



University of Wales Trinity Saint David

To investigate the possibility of generating damper characteristics
from live track data

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Declaration of Authenticity

This thesis is the result of the author's original research. It has been composed by the author and has not been previously submitted for examination which has led to the award of a degree.

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Abstract

A brief explanation is done to how the damper fits into the suspension and its purpose. Research below explains the constructions of dampers and also briefly mentioned the different types of dampers available. Two clear comparable sets of data are shown with all the limitations in the data and how further work can be done for a more detailed analysis. A step by step process is provided to show how all the data was produced.

For the beginning of the report the dynamometer data was seen as the expected damping forces seen by the vehicle, by comparing this to the live track data it was seen that this was not the case.

This report shows the approach made to develop a force-velocity graph from live track data. An approach was made to then be able to predict or even analyse damper characteristics just from live track data.

Acknowledgements

For the contribution toward the progression of this investigation. I would like to take this opportunity to acknowledge the following people for all the help provided:

Roger Dowden (Supervisor) – I would like to thank Roger Dowden for providing direction at the early stages of the report and for planning regular meeting to keep up to date with progression of the repot.

Paul Davies (Technician) – I would like to thank Paul Davies as he helped with all the physical testings done on the dampers and also providing valuable insights to why the damper acts the way it does due to having worked with damper before.

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Introduction

This project will be looking at how the suspension reacts to changes in loads. Hence we will be looking at the motion of components under a load. For this investigation we are just interested in the vertical load generated through the tyres of the vehicle as it goes around the track.

The basis of this report is to show if damper characteristics can be generated by looking at road load data. To start with there are two sets of data gathered for this investigation. The first set of data was gathered by using one of the dampers from the MCR with the damper dynamometer. The second set of data was provided by Rodger Dowden, which is the live road load data of the MCR.

The aim of this project is to look at real track data and how tyre load is affected by damper characteristics. Understanding this will help us detect the effects on tyre/wheel load that comes when you change the damping characteristics. Thus helping us predict the path the car will follow when going around a corner for different damper settings.

Suspension components

Figure 1 below shows the main components in the suspension which play a major part in a spring/damper system. This vertical load is transmitted from the tyres to the spring/damper assembly. The vertical load exerted on the tyres, creates a forces through to the pushrod which is turned through 90° by the rocker and is supported by the springs. The damper is what prevents the oscillation and vibration of the spring. (Seward, 2014)

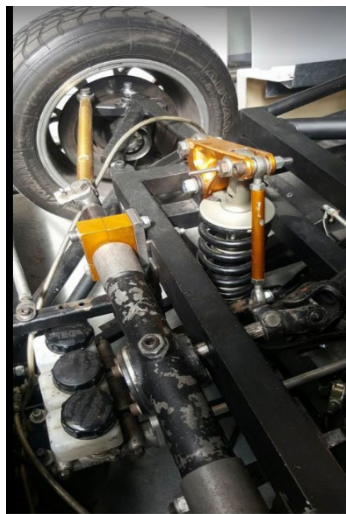


Figure 1 MCR suspension

Pushrod

A pushrod for the suspension is a strut from the lower wishbone which runs diagonally to the upper edge of the suspension, pushing the rocker to activate the spring/damper system. A pull rod is basically just the opposite; the strut runs diagonally to the lower edge of the suspension. The push rod will flex when the wheel is in bump, this is when the wheel is travelling up. (Formula1-dictionary.net, 2018)

Rocker

Also referred to as the bell-crank, which consists of an “L” shaped pivot where the pivot is where the two arms meet. Both of these arms are the same length, forming a 90° angle.

Spring

The energy dissipated by the dampers is the energy stored by the springs. The main purpose of the spring is to compress and store the energy from the vertical motion of the wheel from the contact patch force. There are many different types of springs used in the automotive and motorsport industry; coil springs, leaf springs and torsion springs. (YouTube, 2017)

The springs used on the MCR is a coil spring. A coil spring is a helical compression spring, which is designed to resist compression. The ability of the coil spring to resist compression can be affected by the number of coils, mean diameter of the coil spring, spring material and the thickness of the coils. (Rowley, 2004)

However, no matter what type of spring is used they all obey Hooke's law:

$$F = kx$$

F = force in Newton's (N)

k = spring constant (N/m)

x = extension in metres (m)

Springs produce force with respect to displacement whereas dampers produce force with respect to velocity.

Dampers

A damper consists of a cylinder with pressurised oil and a piston with several holes to allow the oil to pass through it. Dampers are a complex mechanical component that help dissipate the energy smoothly to provide a comfortable ride and better handling. Dampers produce a force with respect to velocity, hence they control the rate of displacement.

Types of dampers

Over years of development, there have been a range of dampers constructed. The three main types of damper used in the automotive industry is the Twin Tube damper, the Monotube Damper and the Monotube Damper with compression piston. These are three basic constructions of dampers used, as you can see in figure 2 below:

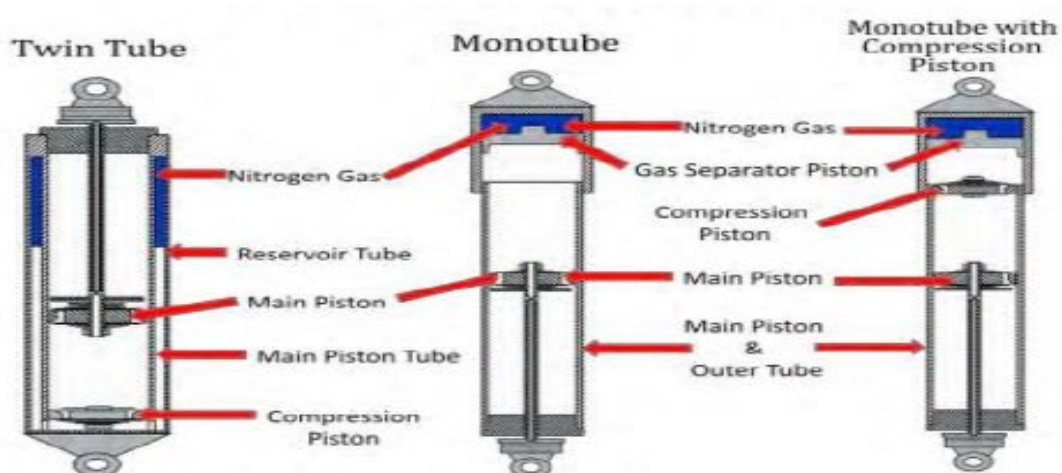


Figure 2 Types of damper constructions (Kasprzak, Understanding Your Dampers: A Guide from Jim Kasprzak, 2011)

Damper construction

Even though there are different construction available for dampers, they are all still similar in design.

Main piston:

The Main piston produces most of the damping forces and also includes primary valving components.

Compression piston:

The compression piston produces a compressive force based on the rod displacement. This provides a balance in pressure during compression to prevent cavitation. Monotube dampers do not have compression piston.

Gas separator piston:

This stops the gas from mixing with oil.

Main piston tube:

This is the area where the main piston operates. The monotube damper it is also the outer tube.

Reservoir tube:

This creates an area for the extra oil and gas pressure, this is represented as the outer tube for the twin tube damper in Figure 2. (Kasprzak, J, 2011)

Monotube Damper

Penske Monotube Dampers are used in the MCR for this investigation. This design of damper, has no compression piston and has a separate oil chamber as shown in Figure 4.

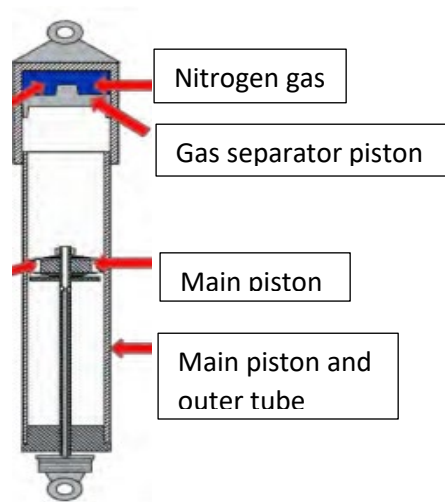


Figure 3 Monotube Damper

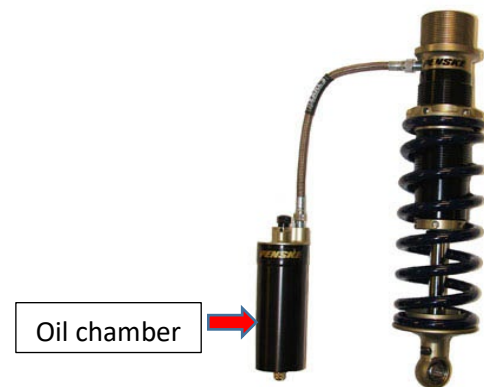


Figure 4 Monotube Damper with separate oil chamber

There are advantages and disadvantages of using a monotube damper:

Advantages

- The oil is separated from the gas.
- Monotube dampers generally have a higher quality construction than twin tube dampers.

Disadvantages

- Due to there being no compression piston, means a higher gas pressure need to be maintained to prevent cavitation.
- Nitrogen gas chamber within the damper can make the damper longer (Kasprzak, J. 2011).

Spring-Mass Damper system

This spring-mass damper system is also known as a shock absorber or coil overs. The shocker absorber helps damping the movement of both the unsprung mass and the sprung mass of the vehicle. Figure 5 below shows a model of one corner of the vehicle.

“Dampers do not change the amount of weight transfer, only the time it takes to transfer this weight” (Kasprzak, J. 2011).

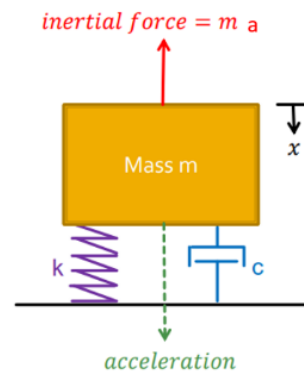


Figure 5 spring mass damper system
(Tim Tudor, 2018)

In a damped spring mass system, there are three main forces acting on the sprung mass:

- Inertia force $F = ma$
- Damping force $F = cv$
- Spring force $F = kx$

Where,

m = Unsprung mass
 c = damping coefficient
 k = spring rate
 x = displacement

The Inertia force generated is from the unsprung mass of the vehicle. Having a lower unsprung mass plays a key role in improving handling. Lowering the unsprung mass means less work is done by the spring and damper to maintain contact with the track. Having a higher unsprung mass means having a higher inertia. Have a larger inertia means more work is done by the spring and damper to maintain contact with the ground. Large inertia leads to difficulty accelerating and decelerating, due to the suspension not maintain a consistent tire load. (Formula1-dictionary.net, 2018)

The three forces can be further expressed mathematically as shown below:

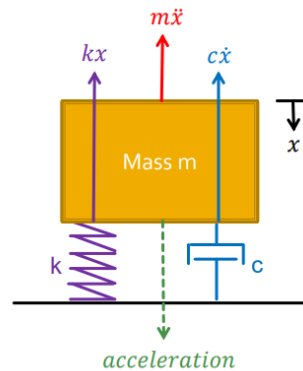


Figure 6 spring mass damper system (Tim Tudor, 2018)

- Inertia force $F = m \left(\frac{d^2x}{dt^2} \right)$
- Damping force $F = C \left(\frac{dx}{dt} \right)$
- Spring force $F = kx$

Where,

x = displacement

$\dot{x} = \frac{dx}{dt}$ = velocity

$\ddot{x} = \frac{d^2x}{dt^2} = \frac{dv}{dt}$ = acceleration

As mentioned before springs produce a force depending on the masses displacement (x), whereas damper produce a force depending on the masses velocity ($\dot{x}$).

Resolving these three forces vertically, you can derive the equation below for the spring mass damper setting:

$$ma + cv + kx = 0$$

Second order differential equation,

$$m\ddot{x} + c\dot{x} + kx = 0$$

Second order differential equation which can also be written as,

$$m \frac{d^2x}{dt^2} + c \frac{dx}{dt} + kx = 0$$

(T.Tudor. spring mass damper system. 2018)

Figure 7 below shows the true representation of a Spring-mass damper system for the vehicle. As it shows the tyres which are also represented as a spring/damper system. This is due to material property of the tyre. It is known that approximately 2% damping is due to the tyre

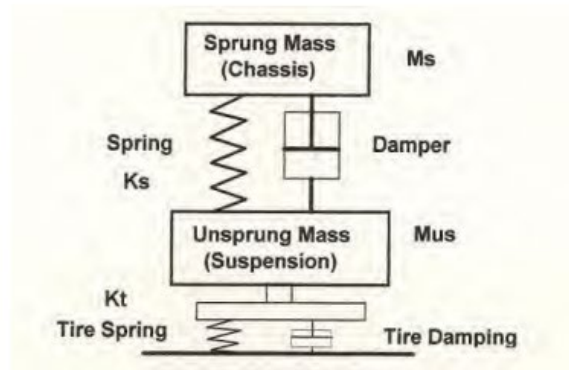


Figure 7 Spring-mass damper system (Kasprzak, J. 2011).

The unsprung mass consists of the tyres, wheels, brake callipers, brake discs and parts of the suspensions arms and suspension links. The sprung mass consists of everything supported by the springs, such as the driver, chassis and bodywork.

Wheel centre rate

Wheel rate is basically the spring rate measured at the wheel itself, which is measured by the equation below:

$$K_w = K_s * MR^2$$

The motion ratios are provided for the MCR:

MR front = 0.81

MR rear = 0.688

Tyre rate

For all the calculations done an assumption was made for the tyre rate. As this is the value for tyre rate used by the MCR Team.

$$\text{Assume } K_T = 256 \text{ Nmm}^{-1}$$

Hard or soft suspension

Equation below shows the motion of the sprung mass in relation to the ground. For which K_T and K_W are two springs which are in series and the joint stiffness is shown as:

$$\frac{1}{K_R} = \frac{1}{K_T} + \frac{1}{K_W}$$

K_R = ride rate (N/mm)

K_W = wheel rate (N/mm)

K_T = Tyre spring rate (N/mm)

For further calculation's make sure that the ride rates are converted from N/mm to N/m.

Assuming SHM (simple harmonic motion), the ride frequency (f_s) due to the vibration of the sprung mass is shown as:

$$f_s = \frac{1}{2\pi} \sqrt{\frac{K_R}{m_s}}$$

f_s = sprung mass natural frequency (Hz)

K_R = ride rate (N/m)

m_s = sprung mass (kg)

Which can also be written as:

$$f_s = \frac{1}{2\pi} \sqrt{\frac{(K_W * K_T)/(K_W + K_T)}{m_s}}$$

f_s = sprung mass natural frequency

m_s = sprung mass (kg)

K_W = wheel rate (N/m)

K_T = Tyre spring rate (N/m)

Table 1 shows the sprung mass ride frequencies for each corner of the MCR:

Table 1 sprung mass natural ride frequencies

Front Left	2.84 Hz
Front Right	3.01 Hz
Rear Left	2.03 Hz
Rear Right	2.17 Hz

It is mentioned later, ranges of values designers aim for the sprung mass natural ride frequencies. Especially when the vehicle having has an aerodynamic downforce acting on it, the ride frequencies are rarely achieved due to ground clearance issues. (Seward, 2014)

And the ride frequency for the unsprung mass is determined by;

$$f_{us} = \frac{1}{2\pi} \sqrt{\frac{K_W + K_T}{m_{us}}}$$

f_{us} = Unsprung mass natural frequency (Hz)

m_{us} = Unsprung mass (kg)

K_W = wheel rate (N/m)

K_T = Tyre spring rate (N/m)

Table 2 shows the unsprung mass natural ride frequencies for each corner of the MCR:

Table 2 unsprung mass natural ride frequencies

Front Left	9.03 Hz
Front Right	9.06 Hz
Rear Left	8.19 Hz
Rear Right	8.19 Hz

The ride frequency is the undamped natural frequency of the sprung mass in ride. When looking at road vehicle usually the higher the frequency the stiffer the ride. Below shows a listing of different ride frequencies of sprung and unsprung masses for different classes of vehicles:

Sprung mass natural ride frequencies

- 1-2 Hz for user friendly road vehicles
- 1.5-4 Hz for NASCAR cup cars
- 5-7 Hz for race cars with high downforce

Unsprung mass natural ride frequencies

- 10-12 Hz for user friendly road vehicles
- 15-17 Hz for NASCAR cup cars
- 23-27 Hz for race cars with high downforce

(Kasprzak, Understanding Your Dampers: A Guide from Jim Kasprzak, 2011)

Low ride frequency have a soft suspension with the lots of mechanical grip. "To maximise mechanical grip the suspension should be as soft as possible". However this does not seem to be the case as race cars have higher frequencies and hard suspension so that the vehicle has a lower ground clearance. Which also lowers the centre of mass which causes larger weight transfer between the wheels, preventing the wheel from lofting during braking, cornering and acceleration. (OptimumG, n.d)

The natural ride frequency does make a difference on the feel of the vehicle:

- 0.5-0.7 Hz - motion sickness
- 0.7-1 Hz - wallow motion
- 1-1.5 Hz – comfortable (user friendly road vehicles)
- 2-2.5 Hz – harsh ride
- 2.5-5 – very harsh (high downforce vehicles) (Tim Tudor, spring mass damper system,2018)

Yes it is useful to calculate ride frequencies to determine initial ride, but it all eventually comes down to the spring rates and damping coefficients. When looking at going fast around corners as they help maintain mechanical grip.

Damping ratio

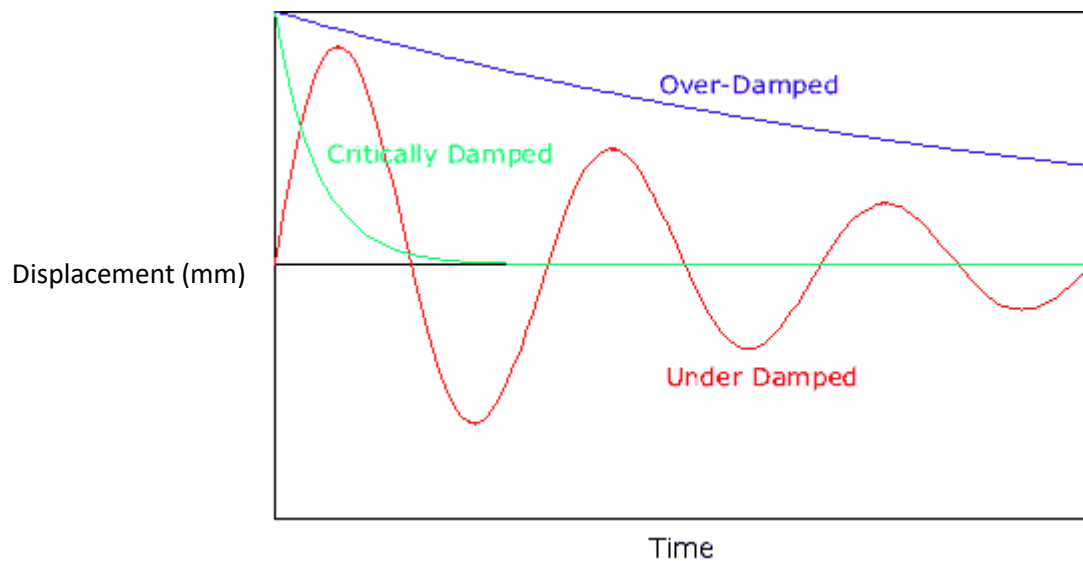


Figure 8 Effects of damping (Anon, 2018)

Theoretically if a mass supported by a spring is displaced, it will oscillate indefinitely at a constant frequency and amplitude. However in real life overtime the oscillations will eventual decay over a finite amount of time due to natural damping from air resistance and internal friction. Which in figure 7 is represented as the red line. A damper is used to modify this behaviour.

How much damping is required? “Just enough” You want to achieve the right amount of damping hence both too much and too little will limit your vehicle ability to corner. The damping coefficient is given the symbol (C), it is the damping force at a velocity of 1m/s. Which is basically the gradient of the force-velocity graph of a damper, this is the measure of the amount the damper damps movement. Critical damping is referred to as the mass returning back to its steady state position with no over shoot, which is represented as the green line in figure 7. The equation is shown below. This is the damping required to stop the oscillation as quick as possible but critical damping in a vehicle does not work as it is too slow from the vehicles perspective. (Seward, 2014)

$$C_{CR} = 2\sqrt{K_s * M}$$

C_{CR} = critical damping coefficient

K_s = system spring rate

M = system mass

In a system if the damping coefficient is larger than the critical, then it is over-damped and will take longer to return to steady state position. This is represented as the blue line in figure 7. Vice-versa if the damping coefficient is less than the critical then the system is under-damped and oscillations around the steady-state position will take place.

The amount of damping required doesn't fit any of the examples above, therefore damping ratios are worked out. This is the damping coefficient (C) in relation to critical damping coefficient (C_{CR}) as shown in the equation below:

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

$\xi = 1$ Critically Damped

$\xi < 1$ Under Damped

$\xi > 1$ Over Damped

A vehicle with both too little and too much damping will have an effect on the performance of the vehicle. As a vehicle with too much damping, therefore being over damped will overall reduce the grip produced by the tyres as there would be a tyre load variation. The ride is harsh as when accelerating and hitting a bump there is very little grip and the tyres feel as they are not maintaining contact with the track.

For a vehicle with too little damping, this is when the vehicle is under damped. The ride will feel very bouncy as the vehicle hits a bump also there would be poor traction over bumps during high speeds. (Motorcyclist, 2018)

The ratio of the damping coefficient (C), to critical damping coefficient (C_{CR}), is known as the damping ratio. This value is what is used to determine if the vehicle has too much damping or too little damping. Usually for user friendly road vehicles you have to compromise between performance and comfort. This value is a little higher for race cars as comfort is not important. The average damping ratios are listed below:

- 0.25 for user friendly road vehicles
- 0.65-0.7 for race cars

(Seward, 2014)

Damper dynamometer

Figure 9 below is the spa damper dynamometer, which is used for damper testing. The damper in this image is the rear right damper which was removed from the MCR itself. This piece of equipment is good for looking at dampers on their own, hence you get no interference with your results due to other forces acting through the damper. Damper dynamometer is used to check and validate results obtained from the live track data.

This piece of equipment is used worldwide by a majority of racing teams; Touring car, Formula one, NASCAR, Motorcycling, PPG Indy car and many more. (Design, 2017)

Setup

1. To set the damper up to the dynamometer you need to mount the damper to two points on dynamometer. The top of the damper is hard mounted to the dynamometer whereas the bottom of the damper is free to move. This allows the bottom of the dynamometer to apply a vertical force to the damper.
2. The hard mount can be adjusted for testing of other different sized dampers.

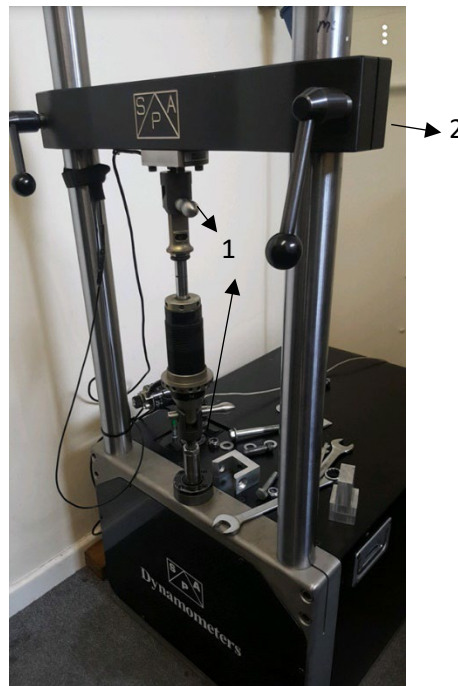


Figure 9 SPA Damper dynamometer

When testing a damper, a known velocity is applied through the bottom of the dynamometer to the damper as a sine wave. An external software records the Force, Displacement and Velocity of the damper. Two main types of graphs are extracted from the external software; force-displacement graph and a force-velocity graph. The damper dynamometer helps provide you with quick yet accurate graphs. These graphs can then be used to look at defects within the damper, valve openings can also be seen on the data. (Design, 2017)

Force-displacement

The graph below shows the force displacement graph produced for the rear right damper of the MCR on the damper dynamometer. The graph shows the damper piston travel against the forced it produces. This is an overlay of two different damping setting; soft and hard damper setting. Soft damping is represented via the colour blue, this is at slower speeds. Hard damping is represented as the colour orange and shows how the dampers works at higher speeds.

This graph itself can be broken down into 4 phases:

1. Phase 1 shows the acceleration of the piston through its bump phase.
2. Phase 2 shows the deceleration of the piston through its bump phase.
3. Phase 3 show the acceleration of the piston through the rebound phase.
4. Phase 4 shows the deceleration of the piston through the rebound phase.

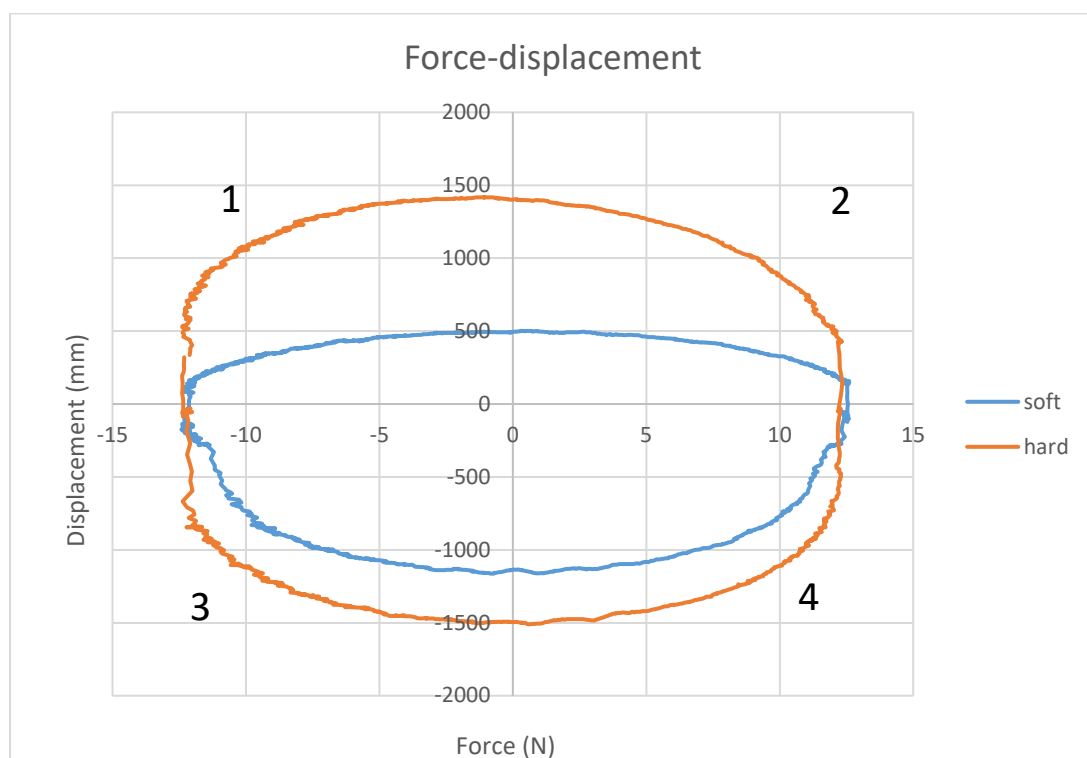


Figure 10 Force-displacement graph

Benefit of producing a force-displacement graph is that you are able spot defects with the damper. Force velocity graphs are also used to check if hysteresis levels are at an acceptable level. Hysteresis is the loss in damping due to the change in direction, this can be when the forces seen during deceleration are higher than the forces seen when accelerating.

As both shapes for soft and hard damping should be very similar, random irregularities seen in only one of the curves show a loss of energy. The effects of hysteresis can be seen in figure 10, section 3 and section 4 for soft damping, this is why there is a significant difference between the damping forces seen for acceleration and deceleration. (Nttires.com, 2018)

Force-velocity

Force-velocity curves have 2 distinct sections. You get a range of positive and negative values of force and velocity, where the positive values represent the forces seen by the damper during the bump phase. The negative values represent the forces seen by the damper in the rebound phase, the negative sign just implies that the direction of travel is in the opposite direction.

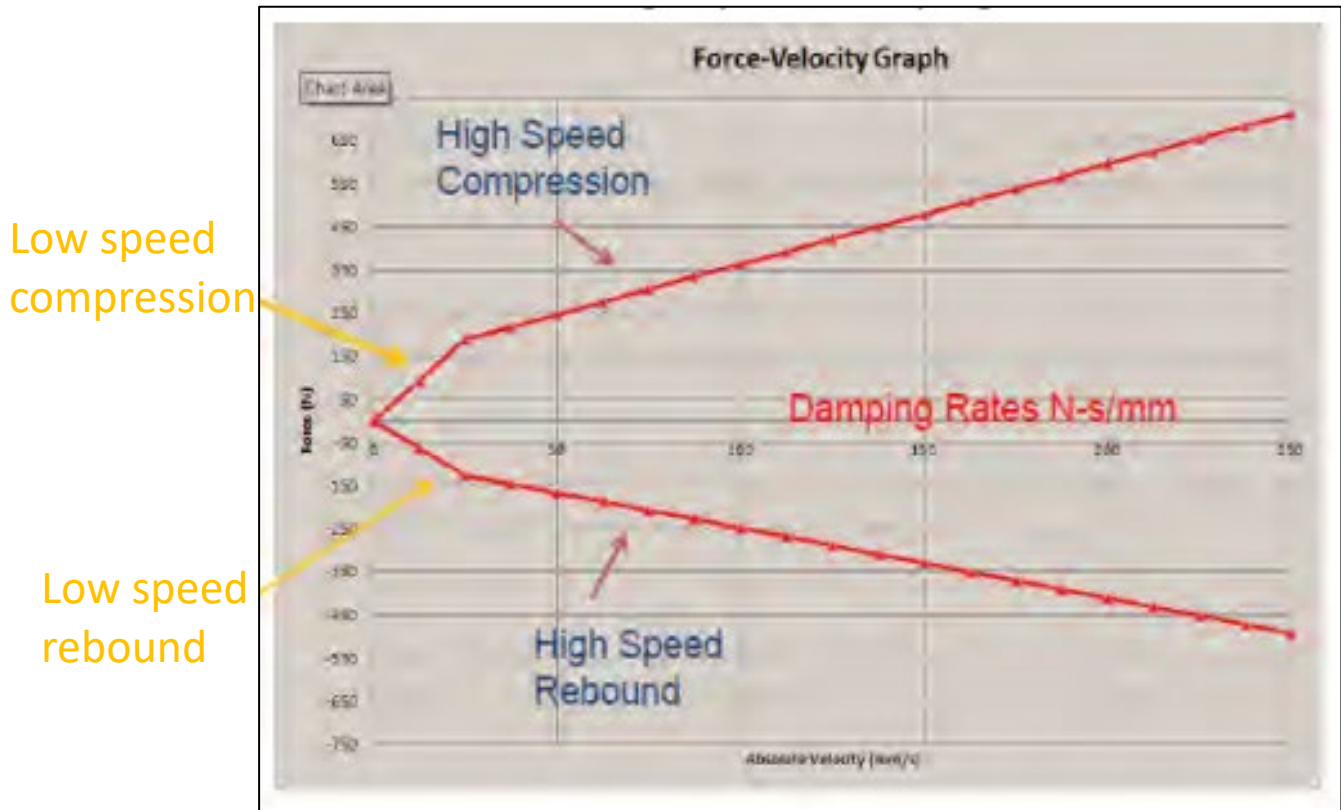


Figure 11 Force-velocity graph

There is a clear change in the gradient of the curve in both the bump and the rebound phases in figure 11. This key point on a force-velocity graph is known as the “knee” point. This is the drastic change in the gradient of the curve due to the compromise required to control grip. This is the point at which the damper switches from low speed bump/rebound to high speed bump/rebound. This is when the shims allow more oil flow through hence changing the rate of damping.

This graph displays the force seen by the damper against velocity, the velocity in this case represents the velocity of the piston in the damper. The maximum velocity occurs as the piston passes through its neutral point (Seward, 2014). This is for simple harmonic motion and the equation for maximum velocity is given below, where the amplitude is half the stroke. The stroke was provided as 25mm and all test done on the dynamometer were done at a frequency of 3 Hz.

$$V_{max} = 2\pi fA$$

Therefore

$$V_{max} = 2\pi * 3 * 12.5 = 235.6 \text{ mm/s}$$

This value calculated for Vmax is correct as this was the maximum velocity seen, when the damper was tested on the dynamometer.

By the use of an equation it is also possible to calculate the acceleration of the piston within the damper at any point whilst oscillating at any point. The equation used is listed below:

$$a = -(2\pi f)^2 x$$

Where,

x = the displacement of the piston = half the stroke = 12.5mm

Therefore,

$$a = -(2\pi * 3)^2 * 12.5 = 4441 \text{ mm/s}^2 = 4.441 \text{ m/s}^2$$

As the force-displacement curve is used to spot defects with the damper, the force-velocity graph is more widely used and can be used to determine damping setup requirements and predict damper performance. Testing different multiple damper setting from hard to soft would help gain a better understanding as it would visually show how each adjustment made to the damper changes the damping curve. Unfortunately the data collected from live track data was from one singular soft damper setting and one singular hard damper setting.

Development of the Force-velocity graph from the Dynamometer

Figure 12 represents the graph produced by the dynamometer when transported to excel.

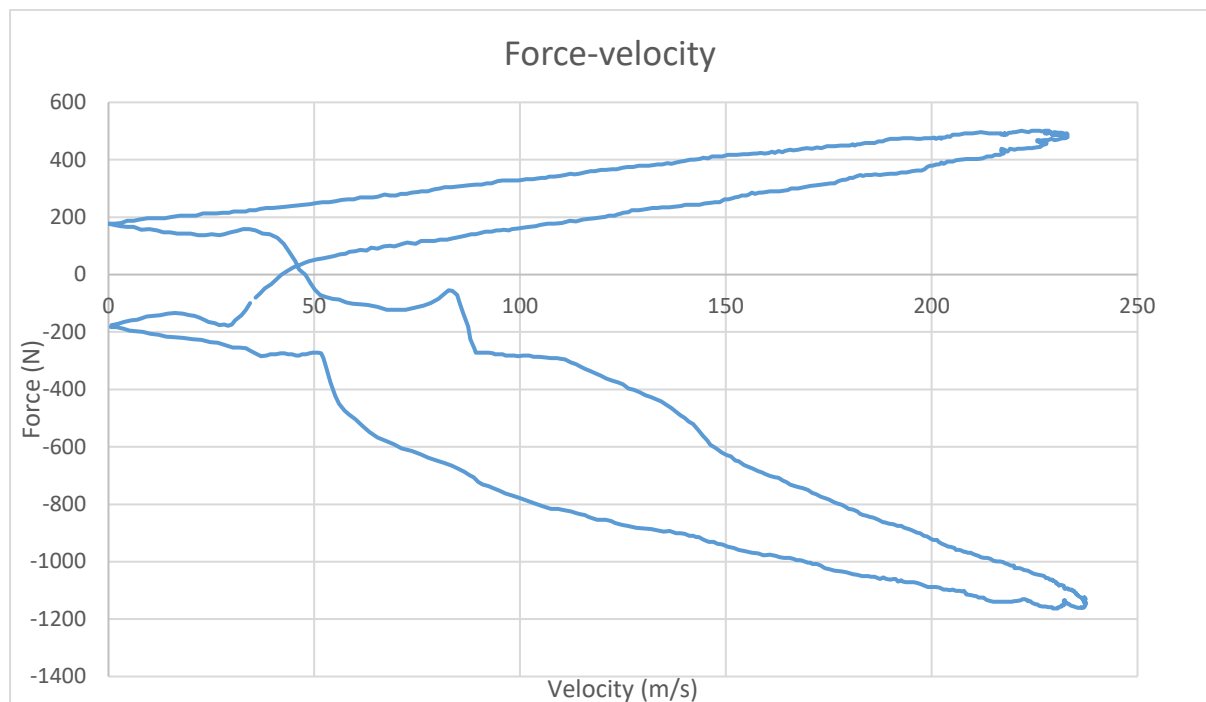


Figure 12 Force-velocity graph

Figure 12 shows the force-velocity graph produced which looked nothing similar to a force-velocity graph in figure 11. Hence the graph was further split into 2 separate graphs from which the average was taken and plotted.

Figure 12 was further broken down into the 2 graphs:

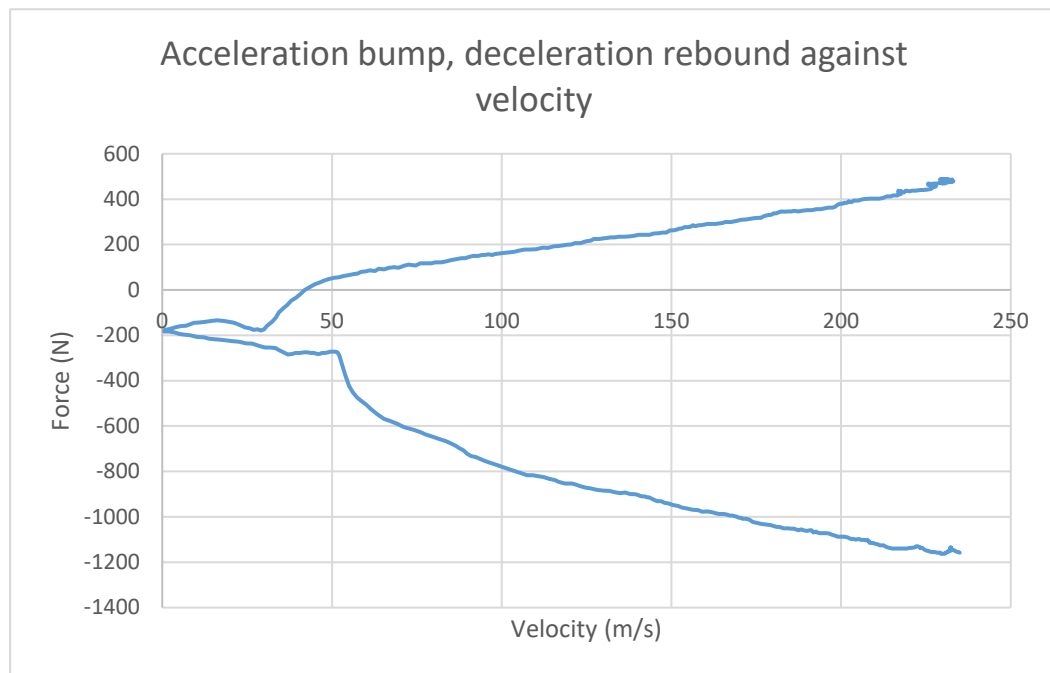


Figure 13 Acceleration bump, deceleration rebound against velocity

Figure 13 represents the acceleration of the piston inside the damper during the bump phase and the deceleration of the piston during the rebound phases.

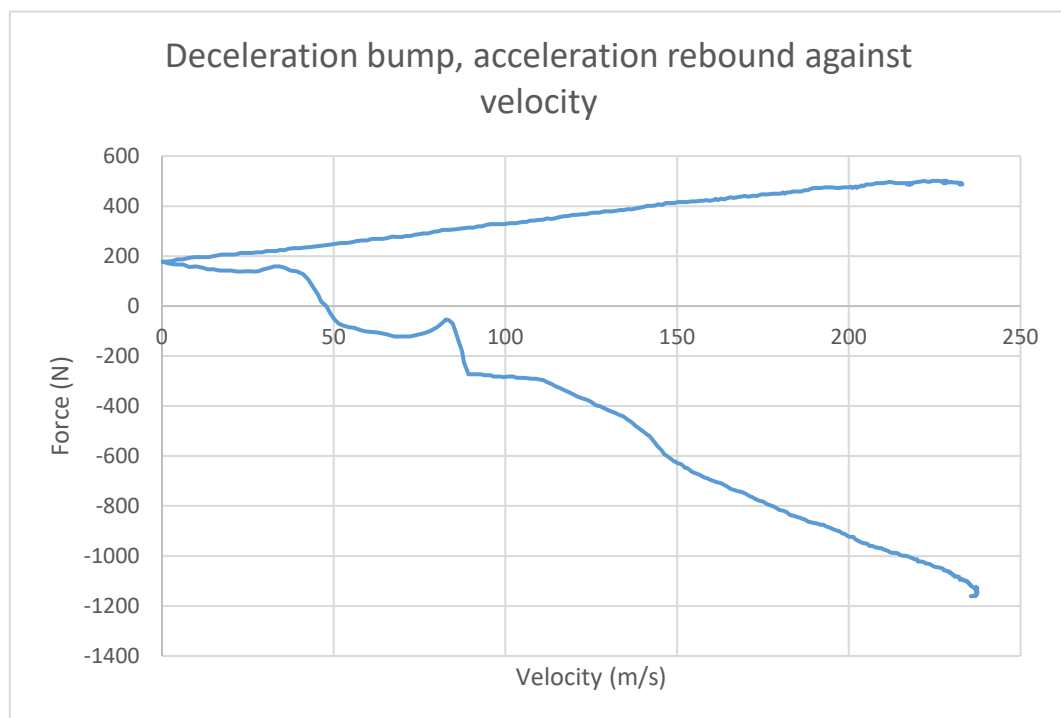


Figure 14 Deceleration bump, acceleration rebound against velocity

Figure 14 represents the deceleration during the bump phase and an acceleration of the piston inside the damper during the rebound phases.

Now the average force was taken from graph 13 and 14 and plotted. The graph below shows the force velocity graph for the rear right when the damper are soft.

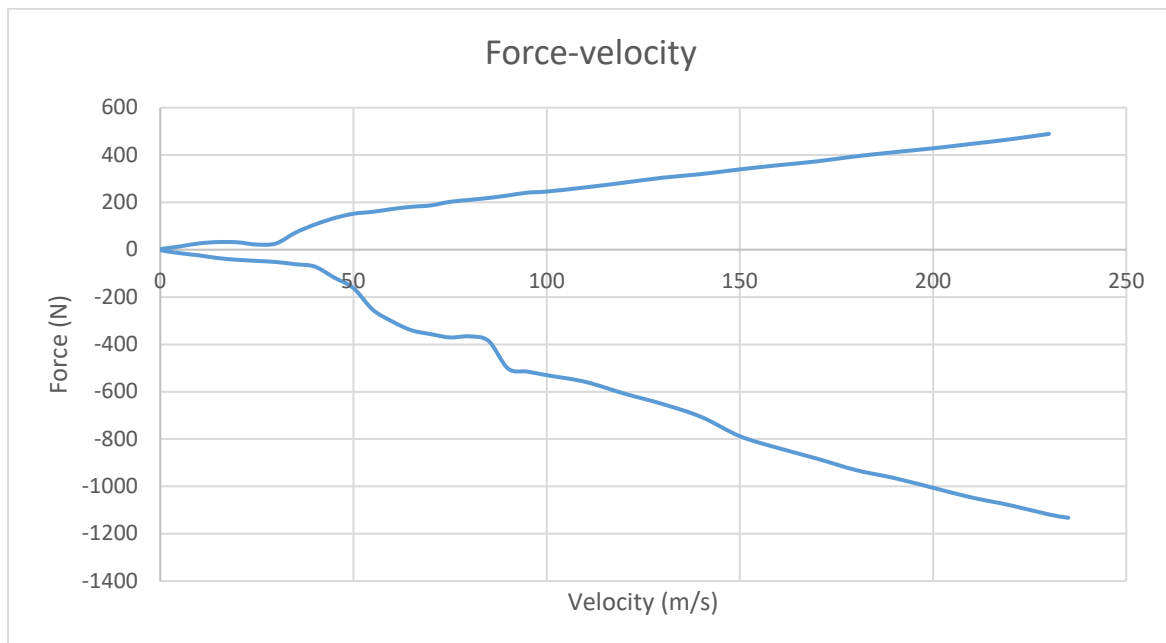


Figure 15 Force-velocity graph

Graph below show the actual force-velocity graph produced for the rear right damper of the MCR on the dynamometer. This graph below shows an overlay of two sets of data, soft and hard damping. Soft damping is represented as the blue curve and hard damping is represented as the orange curve. The change in gradient from low to high speeds is visible in both hard and soft damping.

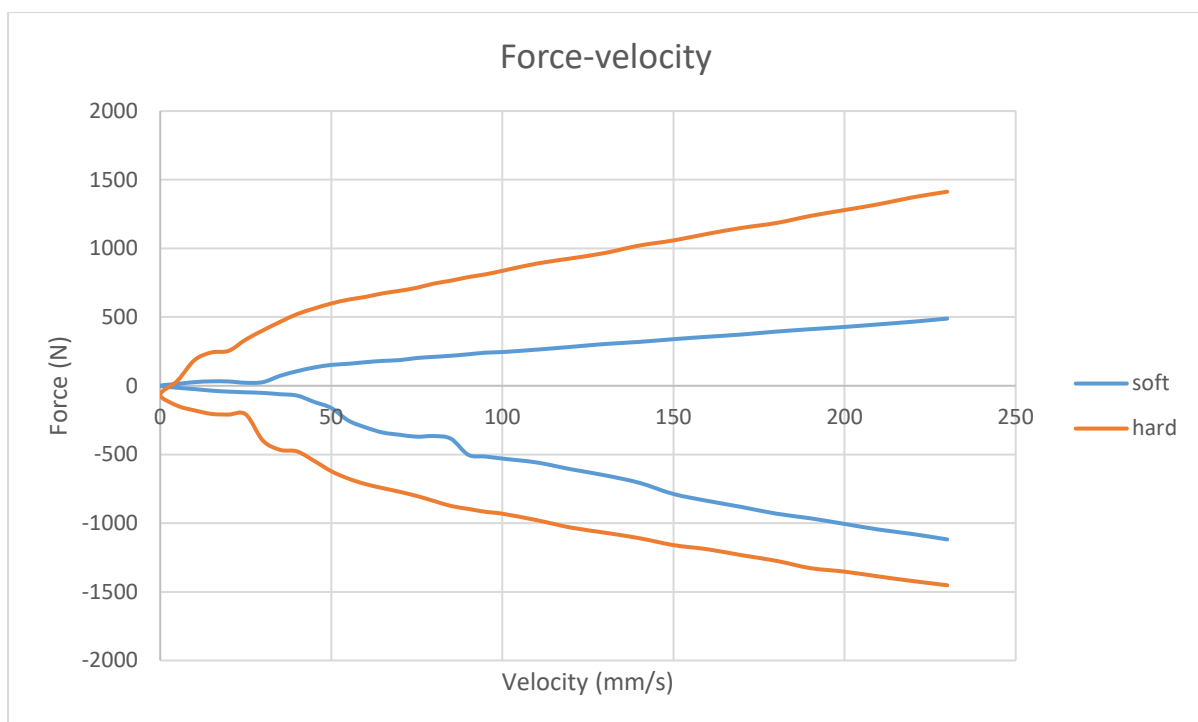


Figure 16 Force-velocity graph

Live track data

As mentioned before there were two sets data collected for this investigation. This second set of data represents the live track testing straight from the MCR. The track chosen for these results was Pembrey and the designated driver for the collection of this data was Roger Dowden. The main reason for the collection of this data was to investigate into the possibility of producing the damper curve as produced from the damper dynamometer.

It was important to make sure than there was a computer available which has a logger supported software installed for the analysis. The software used for this investigation was Motec. The track below shows the piece of track used for this investigation, which last a duration of 9.28 seconds and travels a total distance of 247 meters.

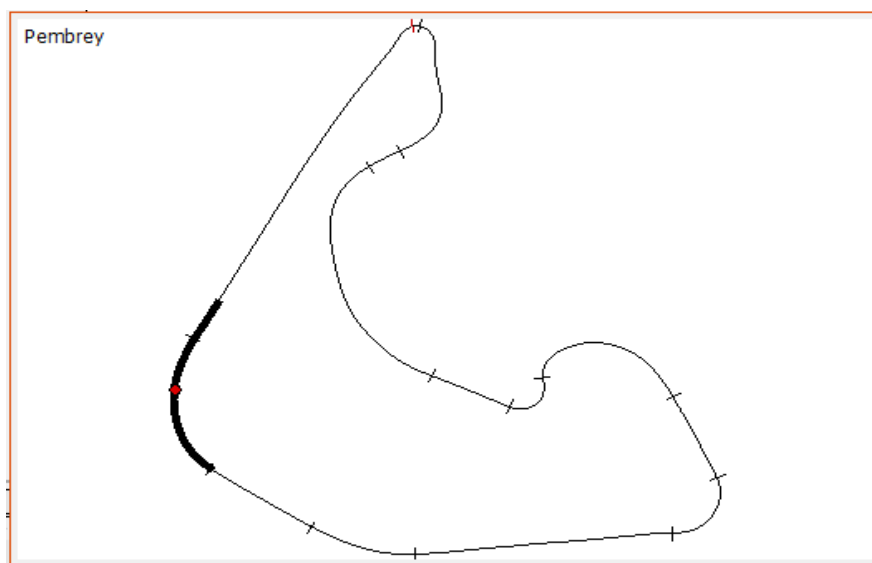


Figure 17 Pembrey circuit

Figure 18 below shows how data is represented on Motec, which in this case is not very useful to us hence why the data was exported to excel for further analysis and data manipulation. Image below show how this data was exported to excel.

Click file – export data

Which then, opens the time/distance time export setup and click the same settings as shows in the screenshot below and then click export.



Figure 18 Exporting Motec Channels to Excel

As soon as the data was successfully exported to excel a graph was produced with the suspension force against damper velocity as shown below in figure 19. The graph clearly looks very random and has no resemblance to a force-velocity graph produced by the damper dynamometer. There is no clear visual representation of bump and rebound forces due to there being no negative forces.

The forces seen by the graph below are exceptional higher than the forces seen by the damper, this is because of all the other forces which are transferred through the pushrod from where these forces are being recorded to which we refer as 'Noise'.

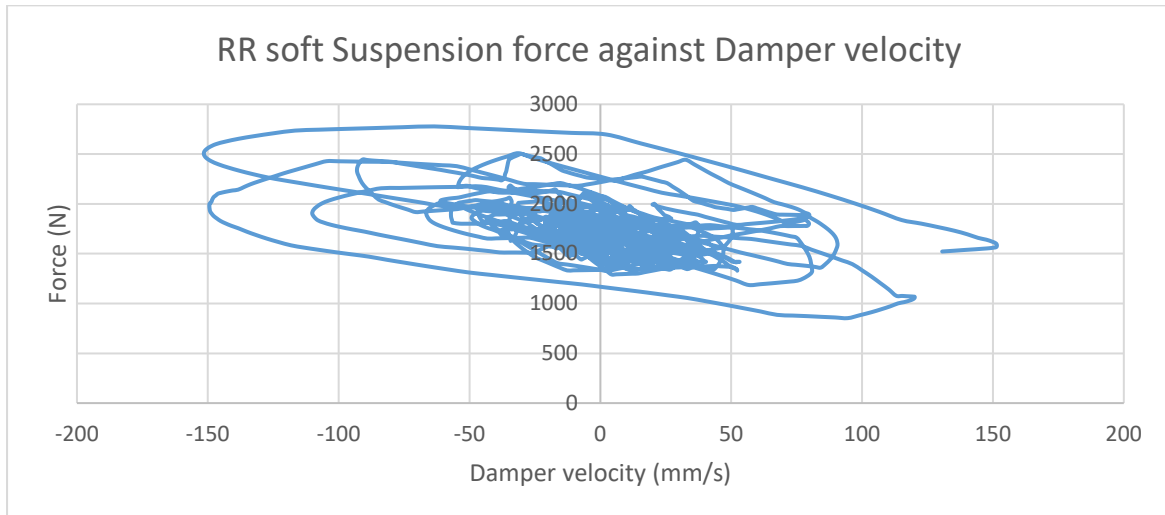


Figure 19 Suspension force against Damper velocity

This data was now arranged by using the sort function in excel. All the damper velocity were sorted by values from the smallest to largest whilst the respective suspension force was still kept with its original damper velocities.

The graph shows multiple readings of the forces generated at same velocities, this is because live track data will never show a repetitive data as it would on a damper dynamometer. As the damper experiences different rates of changes in velocity and acceleration of the piston within the damper. Whereas the data collected from the dynamometer produces a sinusoidal pattern hence the damper experiences the change in velocity and acceleration of the piston which does not truly represent how the damper works around the track.

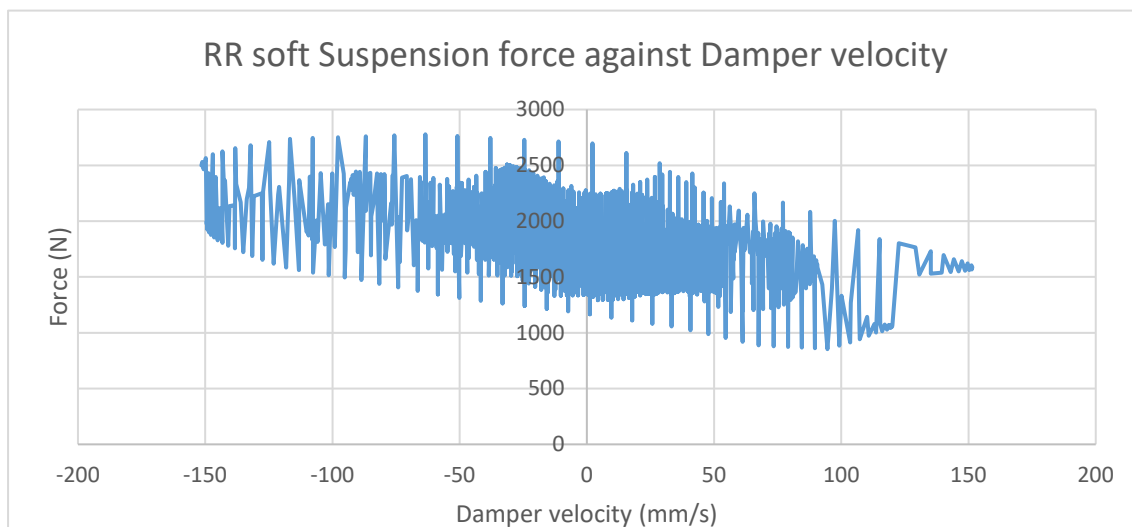


Figure 20 Suspension force against Damper velocity

The average of suspension forces was plotted against a range of damper velocities. Now for the first time the graph showed a slight trend, in figure 21 below you see that as the velocity increases you also see an increase in the suspension forces. In this graph the bump and rebound sections are now clearly visible but still no negative forces are seen by the suspension.

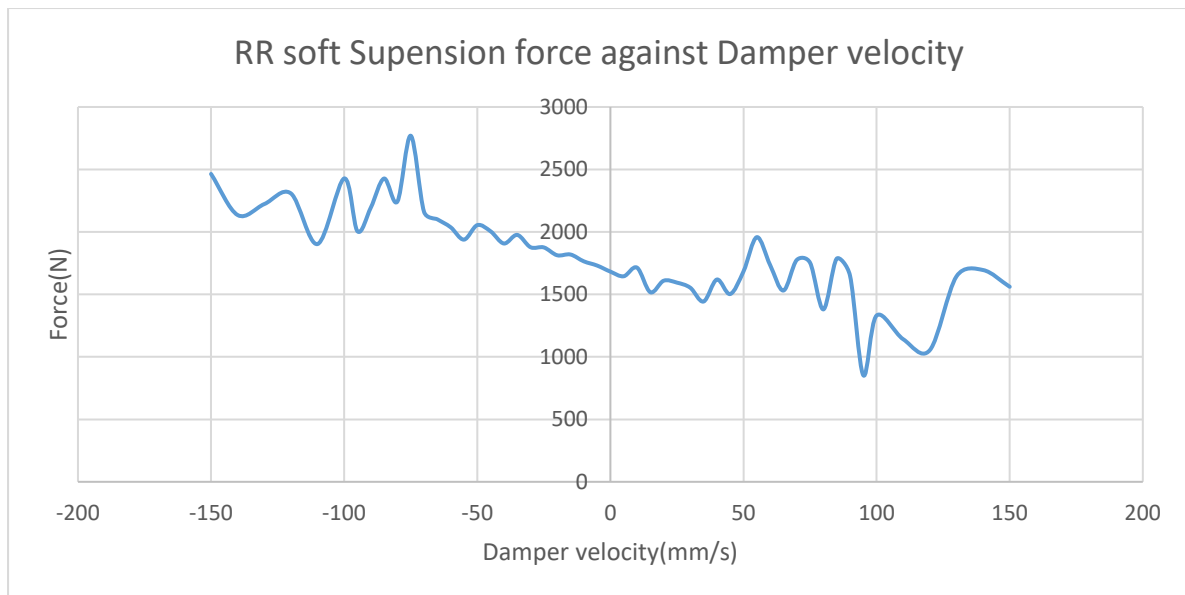


Figure 21 Suspension force against Damper velocity

Noise cancelling

For the basis of this investigation, the force-velocity graphs produced from the live track data have been the suspension force plotted against damper velocity. Damping force is only a proportion of the suspension force recorded by the data logger attached to the pushrod of the vehicle. To be able to see just the forces seen by the damper requires an understanding of all the other forces acting through the pushrod of the MCR when the vehicle is accelerating, braking and cornering. All the other forces acting through the pushrod are referred to as “noise”.

Subtracting Spring force

To be able to extract the spring force from the suspension force, will provide a higher accuracy of the forces acting through the damper itself. The equation used to work out spring force is:

$$F = kx$$

The spring rates to all for springs was provided:

Table 3 Spring rates

FL Spring rate	71.388 N/mm
FR Spring rate	72.141 N/mm
RL Spring rate	80.196 N/mm
RR Spring rate	80.318 N/mm

Whereas the displacement of the spring was not, Motec does provide a channel for each individual damper position for the 9 second track chosen. The damper position is used to represent the spring position as well.

The spring displacement is the damper position from which the static damper position is subtracted. The image below show the static damper positions for all four corners when the vehicle is stationary.

$$\text{Spring displacement (x)} = \text{Damper position} - \text{Static damper position}$$

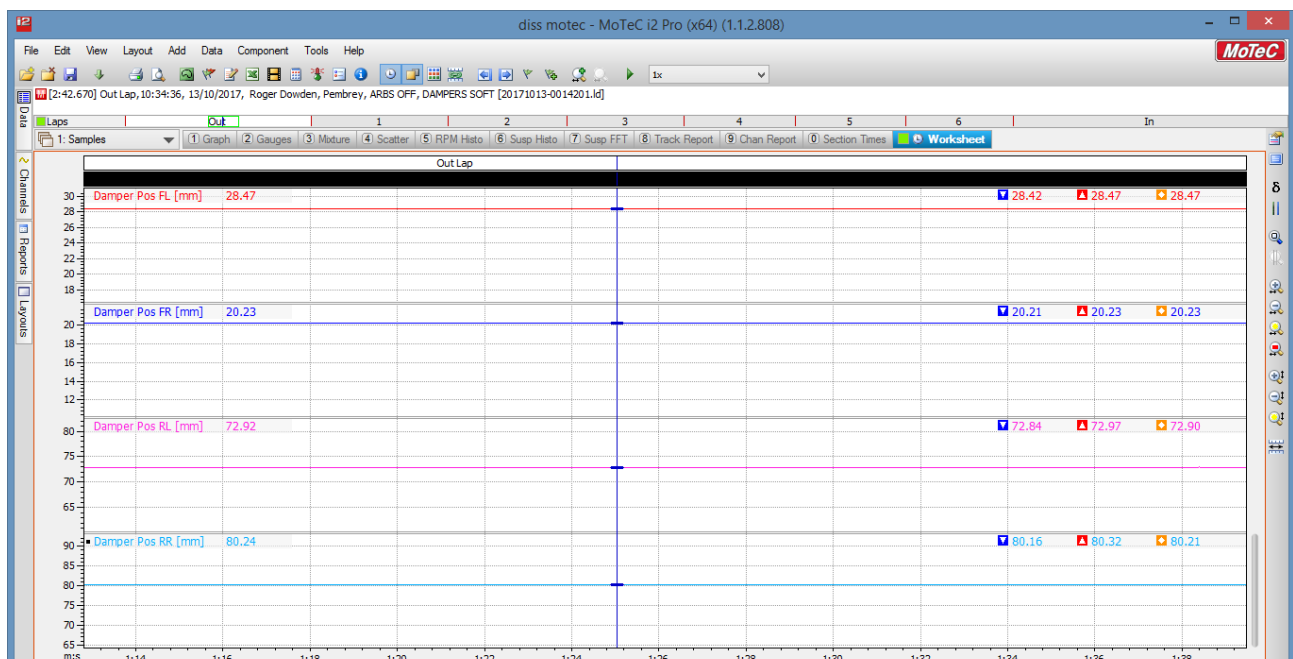


Figure 22 Static Damper position

Table 4 Static Damper position

FL Soft Static damper position	28.47 mm
FR Soft Static damper position	20.23 mm
RL Soft Static damper position	72.90 mm
RR Soft Static damper position	80.21 mm

The graph below represents the damping force against velocity after the spring force is subtracted, which brings us a step closer to looking at the forces acting just through the damper.

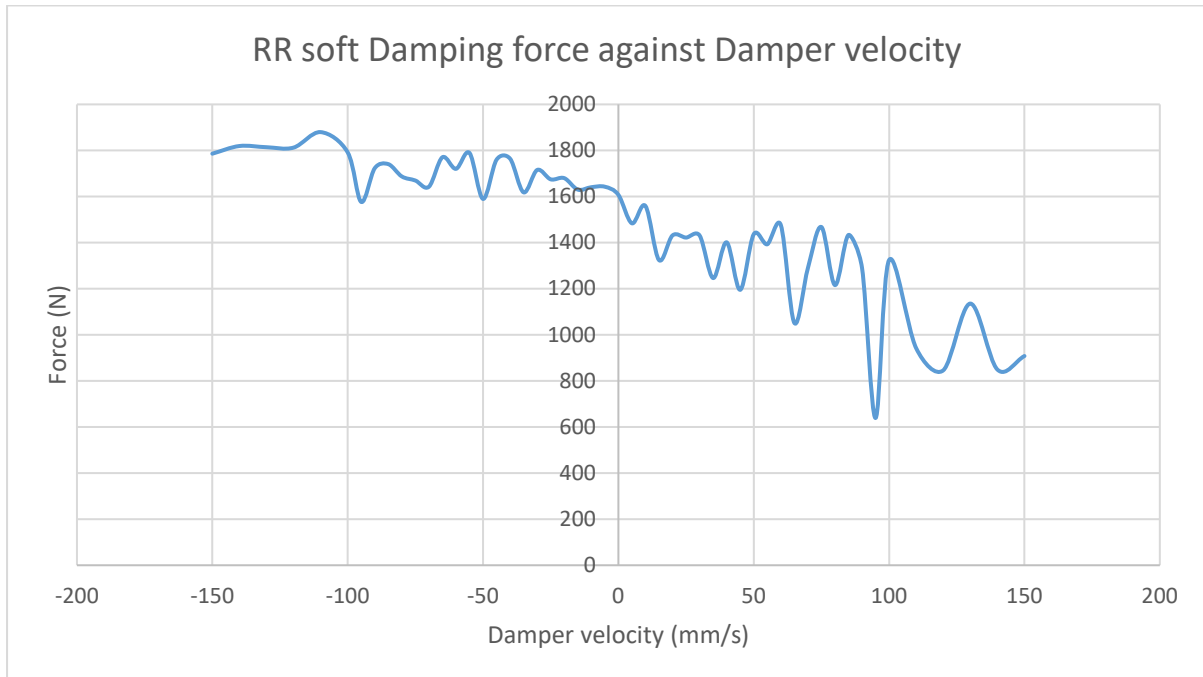


Figure 23 Damping Force against Damper velocity

Subtracting Inertia force

Another force seen through the suspension force is the inertia force. The equation used to work out the inertia force is given below:

$$F = ma$$

Where m is the unsprung mass of the MCR. The unsprung mass measured for all four corners is given below:

Table 5 Unsprung mass

FL Unsprung mass	22.5 kg
FR Unsprung mass	22.5 kg
RL Unsprung mass	24 kg
RR Unsprung mass	24 kg

The acceleration was worked out as the velocity of the damper was provided as a channel from Motec. The acceleration is basically the rate of change in velocity. The screenshot below shows the math channel created in Motec to work out the acceleration.

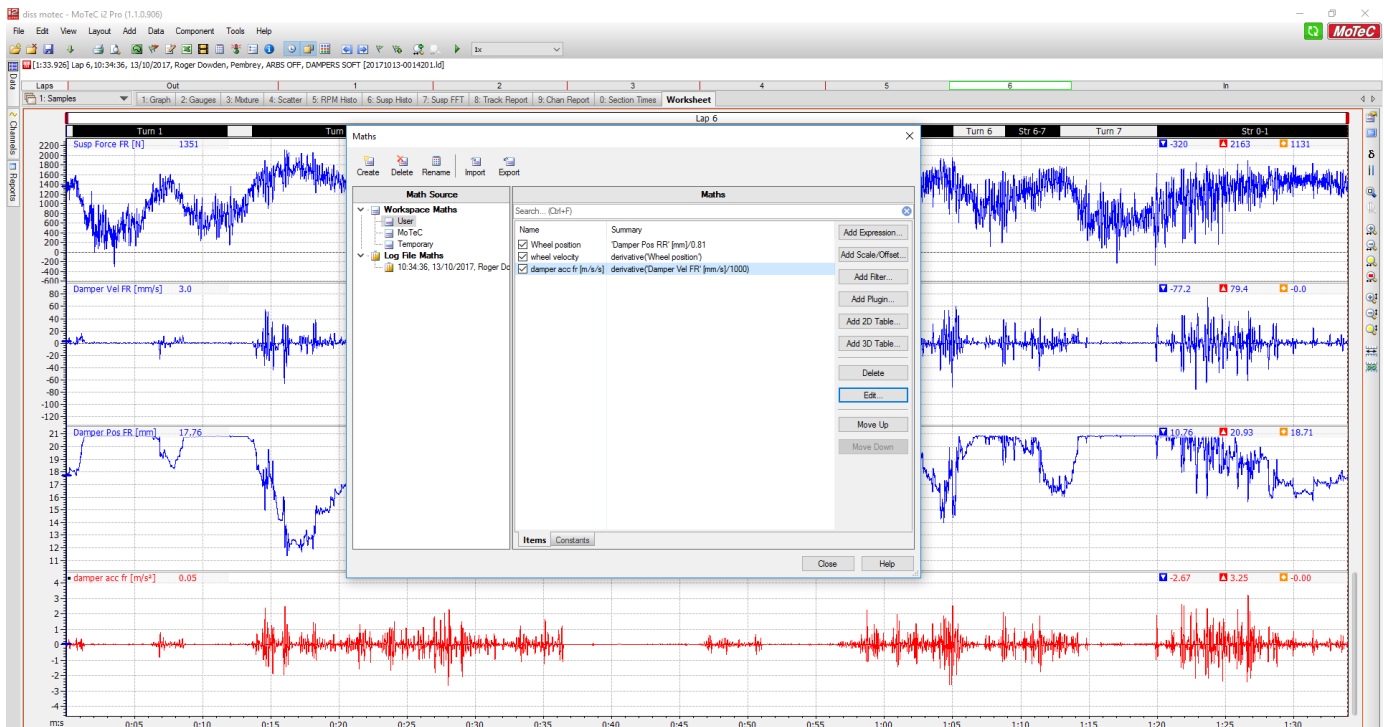


Figure 24 Acceleration Math channel

The math channel created to work out the damper acceleration in m/s^2 :

$$\text{Derivative} \left(\frac{\text{Damper velocity} \left(\frac{mm}{s} \right)}{1000} \right)$$

Figure 25 below shows the graph created after neglecting the inertia force acting through the pushrod. You will notice that subtracting the inertia forces from the graph has not significant effect on the shape of the curve. This is due to the inertia force created due to the acceleration of the piston within the damper is very small.

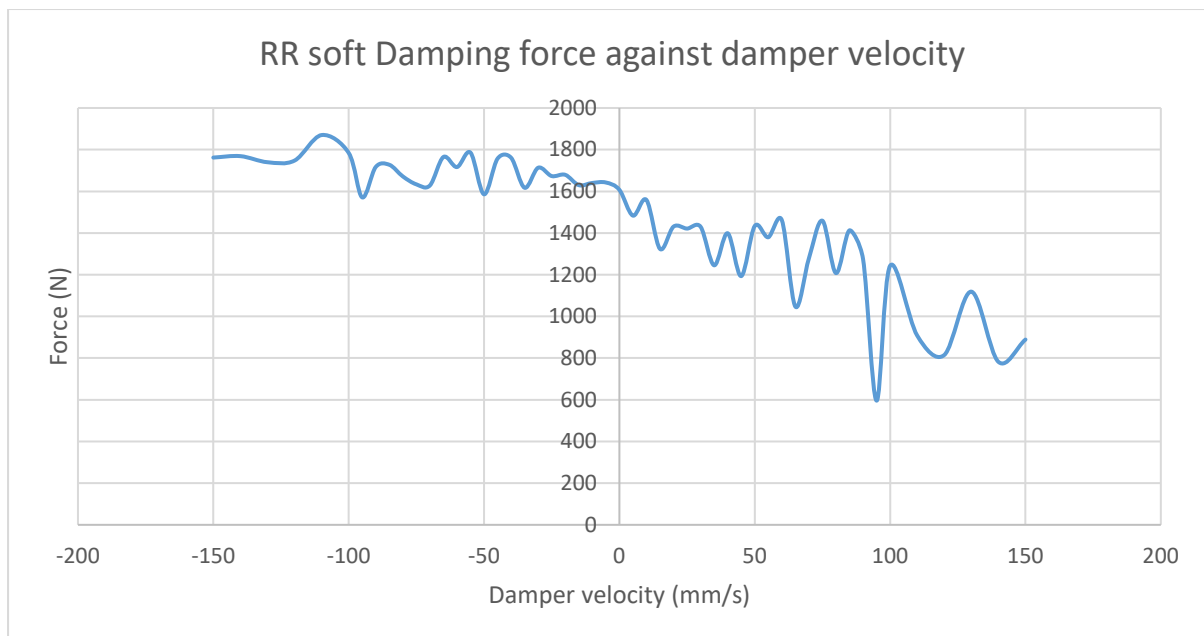


Figure 25 Damping Force against Damping force

Subtracting Static loads

When producing the suspension force against damper velocity was the forces seen by the suspension were higher than expected and did not pass through the zero at all. This is due to not removing the static loads from the suspension forces obtained. The static load acting on the vehicle is the preload on the suspension when the vehicle is stationary, therefore to be able to clearly look at all the other forces the static load need to be neglected from all recorded suspension forces.

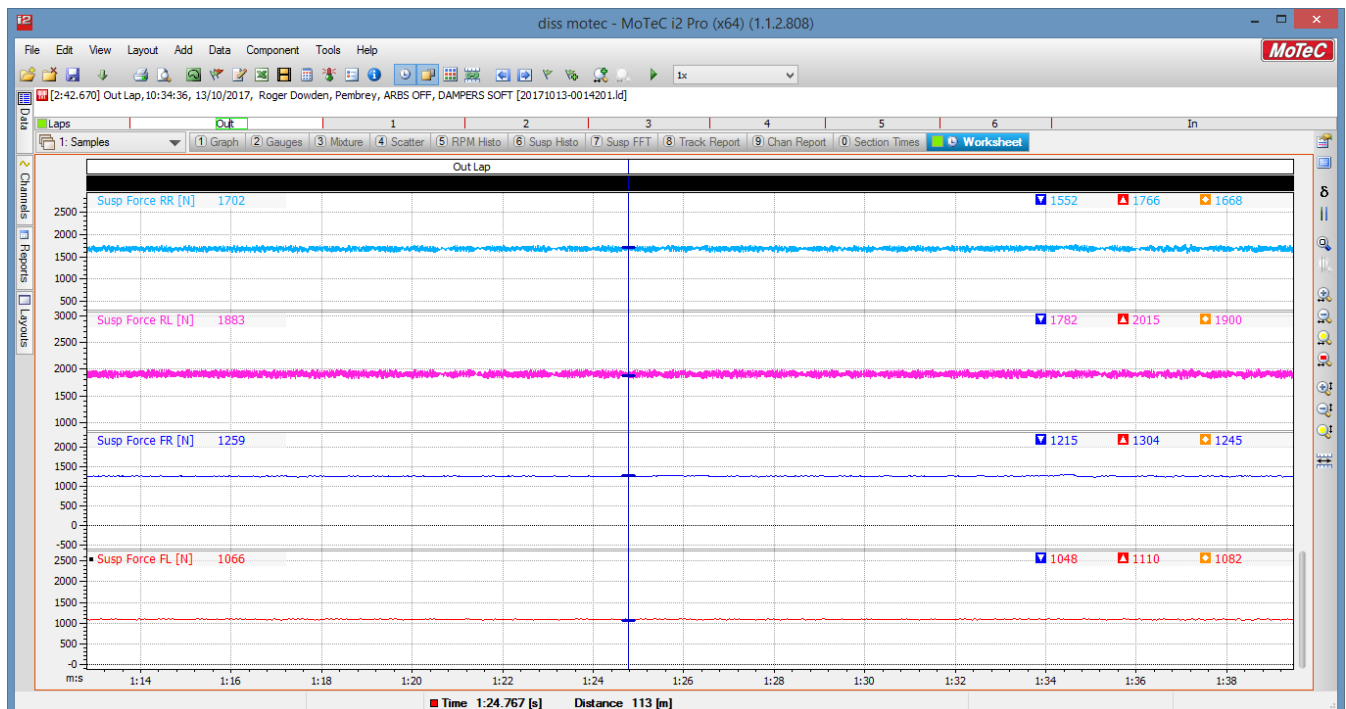


Figure 26 Suspension Static loads

Table 6 Suspension Static loads

FL Soft Static load	1082 N
FR Soft Static load	1245 N
RL Soft Static load	1900 N
RR Soft Static load	1668 N

Table 6 above shows the average static load measured for all four corners of the MCR with a soft damper setting when the vehicle is stationary. The static loads are higher for the rear of the vehicle this due to the rear of the vehicle have a higher mass.

The graph below shows damping force against damper velocity, once the static loads are subtracted from the initial suspension force.

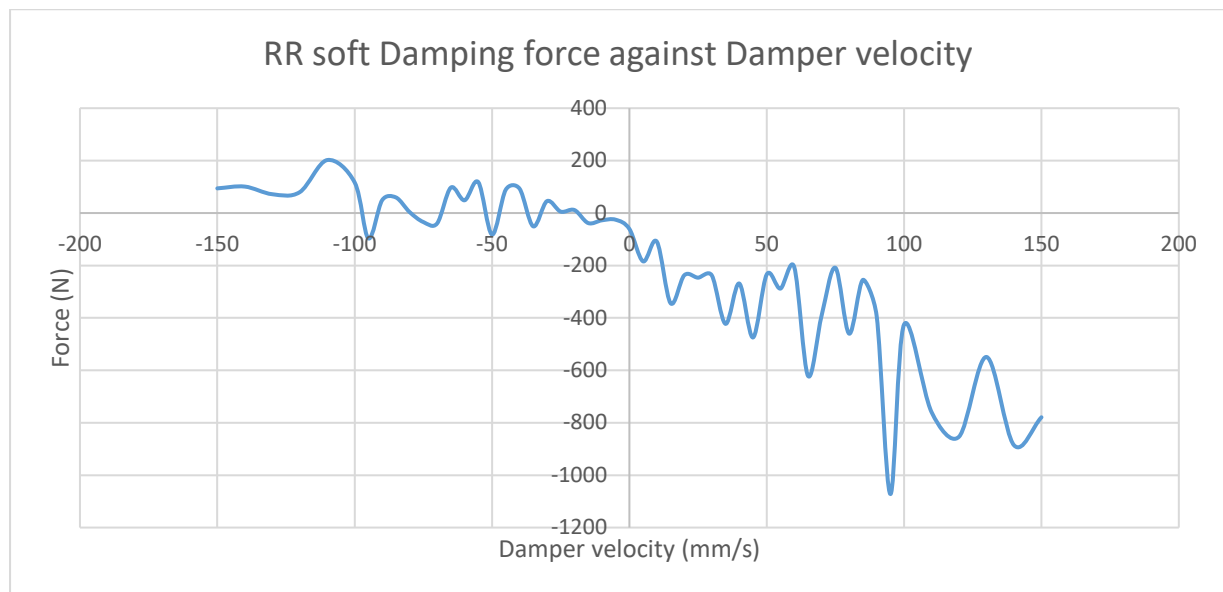


Figure 27 Damping force against Damper velocity

Now looking at this graph there are negative and positive forces seen for the first time, which makes the bump and rebound phases more clear on the graph. Earlier on all the forces created by the damper were positive hence showing that the forces were produced when the damper is moving in one direction(bump phase). This is incorrect, by subtracting the static loads a zero point was established, which now visually shows the damper compressing (bump) and extending (rebound) in the graph above. The graph above show the actual force seen through the damper itself.

To be able to make a comparison between the live track data and the dynamometer data. All velocities were made to be positive as done to the dynamometer data. Figure 27 below shows the graph produced after making all the velocities positive.

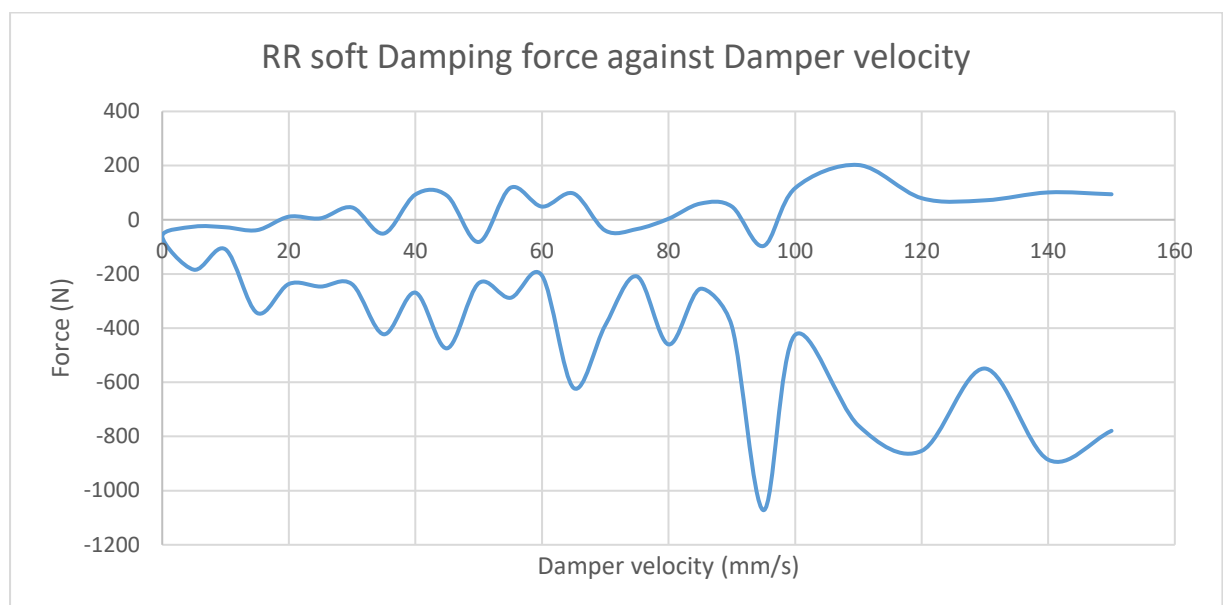


Figure 28 Damping force against Damper velocity

Repeatability

A key aspect for the results obtained is repeatability, this is to see if applying this method again will give us similar results. This is very important as it allows us to spot anomalies with the data, hence increasing the accuracy and understanding of the data obtained.

Rear right soft damping

For soft damper settings for the MCR, the whole process above was repeated for lap 5 and lap 4. The same length and duration of the track is used. The graphs below show an overlay of the two laps and the results obtained from live track data.

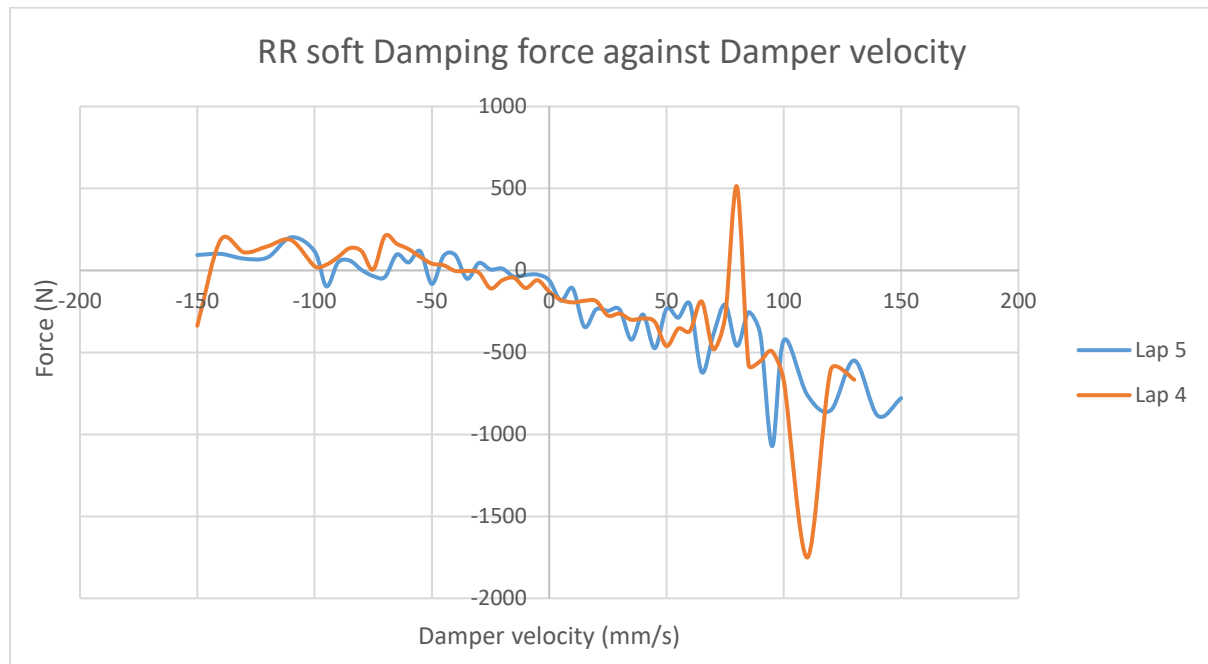


Figure 29 Damping force against Damper velocity

A perfect match of the two laps was not expected but overlaying the data does show a very consistent trend between the two laps. Except for two coordinates, one when the damper velocity is 80mm/s and the other at 110mm/s. These coordinated might differ due to the vehicle travelling over a different patch of track or hitting a bump on the road causing the wheel to accelerate hence altering the force seen by the damper. Discarding the two points mentioned the damping force worked out does seem to have some consistency which does make the data obtained more reliable.

Lap 5 was used for any further analysis.

Rear right hard damping

Only three laps were recorded for the damper with a hard setting, for which 2 laps provided were steady state laps. Which was not helpful for this investigation as it did not provide much data for a range of velocities. Lap 3 provided a higher range of velocities as the driver was not trying to achieve steady state. Still the range of velocities was not even close to range seen for soft dampers. Hence the track length was increased. The image below shows the track length used for when the MCR had a hard damper setting, which lasts a duration of 13 seconds and the distance travelled is 435 meters.

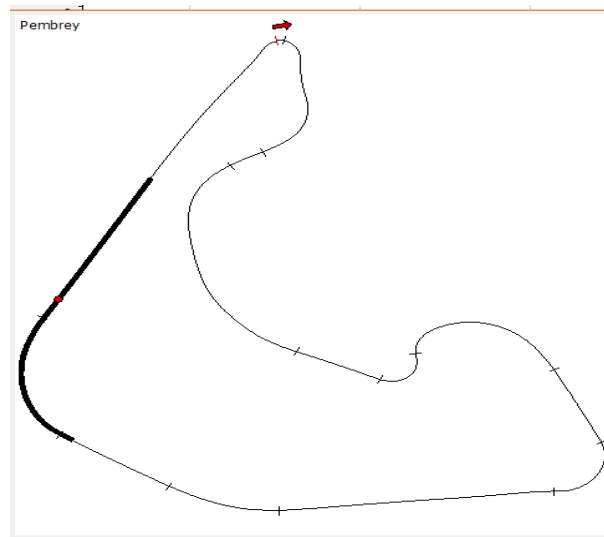


Figure 30 Pembrey circuit

Changing the dampers to hard did cause the preload on the suspensions to change slightly which is expected. The screenshot below shows the static loads for when a hard setting is used on the dampers.

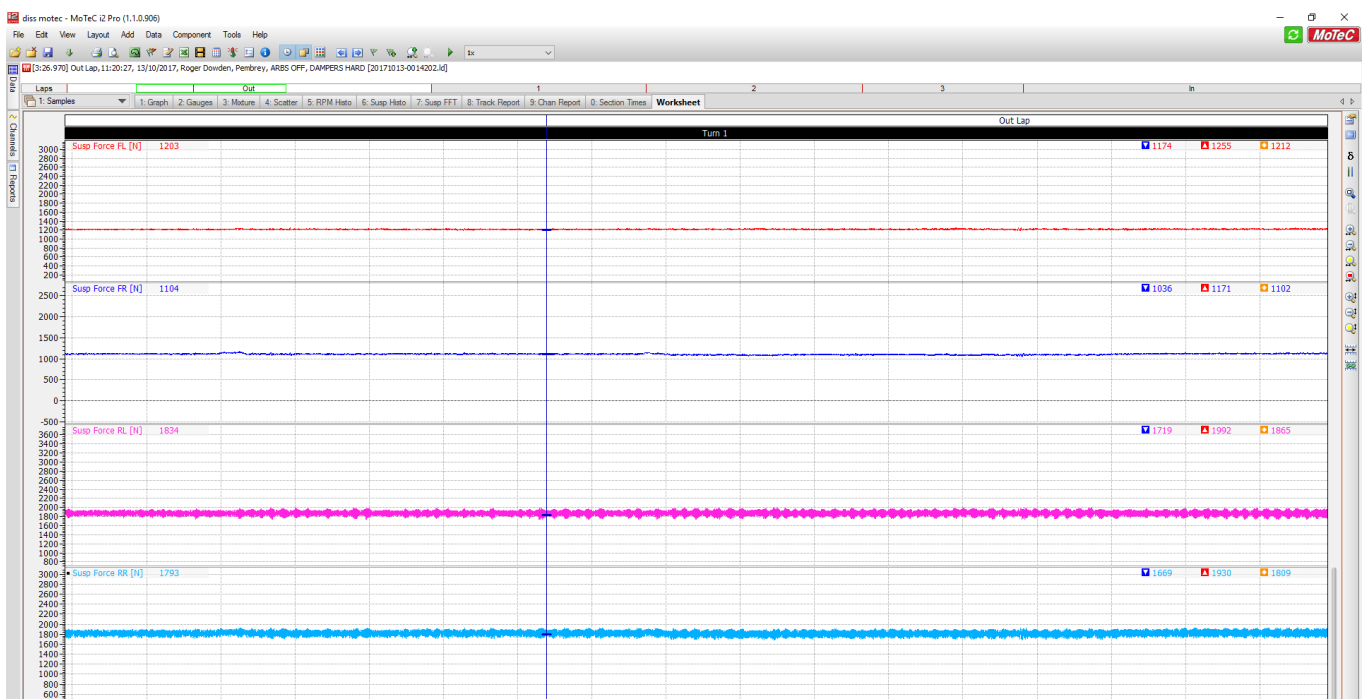


Figure 31 Suspension Static loads

Table 7 Suspension Static loads

FL Hard Static load	1212 N
FR Hard Static load	1102 N
RL Hard Static load	1865 N
RR Hard Static load	1809 N

The table above shows the static load measured for all four corners of the vehicle when the damping setting is set to hard. Static loads are higher toward the rear of the vehicle due to the weight distribution and the rear having a higher mass than the front.

The static position of the damper were measured again for the hard damping as well.

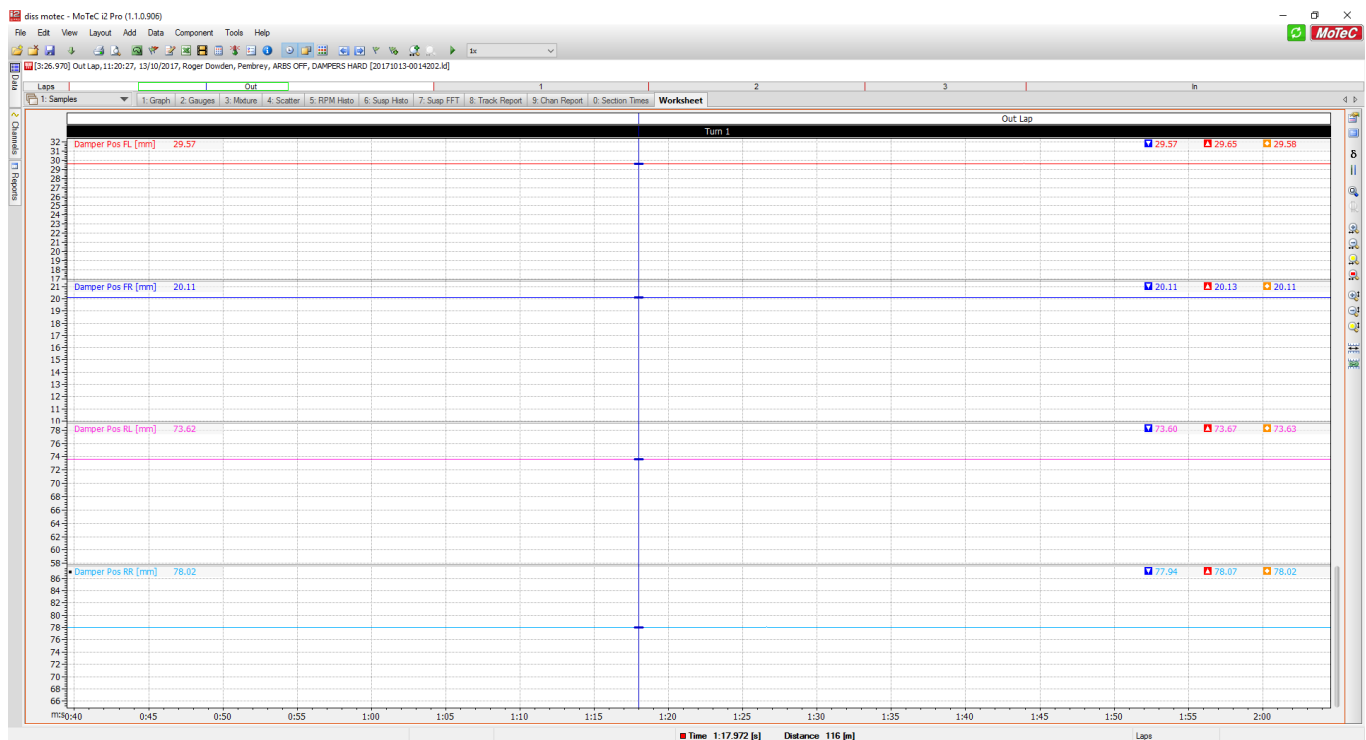


Figure 32 Static damper position

Table 8 Static damper position

FL Hard Static damper position	29.58 mm
FR Hard Static damper position	20.11 mm
RL Hard Static damper position	73.63 mm
RR Hard Static damper position	78.02 mm

Further work include going back to the track gathering a more laps for dampers with a hard settings. Which will make this investigation more reliable. The graph below shows the damping force against damper velocity for the rear right when the dampers are hard.

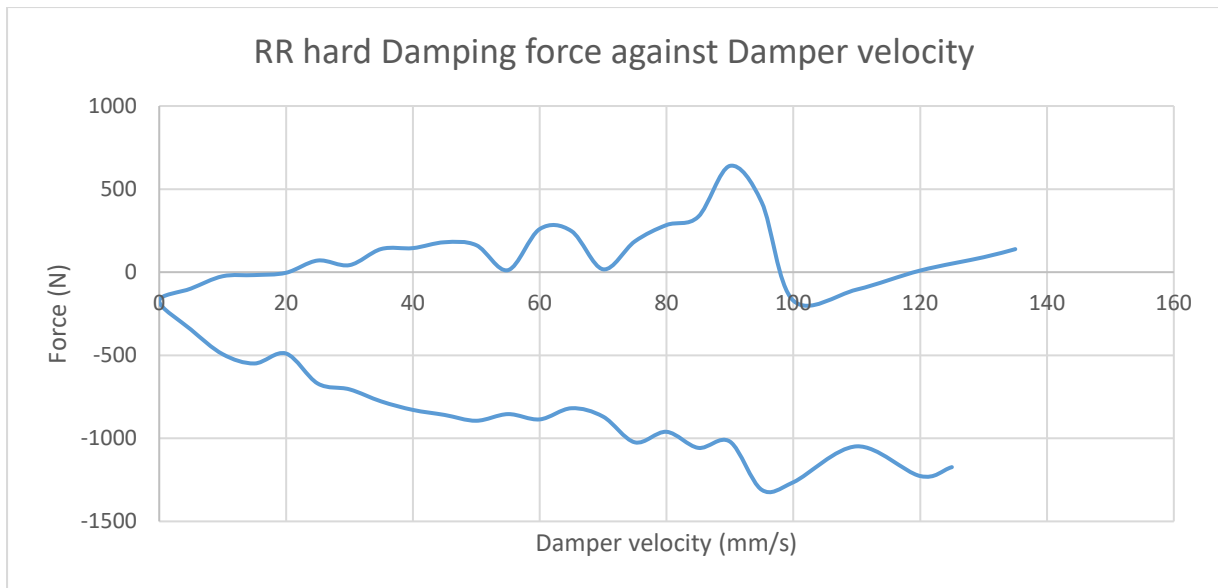


Figure 33 Damping force against Damper velocity

Figure 32 now shows the graph produced when all the velocities seen by the Damper are made to be positive. Changing all the velocities to positive basically just reflects all the negative velocities along the y-axis. Hence producing a similarly shaped graph seen to the one created by the dynamometer.

Results

Now the process mentioned above was repeated for each corner of the MCR for both soft and hard damper settings. Below shows a comparison between a force-velocity graph produced by the dynamometer with an overlay showing both soft and hard damping, and a force-velocity graph produced from the live tract data with an overlay showing both soft and hard damping.

For a clearer analysis all the graphs produced below have been zeroed out. As this helps making a better comparisons between different graphs and would also eliminate the offsets created due to the damper piston having to overcome the gas pressure. The gas pressure within the damper is the preload acting on the piston within the damper, hence why it registers a force acting on the damper even when the velocity is zero. Which is not true as the theoretical assumption of viscous damping is that the damping force is proportional to the velocity. (Bastow, Howard and Whitehead, 2004)

Front left

Damper dynamometer data

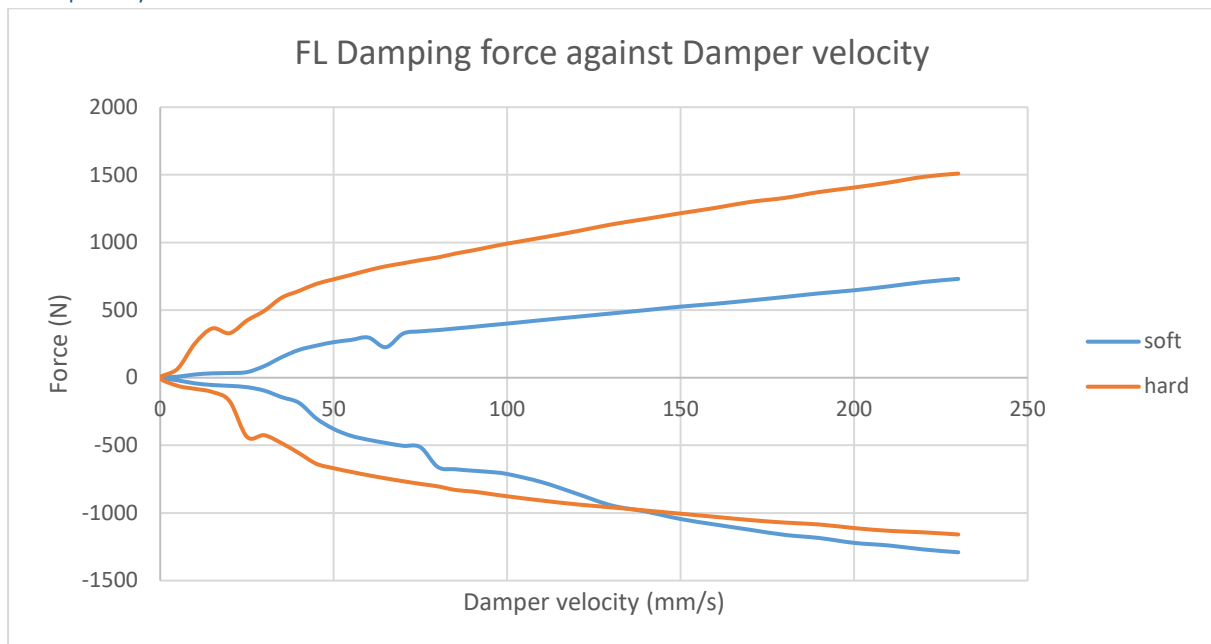


Figure 34 Front left Damping force against Damper velocity

Live track data

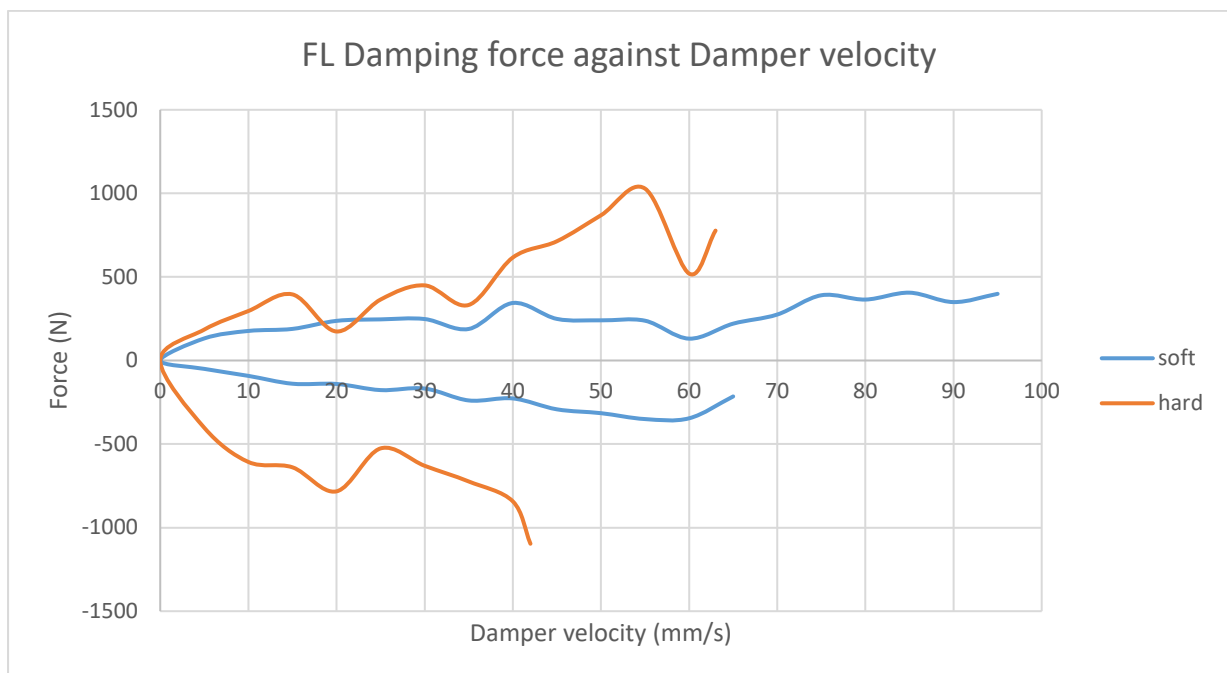


Figure 35 Front left Damping force against Damper velocity

Front Right

Damper dynamometer data

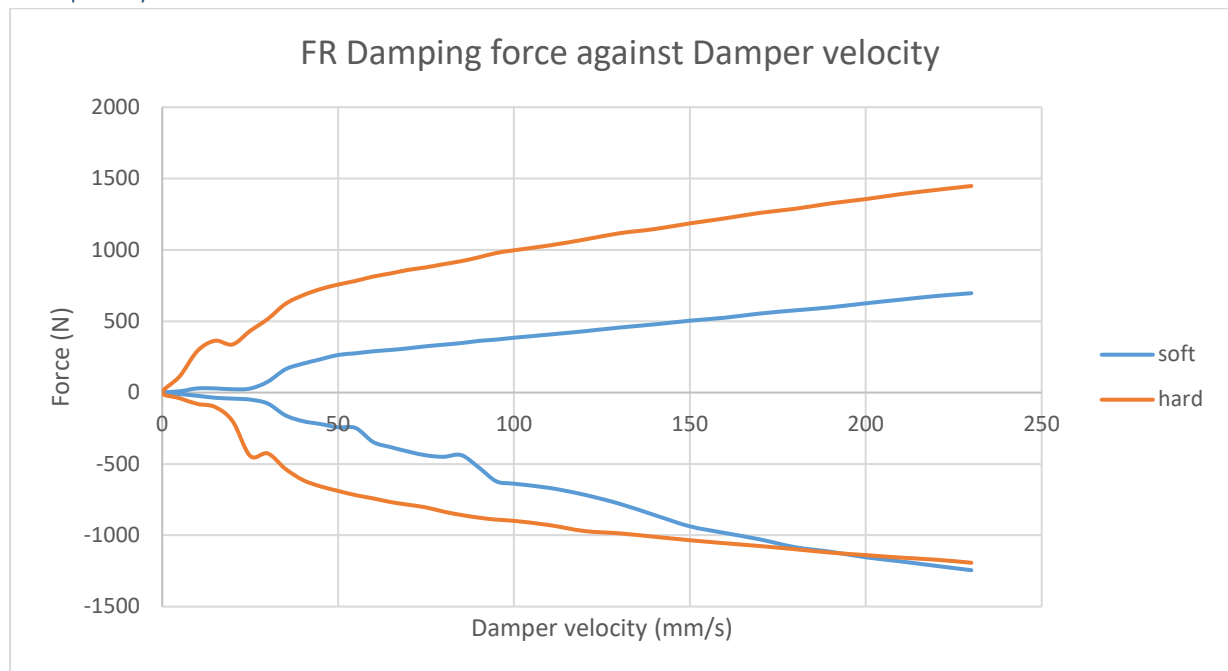


Figure 35 Front right Damping force against Damper velocity

Live track data

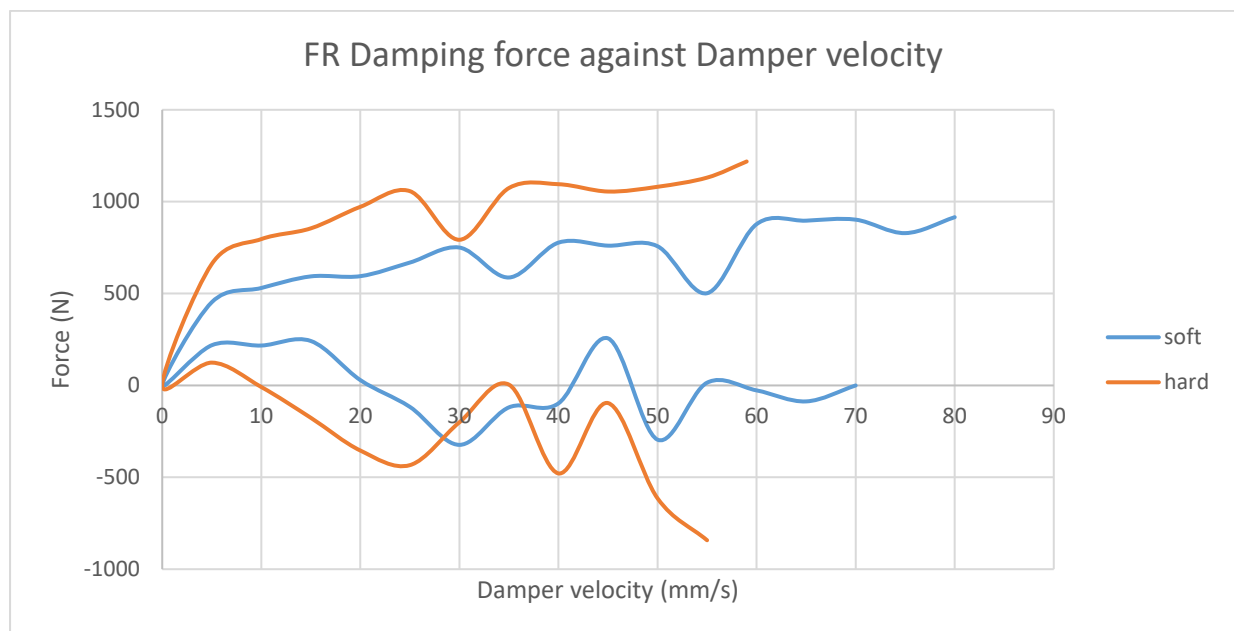


Figure 36 Front right Damping force against Damper velocity

Figure 36 does not follow the general trend that should be seen for a force-velocity graph. There is no clear sign in the distinction between the bump and rebound stages seen by the damper. As all the forces seen by the damper for the bump phases should be positive and all the rebound forces seen by the damper should be negative. Therefore this graph can be noted as an anomaly in data for all the graphs produced

Rear left

Damper dynamometer data

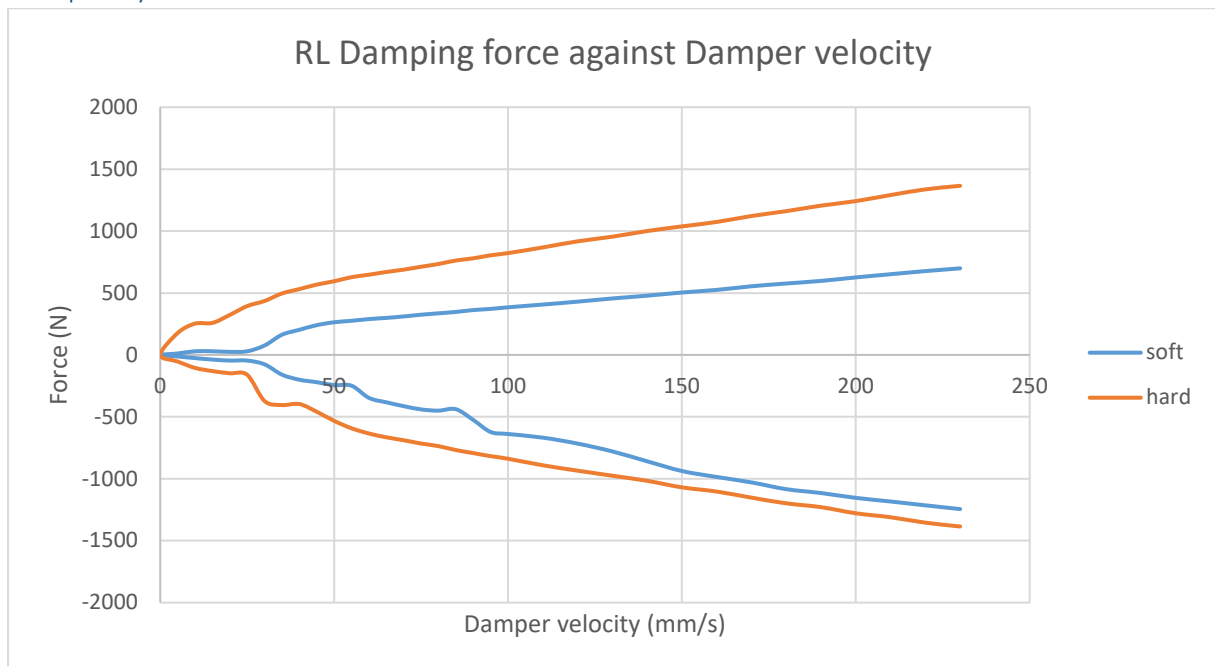


Figure 37 Rear left Damping force against Damper velocity

Live track data

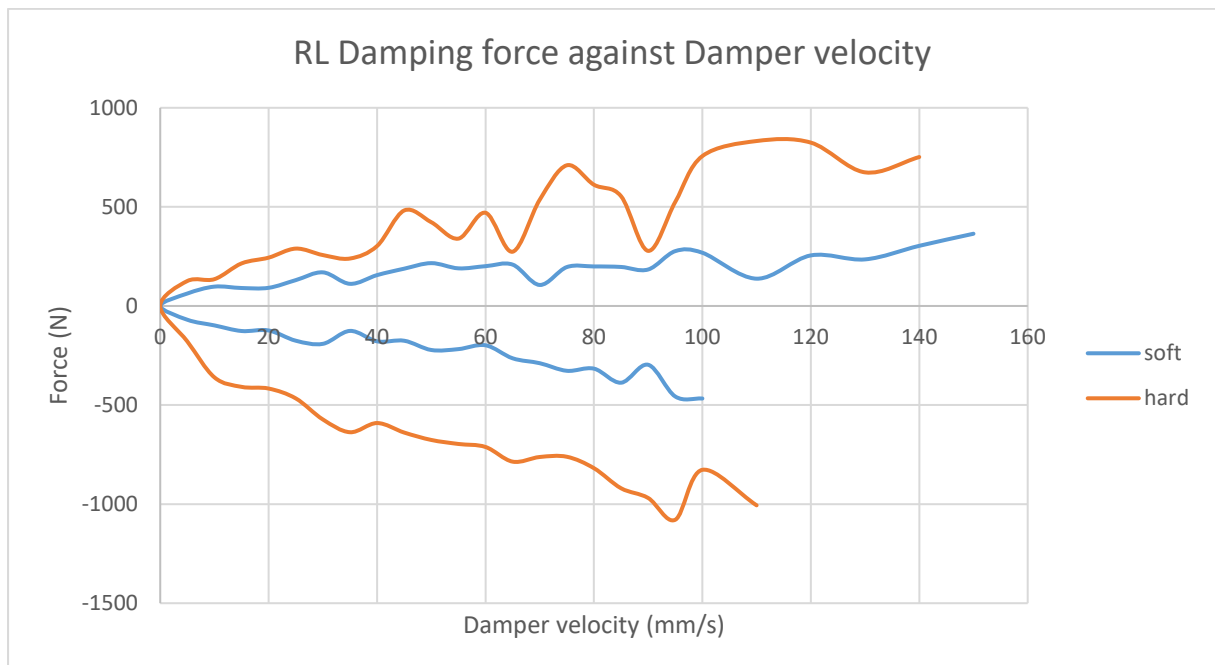


Figure 38 Rear left Damping force against Damper velocity

Rear Right

Damper dynamometer data

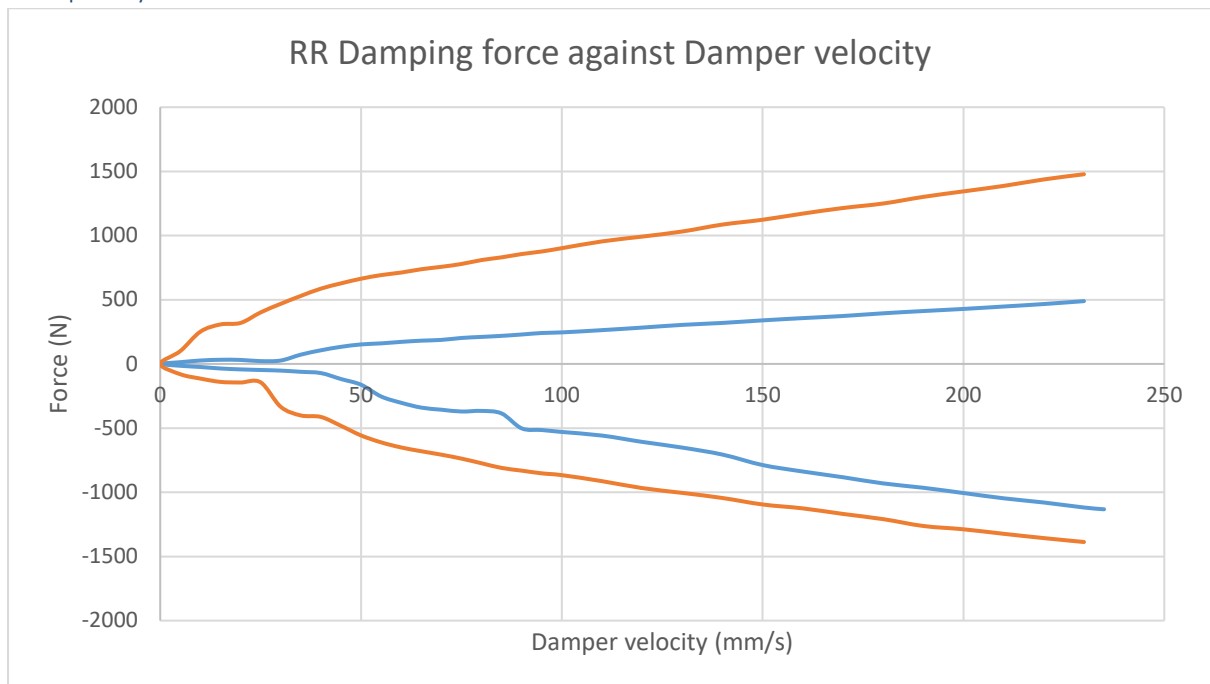


Figure 39 Rear Right Damping forces against Damper velocity

Live track data

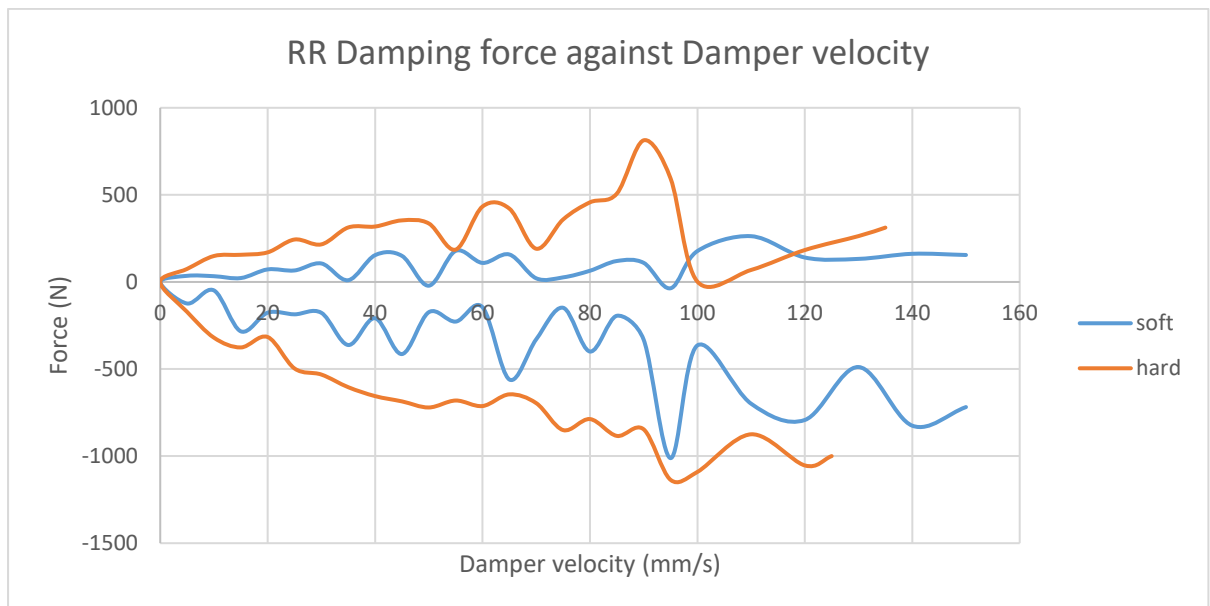


Figure 40 Rear Right Damping forces against Damper velocity

Analysis

For the graphs created using the live track data, these graphs are definitely “Noisy”. But they all are valid looking graphs. There is a clear trend visible, you can see that the damping force generated by the damper increases for all four sets of data as the damper settings are changed from soft to hard. The exact same is seen for all four graphs produced by the damper dynamometer as well, this is because the damper has a greater oil flow restriction for when set to a hard setting.

All graphs produced on the damper dynamometer reach velocities of 230mm/s for both bump and rebound. This is due to the sine wave input from the dynamometer, the crank in the dynamometer inputs a sine wave to the damper. (WHY THERE IS A FORCE VALUE AT ZERO VELOCITY, n.d.)

You can notice by looking at the live track data for all four corners, the velocities seen are nowhere close to 230mm/s hence further work would include using a larger track. However it is also noticeable that the range of velocities seen by the damper is greater in the bump phase than the rebound phase. Which is expected, as you hit a bump in the track it will accelerate the wheel, which then accelerates the damper in its bump phase thus showing us a higher range of velocities. This excitement of velocity will not be seen in the rebound phases.

As all graphs produced on the dynamometer are an average of the acceleration and deceleration forces for the bump and an average of the acceleration and deceleration forces for the rebound. Therefore for an explanation to why for all graphs produced from live track data have that “noisy” characteristic is, as at each velocity the damper could be at any rate of acceleration or deceleration. Figure 41 below shows an overlay of the force-velocity graph produced by the dynamometer for the rear right in the early stages without any data manipulated and the force-velocity graph produced from the live-track data for the rear right.

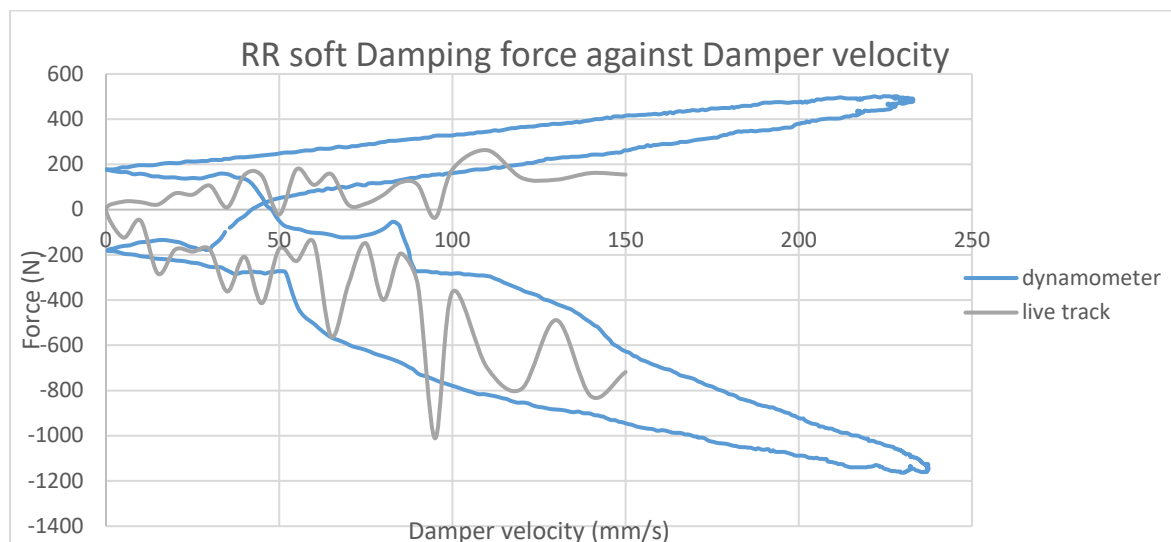


Figure 41 Damping force against Damper velocity for soft setting

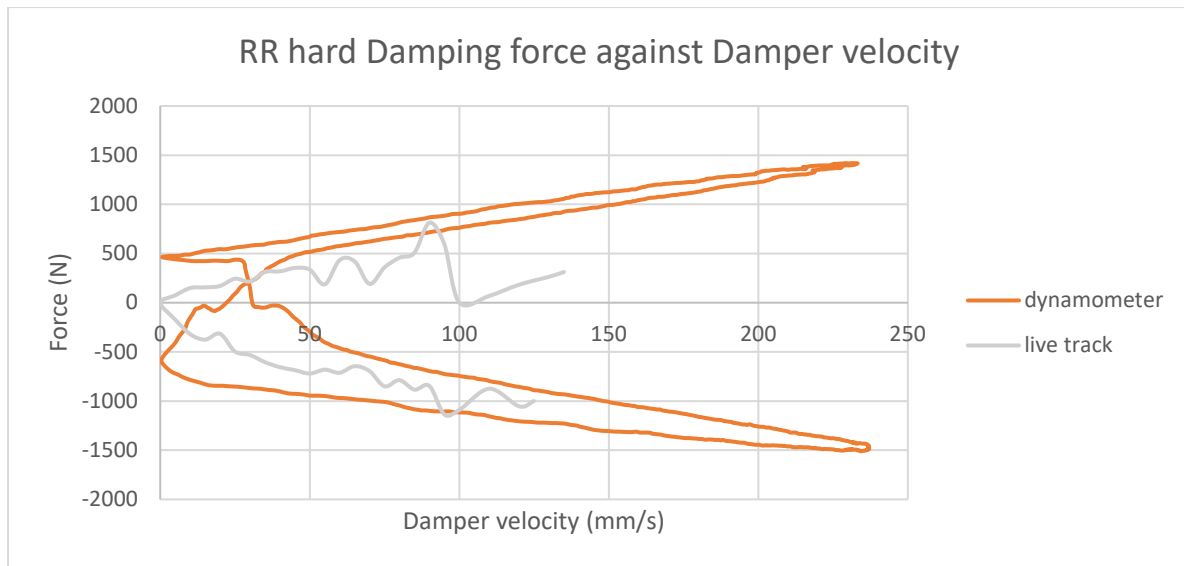


Figure 42 Damping force against Damper velocity for hard setting

Figure 41 and 42 does show a better comparison of data. As it makes slight sense to why the live track data is so squiggly. The live track data somewhat shows the forces seen by the damper for the live track data lie between the acceleration and deceleration range for bump and rebound. As all damping forces produced were plotted in increments of 5mm/s till 100mm/s and then in increments of 10mm/s for any higher velocities. For a clearer and better analysis further work will involve sampling this data at even smaller increments of velocity, so a more accurate characteristic of the force generated could have been plotted for the live track data.

Discussion

For a split second when the damper switches from bump to rebound the dampers will have a velocity of zero and therefore should have a damping for of zero as well. As damping force is directly proportional to velocity. All graphs produced via the dynamometer have a slight offset in its results, this means that the transition from bump to rebound does not happen when velocity is zero which it should be. This is because the frequency the damper generates its force at, is not consistent with the frequency the dynamometer can record the information at.

On the dynamometer if you test a damper at its soft setting, the dynamometer will produce a graph where the graph passes through the zero point on the graph. But if you then change the damper setting to hard and test it again the graph will not zero out this is because the damper is reacting faster. This is due to the low speed damping being blocked and main valves are being opened earlier therefore the dynamometer does not sample at a fast enough rate to measure that change point.

When looking at the piston in the damper, there is a certain area in the piston which have no holes which also generates a force. Between this force being generated and the blocking of the low speed damping the dynamometer moves the start point because at the point that it changes direction and starts to test the damper in the other direction. There is a slight delay where the damper needs to react but the dynamometer is still trying to record the information and it's this discrepancy between the two which causes the offset. (WHY THERE IS A FORCE VALUE AT ZERO VELOCITY, n.d.)

Data collected from the Dynamometer, does not truly represent the forces seen by the damper for the live track data. The rate of acceleration and deceleration you see on the dynamometer is controlled by the sinusoidal characteristic of the dynamometer. Basically the dynamometer produces a steady rate of velocity and acceleration for when testing dampers. The range of accelerations seen by the dynamometer is limited because of the simple harmonic characteristic of the Dynamometer, which was worked out to be 4.441 mm/s^2 .

Whereas the track data does not have this limitation, therefore hitting a bump will accelerate the piston within the damper at a much higher rate than seen on the dynamometer. Table 9 below show the maximum acceleration for each corner of the vehicle for both soft and hard. Most of the ranges are within the acceleration range seen by the dynamometer, except for the rear right corner when the damper are soft. To look at what is really happening on track? What load is the wheel actually experiencing from the damper? You have to take into account the acceleration of the piston in the damper, as for the track data the vehicle is experiencing lots of different rates of accelerations.

Also you can see from Table 9 that the range of accelerations seen by the damper when soft, is significantly higher than the range of accelerations seen by the hard damper. This is because when the dampers are set to soft there is less restrictions in oil flow, therefore the oil flows through the piston at a much faster rate hence why the piston experiencing a higher acceleration.

Table 9 Maximum acceleration for each corner of the vehicle for both soft and hard

	Front left	Front right	Rear left	Rear right
soft	3.58 m/s^2	2.48 m/s^2	3.67 m/s^2	6.73 m/s^2
hard	1.42 m/s^2	1.77 m/s^2	1.82 m/s^2	2.21 m/s^2

Temperatures affects the Damping force majorly, hence why all dynamometer data was collected when the dampers were at a room temperature of 22° . The SPA dynamometer is equipped with a built in thermostat which helps record the temperature of the damper whilst in use. If the dampers were below room temperature the SPA did have a warm up function were it applies a force to the damper casing it to heat up. This heating is caused as viscous oil is forced through small orifices within the damper, and if the damper ever overheated it was just allowed time to cool down.

Whereas the temperature of the damper when testing around the track was not recorded, by using later laps could suggest that the damper could have heated up due to the viscosity of the oil therefore temperature was not consistent in both tests. Which could explain the difference between the two tests done. As there will be energy lost this is known as hysteresis damping. (Encyclopedia Britannica, 2018)

Another explanation to why the dynamometer data may vary from the track data is that the gas pressure is zeroed in the damper on the dynamometer. So the gas pressure is not taken into account but obviously there is gas pressure in the damper which you cannot be take out when obtaining data from track data.

Other applications

Dampers do provide an importance in other application than just vehicles. Damper are widely used by engineers for when designing bridges and buildings:

Taipei 101

The Taipei 101 was the tallest building in 2004. What's interesting about this multi-storey building has a 730 ton ball suspended from the top few floors of the building which acts as a damper. This 730 ton balls has become a tourist location and is also known as the Damper Baby. This helps the building from collapsing by natural phenomena's such as earthquakes and strong winds. (Gizmodo.com, 2018)

Millennium Bridge

Another good example of why damping is important is the millennium bridge. Thousands of pedestrians caused the bridge to rock from side to side, as the movement of the bridge increases people started planting their feet wide pushing out to the side with every step in sync. Which means everyone started walking at the same natural frequency, amplifying the oscillation. The millennium bridge was close and then reopened after been fitted with 91 damper to help dampen both vertical and lateral oscillation which cost \$8.9 million. (ScienceDaily, 2018)

Further work

Throughout the whole report there have been stages which would require further work. In the progression of this investigation a method was developed for producing a damper curve.

- One of the limitations for this analysis is not being able to make a wider comparison between the two sets of data. As testing different multiple damper setting from hard to soft would help gain a better understanding as it would visually show how each adjustment made to the damper changes the damping curve. The behaviour of the damper on track is not the same as it is on the dynamometer.
- Another limitation was not having many laps, when the damper setting was adjusted to hard as mentioned before. Having more laps will help overlay data to make a comparison between the two laps and see how much the track actually effects the damping force.
- Further work would also involve producing a excel spreadsheet which will help sort the dynamometer data to produce the force-velocity graphs as the process can be quite long and repetitive.
- Looking further into all the forces acting through the pushrod and looking into more forces acting through the pushrod. To give a more precise reading for the Damping force seen by neglecting them from the suspension force.
- Due to the dynamometer seeing a much bigger range of velocities, further work would involve repeating the experiment but using a larger piece of track both soft and hard damping to be able to make a clearer comparison for high velocities.

Conclusion

It is explained how the force-velocity graph were produced and all the forces seen through the suspension arm which were subtracted to just give us the damping force. The investigation above shows a clear possibility of being able to extract the damping force from live track data. A concise analysis was done comparing both dynamometer and live track data. The limitations of comparing both data with each other is mentioned.

All graphs shows above only show a comparison of data made between two settings of damping. For a better analysis to then be able to predict how the damper will behave on track further work is needed to test a range of damper settings.

I personally feel as the investigation progressed a correct method of producing a force-velocity graph was developed. As the data collected did have some relevance as gone to explain in the analysis showing that the range of velocities seen by the track fit into the range provided by the dynamometer.

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