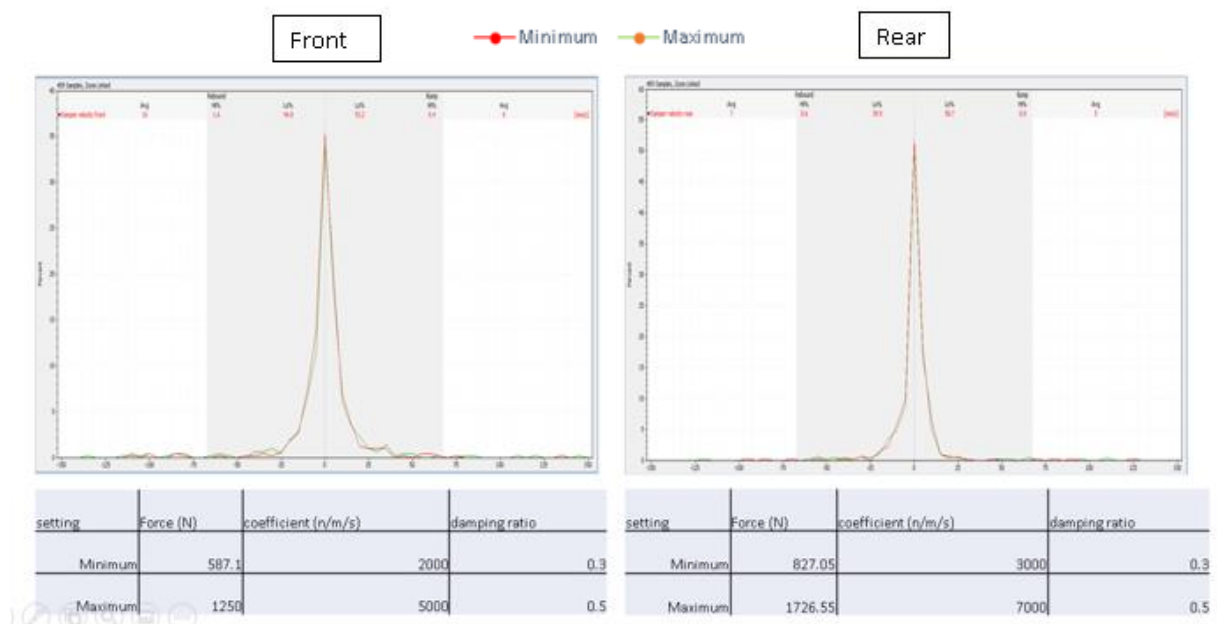


Figure 7-52 Comparing front to rear histograms

7.5.3.2.1 Analysis of histograms

When looking at the real life damper forces with a larger range of adjustment it was difficult to see in the histograms because the dampers spend a lot more time in low speed. As the range is only a 0.3 damping ratio it is difficult to see the difference. Therefore comparing the force vs velocity graph to the data would be reliable as the data could be mismatched as it is difficult to see the difference in high speed from the histograms.

7.5.3.3 Effect on handling

Firm compression can stop the dampers “*absorbing impacts from bumps and kerbs*” (Smith, 1978)

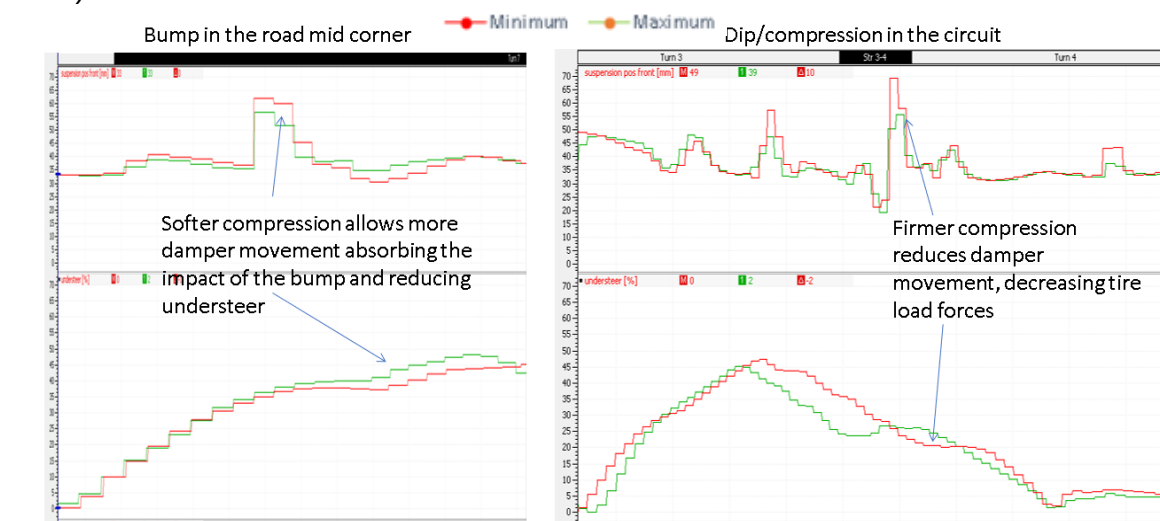


Figure 7-53 Comparing long beach to Silverstone high speed input

Firm compression can cause “*instability over bumps, skips over bumps*” (Smith, 1978)

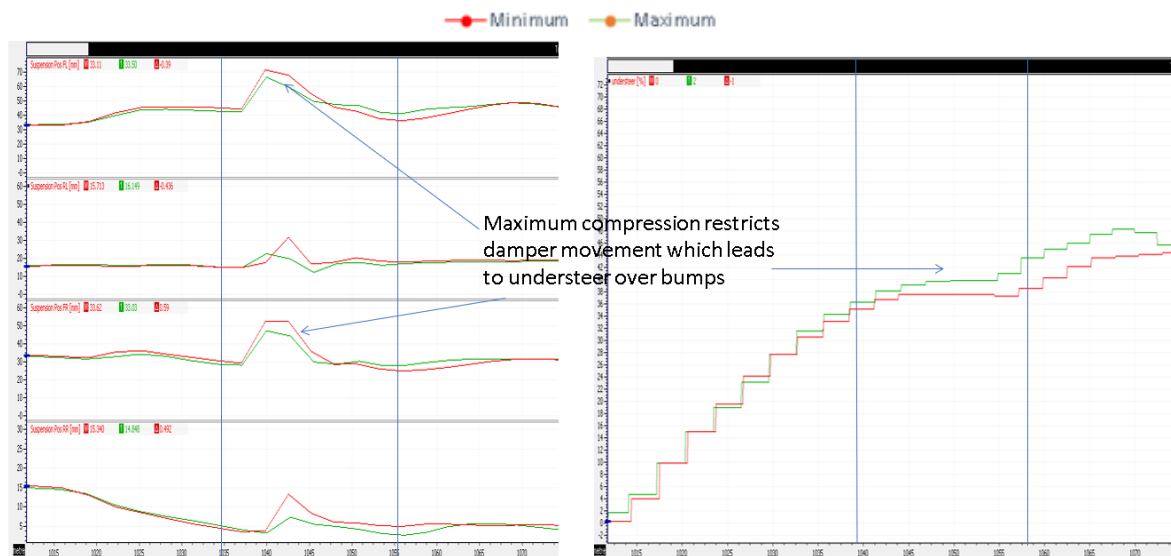


Figure 7-54 Showing high speed input at long beach

7.5.3.3.1 Handling analysis of maximum compression

The data correlates with the real life damper forces as the damping ratio and coefficient values are very similar, real life damping ratio ranging from 0.3-0.6 and updated 0.2-0.5. The data from the new coefficients is more reliable because the coefficients have been defined by the damping ratio so the handling should directly correspond to the damping ratio whereas the real life data coefficients were from the damper dyno that was adjusted by the low speed forces.

From the data a bump mid corner and a dip in the circuit shows that the firmest setting increases understeer by restricting the damper movement, this shows that a damping ratio from 0.2-0.3 would be ideal for the bumps as this is softer than low speed compression which would match the trends of a force vs velocity.

Soft compression could cause the “*car to ground over bumps and compression*” (Smith, 1978)



Figure 7-55 High speed input at Silverstone

Soft compression can give a “floaty feeling over bumps” (Smith, 1978)

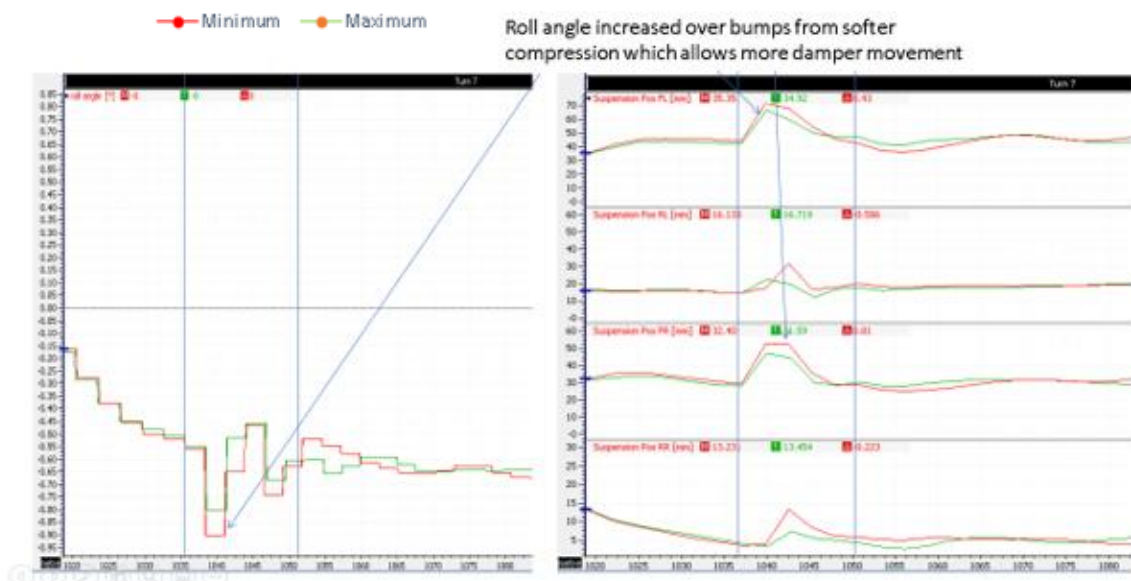


Figure 7-56 High speed input at long beach

7.5.3.3.2 Handling analysis of minimum compression

As the damping ratio is ranging from 0.2-0.5 there is quite a large difference for high speed adjustment and this shows in the data as the ride height drops by 20mm more, this would directly correspond to the extra damper travel. As the softer setting reduces understeer the softest setting would be ideal as long as the car does not ground out which it does not. As the car is mid aero there would not be sufficient aero force to reduce the damper travel over compressions significantly enough to majorly factor into the damper coefficients.

Figure 7-56 shows that the roll angle increases over the bumps because there is more damper travel, this can decrease stability because there is more WT over the bumps and can vary tyre load force from the inside to outside. This means that the damping ratio of 0.2 could be too soft as although the car is not grounding out the roll angle has increased which could be too soft.

7.5.3.4 High speed rebound

7.5.3.5 Force vs velocity graph

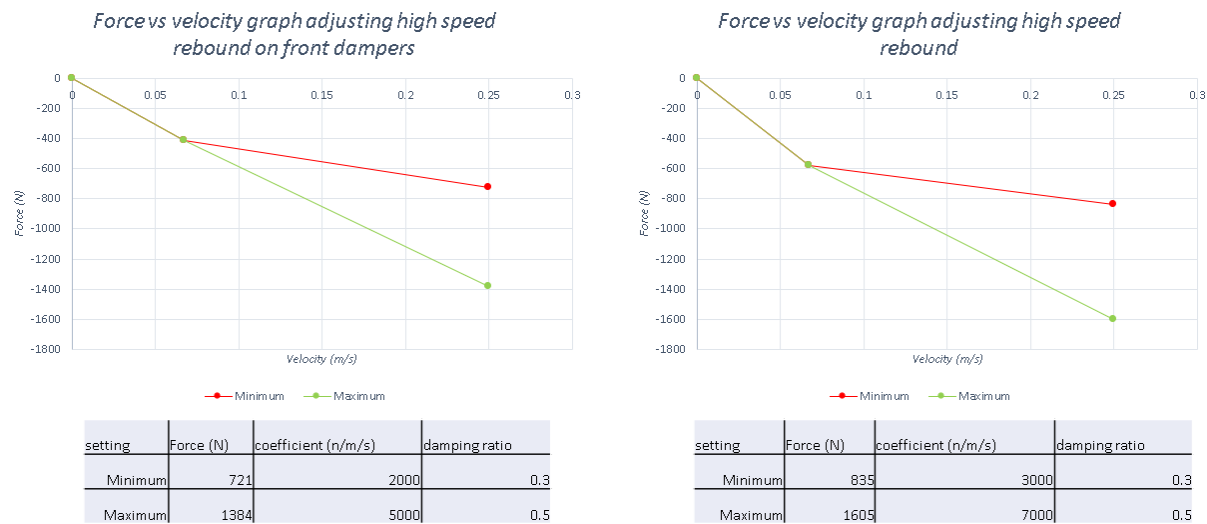


Figure 7-57 Comparing front to rear force vs velocity graphs

7.5.3.5.1 Analysis of force vs velocity graph

As from learned for the high speed compression investigation 0.5 damping ratio was too stiff which could mean that 0.5 would be too stiff for rebound, however as seen from low speed rebound forces at larger which should be the same for high speed as firmer rebound controls the sprung mass.

7.5.3.6 Histograms

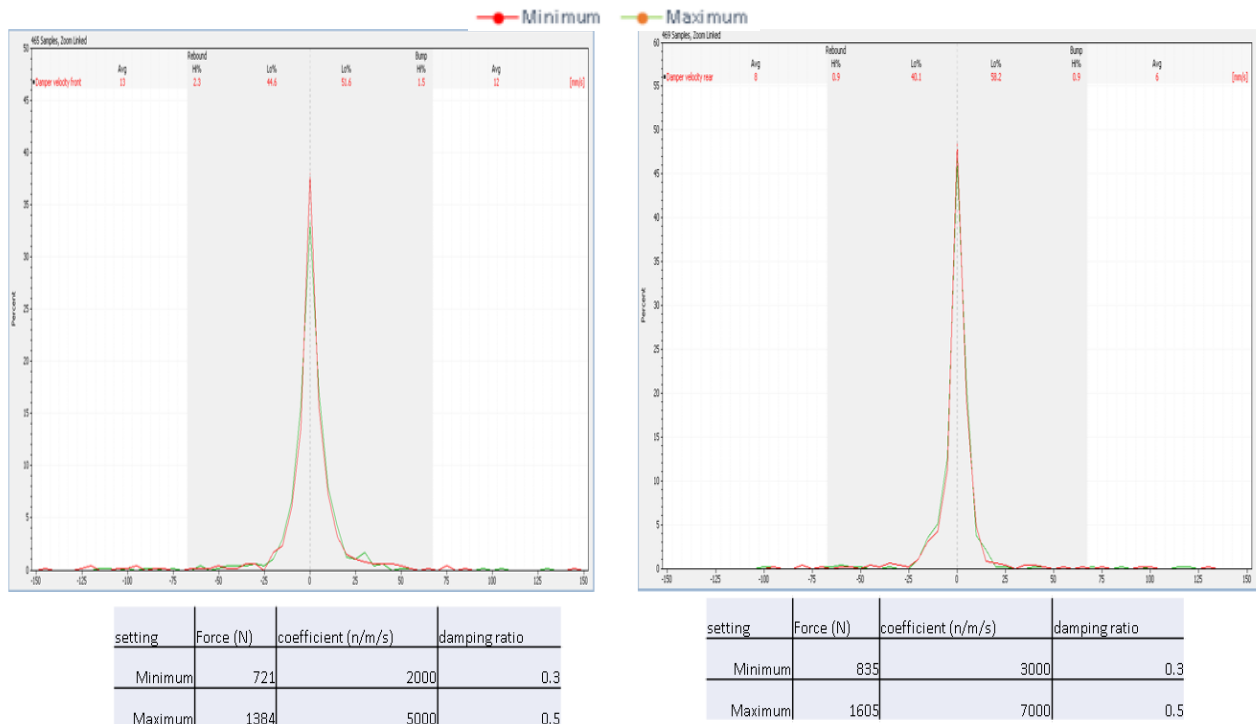


Figure 7-58 Comparing front to rear histograms

7.5.3.6.1 Analysis of histograms

Just like the high speed compression trends the histograms are difficult to see the difference in high speed rebound as the range in damping ratio is only 0.3 which would not be larger enough to notice a difference in high speed as histogram mainly shows the difference in low speed.

7.5.3.7 Handling effects

Slower/firmer effect on the dampers are that “dampers doesn’t return to full stroke to absorb the next impact” (Smith, 1978)



Figure 7-59 High speed input at long beach

Firmer/slower rebound can cause “sliding over bumps” (Smith, 1978)

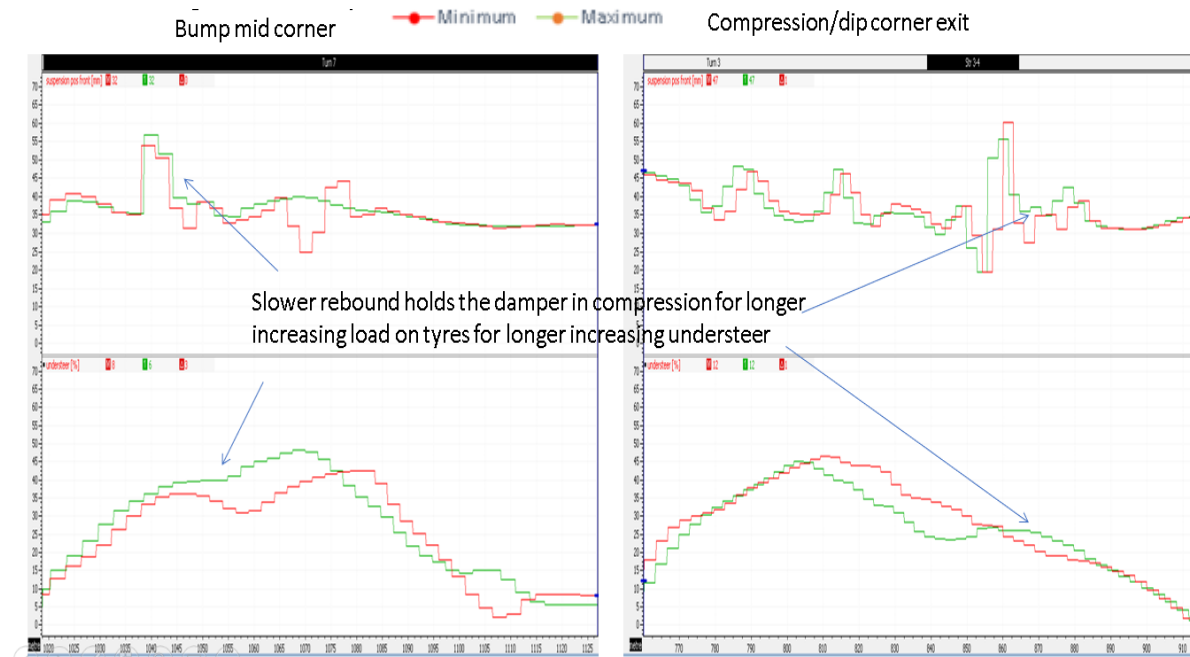


Figure 7-60 Comparing long beach to Silverstone high speed input

7.5.3.7.1 Handling analysis of maximum rebound

0.5 would be too firm for rebound as the data shows the same trends from a bump in the corner and a dip in the circuit which shows the dampers behave the same over 2 different types of high speed impact.

Faster/softer rebound can cause “excessive oscillation over bumps and kerbs” (Smith, 1978)

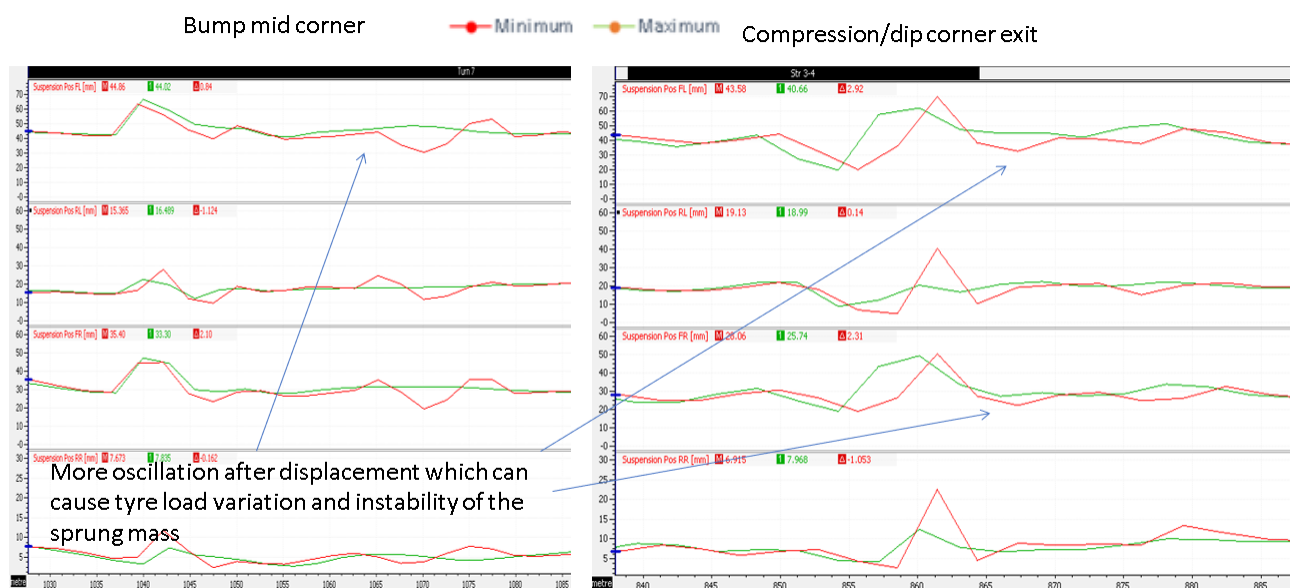


Figure 7-61 Comparing long beach to Silverstone high speed input

Faster/softer rebound can cause “*instability over bumps and corner entry*” (Smith, 1978)

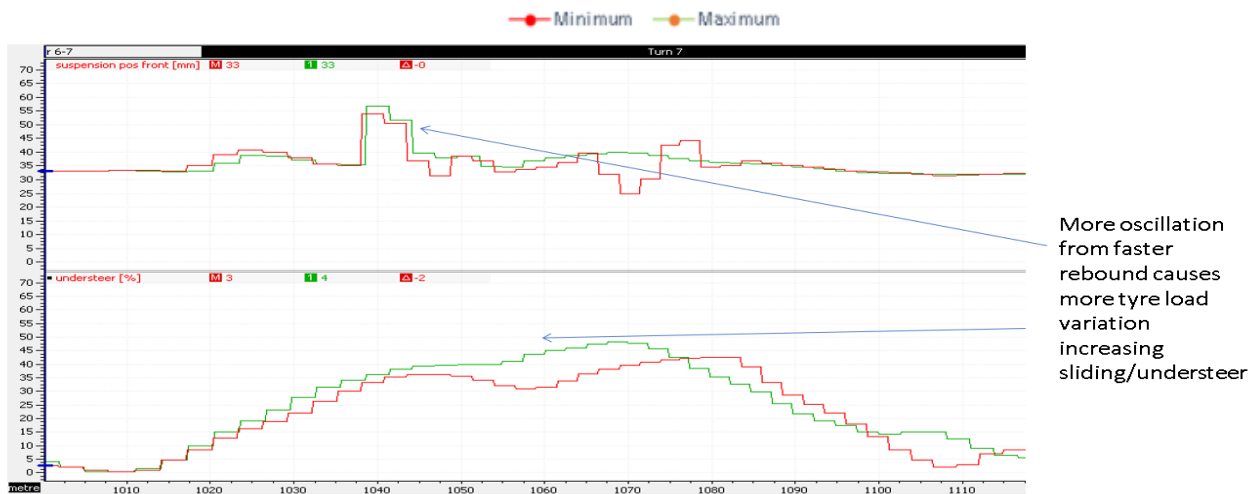


Figure 7-62 High speed input Long beach

7.5.3.7.2 Handling analysis of minimum rebound

Figure 7-61 shows that there is more damper oscillation after the impact of the bump showing that a damping ratio of 0.2 could be insufficient force as although there is less understeer there is more damper movement after the bump causing more tyre force variation. Although this oscillation after the bump was not in the high speed damper range and controlled by the low speed range as the velocity was below the knee point speed, therefore the high speed rebound force could be soft immediately after impact and then when the dampers start to oscillate this would be in the low speed range and the ideal low speed rebound force was between 0.5-0.7 which is significantly stiffer than high speed.

8 Discussion

8.1 Spring set up guide

If the car is understeering when cornering, this could be the front spring being too stiff or soft, therefore the sprung frequency would need to be compared to similar vehicles to see if it's too soft or too stiff. It's more likely that if there is understeer/slip the spring is likely to be more stiff as there not enough load on the tyres, whereas if the spring were too soft that increased slip there would be obvious handling characteristics such as increased roll and pitch.

If the car is oversteering when cornering, this could be because the rear spring is too firm causing more slip unless there is excessive roll or squat on acceleration as this could be because the spring is too soft. Therefore correlating the sprung frequency is good base point to compare but needs to be correlated to the data as ideally you would want as stiff as spring as you could go but excessively would increase slip.

If the car is understeering over bumps, this is very likely that the spring would be too firm at the front as the spring isn't absorbing the bumps and not allowing enough movement of the wheel, this keeps the body of the vehicle more rigid but does increase slip. The only way the spring could be too soft causing understeer would be that the spring rate is excessively soft, which means that the sprung frequency would need to be compared to similar vehicles.

If the car is oversteering over bumps, similar to understeer this would be that the rear is too stiff unless there is excessive damper travel and the damper hitting the bump stops, even still this absorb the bumps compared to hard spring which would increase slip.

Overall the spring rate choice will have the biggest effect on handling as it controls the sprung mass and depending on the compression adjustment can cancel out the damper settings. For example small difference in maximum to minimum compression the spring rate choice will override the handling effects of compression. If the circuit is bumpy a softer spring rate choice will be recommended so the MCR will be more stable over bumps and provide more grip mid corner as long as the car is not grounding out or excessive roll. Therefore choosing the spring rate will need to be a combination of comparing the sprung frequency to the handling data instead of just relying on one section.

8.2 Low speed set up guide

All of the assumptions made about the damping ratio and effect on handling will be based on the spring rate being set correctly with the sprung frequency within 0.5 Hz of similar vehicles, for the MCR this would range between 2-2.5 Hz.

If the car is understeering on entry, this could be that there is insufficient compression force as the outside tyre becomes overloaded and roll angle increases, therefore if the damping ratio was below 0.3 this would be too soft unless the spring rate is very firm but the dampers would be too soft at the front. If the firmest setting on the dampers is not firm enough then using a larger needle valve to restrict oil movement passing through but this would increase oil shear, therefore smaller bleed holes in the main piston will increase the low speed compression coefficient.

If the car is understeering mid corner, this could be because there is not enough rebound force as the front dampers start to extend and reduce tyre force on the front tyres, therefore if the

damping ratio is below 0.4-0.5 there is not enough rebound force. To increase the rebound force smaller diameter holes could be drilled in the main piston to reduce the oil flow.

If there is understeer mid corner this could also be because the compression is too firm and there is not enough force on the front tyres because the dampers are resisting movement, if there is more squat mid corner this will not be enough rebound because the front dampers are lifting but if there is more rake and less roll angle this would be there isn't enough compression and there is less body movement.

If the car is understeering corner exit, this could be because the compression is too soft on the front dampers as the outside tyres are overloaded and roll increased too much. Therefore check the damping ratio is not below 0.3 because this would be too soft or that the rebound is too firm on the front dampers as there holding dampers in compression for too longer. As too firm rebound and too soft compression give similar handling characteristics of a steep rake angle but if there is excessive roll this would be because the front dampers are too soft. If in doubt check the damping ratio to the desired damping ratio so the damper coefficients are in the correct range. Compression is 0.3-0.5 and rebound 0.5-0.7.

If the car is oversteering on entry, this could be because the rear dampers are too stiff and reducing tyre load, especially if it is a corner requiring good traction from power and will likely to increase oversteer. If it's a high speed corner it could be that there is not enough rebound force in the rear dampers and the roll velocity increases too quickly unloading the inside tyre and could decrease stability. To identify if its compression or rebound if the roll rate is too fast and the rear outside dampers load up to quickly this will be there is not enough rebound force and if there is less squat and less roll this will be the rear dampers are too firm. Comparing the damping ratio also shows if the dampers are behaving correctly.

If the car is oversteering mid corner, this will be that the rear dampers are too stiff decreasing tyre load and squat. As the MCR is a rear wheel drive car the rear dampers control all of the traction and the compression controls to force acting on the tyres. Therefore if there is any oversteer in the corner its likely to be more compression as rebound mainly affects entry and exit.

If the car is oversteering on corner exit, this could be that the rebound is too slow at the rear holding the dampers in compression for longer or that the compression is too firm on the rear dampers. To identify whether if it is too much compression or too much rebound will depend on the squat angle on corner exit, if there is more squat this will be too much rebound and if the rake angle is steeper (less squat) this will be the rear compression is too firm. If the rebound is too firm it can lift the front tyres because the weight transfer is further over the rear because there is more squat, this can increase understeer as well as oversteer.

Overall the handling of the low speed will need to compare the damping ratio to the handling data as the damping ratio also relates to the force vs velocity graph. Compression should be softer as the compression controls the unsprung mass whereas the rebound should be firmer to control the sprung mass.

8.3 High speed set up guide

If the car is understeering over bumps, just like spring rate this will be because the compression is too stiff on the front and could be seen in the data if there was not enough damper travel reducing tyre force. Understeer could be increased if the compression were too soft if the dampers were hitting the bump stops but would be more likely that the compression is too stiff. As there is no high speed adjust on the damper with a needle valve it would be down to the shim stack thickness, if there was understeer and the MCR skipping over the bumps the amount of shim stacks will need to be reduced to reduce the force needing to open the shim stacks which reduces the coefficient. Most likely that the spring rate will have a larger

effect over the bumps than the high speed adjust on the MCR as the MCR only has 2 way adjustable dampers which means the low speed force affects the high speed force. Therefore if the car is skipping over bumps and harsh ride over kerbs it would be recommended to drop the spring rate as this would make the car softer over bumps which the low speed damper adjust cannot affect.

9 Conclusions

The spring rate would need to be in an “ideal” range of a sprung frequency for the MCR between 2-2.5 hz as anything above or below would be excessive and increase pitch and roll if it were excessively soft and a harsh ride and slip if it were excessively stiff. In the “ideal” range you would want the spring rate to be as soft as you can go without excessive body roll as there is more force on the tyres and increases grip, this will provide the driver with a more predictable car. The relationship the spring rate has to damper settings is that the spring has a larger impact on handling compared to the dampers as the spring is always providing force and there is not enough range from minimum to maximum in the dampers to have as larger effect as changing spring rate. Therefore it's important to get the spring rate set before adjusting the dampers in order to get the car set up.

Ideally you want also want to run compression as soft as you could go without excessive body roll because there is more force on the tyres but identifying if you need to adjust compression or spring rate you would need to evaluate the force vs velocity graph and sprung frequencies as they provide similar handling characteristics. You would want to configure the compression and spring rate to support mass of the MCR front to rear as a starting point, i.e. stiffer for more mass. For low speed compression you would want a damping ratio ranging from 0.3-0.4 in low speed and high speed 0.2-0.3 with driver preference tweaking the compression from there.

For rebound you want to run a firmer setting of a damping ratio ranging from 0.6-0.7 to increase stability and reduce damper oscillation, the rebound will be more subjective to the driver style as the rebound has a greater effect than compression which affects the responsiveness on turn in and braking and corner exit, however the rebound force should always be above compression to keep the sprung mass more stable, this would follow the same trend as compression as the high speed you would want a damping ratio of 0.3-0.4.

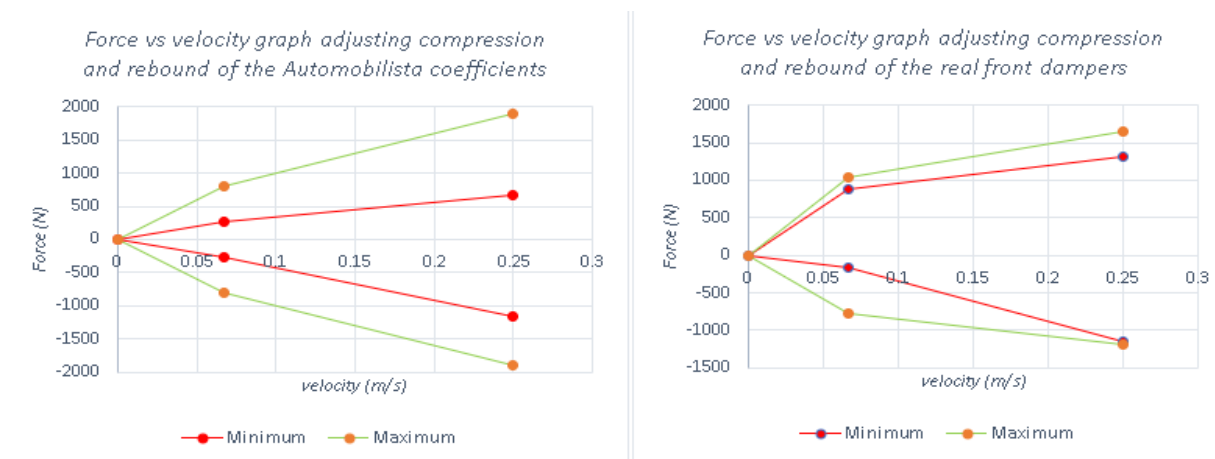
10 Appendices

10.1 Comparison between automobilista and real life

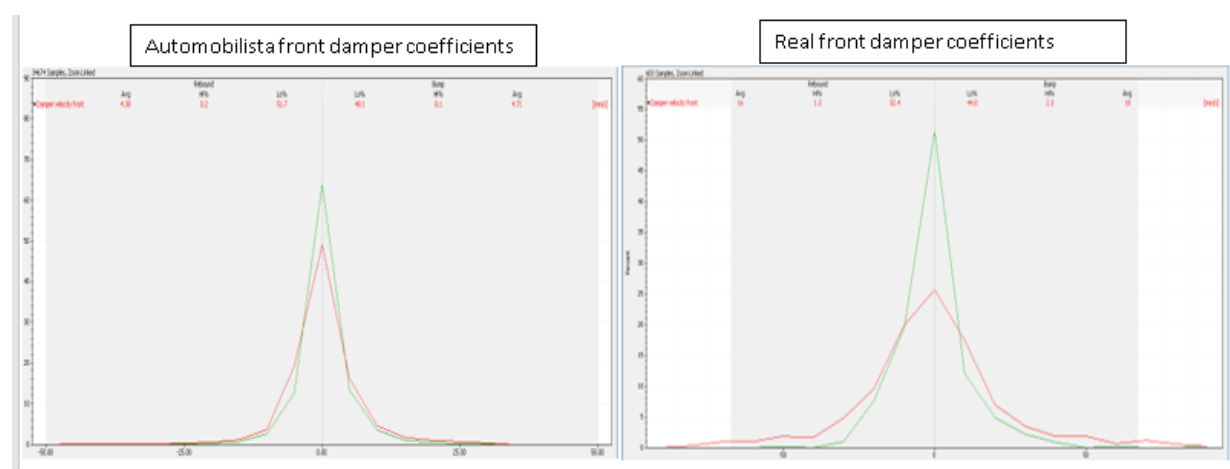
Automobilista coefficient				
	low compression (n/m/s)	low rebound (n/m/s)	high compression (n/m/s)	high rebound (n/m/s)
Minimum	4000	-4000	2200	-4900
Maximum	11999	-11999	6000	-5983

Real damper coefficients				
Front	low compression (n/m/s)	low rebound (n/m/s)	high compression (n/m/s)	high rebound (n/m/s)
Minimum	13134.32836	-2319.402985	2366.120219	-2315
Maximum	15374.62687	-11489.46269	3425.136612	-5368.7

10.2 Force vs velocity graphs



10.3 Histograms



10.4 Sprung and unsprung frequencies of the MCR

front					
mass (%)	unsprung mass (kg)	sprung mass (kg)	spring rate (n/m)	sprung frequency (hz)	critical damping rate (n/m/s)
60	44	228	39000	2.1	8805.225721

rear					
Mass (%)	unsprung mass (kg)	sprung mass (kg)	spring rate (n/m)	sprung frequency (hz)	critical damping rate (n/m/s)
40	32	340	67000	2.1	11541.05714

10.5 New coefficients low speed testing

testing plan											
setting	front spring (n/m)	Rear spring (n/m)	front LSC (n/m/s)	rear LSC (n/m/s)	front LSR (n/m/s)	rear LSR (n/m/s)	front sprung (hz)	rear sprung (hz)	Total (hz)	front damping ratio	rear damping ratio
1	130000	260000	4450	4450	6150	6150	3.8	4.2	4.0	0.3	0.2
2	100000	200000	4450	4450	6150	6150	3.3	3.7	3.5	0.3	0.2
3	75000	150000	4450	4450	6150	6150	2.9	3.2	3.0	0.4	0.3
4	50000	100000	4450	4450	6150	6150	2.4	2.6	2.5	0.4	0.3
5	35000	70000	4450	4450	6150	6150	2.0	2.2	2.1	0.5	0.4
6	50000	100000	3300	4150	6150	6150	2.4	2.6	2.5	0.3	0.3
7	50000	100000	4150	5800	6150	6150	2.4	2.6	2.5	0.4	0.4
8	50000	100000	5000	6650	6150	6150	2.4	2.6	2.5	0.5	0.5
9	50000	100000	4150	5800	5300	7000	2.4	2.6	2.5	0.5	0.5
10	50000	100000	4150	5800	6150	8650	2.4	2.6	2.5	0.6	0.6
11	50000	100000	4150	5800	7000	10300	2.4	2.6	2.5	0.7	0.7
12	75000	150000	3300	4150	5300	7000	2.9	3.2	3.0	0.3	0.2
13	35000	70000	5000	6500	7000	10300	2.0	2.2	2.1	0.6	0.6

adjustment

10.6 New coefficients high speed testing

Damping coefficient						
Log	front compression (n/m/s)	rear compression (n/m/s)	front rebound (n/m/s)	rear rebound (n/m/s)	front damping ratio	rear damping ratio
1	2000	3000	5000	7000	0.2	0.2
2	5000	7000	5000	7000	0.5	0.5
3	8000	11000	5000	7000	0.8	0.8
4	5000	7000	2000	3000	0.2	0.2
5	5000	7000	5000	7000	0.5	0.5
6	5000	7000	8000	11000	0.8	0.8

10.7 Maths channels

Name	Summary
Expressions	
☒ wheel rate front [Nm]	[(Spring Rate FL [N/m]*Motion Ratio FL^2)+Roll Bar Rate Front [N/m]]
☒ wheel rate rear [Nm]	[(Spring Rate RL [N/m]*Motion Ratio RL^2)+Roll Bar Rate Rear [N/m]]
☒ lateral acceleration [m/s/s]	G Force Lat [G]*9.81
☒ wheel deflection	[(Track Front [m]+Track Rear [m])/2]*roll angle [rad]/2
☒ anti roll moment front	(Track Front [m]*2*roll angle [rad]*wheel rate front [N/m])/2
☒ anti roll moment rear	(Track Rear [m]*2*roll angle [rad]*wheel rate rear [N/m])/2
☒ Ride height front [mm]	(Ride Height FL [mm]+Ride Height FR [mm])/2
☒ Ride height rear [mm]	(Ride Height RL [mm]+Ride Height RR [mm])/2
☒ roll stiffness front	[(Track Front [m]*2*wheel rate front [N/m])/2]
☒ roll stiffness rear	[(Track Rear [m]*2*wheel rate rear [N/m])/2]
☒ angle of roll centre [rad]	tan([(Static Roll Centre Height Front [m])/(Static Roll Centre Height Rear [m])*Wheelbase [m]])
☒ Height of CG above roll centre [m]	tan(angle of roll centre [rad])*Wheelbase [m]/2
☒ overturning moment [Nm]	(sprung mass [kg])*lateral acceleration [m/s^2]*(COG height [m]*Static Roll Centre Height Front [m]*Height of CG above roll centre [m])
☒ roll axis [rad]	(overturning moment [Nm])/(roll stiffness front +roll stiffness rear)
☒ Wturf [N]	(unsprung mass [kg]*2*lateral acceleration [m/s/s]*Front tyre rolling radius [m])/(Track Front [m])
☒ Wtusr [N]	(unsprung mass [kg]*2*lateral acceleration [m/s/s]*Rear tyre rolling radius [m])/(Track Rear [m])
☒ Wtfr [N]	Wturf [N]+Wtsurf+Wtusr [N]
☒ Wtrr [N]	Wtusr [N]+Wtsurf+Wtsrr [N]
☒ Actual Wtfr [N]	(Tire Load -FR [N]*Tire Load -FL [N])/2
☒ Actual Wtrr [N]	(Tire Load -RL [N]*Tire Load -RR [N])/2
☒ Actual Wtfr [N]	(((Tire Load -FL [N]*Tire Load -FR [N])+Tire Load -RL [N]*Tire Load -RR [N])*9.81)/lateral acceleration [m/s/s]/(cog height [m]*(Track Front [m])
☒ total roll stiffness [Nm]	roll stiffness front +roll stiffness rear
☒ aero front [N]	Tire Load -FL [N]*Tire Load -FR [N]*(((Static Weight FL [kg]*9.81)+(Static Weight FR [kg]*9.81)))+(1148)*G Force Long [G]*9.81*cog height [m]/Wturf [N]
☒ aero rear [N]	Tire Load -RL [N]*Tire Load -RR [N]*(((Static Weight RL [kg]*9.81)+(Static Weight RR [kg]*9.81)))+(1148)*G Force Long [G]*9.81*cog height [m]/Wtusr [N]
☒ CoLl front	aero front [N]/(0.5*air density [kg/m^3])/(Ground Speed [m/s]^2)
☒ CoLl rear	aero rear [N]/(0.5*air density [kg/m^3])/(Ground Speed [m/s]^2)
☒ total CoLl	CoLl front +CoLl rear
☒ CoP [ratio]	aero front [N]/Total aero load [N]
☒ Wtsrcl	(sprung mass [kg])*2*lateral acceleration [m/s/s]*Static Roll Centre Height Front [m]/(Track Front [m])
☒ Wtsrr	(sprung mass [kg])*2*lateral acceleration [m/s/s]*Static Roll Centre Height Rear [m]/(Track Rear [m])
☒ Wtsf [N]	(wheel rate front [N/m]*Track Front [m]*roll angle [rad])/2
☒ Wtsr [N]	(wheel rate rear [N/m]*Track Front [m]*roll angle [rad])/2
☒ Total wt [N]	Wtsf [N]+Wtsr [N]
☒ front slip [%]	(Slide Pot -FL [%]+Slide Pot -FR [%])/2
☒ rear slip [%]	(Slide Pot -RL [%]+Slide Pot -RR [%])/2
☒ understeer/oversteer [%]	front slip [%]-rear slip [%]
☒ Total aero load [N]	aero front [N]+aero rear [N]
☒ Corner radius [m]	(Ground Speed [m/s]^2)/lateral acceleration [m/s/s]
☒ Theoretical wheel steer [rad]	Wheelbase [m]/Corner radius [m]
☒ Theoretical steer position [%]	Theoretical wheel steer [rad]*(Steering Wheel Pot [%]/Theoretical wheel steer [rad])
☒ understeer [%]	abs(Theoretical steer position [%])-abs(Theoretical wheel steer [rad])
☒ Rake angle [rad]	(tan((Ride height rear [m]-ride height front [m])/Wheelbase [m]))
☒ gyro pitch [rad]	Gyro Pitch Angle [rad]*Gyro Pitch Angle [rad]
☒ Damper velocity front [m/s]	(Damper Velocity FL [m/s]+Damper Velocity FR [m/s])/2
☒ Damper velocity rear [m/s]	(Damper Velocity RL [m/s]+Damper Velocity RR [m/s])/2
☒ Front tire load [N]	(Tire Load -FL [N]+Tire Load -FR [N])/2
☒ Rear tire load [N]	(Tire Load -RL [N]+Tire Load -RR [N])/2
☒ Front to rear weight transfer [N]	Front tire load [N]-Rear tire load [N]
☒ Fsping rear [N]	(Spring Rate RL [N/m]*(Damper pos RL [mm]-Damper pos RL [mm])/2
☒ Fsping FL [N]	(Spring Rate FL [N/m]*suspension pos FL [mm]
☒ Fsping FR [N]	(Spring Rate FR [N/m]*suspension pos FR [mm]
☒ Fsping RL [N]	(Spring Rate RL [N/m]*suspension pos RL [mm]
☒ Fsping RR [N]	(Spring Rate RR [N/m]*suspension pos RR [mm]
☒ Weight distribution front to rear [ratio]	Front tire load [N]/(Front tire load [N]+Rear tire load [N])
☒ Weight distribution left to right [ratio]	(Tire Load -FL [N]+Tire Load -FR [N])/(Tire Load -FL [N]+Tire Load -FR [N]+Tire Load -RL [N]+Tire Load -RR [N])
☒ Natural frequency	sqrt(Spring Rate FL [N/m]/sprung mass [kg])
☒ Ride height [mm]	(Ride height front [mm]+Ride height rear [mm])/2
☒ actual magic number [ratio]	(Actual Wtfr [N])/(Actual Wtfr [N])
☒ Predicted magic number [ratio]	(Wtfr [N])/(Wtfr [N]+Wttr [N])
☒ suspension pos front [mm]	(Suspension Pos FL [mm]+Suspension Pos FR [mm])/2
☒ Suspension pos rear [mm]	(Suspension Pos RL [mm]+Suspension Pos RR [mm])/2
☒ lift vs drag efficiency	total CoLl*/drag coefficient [1]
☒ circumference	3.142*Corner radius [m]
☒ circumference/velocity	circumference */Ground Speed [m/s]
☒ Calculated yaw rate	(2*3.142)/(circumference/velocity)
☒ suspension pos FL [mm]	Suspension Pos FL [mm]-10.59
☒ suspension pos RL 2 [mm]	Suspension Pos RL [mm]-11.23
☒ suspension pos RR 2 [mm]	Suspension Pos RR [mm]-11.23
☒ Damper coefficient front	((Damper velocity front [m/s]> 0)& (Damper velocity front [m/s]<= 0.067))/Low speed compression Front [N/m]+((Damper velocity front [m/s]> 0)& (D
☒ Damper force front [N]	Damper velocity front [m/s]*Damper coefficient front
☒ Damper coefficient rear	((Damper velocity rear [m/s]> 0)& (Damper velocity rear [m/s]<= 90))/Low speed compression Rear [N/m]+((Damper velocity rear [m/s]> 0)& (D
☒ Damper force rear [N]	Damper velocity rear [m/s]*Damper coefficient rear
☒ Roll velocity [mm/s]	derivative(roll angle [rad])
☒ dynamic ride height [mm]	suspension pos front [mm]-Suspension pos rear [mm]

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