



Investigating Damper Calibration on a Live Test Vehicle, to Understand Real-Time Damper Characteristics

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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

This report contains details of the investigation undertaken to understand damper calibration, and the ability to test for, analyse and develop damper data curves. Dampers are currently set-up using damper dynamometers which are not always easily available. The idea of being able to manipulate on-board logged data to produce damper curves is the basis for this investigation. Utilising on-board data logging systems, the purpose of the investigation is to develop a method for determining damping force from recorded live track data. The ability to record damper velocities and positions, plotted against damping force, provides a method for producing force-velocity and force-displacement graphs per damper.

Research details the fundamentals of damper construction. Current damper dynamometer methods are explored, to gain an already proven method to abide by when developing a method from vehicle recorded data. Data collection and analysis follows based on the MCR Sports 2000 race car, Base Performance simulator, and damper dynamometer equipment made available by UWTSD, Swansea.

Technical developments made from analysing final results are detailed. Concluding points outline the positives and negatives of work undertaken. Lastly, there is a section dedicated to describing the possible future developments that could be undertaken to advance the project further.

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IV: Abbreviations

ARB	=	Anti-Roll Bar	FR	=	Front Right
CSV	=	Comma-Separated Values	F-V	=	Force-Velocity
Deg	=	Degree	MR	=	Motion Ratio
Dyno	=	Dynamometer	RL	=	Rear Left
F-D	=	Force-Displacement	RR	=	Rear Right
FL	=	Front Left			

V: Nomenclature

C	=	Damping Constant	m	=	Mass
C_{cr}	=	Critical Damping Coefficient	m	=	Metre
d	=	Wire Diameter	mm	=	Millimetres
D	=	Mean Coil Diameter	ms^{-1}	=	Metres per Second
D_c	=	Sum of Damping Coefficients	mms^{-1}	=	Millimetres per Second
dx/dt	=	1 st Order Differential	n	=	Number of Active Coils
d^2x/dt^2	=	2 nd Order Differential	N	=	Newton
F	=	Force	Nmm^{-1}	=	Newton's per Millimetre
g	=	Gravitational Constant	N/ms^{-1}	=	Newton's / Metre per Second
G	=	Shear Modulus	PSI	=	Pounds per Square Inch
k	=	Spring Stiffness	R	=	Spring Constant
Kg	=	Kilograms	V	=	Velocity
K_s	=	Spring Rate	ω	=	Frequency
K_w	=	Wheel Rate	x	=	Deflection
			$^{\circ}$	=	Degree
			ζ	=	Damping Ratio

1: Introduction

As vehicle technology advances, the demand for further understanding and improvement drives the industry for innovation. This is no different when looking at vehicle dynamic behaviour, more specifically, damper calibration. Dampers have been dubbed a 'black art' (Milliken & Milliken, 1995) due to their complexity. Currently, the industry accepted standard for calibrating dampers is by means of a Damper Dynamometer, of which are a proven method for damper calibration. However, this is a general method used for all types of damper, no matter their intended application. What if it was possible to calibrate the dampers directly on the vehicle of use? What if that vehicle could then be used to calibrate dampers for specific circuits? Would live track data produce enough information to allow for a comprehensive damper calibration without the use of a dynamometer? These are just some of the questions aimed at investigating through research and testing.

Within motorsport, there is a huge financial variation between professional high-end teams and the smaller motorsport enthusiast whom enters club motorsport events. This means that specialised equipment is mostly only accessible to those with large financial backing. Thus, being the purpose of trying to potentially exploit the capabilities of recorded live track data. Whilst damper dynamometers may not be easily accessible, nor cost effective in most cases, data logging equipment is becoming much more accessible to even the smallest of motorsport enthusiasts. Thereby, the basis of this investigation is to utilise a data logging system to record test vehicle track data, to analyse and refine damper data outputs.

Damper calibration is conducted by looking at the force produced by the damper dependent upon the applied bump and rebound setting, and the pistons velocity. Damper dynamometers achieve this by producing two types of graphical outputs; one being a Force-Velocity graph, two being a Force-Displacement graph. The success of the data logger technique will be determined by producing accurate graphs that replicate those created by the dynamometer. From which, analysis of such graphs will provide understanding on the validity of the technique.

This investigation will initially consider the fundamentals of damping, types of dampers, and aspects which influence damping, with the forward view of developing upon the research to define a methodology for determining damper force from track data collected. This project has utilisation of; an MCR Sports 2000 test vehicle with Motec data logging installed, a Base Performance Simulator with the MCR car model and multiple test tracks, and a damper dynamometer.

2: Literature Review

Developing understanding and applications of dampers firstly requires an initial background investigation into the fundamentals of damping and its required purpose. Further considering the methods used to tune dampers, and the theory behind vehicular damper systems.

2.1: Damper Construction & Operation

Years of development have produced multiple damper constructions, mostly similar in operation but often having noticeably effective re-engineered components. The principles upon which they have been based on have also been changed and adapted, from the Koni 'double-tube damper' to the Girling 'Monotube Principle', and the adapted Woodhead-Monroe 'Monotube Principle' which incorporates aeration of the oil (Campbell, 1981).

Three types of damper are commercially used in the motoring industry; Mono-tube, Mono-tube with Compression piston, and Twin-tube. Figure 1 displays the internal construction of each type of damper mentioned.

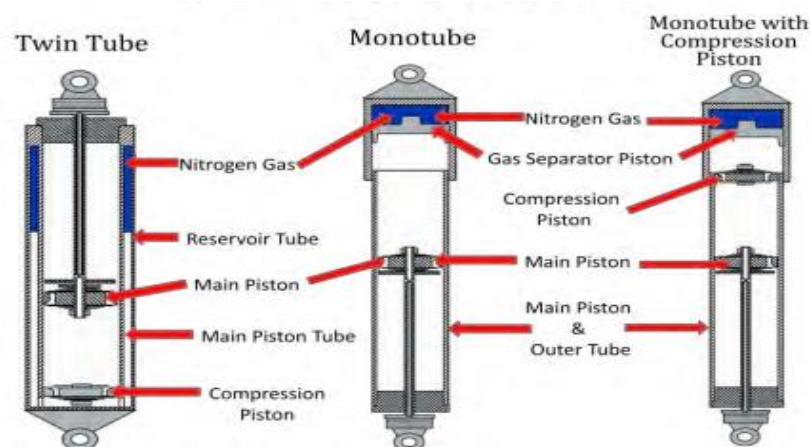


Figure 1 Three Main Types of Damper Construction

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

Fundamentally each type of damper has similar internals, where the main piston contains the primary valves. Damping force is produced as the piston moves up and down the chamber. Oil is forced through the valves during piston compression (Bump) or extension (Rebound), thereby applying a damping force. The damper oil and nitrogen gas are the fluids causing the damping effect due to their compressibility. The velocity of which the piston moves through the fluid, and the valve gap, determine the damping force produced. This means that the damping force is directly proportional to the velocity of the piston. The valves incorporated into the piston design determine the gap size the fluid can flow through. A smaller gap greatly increases the damping force due to the greater limitation of oil being able to flow through the valve. This is commonly referred to as "hard" damping. Vice versa,

a large gap greatly increases fluid flow, thereby reducing damping force produced. This is commonly referred to as “soft” damping. These are the minimum and maximum settings applicable to damping, where optimum damping lie’s in-between. The amount of damping required is summed up appropriately as follows:

‘Although our aim is to damp out as quickly as possible each vertical impulse given to the wheels, a measure of compromise is necessary.’ pp.125 (Campbell, 1981)

This means that whilst the primary objective to damping is to reduce the motion of an oscillating object quickly, the ideal effect of damping for vehicles is to keep the tyre within optimum contact of the ground. Jim Kasprzak backs-up this statement by describing the amount of damping required to be ***‘Just enough!’*** pp.2 (Kasprzak, Understanding Your Dampers: A Guide From Jim Kasprzak, 2011). Reasons for this include; optimising grip by controlling the contact patch between tyre and road surface, reducing vertical loading variations on each tyre from which improves tyre wear, and for a suitable ride condition that does differ dependent upon the intended application of the vehicle.

Most damper constructions will include valve adjusters which allow the operator to change the valve gap. Along with the valves, dampers are constructed with shims stacked together. These shims can flex, opening ports that allow the fluid inside to flow through. This occurs during the high-speed operation of the piston. Shims are applied for both compression and rebound phases. During piston compression, rebound dedicated shims stay closed whilst compression shims flex and allow oil to flow through the ports. Vice versa occurs in rebound (Levy, 2013). Shim stack construction is usually pre-developed by the manufacturer and implemented into the damper.

The adjustability of a damper unit is dependent on the manufacturers’ specifications, however most will either have one, two or four-way adjustability. The valves control two variables; ‘Bump’ and ‘Rebound’. When the damper piston compresses the fluid, the vehicle is said to be in ‘bump’. ***‘Bump damping controls the unsprung weight’*** of the vehicle i.e. wheels and suspension (Milliken & Milliken, 1995). As the piston extends back out, the vehicle is said to be in ‘rebound’. ***‘Rebound controls the transitional roll’*** (Milliken & Milliken, 1995), meaning that this is where the damper controls the sprung mass of the vehicle i.e. chassis and body. The greater adjustability of a dampers bump and rebound valves, allows for increased control of a vehicles movement when it is subjected to various loading conditions i.e. surface changes, cornering, accelerating and braking. Dampers with less adjustability have reduced valve gap variation, thereby reducing the control factor the damper can provide due to a reduction in damper refinement capabilities.

Bump and Rebound can be separated into two aspects: Slow and fast bump, along with slow and fast rebound. Slow velocity can be interpreted to be below 70mms^{-1} , whereas fast velocity is considered to be above 100mms^{-1} (Vaio, 2010). However, this can be split further by including a mid-range bump and rebound. In such cases; low speed is between 0 & 50.8mms^{-1} , mid-range is 50.8 to 203.2mms^{-1} , and fast speed is 203.3mms^{-1} (Kasprzak, Understanding Your Dampers: A Guide From Jim Kasprzak, 2011). These become applicable once analysing force-velocity graphs produced from damper testing.

2.2: Damper Dynamometers

Tuning a damper often involves the use of a dynamometer a.k.a. dyno. A damper dyno is generally a mechanical piece of machinery that utilises a motor to rotate a crank. The rotational motion of the crank is then used to supply a vertical motion to the damper piston.

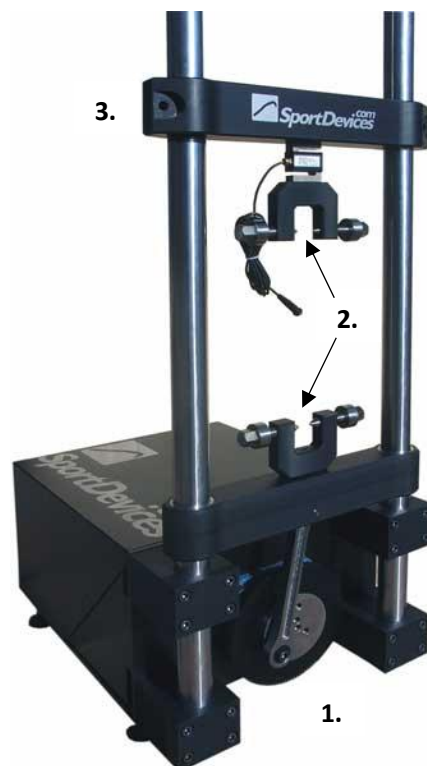


Figure 2 Crank Damper Dynamometer by Sport Devices. (Devices, 2007)

Referring to Figure 2, the operation and damper attachment can be shown.

1. Motor driven crank with connecting rod. The electronically controlled rotational motion of the crank is converted into vertical motion through the rod.
2. Damper mounting points. Commonly the damper is hard mounted at the top, with the damper piston attached at the bottom, where it is free to move with the vertical motion of the connecting rod from the crank.
3. Adjustable top mount. The top hard mount for the damper is adjustable to allow the dyno to incorporate varying sized dampers. It also allows for different stroke lengths to be tested.

The purpose of the dyno is to apply a known velocity to the damper piston. The crank has a sinusoidal motion which is applied to the piston, providing a controlled test frequency. The AC motor controls the input velocity. The piston speed, as previously mentioned in 2.1: Damper Construction & Operation, is directly proportional to the damping force produced. External software records the force and velocity data once in motion and produces two types of graph; a force-velocity graph, and force-displacement graph. A data engineer can then utilise these graphical outputs to determine if a damper has congruity within the unit. This is especially useful when comparing multiple units of apparent similarity. Outliers

in the data readings could suggest problems such as: Contaminated oil, wearing seals and/or parts i.e. valves and shim stacks (Roehrig Engineering, 2016).

Advantages:

- Highly accurate data output.
- Good adjustability for various test types.
- Free from hysteresis losses. (Venkat, 2014)

Disadvantages:

- Commonly large machines, requires adequately planned space for operation.
- Expensive. Research into multiple selling sites found they can range from £2000 to £10,000.
- Sensitive to overloading and mechanical impacts. (Venkat, 2014)

2.3: Data Logging Systems

As vehicular electronic control systems become more sophisticated, an increased amount of data is available to record. Data loggers have vastly become incorporated into competition vehicles, as it provides teams with information that allows engineers to identify potential issues and areas for improvement. It is such capability that can make the difference on race weekends.

There are multiple options when choosing a data logging system. Most of which have differing packages available dependant on the logging sophistication the user requires. Systems available include Motec, PI Toolbox, Aim dashboard data logger series, and VBox Sport just to name a few. UWTSD Team MCR utilise Motec for their data logging requirements. Motec is also the simulation software of choice for the Base Performance Simulator provided by UWTSD, Swansea. Therefore, learning and operating the software was an important demand to effectively analyse test data.

Advantages:

- High logging rates are available. The higher logging rates increase data collection in a specified period.
- Cheaper than Damper Dynamometers. Research suggests data logging systems can cost between £100 & £3000.
- Installed to the vehicle directly, allowing track specific data to be gained.
- Dampers do not require removal from the vehicle.
- Analysis software is available for free.

- Software allows for extra math channels to be created, making it possible to refine data.

Disadvantages:

- Set-up time is longer when compared to dynamometers.
- Require regular maintenance to ensure suitable recordings from all sensors.

2.4: Spring-Mass Damper Systems

Vehicles are constructed with suspension units that contain springs and dampers, with a mass of components mounted above. This is known as a damped spring and mass system (Milliken & Milliken, 1995). Cars have four of these systems, one in each corner of the vehicle. This means theoretically the car can be split into a single system with one differential, or added together with up to four differentials. The more differentials involved in analysing the system, the greater mathematical complexity required to find a solution.

Spring-mass damper systems are typically depicted as shown in Figure 3. There are three aspects to these systems; inertial force, damping force, and spring force. Combined they provide the total force acting per corner of a vehicle.

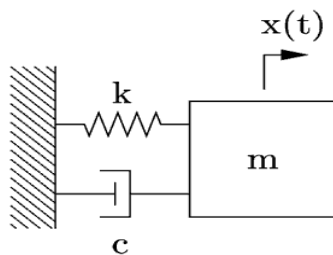


Figure 3 Spring-Mass Damper System (1 Differential)

(M.I.T., 2017)

Each force is related to the other, as inertial is accelerated mass, damping is velocity based, and spring is displacement based. Mathematical techniques such as integration and differentiation can be applied to find displacement, velocity or acceleration from the other. The following equations are the premise for the differentials.

$$\text{Inertia Force} = m(d^2x/dt^2)$$

$$\text{Damping Force} = C(dx/dt)$$

$$\text{Spring Force} = k(x)$$

Equation 1 Spring-Mass Damper Force Formulae

pp.787 (Milliken & Milliken, 1995)

3: MCR Data Collection

Before testing dampers and analysing various types of data outputs provided by dynamometers or data loggers, test vehicle information and calculations are required. This collection of data will later be used to analyse the data traces output from the vehicles data logger. Table 1 outlines the measured data collected from UWTSD's MCR team.

Table 1 MCR Data Collection

Vehicle:	MCR Sports 2000
Type:	Open Cabin Sports Prototype Race Car
Weight Split:	50:50
Wheelbase [m]:	2.588
Tyre Rolling Radius Front [m]:	0.25783
Tyre Rolling Radius Rear [m]:	0.27662
Total Mass [Kg]:	705.4
FL Corner Mass [Kg]:	115.6
FR Corner Mass [Kg]:	157.6
RL Corner Mass [Kg]:	227.5
RR Corner Mass [Kg]:	204.7
FL Unsprung Corner Mass [Kg]:	22.5
FR Unsprung Corner Mass [Kg]:	22.5
RL Unsprung Corner Mass [Kg]:	24
RR Unsprung Corner Mass [Kg]:	24
FL Spring Rate [Nmm ⁻¹]:	71.8031
FR Spring Rate [Nmm ⁻¹]:	71.8031
RL Spring Rate [Nmm ⁻¹]:	80.1958
RR Spring Rate [Nmm ⁻¹]:	80.1958

Following the collection of data from the test vehicle, mathematical formulae can be used to calculate further required parameters about the vehicle. Parameters include motion ratios, wheel rates, and ride frequencies.

3.1: Motion Ratio

A motion ratio is a mathematical percentile of one components movement in relation to another. In terms of suspension and damper movements there are two motion ratios' that are of interest. The spring to wheel displacement, and the damper to wheel displacement as depicted in equations 2 and 3 (Eibach, 2017).

$$MR_s = \frac{\text{Spring Displacement}}{\text{Wheel Displacement}}$$

Equation 2 Spring Motion Ratio

$$MR_d = \frac{\text{Damper Displacement}}{\text{Wheel Displacement}}$$

Equation 3 Damper Motion Ratio

Calculating the motion ratio for the MCR means that data recorded from the wheel can then be multiplied against the ratio to gain an understanding of the loads and displacements being seen at the springs and dampers.

The following methodology was used to gain the motion ratio for the MCR:

1. Scales are fitted under each wheel and zeroed.
2. Measure spring/damper length.
3. Utilising a car jack, lift one corner of the vehicle at specified increments i.e. 5mm.
4. Re-measure the spring/damper length and record each wheel loadings registered on the scales.

A graph of spring/damper displacement against wheel lift is plotted, with the gradient being taken to be the motion ratio for the vehicle.

$$MR = 1:1$$

3.2: Wheel Rate

There are two proven methods of gaining the wheel rate of a vehicle. The first method utilises the testing outlined in 3.1: Motion Ratio, where the load changes are plotted against wheel displacement. The gradient of the resultant graphical output would provide the wheel rate. Secondly the wheel rate could be gained mathematically utilising the following equation:

$$K_w = K_s \times MR^2$$

Equation 4 Wheel Rate

(Kasprzak, Seminar Downloads: Kaz Technologies, 2011)

Utilising the data provided by Team MCR, there are two different spring rates used at the front and rear. This will provide two different wheel rates front to back. The wheel rates are:

$$\text{Front } K_w = 71.8031 \times 1 = \mathbf{71.8031 \text{ Nmm}^{-1}}$$

$$\text{Rear } K_w = 80.1958 \times 1 = \mathbf{80.1958 \text{ Nmm}^{-1}}$$

3.3: Spring Rate

Due to the spring's interaction along with the damper, it is highly likely that a spring force will be a contributing factor to total corner forces. Cars have cylindrical helical springs, with a spring constant calculated as follows:

$$R = \frac{G \times d^4}{8 \times D^3 \times n} \text{ Nmm}^{-1}$$

Equation 5 Spring Rate

pp.244 (Bosch, 2011)

The spring constant is also known as the spring stiffness which is given the symbol 'k'. Spring stiffness is already a known quantity for both front and rear springs. In terms of data analysis and developing an understanding of the forces acting at the corner of the vehicle, the spring force is more relevant than the spring constant. For this, Hooke's law is used to find spring force.

$$F = -kx$$

Equation 6 Hooke's Law

(Holzner, 2017)

Rather eloquently the equation tells us that spring force is equal to the spring stiffness multiplied by the deflection of the spring. The negative sign simply determines the direction the force is directed in. Utilising the Hooke's law equation, the force at each recorded deflection point can be calculated on the Motec data analysis software. Spring deflection is a known variable as each damper has a potentiometer installed in-line with its deflection direction. The potentiometer recordings provide the 'x' variable. Combined with the spring stiffness values, spring force at each logged data point can be calculated.

3.4: Critical Damping and Damping Ratio's

Damping occurs at multiple different frequencies dependent upon its damper construction. Mathematically, damping coefficients can be calculated to represent the how quickly the mass of a system returns to its steady state due to the amount of damping applied (Kasprzak, Seminar Downloads: Kaz Technologies, 2011). The following formulae is used to calculate critical damping coefficients:

$$C_{cr} = 2\sqrt{k \times m}$$

Equation 7 Critical Damping Coefficient Formulae

(Kasprzak, Seminar Downloads: Kaz Technologies, 2011)

The MCR test vehicle has two different spring rates from front to rear. This means that the calculated critical damping coefficient at the front will be different to the rear. The formula was applied as follows:

$$Front C_{cr} = 2 \sqrt{71.8 \times 10^3 \times \left(\frac{1}{4} \times 705\right)} = 7114.703 N/ms^{-1}$$

Equation 8 Front Critical Damping Coefficient

$$Rear C_{cr} = 2 \sqrt{80.2 \times 10^3 \times \left(\frac{1}{4} \times 705\right)} = 7519.37 N/ms^{-1}$$

Equation 9 Rear Critical Damping Coefficient

There are three types of damping; underdamping, overdamping and critical damping. All three have their benefits dependent upon the intended application of the damper, however typically for a vehicle an ideal amount of damping is required. This ideal state doesn't fit directly into any of the three cases mentioned, therefore damping ratios are calculated. These are a relationship of the damping coefficients in relation to the critical damping coefficient. Equation 10 represents the formula outlined by (Pacejka, 2006) to calculate the damping ratio, however it can be expressed simply utilising damping coefficients (Equation 11).

$$\zeta = \frac{D_c}{2m\omega_0}$$

Equation 10 Damping Ratio

pp.31 (Pacejka, 2006)

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

Equation 11 Coefficient Damping Ratio

(Kasprzak, Seminar Downloads: Kaz Technologies, 2011)

Before the damping ratio can be calculated, the damping constant 'C' needs to be calculated. The damping constant is the gradient of a force-velocity line graph. Equation 12 is the damping constant formula.

$$C = \frac{F}{v}$$

Equation 12 Damping Constant

(Williams, 2016)

Applying the calculated damping constant to the damping ratio formula resulted in the following ratios:

$$\varepsilon_{Front} = \frac{4699.606}{7114.703} = 0.661$$

Equation 13 Front Damping Ratio

$$\varepsilon_{Rear} = \frac{4699.606}{7519.37} = 0.625$$

Equation 14 Rear Damping Ratio

4: Damper Testing

Mathematical data collection based on the test vehicle outlined in [section 3](#) will become vital to damper analysis later. The next step is to acquire test data. Three forms of data are available for recording via damper dyno, track and simulation testing.

4.1: Dynamometer Testing

The purpose of this investigation is to determine whether accurate damper plots can be produced just by utilising logged data from the test vehicle. As previously stated, damper dyno testing is the industry accepted standard for damper performance analysis. For this reason, it is important to test the current MCR dampers on a dyno. The results of the test will provide suitable force-displacement and force-velocity graphs to refer too. In developing the method to produce force-displacement and/or force-velocity graphs from logged data, the dyno damper plots will be constantly referenced back too to determine the accuracy of the method utilised.

Collaborating with Team MCR and UWTSD technician Paul Davies, a damper was tested on the dyno with force-displacement (Figure 4) and force-velocity graphs (Figure 5) plotted from the recorded data.

4.1.1: Force-Displacement

A force-displacement graph depicts the damper piston travel against the force that is produced. The graph shows the displacement of the damper piston in both bump and rebound phases at two set piston speeds. Slower speed is shown via the inner elliptical trace. Higher speed is shown via the outer circular trace.

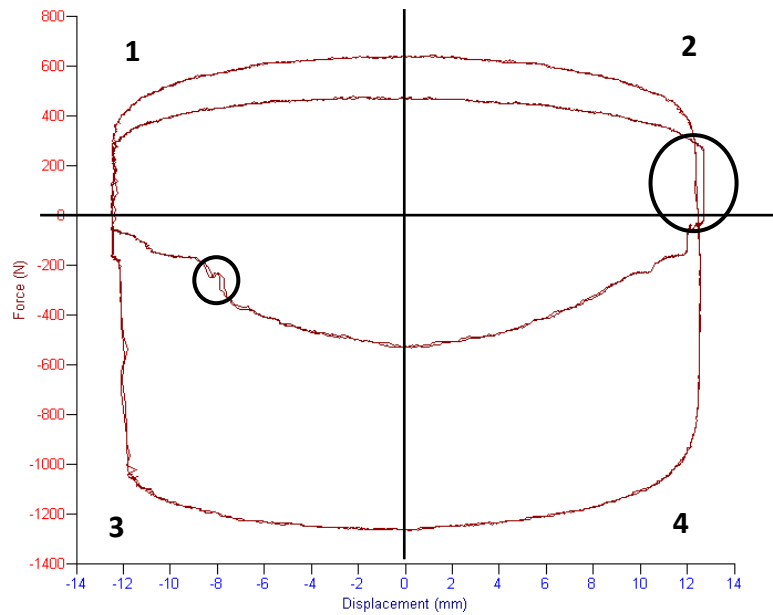


Figure 4 Penske Damper Force-Displacement Graph

To aid description of the graph shown in Figure 4, it has been split into four segments. Each segment visually depicts the following:

1. The damper piston is being accelerated through its bump phase.
2. The damper piston is decelerating through its bump phase.
3. The damper piston is accelerated through its rebound phase.
4. The damper piston is decelerating through its rebound phase.

(NTT Racing, 2008)

Force-displacement plots are particularly useful for determining whether a damper has any defects that effect its overall operation. When analysing the plot, it is important to look for any irregularities within the curves, and to identify whether there is a significant difference in force produced when comparing the acceleration phases to their respective deceleration phases.

Analysis of the force-displacement graph in Figure 4 have determined there to be a few small irregularities being shown within the damper. Circled in segment 2 is a section of the decelerating bump phase, where it can be seen that the force is greater on the slow speed run when compared to the fast speed run. This could be an early sign of hysteresis. Hysteresis occurs when a proportion of the kinetic energy absorbed is converted into heat energy (Campbell, 1981). Hysteresis can occur at minimal levels within a damper, however the greater energy transfer lost as heat, reduces the performance of the damping. Graphically hysteresis is determined by looking for any force differences in the acceleration phases in comparison to their relative deceleration phases (NTT Racing, 2008). No

apparent force differential occurs within Figure 4, therefore other causes must be considered. Reasons for damper failures and/or reduced performance include:

- Wear and Tear. Dampers include mechanical parts that over time are likely to wear out and require replacement. Likely cause of the irregularity circled in segment 3 of Figure 4.
- Oil and/or gas leaks. Seals commonly dry and crack over time thereby causing an imperfect seal, allowing fluid inside to leak.
- Foreign objects. Not a common issue, however sometimes an unexpected piece of grit can get trapped within the unit. This is especially applicable during damper rebuilds.

4.1.2: Force-Velocity

Alternatively, a damper dyno will produce a force-velocity graph. It is similar to the before mentioned force-displacement graph, however the pistons velocity is plotted against the force applied. On a dyno, the velocity range recorded can be predetermined. A larger velocity range will have increased data increments, whereas reduced velocity ranges will have finer data increments.

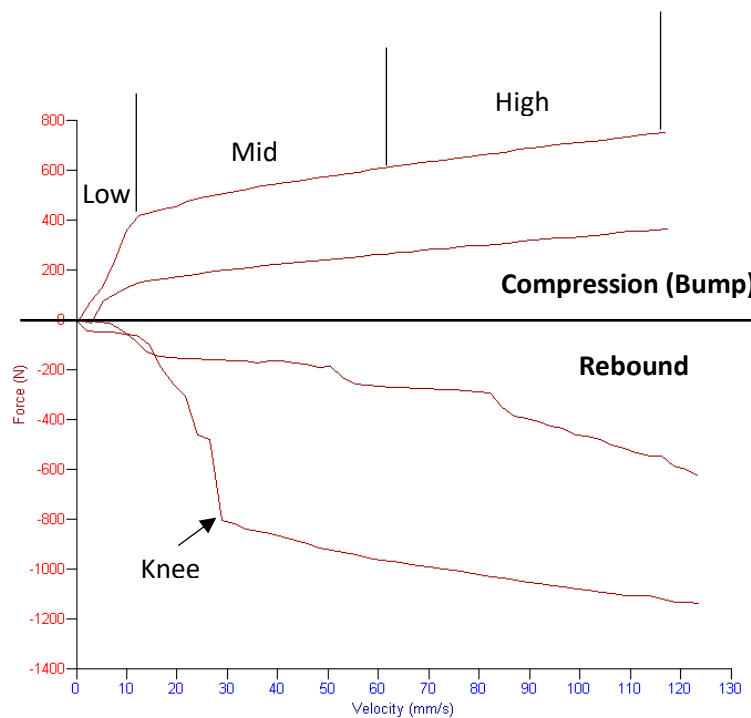


Figure 5 Penske Damper Force-Velocity Graph

Unlike the force-displacement curve, force-velocity curves have two distinct sections. Positive force values depict the damper in bump, and negative force values depict the damper in rebound. Negative force simply implies that the direction of travel is opposite. As described in section 2.1, the traces can be split into the three speed differentials of low, mid and high range, as outlined in Figure 5.

Noticeably the low speed sections of the curves have a significantly different gradient to that seen in the mid and high ranges. The point at which these change is known as the 'knee' of the curve. Due to the varying low and high speed transitions a vehicle will undertake, compromise is required to control grip levels, tyre degradation, and ride (NTT Racing, 2008). Therefore, this two-step curve is engineered into the damper to reduce the amount of compromise required. This provides a suitable basis for a balanced set-up.

Force-velocity are more commonly used to determine damping performance and set-up requirements. Most dampers incorporate bump and rebound adjustment systems. Testing a damper on each different adjustment can be clearly analysed and compared utilising force-velocity graphs. This type of analysis is particularly useful for damper tuning as it visibly shows how each type of adjustment changes the damping curve produced. Gaining an understanding of how tuneable your damper set is, will be a huge advantage for on-track performance. Dialling in a damper tune will improve vehicle driveability, predictability and driver confidence resulting in improved lap times in a motorsport application. Clearly the ability to produce force-velocity graphs from live vehicle logged data could prove pivotal in improving performance, plus providing the possibility of gaining an advantage over the competition.

4.2: Track Testing

The second form of testing is to utilise live track data recorded directly from the test vehicle. Such tests will require the vehicle to be available and a pre-planned test circuit of the users' preference. The following checks should be carried out on the test vehicle before it is taken for track testing:

- Simple mechanical checks i.e. Spanner check, tyre inspection, camber, castor, toe settings etc.
- Pre-booked test track/ track day/ racing event.
- Sensor Calibration! Very important to check that all sensors and logging devices are properly calibrated beforehand. This will reduce set-up time at the trackside, which is particularly important where track running time is limited.
- Ensure correct mounting of dampers. Apply a base set-up ready for immediate testing.
- Outline a test plan. Adhering to a pre-planned test schedule will provide a logical order to testing, and help keep up-to-date with information collected.
- Ensure a computer is available with logger supported software installed for data analysis after downloading track data!

It is important to remember that the purpose of live track testing is to develop a method and understanding to whether data logging equipment can produce damper curves, similar to those output from dynamometers. Therefore, it is credible to assume that the test type needs to be similar to that utilised on a dyno. There are a couple of rules to testing that can be applied to increase data accuracy and complexity. Such rules include:

- Variables. The variables available on a dyno are mostly available on the actual test vehicle. Therefore, utilise the valve adjusters, driver inputs and vehicle speeds to adapt a suitable test (Vaughan, 2011).
- Start with the extremes. In terms of damping, the extremes are considered to be soft and hard bump and rebound. Gain an upper and lower limit before increasing complexity by making fine adjustments (Vaughan, 2011).
- Repeatability. A key aspect to data collection is repeatability. Repeatable data will allow for comparisons to outline whether the same trends are seen. Averages can also then be applied to increase accuracy. Note however, variables such as tyre pressures and track temperature can influence results. Ensure variable differences are kept to a minimum for better results.

Team MCR took the race car to Pembrey race track for testing, from which a general set of track data set was recorded and provided for analysis. This was the first set of live track data provided and is to be extracted and manipulated to start developing a method for producing accurate damper plots.

4.3: Simulation Testing

The UWTSD Base Performance simulator provides a third test type available to this project. Simulation testing is a computer based test type that utilises vehicle models and calculations to simulate the performance as if it was in a real-life situation. Whilst the output data is purely theoretical, based on the models designed into the software, advances in simulation systems have meant that output data is more closely relatable to actual track performance. It has been used to successfully improve vehicle performance, as proven by Danny Nowlan's simulation techniques used for time attack (Nowlan, 2017).

Advantages:

- Less initial set-up time required in comparison to track testing.
- Computerised vehicle changes also speed up between test set-ups, increasing the number of different tests possible in the same time frame.
- Accurate results.

- Good for analysing vehicle behaviour across multiple tracks. An ability particularly useful as it can be used to test if the damper curve methodology developed for data loggers can be used to tune dampers to specific tracks.

Disadvantages:

- Very expensive equipment and would not be a suitable investment for all forms of motorsport.
- Limited to the designer software models implemented into the simulation.
- Simulation results can be difficult to interpret (Xiannong, 2002).

4.4: MCR Drop Test

Lastly, a fourth test involves lifting either the front or the rear of the test vehicle, and allowing it to drop from a height. The purpose of this test is to record suspension data with the onboard data logging equipment, and analyse the results to determine where/how damping occurs. The drop of the vehicle applies a velocity and force to the suspension. The damper will damp out the oscillation of the system which should be picked up on the data recorded. The following method was adhered too, to gained drop test data:

1. MCR test vehicle was set up in an adequate environment with plenty of space.
2. Damper set to either its min, mid or max damping ranges.
3. Tyre pressures checked and equalled. For this test, pressures were at a high 40psi.
4. Data logger is switched on and set-up ready for data acquisition.
5. Vehicle is then lifted. The rear was the chosen end for testing. In the case of the MCR the height was limited due to the limited height the front splitter could drop through.
6. The vehicle is then allowed to drop straight off the lifter. Note, foot protection must be worn when undertaking this task.
7. Data is downloaded from the logger and checked using the Motec data analysis software.
8. Repeat the test for each damper setting.

5: Data Analysis

Completion of testing has provided multiple sets of data from which a method for producing damper plots can be developed. Outlined, will be the methodology for producing force-velocity graphs from logged data, including developments made to refine the data for increased accuracy. A mixture of live track and simulation data has been used to develop the method.

5.1: Damper Data Methodology Developments

Initially the Pembrey track data provided by Team MCR was analysed. As the purpose of the investigation was to understand if such damper plots could be produced from actual track data, this seemed to be the logical starting point. However, it soon became clear that this data set alone would not suffice.

5.1.1: Pembrey Data Initial Analysis

The starting point was to understand the data sets that had been recorded from the on-board data logger during the test session. Multiple variables were logged including tyre loads, suspension position, damper velocities, G forces, lateral forces etc. One variable that data logging doesn't record however, is the damping force. Therefore, it was clear that this variable would require calculation from the remaining logged variables.

Overviewing the data, it was clear that the sensors at the front of the vehicle had either been incorrectly calibrated or failed. This data was insufficient and was immediately disregarded. The rear data was deemed sensible and is the focus for all further analysis and data manipulation developments.

5.1.2: Initial Force-Velocity Graph Developments

Utilising the useable Pembrey track data, initial force-velocity graphs were attempted. As damping force wasn't recorded, suspension force was substituted in its place. This was purely a theoretical exercise to find if a force-velocity curve was possible. Damper velocity was a recorded variable and could be directly used against the force produced. Initial attempts produced graphs as seen in Figure 6.

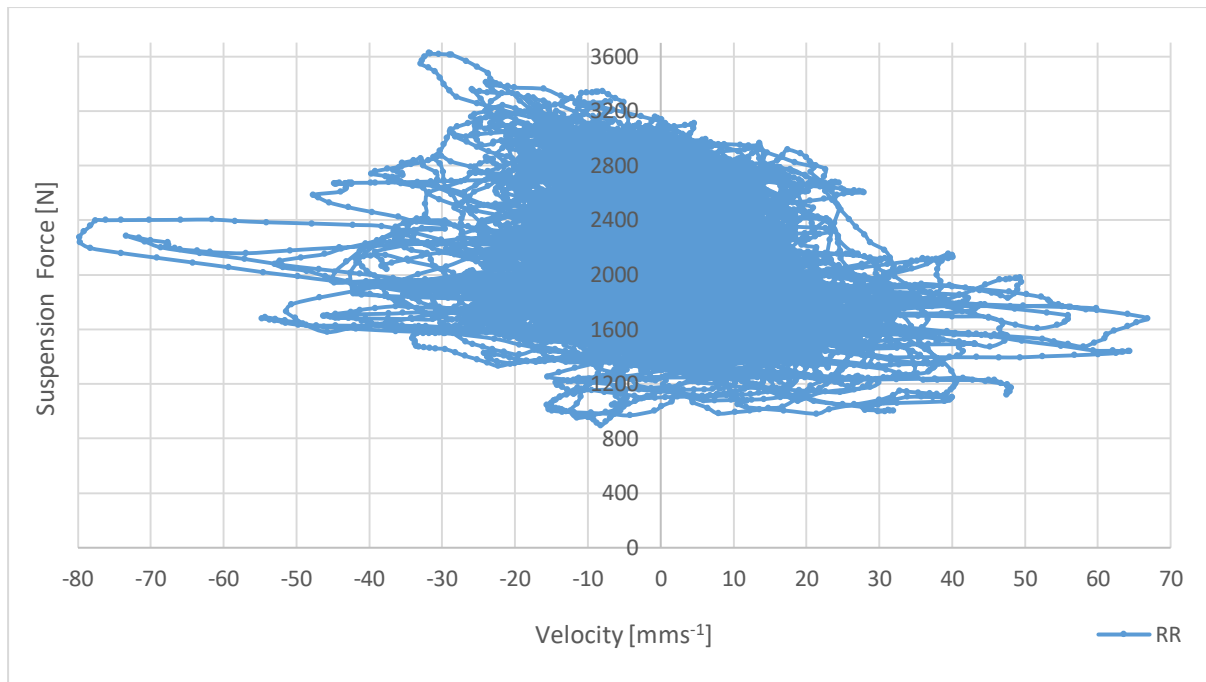


Figure 6 Attempt 1 of a Force-Velocity graph from track data

This graph was produced through exporting the logged data from Motec into Excel as a CSV file. Straight away it's obviously extremely sporadic and does not even come close to producing force-velocity graphs as commonly created from dynamometers. However, there were a few key aspects to this graph that provided insight into where changes were required. Graphical analysis concluded:

- Extremely sporadic data, meaning damper velocity data required ordering to reduce/neglect the wave form caused by the oscillation of the dampers piston.
- Very high forces in terms of damping, therefore suspension force as expected would require other force variables to be neglected to refine the actual damping force seen.
- There is no clear differential between bump and rebound force. Again, data sorting would be the key to separating these out.
- Positive and negative velocity was recorded meaning a distinction between bump and rebound could be found as opposing signs show opposing directions.
- At no point did the force record zero. Therefore, it was concluded that the sensor calibration point wasn't at zero. Data adjustment would be required to find the point at which the direction changes.

5.1.3: Damper Velocity Development

After reviewing the initial graph, the next important step to develop upon was the damper velocity arrangement. Identified as the cause of the sporadic data outlay, it was clear that the velocity data required sorting.

There are several sorting functions available within Microsoft Excel spreadsheet software. These include an array of values from smallest to largest or vice versa (Figure 7). There is also the ability to sort via columns or rows (Figure 8). All data was sorted from smallest to largest when utilising this method.

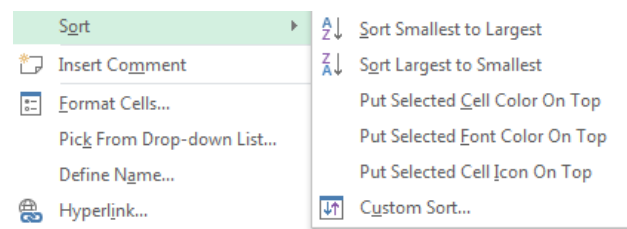


Figure 7 Excel 'Sort' Functions

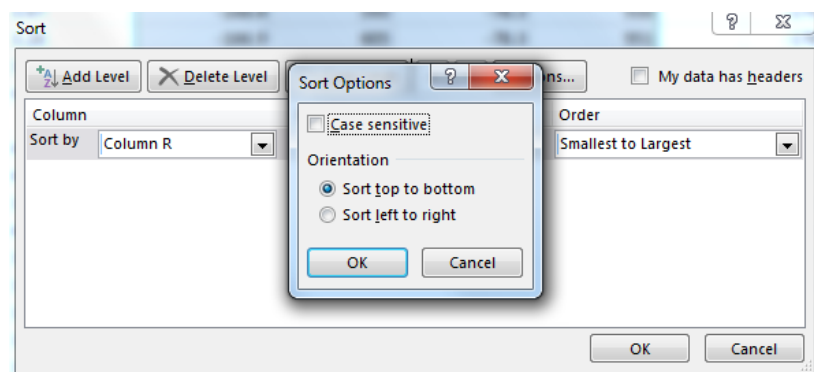


Figure 8 Excel Column and row 'Sort' functions

When first applying the smallest to largest sort function to the damper velocity data, the respective force value was kept with its original recorded value. A graphical representation of the sorted velocity data against suspension force can be seen in Figure 9.

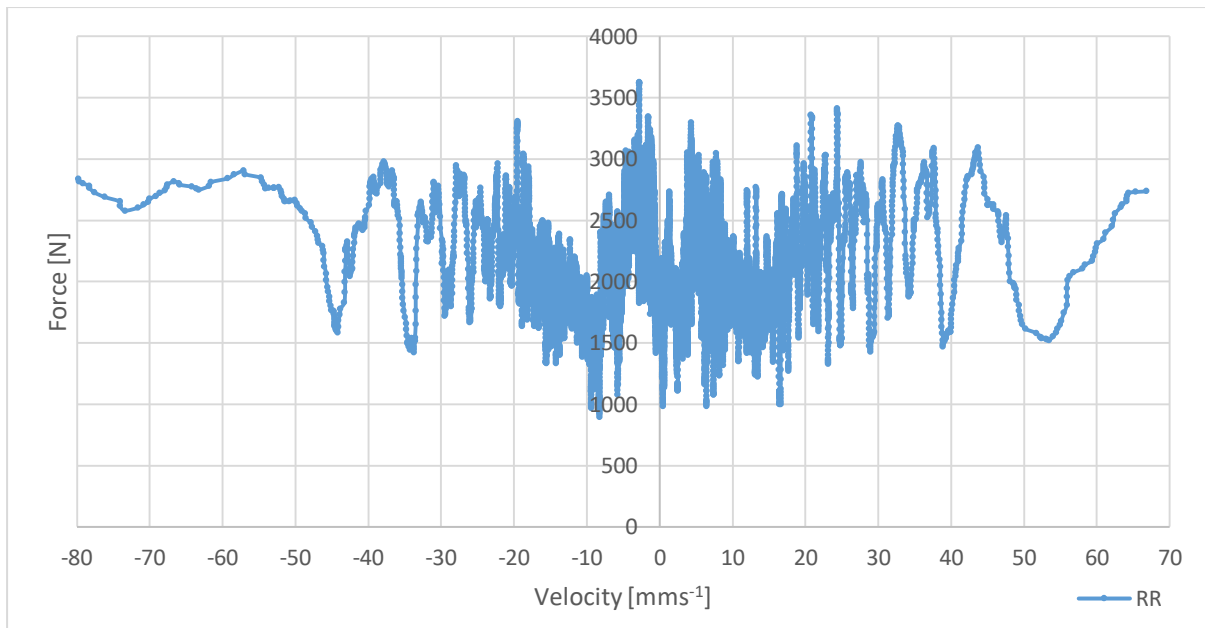


Figure 9 F-V diagram with damper velocities sorted from smallest to largest

Overall this produced a slight improvement over the initial result shown in Figure 6. Now, rather than having sporadic data across the axis, the graph is displaying one continuous waveform. This waveform could not have been produced due to velocity as it is now in an ordered array. Therefore, the waveform displayed must be being produced because of the varying forces recorded at each velocity value. Further development was then implemented on the force values to change the oscillation of the data into linear data.

5.1.4: Linear Force Development

The next step involved refining the force data in an attempt to output damper force values of reasonable expectation, using dyno results as a guideline. It was also required to develop a linear data trace, as would be seen in a damper dyno graph. Utilising the same sorting methods used for damper velocity data, the suspension force data was also sorted from smallest to largest. Originally this kept the related damper velocity value from when it was recorded, however this produced a graph similar to Figure 9 but with the oscillation about the y-axis. Eventually it was found that by sorting both velocity and force in the same direction with neither influencing the other, a linear data trace was possible. Figure 10 displays the output produced using this method for both rear MCR dampers during the Pembrey track test.

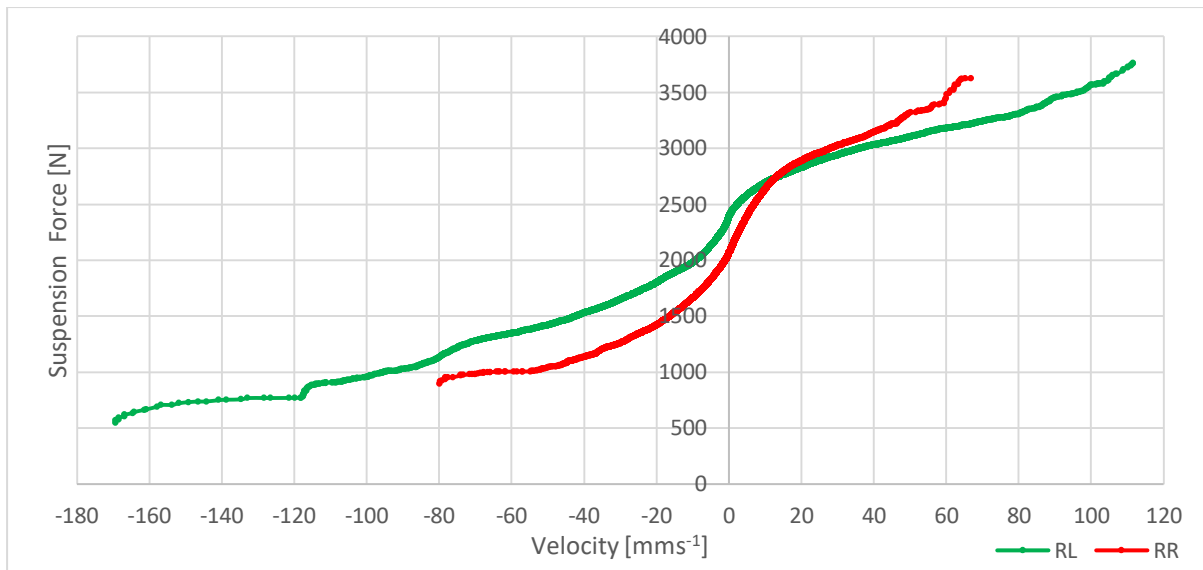


Figure 10 An F-V diagram with sorted force data

Now it can clearly be seen that the ability to develop force-velocity traces from on track logged data is a possibility. A couple of key trace aspects are now visible. Both traces clearly have differing low speed and high speed gradients, with a 'knee' point in-between. Oppositely, a couple of points arose after analysing the development. These include:

- Obviously, suspension force is the total force measured at the tyre from the suspension unit. The actual force produced from the damper a contributing factor to the total force recorded, meaning other force contributing factors need to be negated to gain damping force.
- The data does not centre at zero. Through data manipulation and neglecting of other force variables i.e. static load, the traces should reduce closer to a null centre.
- Dis-similar to a common force-velocity graph, the lower section of the curve is being displayed on the negative side of the y-axis.
- It is still unclear as to which sections of the curves are representing bump and rebound.

5.1.5: Force Manipulation

Suspension force values have till now provided a good basis for developing a sorting method. However, damping force is only a proportion of the full suspension force recorded by the data logger. Therefore, to refine the force data to show damping force, an understanding into what contributed to the total corner tyre force was required.

As a starting point, the force acting at a corner of the vehicle was thought of as one singular system. Expanding upon this, the idea that static measured loads acted as a type of preload on the suspension

occurred. Utilising the MCR data previously collected, the static loads were neglected from each suspension force value.

Before the static corner masses could be neglected from the suspension forces, the corner masses needed to be multiplied by the gravitational constant of 9.81ms^{-1} (Table 2). Providing static corner forces by applying newton's law of, force is equal to the mass of an object multiplied by the Earth's gravitational constant $\{F = mg\}$ (University of Winnipeg, 1997).

Table 2 Static Corner Forces

	<u>Mass [Kg]</u>	<u>Force [N]</u>
FL Corner:	115.6	1134.036
FR Corner:	157.6	1546.056
RL Corner:	227.5	2231.775
RR Corner:	204.7	2008.107

Using the exported suspension force data, the static load relative to the corner of the vehicle, is negated from each recorded value. The idea behind this is to establish whether subtracting static load can provide an alternative direction for force, as all values are currently positive, suggesting the force acts only in one direction which is known to be false. By negating this force, there's an ability to establish a zero point from which the damper compresses and extends.

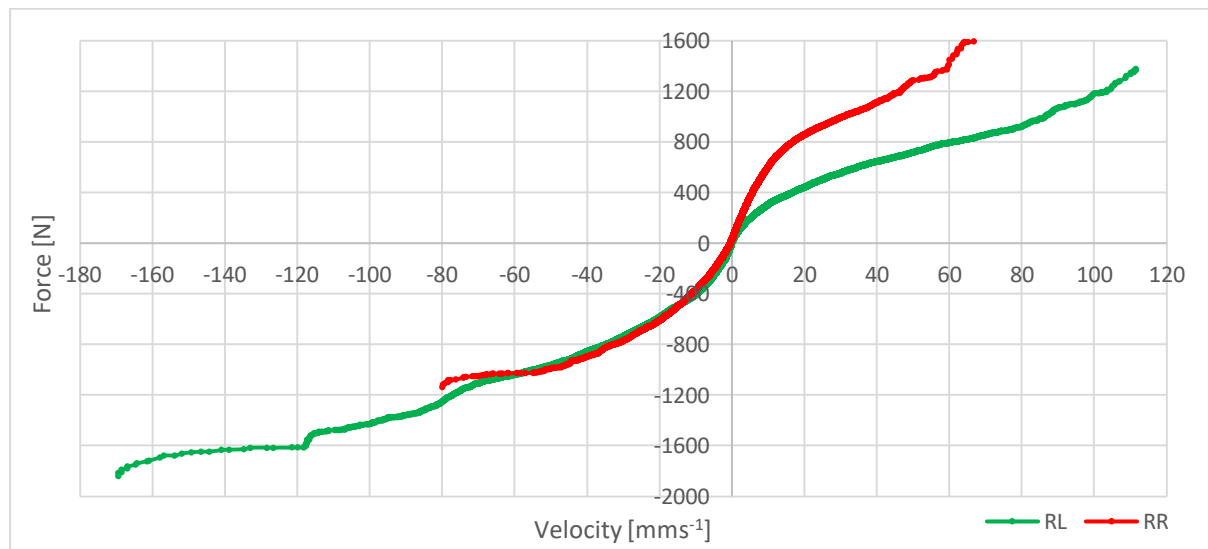


Figure 11 F-V diagram where total suspension force has static corner forces negated

Overall the total effect of reducing all recorded force values by the static load, significantly lowered the data traces as shown in Figure 11 compared with the original results in Figure 10. Importantly the trace shape was retained! Originally the static values recorded in Table 2 were used to reduce the force values, however it was noted that the centre of each trace didn't flow through point (0, 0), but

were close. This arose the question of whether the sensor recorded static load from track testing was different to the workshop tested static load measurements.

As suspected the static load recorded from the data logger was different to those measured in the workshop. The difference was about 100N but it produced a more representative graph as an end result. Figure 11 is the result of reducing values via data logged static loads. The traces are almost perfectly centralised. Figure 12 shows how the static load was negated from the original measured load highlighted in blue. The drag and drop feature in excel enabled the calculation to be applied to the remaining data points.

-169.3	547	-79.8	896	=P6-2385	-1136
-169.3	571	-79.7	916	-1814	-1116
-168.5	576	-79.2	924	-1809	-1108
169.4	595	79.3	926	1790	1096

Figure 12 Measured Load minus Static Load in Excel

An example of how the static load was gained from Motec is shown in Figure 13. Using the tyre load channels in this example, the user needs to find a point at which the data logger is recording whilst the car is standing still, or possibly moving at very low speed when the other is not possible. Values taken for developing the method were gained by moving the blue timing line over the static section of the traces, and recording the force values shown in the top left corner.

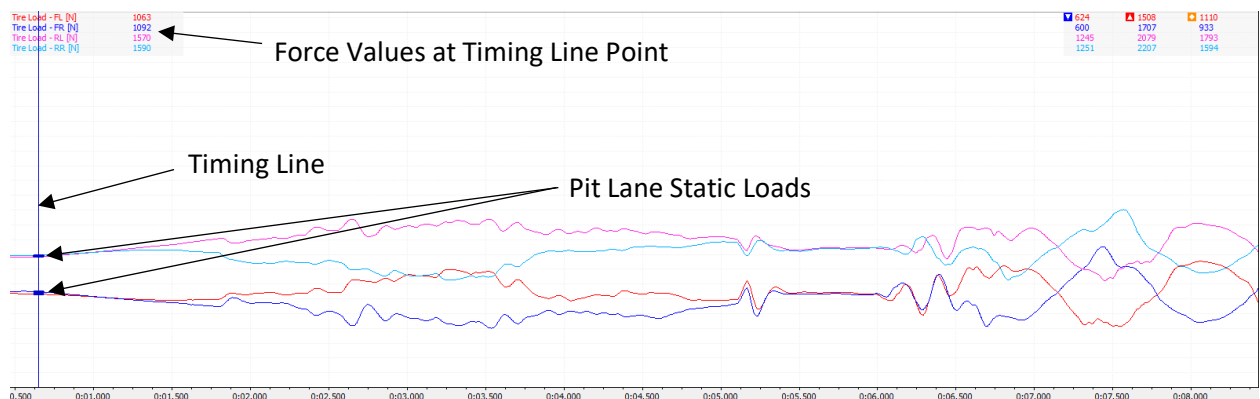


Figure 13 Taking Static Loading from Motec

This was a good step towards finding an end product. The maximum and minimum forces displayed were still large in comparison to results produced from the damper dynamometer. The single system idea now needs to be broken down into component specific variables i.e. spring force and ARB force. Further testing was required through utilisation of the Base Performance simulator.

5.2: Simulation Data Analysis

Due to the limited amount of data recorded from the on-track Pembrey data supplied by Team MCR, further testing was required. The Base Performance simulator incorporates an MCR vehicle model and Pembrey race track into its software. The purpose of using the simulator was to develop an understanding as to whether this technique would show different force-velocity graphs for settings changes applied to the dampers. The data collected from the simulator would also be used to continue refining the force output.

5.2.1: Simulated Pembrey Test Data

Prior to testing, a test plan was outlined (*Refer to Appendix 3*). Following through with the test plan, the simulator was organised and set-up with the MCR vehicle model and Pembrey circuit model. A base vehicle set-up for camber, castor, toe and other such variables was applied (Table 3) Table 3 MCR Simulator Base Settings. The only variable changed throughout testing was the damper bump and rebound settings. Bump and rebound settings were coded into the software in the form of clicks, where 1 was soft and 20 was hard.

Table 3 MCR Simulator Base Settings

	<u>FL</u>	<u>FR</u>	<u>RL</u>	<u>RR</u>
Tyre Pressure [kPa]	95	95	95	95
Spring Rate [Nmm⁻¹]	70	70	80	80
Slow Bump [N/m/s]	1-20	1-20	1-20	1-20
Slow Rebound [N/m/s]	1-20	1-20	1-20	1-20
Fast Bump [N/m/s]	1-20	1-20	1-20	1-20
Fast Rebound [N/m/s]	1-20	1-20	1-20	1-20
Ride Height [cm]	4.4	4.4	5.7	5.7
Camber [deg°]	-3.0	-3.0	-2.5	-2.5
Caster [deg°]	5.0	5.0	-	-
ARB Front [Nmm⁻¹]	Detached	Detached	Detached	Detached
ARB Rear [Nmm⁻¹]	Detached	Detached	Detached	Detached

It was chosen that the tests would incorporate a couple of track runs separated out to change the damper settings. Damper extreme settings of full soft and full hard bump and rebound were the subject for testing. The premise being that the upper and lower limits to the damper capabilities should show clear differences when reviewing their respective force-velocity graphs. If a clear difference can be seen using the analytical method developed, it will provide some initial validity to

the work carried out. A review of the curves against damping coefficient lines will also provide knowledge into the validity of the technique. Finally, utilising Motec's data analysis software, the test data is reviewed to check for any irregularities, and to choose the data set which will provide stable data traces.

5.2.2: Motec Math Channels

So far, all analysis conducted using the live Pembrey track data involved exporting the data into excel before applying calculations to refine the variables. This method is extremely time consuming as it requires the user to applying changes to each data point recorded. The spreadsheet software does have useful functions programmed to make such calculations and changes easier/quicker, but users with limited knowledge of the software would find it frustratingly complex and not intuitive.

Motec's data analysis software includes multiple functions, one of which is the ability to create new channels that display data traces based on the calculation input. This method of applying formulae to the recorded data is much quicker to implement, simpler and visually easier to analyse in Motec compared to Excel. This is helpful when checking that a math channel is implemented correctly. For example, damping force is obtained through the subtraction of other force variables such as spring and ARB forces, therefore whilst the values will change, the trace shape should stay the same. The example is visualised in Figure 14.

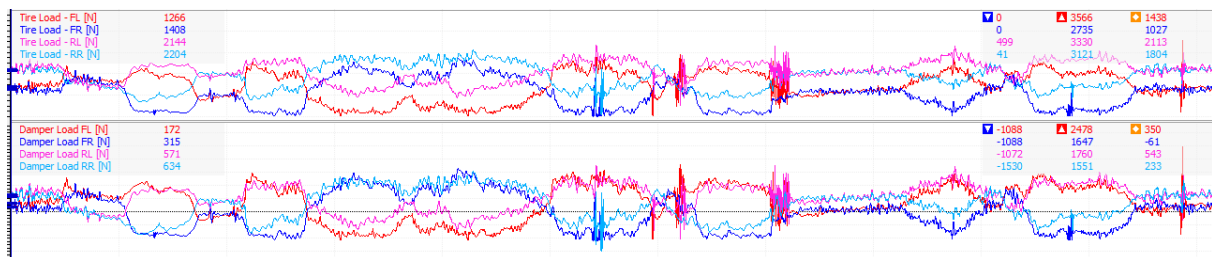


Figure 14 Tyre and Damper data trace comparison

The data analysis software has a function known as 'Math'. This function allows the user to create new 'math channels'. A math channel is used to implement a mathematical function and/or equation. Channels can include new variables, plus those recorded by the data logger. The benefit of using this function is that the resultant calculated data can be displayed, compared and/or analysed on the software worksheets. Figure 15 lists the damper related math channels created to extract damping force from the 'Tyre Load' channels recorded from the simulator testing. A complete list of math channels is available under appendix 5.

☒ Motion Ratio	1
☒ Damper Load FL [N]	('Tire Load - FL' [N]*'Motion Ratio')-'Static Corner Load FL' [N]
☒ Damper Load FR [N]	('Tire Load - FR' [N]*'Motion Ratio')-'Static Corner Load FR' [N]
☒ Damper Load RL [N]	('Tire Load - RL' [N]*'Motion Ratio' [N])-'Static Corner Load RL' [N]
☒ Damper Load RR [N]	('Tire Load - RR' [N]*'Motion Ratio' [rad])-'Static Corner Load RR' [N]
☒ Static Corner Load FL [N]	1088
☒ Static Corner Load FR [N]	1088
☒ Static Corner Load RL [N]	1570.5
☒ Static Corner Load RR [N]	1570.5
☒ Spring Rate Front [N/mm]	70
☒ Spring Rate Rear [N/mm]	81
☒ Spring Force FL [N]	('Suspension Pos FL' [mm]*'Motion Ratio')*'Spring Rate Front' [N/mm]
☒ Spring Force FR [N]	('Suspension Pos FR' [mm]*'Motion Ratio' [N])*'Spring Rate Front' [N/mm]
☒ Spring Force RL [N]	('Suspension Pos RL' [mm]*'Motion Ratio' [N])*'Spring Rate Rear' [N/mm]
☒ Spring Force RR [N]	('Suspension Pos RR' [mm]*'Motion Ratio' [N])*'Spring Rate Rear' [N/mm]
☒ DL FL [N]	('Tire Load - FL' [N]*'Motion Ratio')-'Spring Force FL' [N]
☒ DL FR [N]	('Tire Load - FL' [N]*'Motion Ratio')-'Spring Force FR' [N]
☒ DL RL [N]	('Tire Load - FL' [N]*'Motion Ratio')-'Spring Force RL' [N]
☒ DL RR [N]	('Tire Load - FL' [N]*'Motion Ratio')-'Spring Force RR' [N]

Figure 15 Damper related Motec math channels

5.2.3: Spring Displacement

A problem highlighted from the work completed on the live track data was that the forces were still excessive compared to results output from the dynamometer. Now, the suspension unit for the test vehicle needed to be broken into specific components. Two components that would have a large influence on tyre loads are the springs and anti-roll bars.

In section 3.3: Spring Rate, it was discussed how Hooke's law would apply to the total tyre load recorded. Utilising the Hooke's law formula, spring force was calculated for both the front and rear of the vehicle and entered into Motec as a math channel. The equations were as follows:

$$(\text{Suspension Position Front} \times \text{Spring Motion Ratio}) \times \text{Spring Rate Front}$$

Equation 15 Front Spring Position

$$(\text{Suspension Position Rear} \times \text{Spring Motion Ratio}) \times \text{Spring Rate Rear}$$

Equation 16 Rear Spring Position

Suspension position is a logged data variable from testing measured from the pushrod movement. By multiplying this by the motion ratio we can find the deflection of the spring in relation to the deflection of the pushrod. Thereby obtaining the 'x' variable in Hooke's law. Finally, the formula requires the displacement of the spring to be multiplied by the spring constant/ stiffness 'k'. Figure 16 shows each math channel created for spring force at each corner of the vehicle.

☒ Spring Force FL [N]	('Suspension Pos FL' [mm]*'Motion Ratio')*'Spring Rate Front' [N/mm]
☒ Spring Force FR [N]	('Suspension Pos FR' [mm]*'Motion Ratio' [N])*'Spring Rate Front' [N/mm]
☒ Spring Force RL [N]	('Suspension Pos RL' [mm]*'Motion Ratio' [N])*'Spring Rate Rear' [N/mm]
☒ Spring Force RR [N]	('Suspension Pos RR' [mm]*'Motion Ratio' [N])*'Spring Rate Rear' [N/mm]

Figure 16 Spring Force Math Channels

Lastly, the spring force along with the static load were negated from the tyre load channels. A second variable was mentioned: Anti-roll bar. However, as the simulation testing had the ARB detached from the vehicle, it would not have had any effect on the final results. If the ARB was attached however, the equation required to be implemented into a math channel would be as follows:

$$\text{Suspension Position} \times \text{ARB Motion Ratio} \times \text{ARB Rate}$$

Equation 17 ARB Position

Initial attempts at incorporating the spring force did reduce the total force, but way too much! Generally, it was found that once the spring force was negated, it showed the movement to only be in one direction. This was a similar issue seen previously when initially analysing the recorded tyre load data. This suggested that the spring force also required a static point. To achieve this, the method that incorporated finding a stationary or very slow section of on-track data described in section 5.1.5: Force Manipulation was re-used. This time the stationary position values were taken from the suspension position data. Figure 17 is the stationary section taken from the beginning of the out-lap.

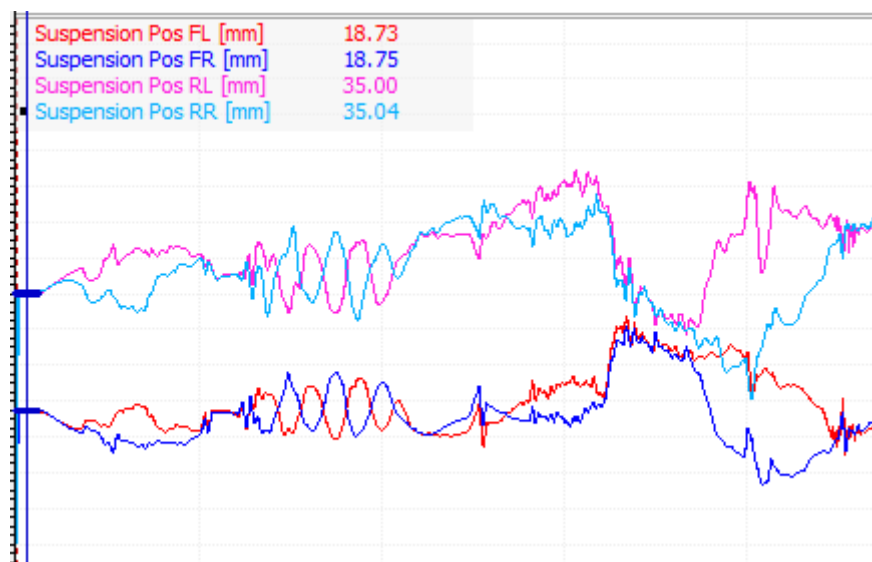


Figure 17 Stationary Suspension Position

Incorporating the static suspension position now provided a central point from which direction of movement could be determined. Finally, the math channels were edited to incorporate this new variable (Figure 18).

☒	Static Suspension Position FL [mm]	18.73
☒	Static Suspension Position FR [mm]	18.75
☒	Static Suspension Position RL [mm]	35.00
☒	Static Suspension Position RR [mm]	35.04
☒	Spring Force FL [N]	$((\text{Suspension Pos FL}' [\text{mm}] - \text{Static Suspension Position FL}' [\text{mm}]) * \text{Motion Ratio}') * \text{Spring Rate Front}' [\text{N/mm}]$
☒	Spring Force FR [N]	$((\text{Suspension Pos FR}' [\text{mm}] - \text{Static Suspension Position FR}' [\text{mm}]) * \text{Motion Ratio}') * \text{Spring Rate Front}' [\text{N/mm}]$
☒	Spring Force RL [N]	$((\text{Suspension Pos RL}' [\text{mm}] - \text{Static Suspension Position RL}' [\text{mm}]) * \text{Motion Ratio}') * \text{Spring Rate Rear}' [\text{N/mm}]$
☒	Spring Force RR [N]	$((\text{Suspension Pos RR}' [\text{mm}] - \text{Static Suspension Position RR}' [\text{mm}]) * \text{Motion Ratio}') * \text{Spring Rate Rear}' [\text{N/mm}]$

Figure 18 Corrected Spring Force Math Channels

This largely improved the issues occurred upon earlier. The data including the new math channels was exported and a force-velocity plot was created.

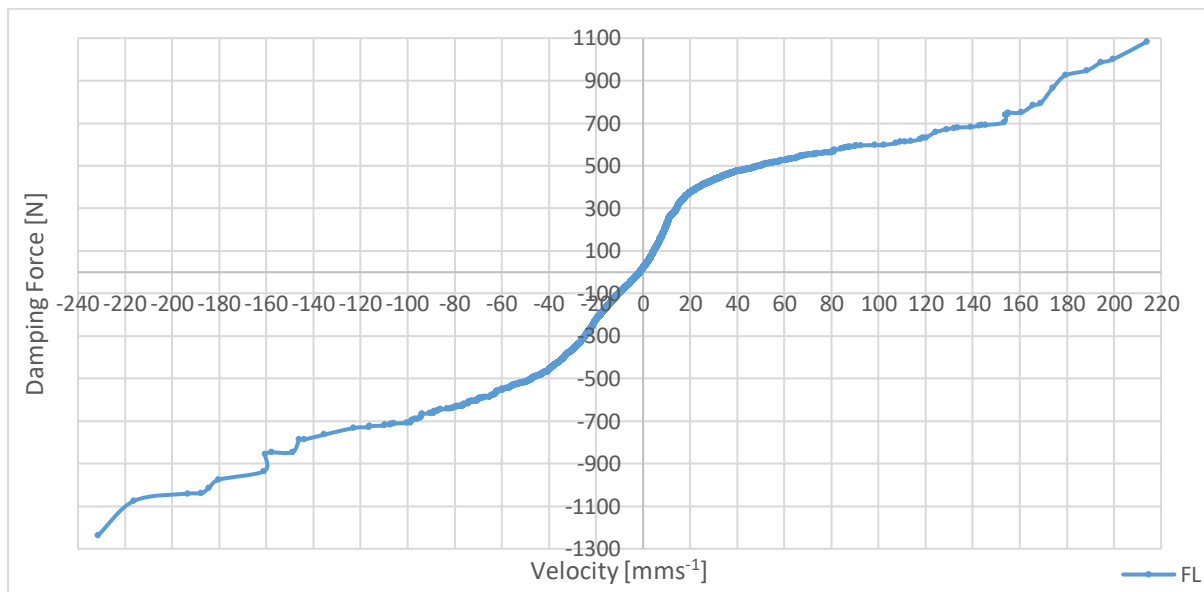


Figure 19 FL Damper F-V Graph with Spring Force Negated

Analysing the graph concluded:

- The trace needed zeroing before comparison graphs can be produced.
- Force values are now very close to expected values.
- Possibly a few anomalous values at the high velocity ends due to unforeseen errors during testing i.e. spins, running off track and/or due to running over extreme curb heights.
- Clear knee point between low speed and high speed damping.
- Clearly shows there to be a greater force application in rebound when compared to bump.

5.2.4: Soft vs. Hard

Simulation testing involved collecting two sets of data. One being for the dampers set to full soft, whilst the second had the dampers set to full hard. The next step required applying the developed force-velocity method to both data sets in the interest of finding whether a distinct force-velocity curve difference would be output.

Using the motec math channels created and the ability to export the final calculated data into excel, force-velocity graphs were produced for each corner of the vehicle. Each vehicle corner had a soft and hard damping curve produced, from which were overlaid respectively. Figures 20 – 23 are the graphical results produced from the exported data.

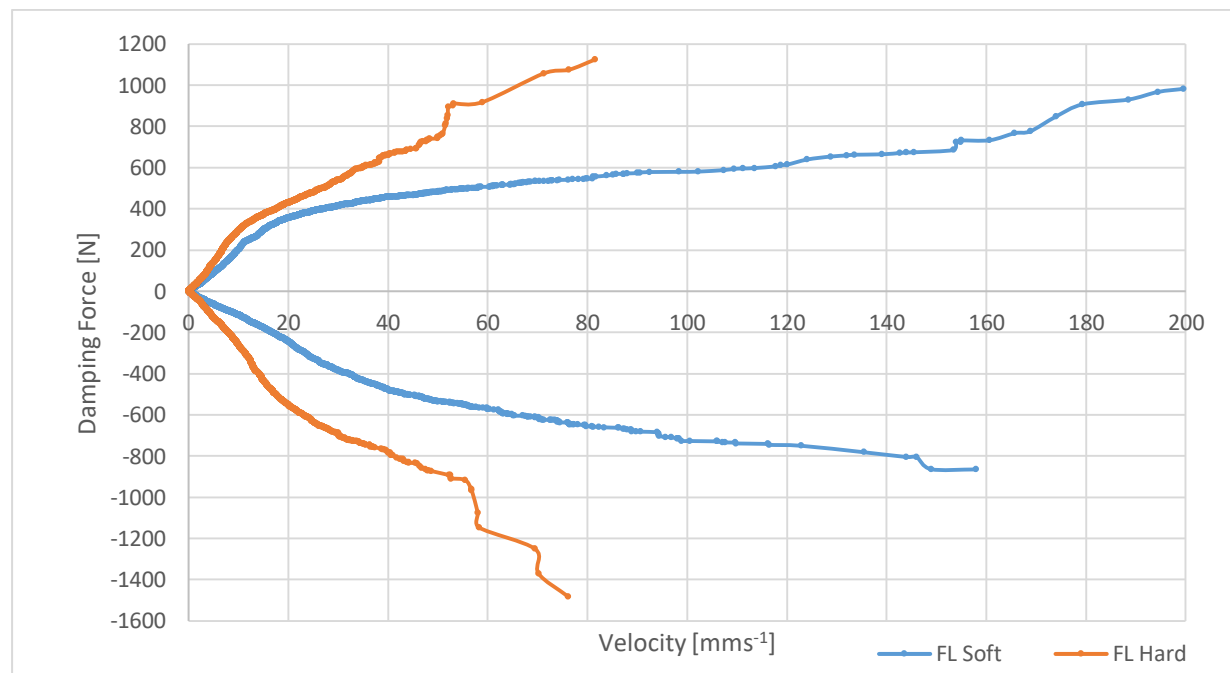


Figure 20 FL Soft vs Hard F-V Graph

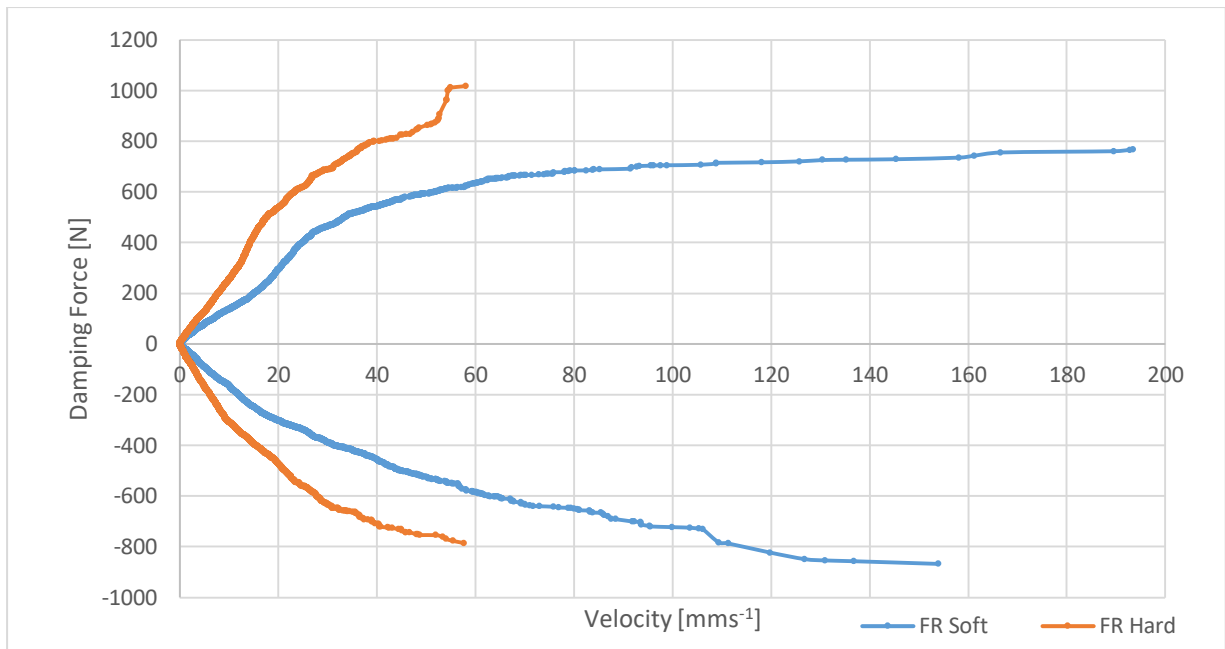


Figure 21 FR Soft vs Hard F-V Graph

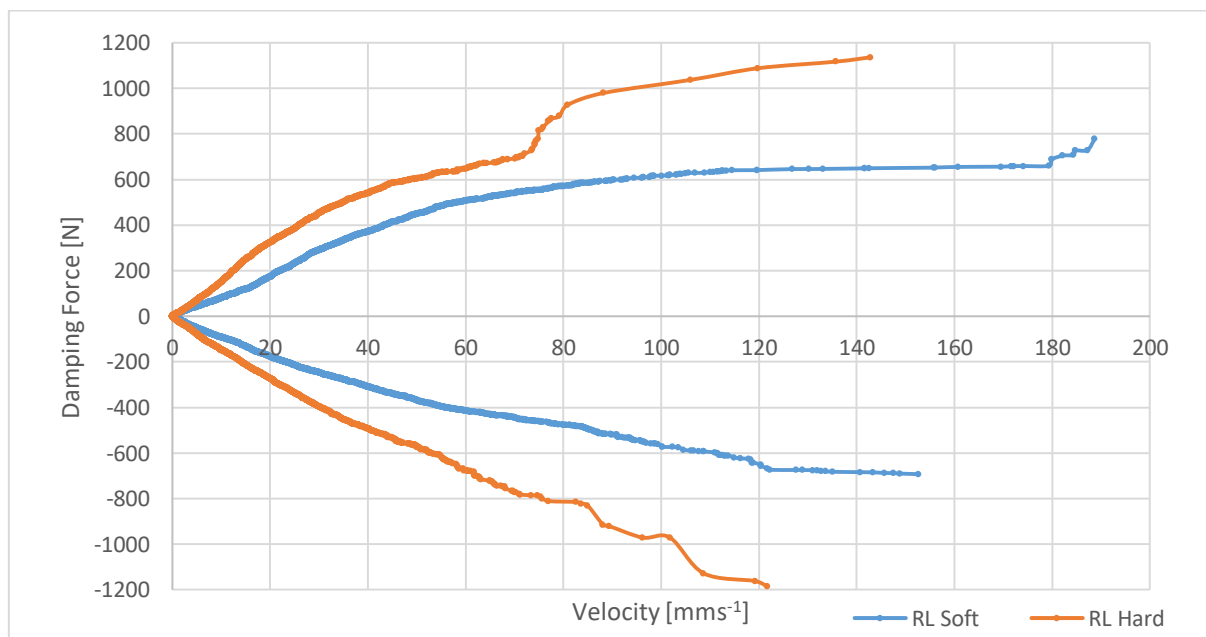


Figure 22 RL Soft vs Hard F-V Graph

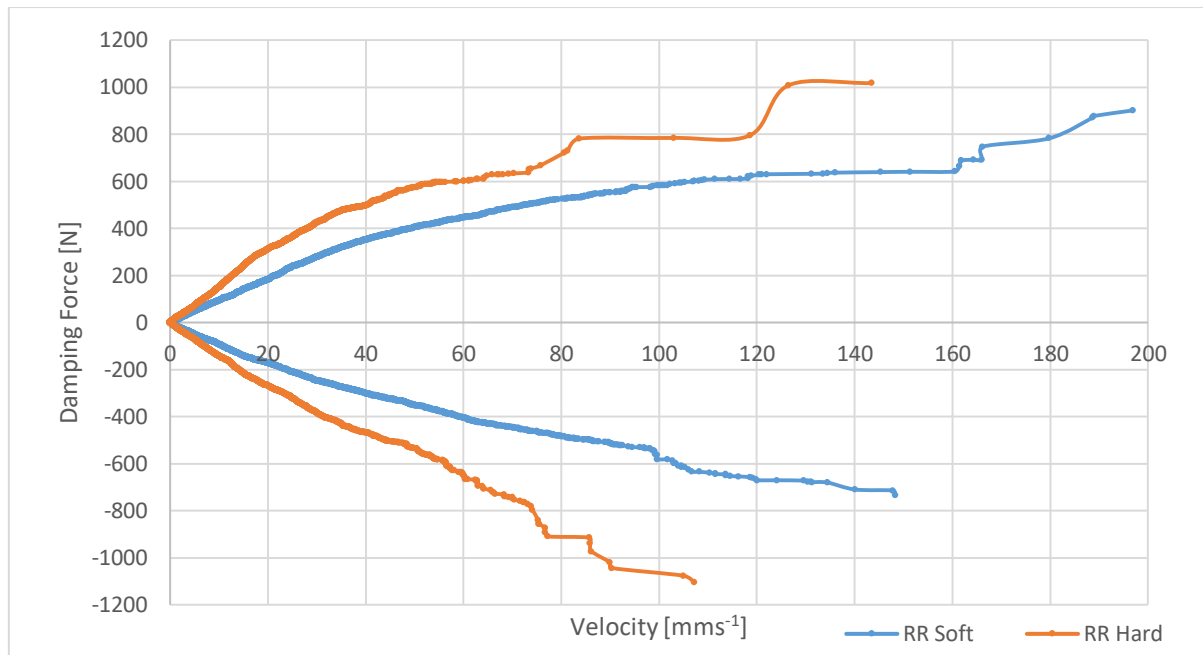


Figure 23 RR Soft vs Hard F-V Graph

The first thing to note is that an extra step was included to represent the graphs in a similar matter to those output from damper dynamometer testing. Throughout the method development process, bump values were shown to have positive damper velocities, and rebound values were shown to have negative damper velocities. Such direction distinction is unnecessary for damping velocity as it can be determined from the damping force axis. Therefore, within excel, each negative damper velocity value was multiplied by '-1' to convert them to positive values. Simply, this allowed the rebound curve to be shown directly under the bump curve.

Through graphical analysis of all four corner dampers mounted to the MCR test vehicle, comparisons could be made front to back, plus corner to corner. Due to the set-up of the vehicle, it was expected that there would be similarities at the fronts, as well as, the rears. The question was asked, is the graphical output consistent? Producing a set of force-velocity curves for each vehicle corner, allowed for some interesting conclusions to be made.

- Greater force curves were generated for full hard damping in comparison to full soft damping. This is as expected, as the damper would have a greater oil flow restriction applied on full hard than full soft.
- The method used can distinguish between setting changes.
- Mostly has a distinguishable change between low speed and high speed damping.

- The 'knee' of the curves mostly occurred at 30mms^{-1} damper velocity, like the dynamometer result. The programmed damper model has an influence here but it is clearly close to the actual results.
- Maximum force values do not exceed an expected range, however could be better. Clearly some more variables are influencing the total force logged at the tyre i.e. frictional forces in the system.
- Both front damper curves (Figure 20 & Figure 21) when compared are similar in shape. They both have similar damping velocities and forces for soft and hard. An expected result due to each front corner having the same components and rates applied.
- The front left hard rebound curve (Figure 20) does show to produce more force than the front right (Figure 21), however the trace after -1000N becomes inconsistent and can be declared as anomalous in terms of the final result.
- Both rear damper curves (Figure 22 & Figure 23) when compared are similar in shape. They both have similar damping velocities and forces for soft and hard. An expected result due to each rear corner having the same components and rates applied.
- The curves tended to become less accurate the higher the damping velocity. A reason for this is that less data is logged at these damping speeds. This could be due to the nature of the track; however, it was found that if more data were logged per velocity unit, a more accurate result was generated. Averages increased the accuracy!

As a final validity check to the method developed, damping coefficients were used to produce a basic force-velocity curve. The closer the curves produced from the method are to the damping coefficient curves, determines the success of the developments made.

5.3: Drop Test Data Analysis

As a final test, a drop test was carried out at the rear of the MCR test vehicle. To recap, the drop test involves raising one end of the vehicle and allowing the vehicle to drop from height, using the data logger to record the signal outputs generated. Testing was conducted initially on a full soft damper setting, before being repeated for both medium and hard damper settings. It should be noted that there is no definitive medium setting, and it was established by adjusting the valves by half the total turns between full soft to full hard. Drop testing the MCR produced some interesting data traces (Figure 24).

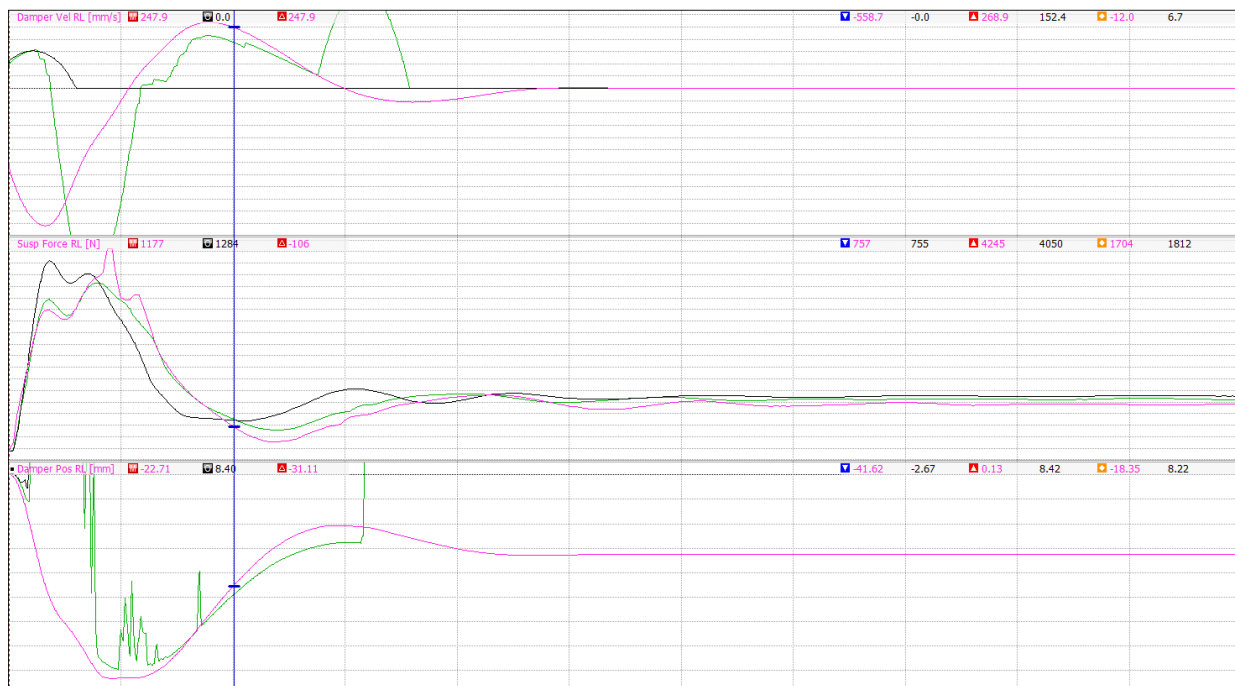


Figure 24 Drop Test Data Traces

As Figure 24 shows, there are three logged channels. The top channel displays damper velocity, middle channel displays suspension force, and the bottom channel displays damper position. On each channel, three data traces are displayed. The pink trace represents soft damping, green trace represents medium damping, and finally the black trace represents hard damping. Using Motec's function to add lap beacons, enabled for each recorded data trace to be set at the point where the car initially shunts into the ground. Following that, each trace was overlaid to provide a direct comparison.

5.3.1: Damper Velocity

Looking at the damper velocity channel, there is a distinct difference in damper compression and rebound movement as damping is increased from soft to hard. Referring to Figure 25 the following conclusion were made:

- It can be seen how the soft dampers followed a sinusoidal wave function as is slowly returned to its equilibrium point
- The medium damper setting also follows a sort of sinusoidal wave function.
- Medium damping produced the greatest peak velocity, however it returned to the equilibrium point in almost half the time compared to soft damping for one oscillation.
- Hard damping produced the smallest peak velocity.
- Hard damping does not oscillate, as it returns to equilibrium and stays.
- Over the complete test velocity reduces the more damping is increased. This was the expected result as the piston has more force to overcome with the same drop height/ speed.

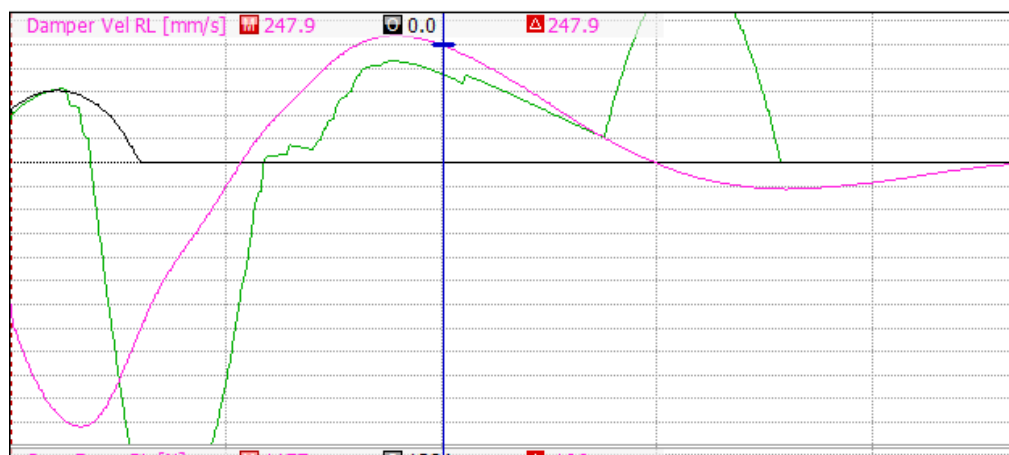


Figure 25 Drop Test Comparison: Damper Velocity

This, at the very least, proves that the data logger is capable of showing the differences in damper outputs that occur when settings are changed. The fact that each test involved dropping the vehicle from the same height, suggests that the velocity and force data manipulated to produce the force-velocity graphs seen in 5.2.4: Soft vs. Hard, have a validity. This is due to how this test shows piston velocity to reduce as damping is increased for a similar force input. Therefore, hard damper force-velocity graphs would show a range of higher forces than soft damping as displayed in Figures 20 – 23.

5.3.2: Suspension Force

To prove that a similar force was applied through the suspension unit, the suspension force channel is displayed. Soft and hard damping have similar force peaks, whereas medium is slightly lower. This suggests that an error occurred in testing, where the car wasn't quite at the same height when dropped. This mostly occurred due to human error in releasing the jack from under the vehicle. However, this error did not have any noticeable effect for analysis purposes.

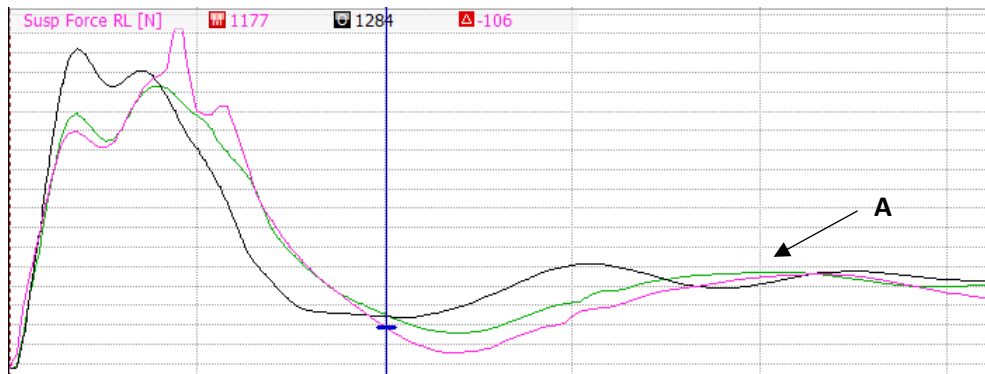


Figure 26 Drop Test Comparison: Suspension Force

Clearly, as the vehicle initially hits the ground it experiences its peak force. All three tests show a slight fluctuation before rather sharply reducing into a smoother damped oscillation. Interestingly at point A on Figure 26 is where damping velocity and position can be seen to have stopped for all settings, even though the suspension force oscillation continues as shown in Figure 27.

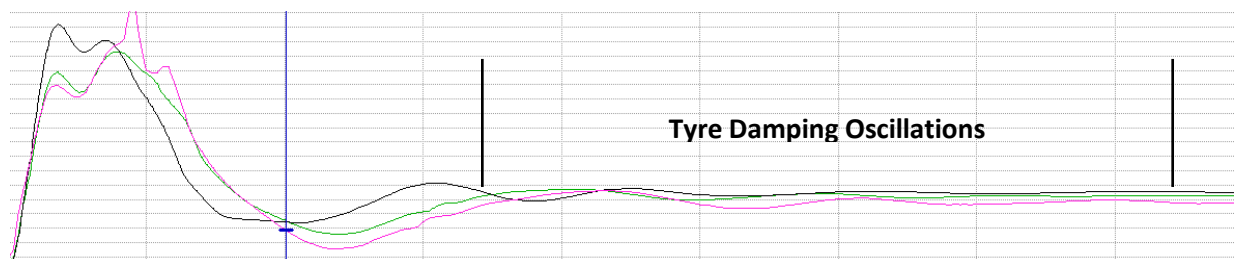


Figure 27 Tyre Damping Oscillation

The only possible conclusion that can be made from this observation, is that the tyres damping effects are also being picked up by the data logger. The traces still show a quicker reduction in suspension force the more damping is increased from point A, but on closer inspection it appears that the damping oscillation recorded on all three tests is similar in frequency. This means that as damping is increased, the system reaches tyre damping effects earlier. Tyre damping effects have not been included in the force-velocity methodology development and would require incorporating to further develop the accuracy of the method.

5.3.3: Damper Position

The final channel shows the damper/spring potentiometer recordings for damper position. Here the amount of movement the damper deflects through is largely affected by the setting applied. Soft damping smoothly oscillates at a large frequency. The medium damping oscillates a little more violently. A theory for this occurrence is that the car may have jumped minutely after the initial bounce off the ground. This cause the wheel to droop/ move freely in mid-air. The trace still shows the position of the damper to reduce in a half wave before returning to its initial resting position. Hard damping barely displayed any damper position. This is highlighted in the red box on Figure 28. The data logged suggests that producing force-displacements graphs would be possible, however the harder damping setting applied, the more data required as this data alone from the drop test would not suffice. The medium data trace also highlights how an uneven surface or jump could significantly disturb the data logged. Therefore, it is suggested that for force-displacement graphs, a steady state lap would most likely increase the accuracy of the output. This is an area highlighted for further development.

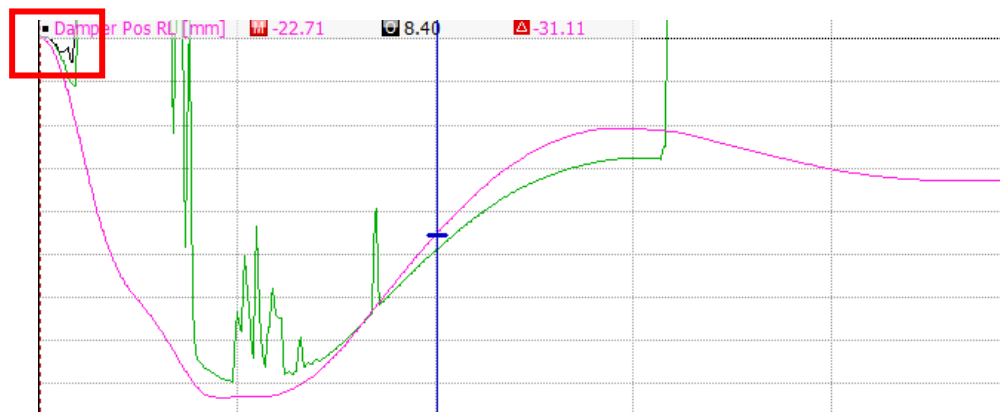


Figure 28 Drop Test Comparison: Damper Position

6: Further Development

The progress made throughout the investigation has constantly enlightened upon other contributing factors to the forces being recorded by data loggers. Naturally, even though a method for producing damper curves was developed, there are several further developments possible. Each development made here on in would improve accuracy and continuing building the model complexity, before defining a simplified method for people to follow. Highlighted issues for progression include:

- Developing a simple model for tyre damping to negate from the suspension forces logged.
- Inclusion of aerodynamic loadings would refine the damper data further.
- Develop a method for damper-displacement graphs. Using similar techniques to develop the force-velocity graphs should provide a good basis.
- Start testing the dampers at smaller incremental setting changes. Use a dynamometer to compare final results.
- More track testing! Initially more track testing could be used to progress accuracy. Eventually through testing at multiple different tracks, it could be determined whether the developed method for damper analysis can conduct damper tuning specific to track layout.

7: Conclusion

An industry striving for the next innovative ideas, where in terms of damper tuning, the first steps have been taken to develop a new type of analysis technique. Damping has been called a 'black art', meaning that you had to be a gifted individual to understand how they worked. The development that has been made over the past 30 years, has clearly dispelled that quote to be nothing more than a myth. Progress made throughout this investigation has begun the process to allowing many motorsport enthusiasts to take control of their own damper tuning track side. Whilst further development is required to refine the techniques built to date, the premise for being able to tune damper for track specific purposes looks to be a promising ability.

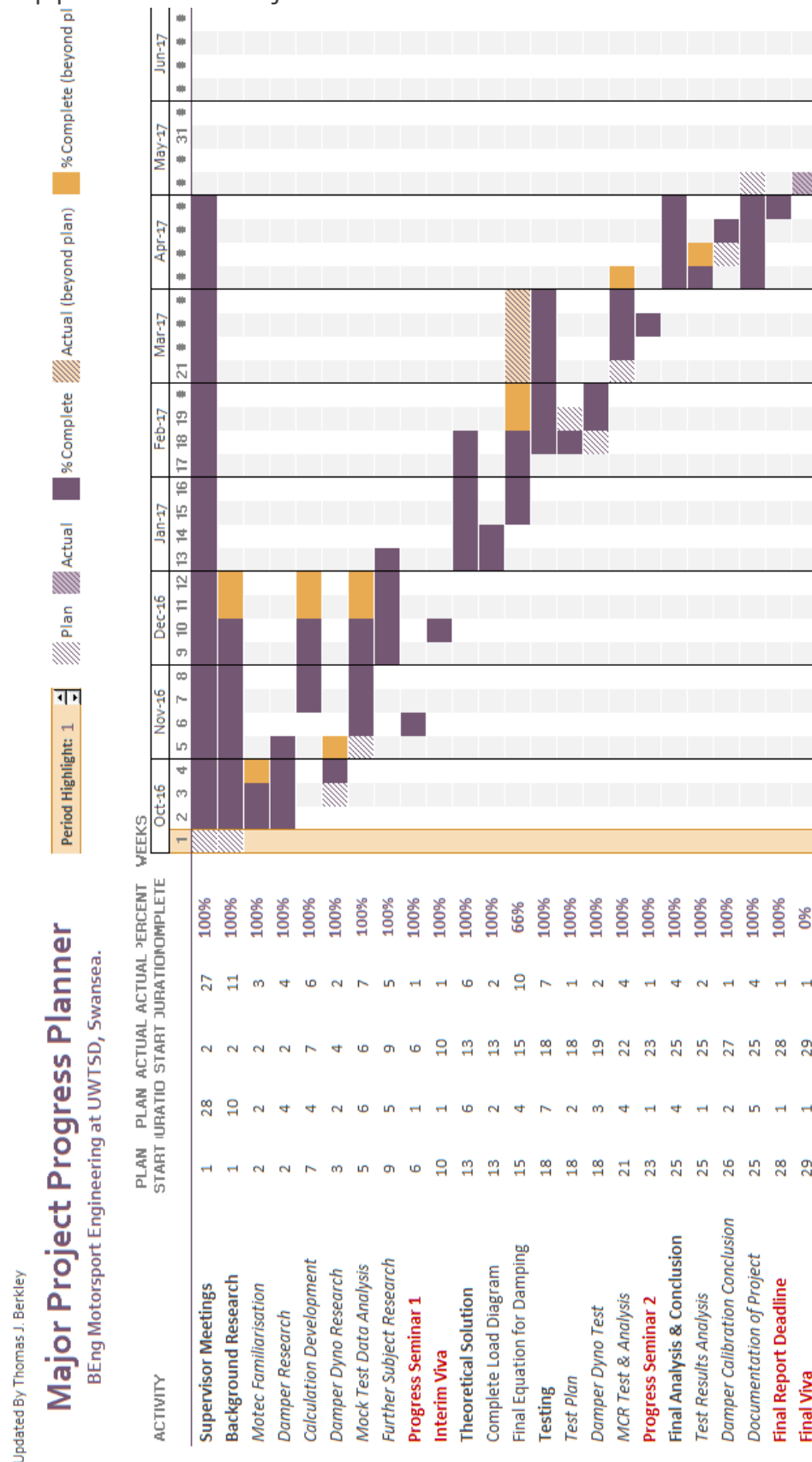
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Appendices

Appendix 1: Project Gantt chart



Appendix 2: Track Test Plan

Driver: Roger Dowden	Track: Pembrey	Vehicle: MCR Sports 2000	Course: BEng Motorsport Engineering
Live Track Testing	Focus: Dampers	Department: UWTSD Pembrey Track Building	

GOALS AND OBJECTIVES

- Acquire track data for damper setting changes.
- Test for damping extremes i.e. low, mid, high.
- Obtain steady-state laps and fast laps for data analysis later.

Test Plan

- Team MCR will set-up the vehicle as required for track warm-up.
- Have driver run for 5 laps warm up on the standard vehicle set-up.
- Run the car on a steady state lap.
- Collect data and determine its validity.
- Repeat steady state run up to three more times. Collecting data after each run.
- Run the car on a fast lap.
- Collect data and determine its validity.
- Repeat fast laps runs up to three more times. Collecting data after each run.
- Make final checks of collected data, and complete any additional runs if necessary.

Appendix 3: Simulator Test Plan

Driver: Tim Tudor	Track: Pembrey	Vehicle: MCR Sports 2000	Course: BEng Motorsport Engineering
Base Performance Simulator	Focus: Dampers	Department: Cadogan Building	

GOALS AND OBJECTIVES

- Acquire track data for damper setting changes.
- Test for damping extremes i.e. low, mid, high.
- Obtain steady-state laps and fast laps for data analysis later.

Test Plan

- Set-up simulator with Pembrey as the track and the MCR car model.
- Have driver run for 5 laps warm up on the standard vehicle set-up.
- Soft Damping: Adjust all bump and rebound settings to full soft for each damper, keeping everything else the same.
- 5 laps of soft damping set-up. Collect data straight after run.
- Medium Damping: Adjust all bump and rebound settings to medium damping, keeping everything else the same.
- 5 Laps of medium damping set-up. Collect data straight after run.
- Hard Damping: Adjust all bump and rebound settings to hard damping, keeping everything else the same.
- 5 Laps of hard damping set-up. Collect data straight after run.
- Make final checks of collected data, and complete any additional runs if necessary.

Appendix 4: Simulation Testing Feedback sheets

<i>Course Title</i>	<i>[Course Title]</i>
<i>Driver</i>	Tim Tudor (Lecturer)
<i>Date</i>	24/02/2017
<i>Location</i>	UWTSD, Swansea. Cadogan Building
<i>Vehicle</i>	MCR Sports 2000
<i>Track</i>	Pembrey

Soft Damper Set-up

Very 'Lurchy'.

Lots of pitch change.

Noticeable balance change on/off throttle.

Transient motions are time consuming.

The vehicle is fine once settled.

Hard Damper Set-up

Less vague. More feedback from the vehicle.

Still a balance change on/off throttle but less than before.

Vehicle was much quicker to react.

Vehicle felt twitchier due to its quicker reactions.

Considerable increase in bumps felt.

<i>Course Title</i>	<i>[Course Title]</i>
<i>Driver</i>	Matthew Harvey (Student)
<i>Date</i>	03/03/2017
<i>Location</i>	UWTSD, Swansea. Cadogan Building
<i>Vehicle</i>	MCR Sports 2000
<i>Track</i>	Pembrey

Soft Damper Set-up

Large pitch. Had to wait for it to settle as it wallowed.

Understeer occurred. Especially on through turns 3/4.

Very throttle sensitive.

Had to use the throttle to bring balance back to the vehicle.

Hard Damper Set-up

Preferred set-up. Much more comfortable and predictable.

Less pitch. Braked, turned, and accelerated more predictably.

Still throttle sensitive but less than the softs.

Initial turn in was less accurate but was more settled during cornering.

Appendix 5: Motec Math Channels

☒	Lat Accel [m/s/s]	'G Force Lat' [G]*9.81
☒	Camber Radius [m]	((('Corr Speed' [km/h]*0.277778)^2)/('G Force Lat' [G]*9.81))
☒	Ackerman Steer [deg]	(3.112/'Camber Radius' [m])*57.2958
☒	Ackerman Pct	'Ackerman Steer' [deg]*(-37.02/5.96)
☒	Understeer	abs('Steering Wheel Pct' [%])-abs('Ackerman Pct' [N])
☒	WTsfrc [N]	((280.704*('G Force Lat' [G]*9.81)*0.0483)/1.632)
☒	WTsrc [N]	((304.096*('G Force Lat' [G]*9.81)*0.0377)/1.5)
☒	WTusf [N]	(49.536*('G Force Lat' [G]*9.81)*0.2888)/1.632
☒	WTusr [N]	(53.664*('G Force Lat' [G]*9.81)*0.2888)/1.5
☒	WTsf [N]	(280000*1.632*'Roll Angle Lat' [rad])/2
☒	WTsr [N]	(280000*1.5*'Roll Angle Lat' [rad])/2
☒	Load Split Front [N]	('Tire Load - FR' [N]-Tire Load - FL' [N])/2
☒	Load Split Rear [N]	('Tire Load - RR' [N]-Tire Load - RL' [N])/2
☒	WTf total [N]	'WTsf' [N]+WTsfrc' [N]+WTusf' [N]
☒	WTr total [N]	'WTsr' [N]+WTsrc' [N]+WTusr' [N]
☒	WTtotal [N]	('WTsf' [N]+WTsr' [N]+WTsfrc' [N]+WTsrc' [N]+WTusf' [N]+WTusr' [N])
☒	Actual WT Lat [N]	('Load Split Front' [N]+Load Split Rear' [N])
☒	WT Total Lat [N]	'WTr total' [N]+WTf total' [N]
☒	Actual WT Long [N]	((('Tire Load - RR' [N]-Tire Load - FR' [N])/2)+((('Tire Load - RL' [N]-Tire Load - FL' [N])/2))
☒	Roll Angle Lat [rad]	(584.8*('Lat Accel' [m/s/s])*0.16507)/((((1.632^2)*280000)/2)+(((1.5^2)*280000)/2))
☒	Actual Roll Angle Front [rad]	tanh((((('Ride Height FL' [mm]-Ride Height FR' [mm])*0.001)/1.632))
☒	Actual Roll Angle Rear [rad]	tanh((((('Ride Height RL' [mm]-Ride Height RR' [mm])*0.001)/1.5))
☒	Magic Number	'WTf total' [N]/('WTf total' [N]+WTr total' [N])
☒	Actual Magic Number	'Load Split Front' [N]/('Load Split Front' [N]+Load Split Rear' [N])
☒	Roll Stiffness Front	(1.632^2*280000)/2
☒	Roll Stiffness Rear	(1.5*280000)/2
☒	Roll Stiffness Total	'Roll Stiffness Front'+Roll Stiffness Rear'
☒	WTslrc Long [N]	(292.4*('G Force Long' [G]*9.81)*0.043)/3.112
☒	WTusl Long [N]	(51.6*('G Force Long' [G]*9.81)*0.2888)/3.112
☒	Roll Angle Long [rad]	(584.8*('G Force Long' [G]*9.81)*0.16507)/(((3.112^2)*280000)/2)
☒	WTsl Long [N]	(280000*3.112*'Roll Angle Long' [rad])/2
☒	WT total Long [N]	-('WTsl Long' [N]+WTslrc Long' [N]+WTusl Long' [N])
☒	Motion Ratio	1
☒	Damper Load FL [N]	('Tire Load - FL' [N]*Motion Ratio)-Static Corner Load FL' [N]
☒	Damper Load FR [N]	('Tire Load - FR' [N]*Motion Ratio)-Static Corner Load FR' [N]
☒	Damper Load RL [N]	('Tire Load - RL' [N]*Motion Ratio)-Static Corner Load RL' [N]
☒	Damper Load RR [N]	('Tire Load - RR' [N]*Motion Ratio)-Static Corner Load RR' [N]
☒	Static Corner Load FL [N]	1088
☒	Static Corner Load FR [N]	1088
☒	Static Corner Load RL [N]	1570.5
☒	Static Corner Load RR [N]	1570.5
☒	Spring Rate Front [N/mm]	70
☒	Spring Rate Rear [N/mm]	81
☒	Static Suspension Position FL [mm]	18.73
☒	Static Suspension Position FR [mm]	18.75
☒	Static Suspension Position RL [mm]	35.00
☒	Static Suspension Position RR [mm]	35.04
☒	Spring Force FL [N]	((('Suspension Pos FL' [mm]-Static Suspension Position FL' [mm])*Motion Ratio)*Spring Rate Front' [N/mm])
☒	Spring Force FR [N]	((('Suspension Pos FR' [mm]-Static Suspension Position FR' [mm])*Motion Ratio)*Spring Rate Front' [N/mm])
☒	Spring Force RL [N]	((('Suspension Pos RL' [mm]-Static Suspension Position RL' [mm])*Motion Ratio)*Spring Rate Rear' [N/mm])
☒	Spring Force RR [N]	((('Suspension Pos RR' [mm]-Static Suspension Position RR' [mm])*Motion Ratio)*Spring Rate Rear' [N/mm])
☒	DL FL [N]	('Tire Load - FL' [N]*Motion Ratio)-Spring Force FL' [N]
☒	DL FR [N]	('Tire Load - FL' [N]*Motion Ratio)-Spring Force FR' [N]
☒	DL RL [N]	('Tire Load - FL' [N]*Motion Ratio)-Spring Force RL' [N]
☒	DL RR [N]	('Tire Load - FL' [N]*Motion Ratio)-Spring Force RR' [N]

[illegible]