

Research and Development of an Active Damping System

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‘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 lays out a background research into spring mass damper systems, their application and calculation. Active damping technology is explained and the benefits of it for and active damper system are stated. Various sensors appropriate for such a system and their operation are listed. Passive damper test results are compared between two different types of damper, a fixed road damper and an aftermarket adjustable sport damper. Details of electrical data transfer systems used in modern automobiles are included and their applications stated. This report provides a platform for further studies into active damping systems, whether that is the implementation of the technology on a vehicle or the study of the effects and potential benefits or downsides of asymmetric damping on vehicle handling.

ACKNOWLEDGMENTS

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1 INTRODUCTION

Active damping enables an engineer to eliminate the need for compromise when determining how the vehicle is likely to behave during transient cornering. When set up right, it can calibrate itself for any scenario and respond to unpredictable variables in a fraction of a second. Which is beneficial to a driver, and more importantly it can maximise grip and aid safety. A damping system designed as a compromise between stiff and soft ride damping will either excel at one and underperform at the other, or be average at both. However the adaptable system can prepare itself to be proficient in both.

1.1 AIMS

- Provide an overview of spring mass dampers and their application
- Show calculations for characterising damper behaviour
- Describe the testing procedure of dampers and discuss the results
- Suggest dampers and electrical components suitable for an active system
- Provide a basic damping system simulator
- Give a platform to begin the construction of an active suspension system
- To indicate how the active system could act to increase handling performance

2 LITERATURE REVIEW

2.1 DAMPERS

Springs have been used on automobiles to improve ride since the days of horse drawn carriages, it wasn't until the early 1900's that a basic damper was introduced between the sprung and unsprung masses to suppress oscillation and control motion caused by lateral and longitudinal acceleration. In the early 1930's, Monroe created a hydraulic shock absorber, which uses resistance due to forcing oil through orifices. This has become the most popular method for damping. (Milliken & Milliken, 1995)

Since then, many adaptations and improvements have been introduced. Improved seals and valves to enable control of piston velocities, adjustable orifice sizes, springs, gas pressure to reduce cavitation in the oil and displacement due to the piston rod. Many of these are used separately in the same systems to control bump and rebound.

2.1.1 Damping Characteristics

Viscous damping theory states that damping force is proportional to the velocity. To achieve this, fluid flow through the damper piston must be controlled by a valve with the desired characteristics. Under extreme conditions such as military vehicles and rally cars, piston velocities can reach speeds of up to 0.5 m/s. In more conventional applications such as road cars, normal piston velocities are generally considered to reach only 0.16 m/s. Fluid flow at small velocities is caused by leaks, which are to some extent unavoidable, however often intentional. Due to the very small areas that cause these leaks, the flow through them is turbulent. Therefore pressure and damper load will vary as the square of the velocity. At high velocities of above 0.16 m/s, flow is likely to become turbulent again. A characteristic unsuitable for suspension damping. Damper characteristics are designed to be as close to the viscous damping line over the required velocity range, as seen in the larger part of the curve Figure 1. This requires the design to incorporate valves which open gradually, along with the valve being designed in proportion to its surroundings. The sharp rise in damper force at low velocities is due to leakage flow. (Donald Bastow, 2004)

Damping is considered linear and proportional to velocity, this adds mathematical convenience and therefore can be implemented into computer simulation easily. When determining the desired damping characteristics for a vehicle, it is more of an art than a science. Engineers will work over a period of damper tuning. This tuning is done with dampers where disassembly and reassembly are

easily done. The process involves assessment and valve changes, which continues until the desired vehicle behaviour is approached. (Donald Bastow, 2004)

When damper tuning takes place, engineers take into account the following considerations with reference to Figure 1. The sharp rise at the beginning of section [a] (below 0.013 m/s), in the rebound curve is mostly due to uncontrolled leakage. But beyond 0.013 m/s up to the “knee point” [1] is mostly due to the controlled leakage. As piston velocities move into the [b], [c] and [d] ranges, the curve gradients are a function of the blow-off characteristics of the valve. These characteristics are controlled by valve orifice sizes and spring loadings in a conventional hydraulic spring damper. At higher velocities, damping forces around [4] and beyond are controlled by orifice throttling effect in the valves. (Donald Bastow, 2004)

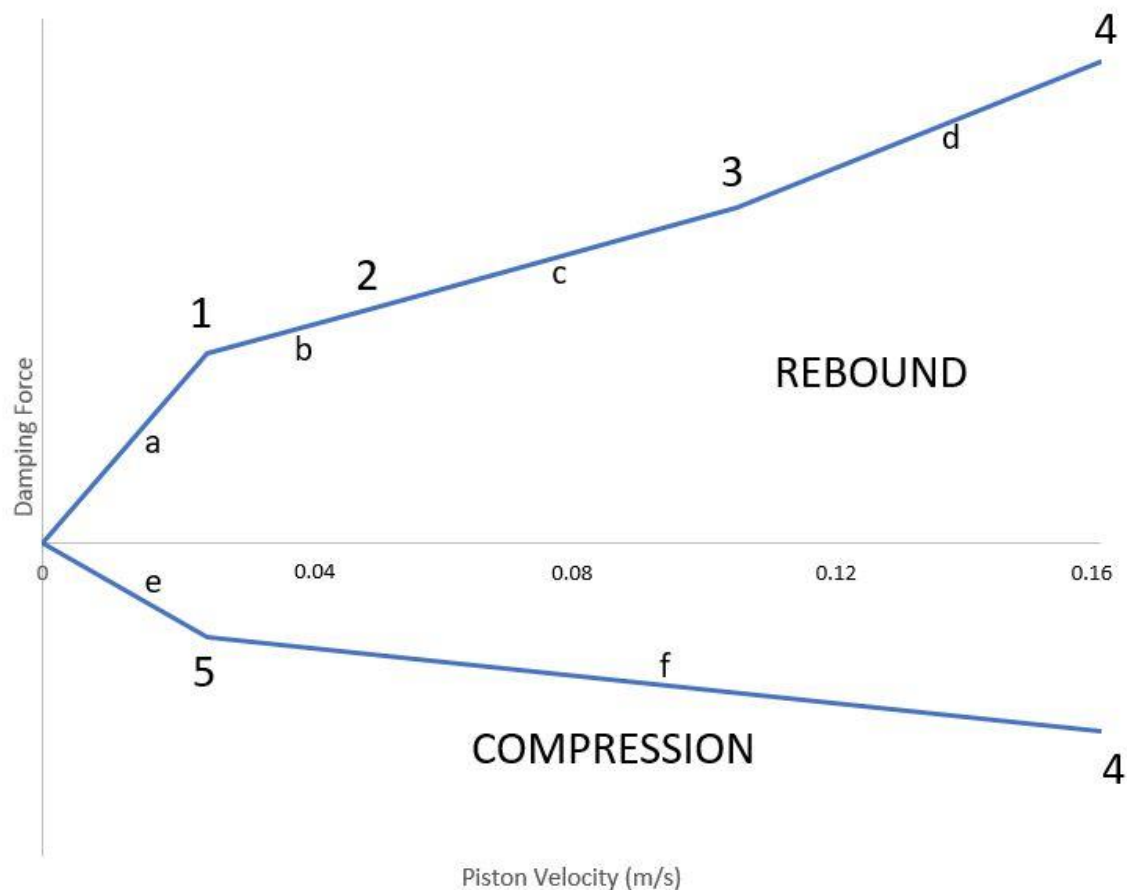


Figure 1 - Characteristic Curves for a Suspension Damper (Annotations discussed in text)

In theory, compression damping is unwanted, as compression damping increases the force required to compress the suspension, thus reducing the ability of the suspension to absorb bumps. However, it is necessary to stop the compression from overshooting, to prevent the suspension from making

contact with bump stops and reduce pitching under braking and accelerating. Also to control roll velocity during transient cornering. If compression damping is too much, the ride of the vehicle will be harsh. If compression damping is too low, this could lead to wheel hop, wheel patter or excessive sensitivity to wheel imbalance. (Donald Bastow, 2004)

If the gradient of [e] is increased, and the height of point [5] raised, this could improve steering response (better turn-in), but result in increased ride harshness. Balancing compressions damping, through [e] and point [5] between front and rear suspensions will alter the ratio of front to rear lateral weight transfer, and therefore the understeer balance through transient cornering. (Donald Bastow, 2004)

Ride behaviour is often referred to as primary and secondary ride. Primary ride is describing how the vehicle suspension reacts to high amplitude, low frequency event, like a large bump. These are often road imperfections that can be seen by a human eye. Secondary ride describes events that involve low amplitude, but high frequency. (Wren, 2012)

The rebound damping alterations have a greater impact on ride than compression damping alterations. The height of point [1] has the largest influence on primary ride. Reaching a balance for front and rear damping at [1] is vital for controlling pitch. If point [4] is too low in relation to lower velocity damping, it can lead to jerkiness in the ride (Jerk is third differential of displacement, therefore the rate of change of acceleration). This is influenced by the gradients along the curve between [b], [c] and [d]. Which would normally be a progressive increase. Therefore, raising the height of point [3] in relation to points [2] and [4] may prevent jerkiness. (Donald Bastow, 2004)

2.1.2 Damping Coefficient and Natural Frequency

Any spring mass damper system will have a damping coefficient and a natural frequency. The damping coefficient is due to resistance in the system. This could be intentional, installing dampers for example, or unintentional, air resistance, gravity, friction etc. If the damping coefficient was zero, the spring would oscillate infinitely with a constant amplitude. Examples of undamped and damped systems can be seen in Figure 2 where 0 refers to zero damping and 1 refers to critically damped (critical damping is described in section 2.1.3). The amplitude of the spring displacement can be seen to reduce as resistance to the oscillation is applied. An undamped system is almost purely hypothetical as it is not possible to eliminate all resistance to motion.

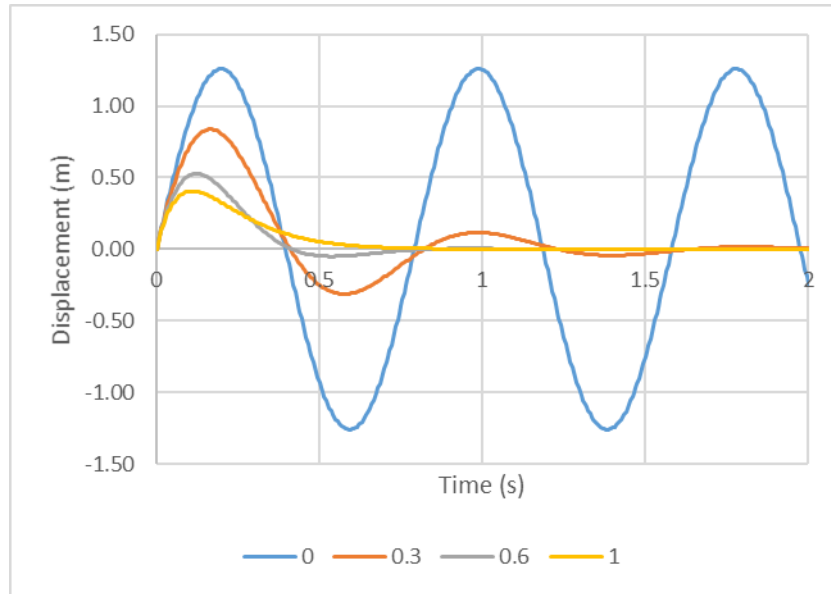


Figure 2 - Varying Damper Characteristics with Varying Damping Ratios

The damping coefficient (C) (N/m/s) of a system can be calculated by

$$C = \frac{\partial \text{Force}}{\partial \text{Velocity}} \quad (1)$$

(Nowlan, 2013)

Natural frequency is the measure of how many oscillations the system will experience without a disturbing force per second. It is measured in radians per second, which can be divided by 2π to give this in cycles per second, or Hertz. This can be determined as an undamped or damped frequency.

Undamped natural frequency (ω)(Hz) can be calculated by

$$\omega = \frac{1}{2\pi} \sqrt{\frac{k_s}{m}} \quad (2)$$

Where

k_s = System spring rate in N/m

m = Sprung mass in kg

(Kasprzak, 2012)

2.1.3 Critical Damping

The definition of critical damping is the minimum amount of damping required after an applied force to prevent an oscillation from passing the point of origin before the force was applied to the spring mass. It is used as a theoretical point of which the damping forces are compared to the critical values.

For the majority of applications, critical damping is far too severe. When tuning for comfort and performance, the best damping falls somewhere between 15-50% of the critical value. Depending on the system and desired ride qualities.

In modern cars, the dampers function is to reduce deflection amplitude over bumps to approximately one fifth of its initial value within two oscillations of the system's natural frequency. (Donald Bastow, 2004)

Critical damping coefficient (C_c) (N/m/s) can be calculated by

$$C_c = 2\sqrt{k_s \times m} \quad (3)$$

(Kasprzak, 2012)

2.1.4 Damping Ratio

As mentioned in 2.1.3, damper characteristics are chosen as a percentage of the critical damping value. This is often referred to as the damping ratio, a value between 0-1 which defines the behaviour of the spring oscillations. A list of general damping ratios and their suitable applications can be seen in appendix A.

The damping ratio (ξ), can be calculated by

$$\xi = \frac{C}{C_c} \quad (4)$$

Where

C = Damping coefficient in N/m/s

(Kasprzak, 2012)

The damping ratio goes to define the type of damping the system will experience, this falls under three categories, overdamped, critically damped and underdamped. Over damped is when the amplitude reduces to zero without any oscillation. Critically damped is when damping ratio is 1, and has been defined in 2.1.3. Underdamped is what the majority of systems experience, where the system will oscillate before returning to the point of origin.

If $\xi = 0 - 1$ the system is underdamped. If $\xi = 1$ the system is critically damped. If $\xi = > 1$ the system is overdamped.

2.2 HANDLING

When engineering projects are undertaken, the engineers set out a number of targets, which indicate the success of the project if these are achieved. However, vehicle handling behaviours are often understood by analysing the behaviour rather than judging it by an objective scale. Therefore, handling behaviour in terms of good and bad has been largely down to judgements made by individuals such as engineers or drivers, in the form of verbal descriptions of handling. Data acquisition goes some way in determining what is happening to the vehicle as these handling characteristics are described. Thus for a given application, the car could be set up to replicate what has been set out by experience to be good handling for that scenario. This experience has been a convergence of various lines of thought. However any scientific determination of what constitutes good handling is rare or non-existent. (Metz, 2004)

During a steady state situation, good handling corresponds with a car that as close to neutral steer as possible. However, dampers are used in vehicles primarily to control transient handling. Good transient handling corresponds with a linear behaviour between steady state values and varied speed and control input amplitude. Constant or near constant damping as speed and control input amplitude increases. Ensuring that yaw and roll resonant frequencies cannot be reached by the drivers, and critical or near critical damping at the frequencies the drivers can reach. (Metz, 2004)

2.3 ACTIVE DAMPING

Semi-active damping is common in modern road cars, where the driver can select a mode such as “sport” or “comfort” for example. These settings would prompt the dampers to adjust themselves to a generally firmer or softer ride for sport and comfort respectively. An active system would operate itself independently without the need for driver input. By having these systems in place, it eliminates the need for compromise in the design of damper configuration, an active damper operates to optimise performance in high and low speed damping, bump and rebound. This provides safety, keeping the vehicle more stable at high speeds and maximise grip during manoeuvres. The ability to maximise grip also allows the vehicle to be driven faster when required. From a business perspective, it is very marketable for car manufacturers such as Audi who sell their cars with their “magnetic ride” option.

During the 1980's and early 1990's, Formula 1 cars had become the most high-tech they had ever been. The first Formula 1 car to use an active suspension was the Lotus Type 79 in 1983, it wasn't for another four years until active suspension system achieved its first race win and nine years before it won its first championship. The late Colin Chapman was the driving force behind the development of the technology. December 16 1982, six months into the Lotus systems development, he died of a sudden heart attack, and the push to get the system performing to a race-winning standard was lost with him. Two races later and Nigel Mansell complaining that it added weight and stole power. The Active system was removed from the car in favour of a conventional passive suspension, with which the car continued to perform poorly. (Howard, 2001)

It wasn't until 1987 that Lotus reinstated active suspension. But by this point Williams had also developed their own system which was out performing the Lotus. The Williams system was designed by Frank Dernie and was technically less advanced but more focused on pure aerodynamic performance and less about driver comfort. (Howard, 2001)

"I didn't give a damn about ride quality. The driver was paid to deal with bumps, all I wanted to do was make the car quicker."

- Frank Dernie

Dernie's conclusion was that in order to maximise aerodynamic performance, it was necessary to have a suspension system that could maintain a ride height that was within millimetres of optimum. Unlike the Lotus design, Dernie's design could not enhance wheel kinematics to aid ride quality and dynamic tyre loading. (Howard, 2001)

Williams described their system as 'reactive', where the ride height was maintained by a system of powered hydraulic connections communicating across the vehicle. The dampers control valves were mechanical, operated by a rod from the suspension arm. This included a mass spring damper setup which works as a mechanical low-pass filter. This prevents high frequency bump reactions, but responds to low frequency scenarios such as car body pitching and rolling. The system was introduced in the latter part of the 1987 season and was successful, however this success could not be replicated in the 1988 season. (Howard, 2001)



Figure 3 - 1992 Williams FW14B (Anon, 1992)

Williams dropped the active suspension for the 1989, '90 and '91 seasons, but in 1992 they returned with an enhanced version of the system. This was so successful that the FIA subsequently banned the technology. (Howard, 2001)

2.4 MAGNETO-RHEOLOGICAL DAMPERS

Magneto-Rheological (MR) or “magnetically responsive” dampers are mechanically very simple by design. This provides reliability due to less small parts within the construction. They also provide a very wide range of damping characteristics with the use of a magnetic field, which takes up very little power, making them ideal for an active damping system.

An MR damper is a simple hydraulic damper, the piston in the cylinder has orifices where the fluid will pass through. Inside the piston is a magnetic coil, which when a current is applied through this coil, the yield shear stress of the fluid passing through the orifices is increased, thus raising the damping force required to resist damper piston velocity. (Lord Corporation, 2007)

The MR fluid consists of a liquid carrier, in which there are micron sized magnetic particles and formulation additives. The carrier is a mixture of mineral oils, synthetic hydrocarbon oil, silicon oil, water and ethylene glycols. The magnetic particles are carbonyl iron, iron/cobalt alloys and nickel alloys, they make up 20-40% of the fluid volume and are typically sized between 1-10 microns. The formulation additives are suspending agents, thixotropes, anti-wear and anti-corrosion additives and

friction modifiers. The formulation will vary depending on the application of the MR fluid. (Lord Corporation, 2007)

When the magnetic field is not applied, the fluid is in an “off” state, during which the fluid will behave as a Newtonian fluid where shear stress is equal to the product of viscosity and shear rate, and is linear. As a magnetic field is applied, the fluid changes from a liquid state, to a semi solid state, which is proportional to the strength of the applied field. It behaves as a Bingham plastic, where the yield stress of the fluid must be exceeded for it to behave as a fluid with linear properties. The yield shear stress is controlled by the applied field and the viscosity of the fluid is unaffected. (Lord Corporation, 2007) Figure 5 Shows the how the particles in the Audi Magnetic Ride system behave when they are in an “on” and “off” state.

Bingham fluid characteristics can be described mathematically by

$$\tau = \tau_y + \mu \times \dot{\gamma} \quad (5)$$

Where

τ = Shear stress

τ_y = Yield shear stress

μ = Fluid viscosity

$\dot{\gamma}$ = Shear rate

(T. Butz, 1998)

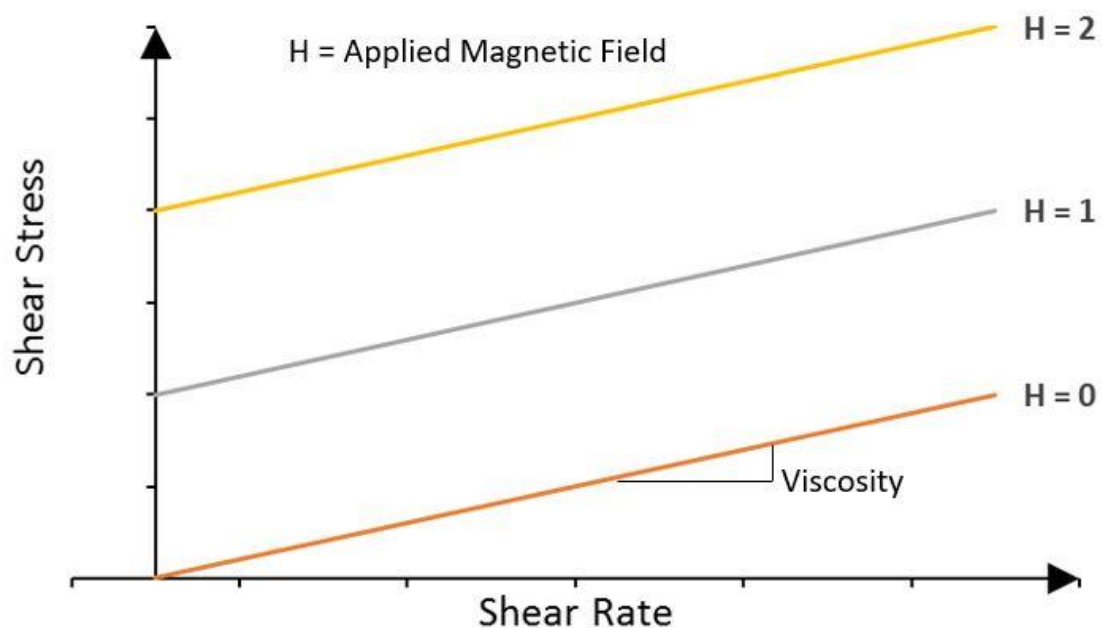


Figure 4 - Bingham Plastic Behaviour with Changing Magnetic Field

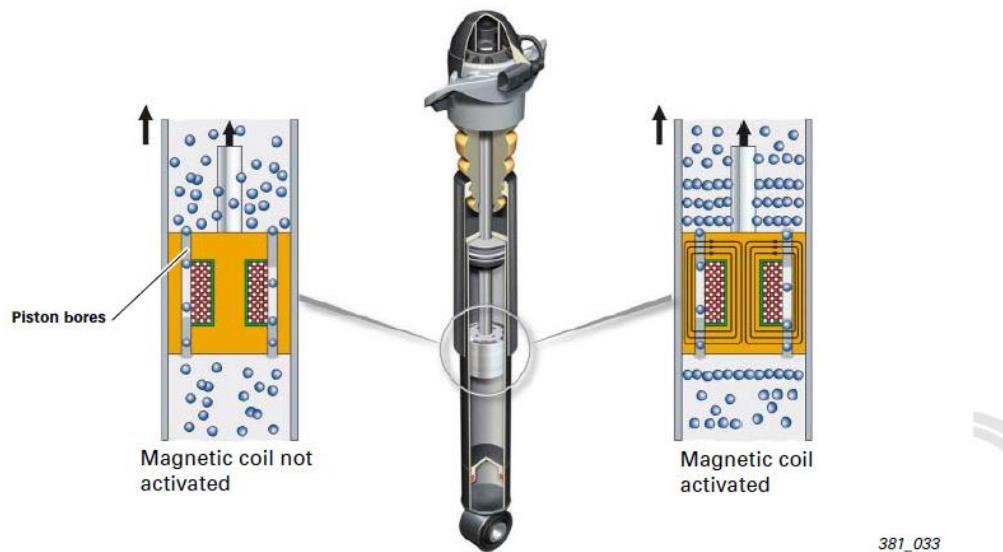


Figure 5 - Audi Magnetic Ride (AUDI AG, 2007)

2.5 ELECTRICAL SYSTEMS

CAN (controller area network) bus systems have been in use in automobiles since 1991, and since then have been used as the standard system in the automotive industry. It is used for various domains of a motor vehicle, some require differing data processing speeds. Therefore it is necessary to have a high-speed type bus and a low-speed bus. This allows for cost saving in vehicle design where the high speed networks are not necessary. The high-speed CAN (CAN-C) operates at 125 kBit/s to 1 Mbit/s. This is used for functions where real-time responses are required, such as engine management and stability systems etc. The low-speed CAN (CAN-B) operates at 5 to 125 kBit/s. These are more commonly used for comfort systems such as seating position adjustments, windows and lighting etc. CAN bus systems are commonly used for diagnostics, where the electronic control unit (ECU) is connected directly to the CAN bus, therefore receives immediate information regarding diagnostics. (Bosch, 2007)

Network nodes in the CAN system generally comprise of three elements. The microcontroller, which runs the application program, it controls the CAN controller, evaluates received data and prepares data to be sent. The CAN controller transmits and receives signals, generates bit streams from binary data to be transmitted to the transceiver. The CAN transceiver amplifies signals and generates voltage for differential data transfer. CAN systems are used to transfer data from sensors to controllers and controllers to actuators. (Bosch, 2007)

Sensors convert physical or chemical variables into electrical variables. The electrical sensors can output signals in voltages and currents. They can also output voltage and current frequencies, amplitudes, pulse widths, phases and cycles or periods of an electrical oscillation, or as electrical parameters, capacitance, resistance and inductance. (Bosch, 2007)

3 METHODOLOGY

3.1 PROJECT

Once the aims of the project had been determined, a plan for the project needed to be prepared, this was done with the use of a Gantt chart, which can be viewed in appendix B and C.

The early stages of the project were purely research based, research was carried out into different types of damping, damping characteristics and active damping methods. This research continued throughout the duration of the project in order to gain a deeper knowledge of specific parts of the technology. Control strategies were reviewed in the early stages but revisited later on in the project.

Testing of dampers was necessary for data collection. Therefore there was a need to be taught how to operate a damper dynamometer and correctly analyse the results. This training was carried out by Paul Davies.

Collaborating with a business was considered early on in the project. This was an Air Suspension company based in Surrey known as Intermotiv. The prospect was to design an active system to work alongside the adjustable air springs, resulting in a vehicle that would have adjustable damping and spring rates. However, upon discussion it became apparent that their aims for the project were not suited to an academic project. So it was mutually decided not to proceed with this partnership.

An appropriate damper for the application was sourced from Lord Corporation, the RD-8041-1 damper, as seen in Figure 6. Lots of information was available for these dampers, operating ranges and control information. They are designed for automotive application and the range comfortably covers any normal forces likely to be experienced by a car, and are very simple by design. Therefore appropriate for this application. Unlike more complex designs from Audi, which are more expensive and harder to control without the use of Audi control units. (Lord Corporation, 2015)



Figure 6 - RD-8041-1 Damper (Lord Corporation, 2015)

Information provided by Lord Corp for the MR damper and the MR fluid can be found in appendix H and I

3.2 TESTING

The car used as a base model for measuring and using as a data reference in the design of the system, and gaining an understanding of vehicle dynamics and simulation was a 2005 Ford Mondeo ST220. In order to gain an understanding of what damping characteristics Ford was aiming to achieve for their high performance model of the Mondeo. It was necessary to test the dampers. The car already had a set of aftermarket adjustable dampers designed by BC Racing intended for track and fast road use, so these gave a good reference for a target range for the active damping system. Road dampers were available uninstalled from the vehicle which made it simpler for testing, once an adapter had been fabricated for the damper dynamometer. The data for the standard road dampers was then obtained by the damper dynamometer. The desired data and spring rates for the aftermarket dampers were provided by the manufacturer, making removing them from the vehicle unnecessary.

In order to accurately simulate damping characteristics, realistic figures need to be obtained to input into calculations. Therefore the car was put on corner weights and the springs were tested to get a standard spring rate. The unsprung masses needed to be taken away from each corner mass, therefore the wheels were removed and weighed. Then the wheel hub assembly masses were obtained by contacting suppliers for the weight of each part, as they store this information for shipping purposes.



Figure 7 - Ford Mondeo ST220 on the Nordschleife (Stevens, 2017)

3.2.1 Mondeo Damper Testing

Due to the damper being fixed from above on the Mondeo, this meant it could not easily be mounted to the damper dynamometer, therefore an adapter needed to be fabricated. The dynamometer measures the displacement and force required actuating the damper at various velocities, in Hz from 1 to 9 Hz. This is outputted to image file force/velocity and force/displacement graphs, and an exportable table of data. The data was the most useful as it could be opened in excel, then could be put into graphs to compare directly with the aftermarket damper graphs.

During testing it was essential to ensure the dampers were at a normal operating temperatures, as the fluid in the cylinders properties will changes with varying temperature levels. Therefore a thermometer was attached to the damper housing surrounding the working area inside the damper (where the piston passes and causes heat from friction). So the damper had to go through a warm up phase until the temperature had risen to 21°C. Then ran through varying speeds, while the speeds set were in Hz, the output to graphs were measured in mm/s.

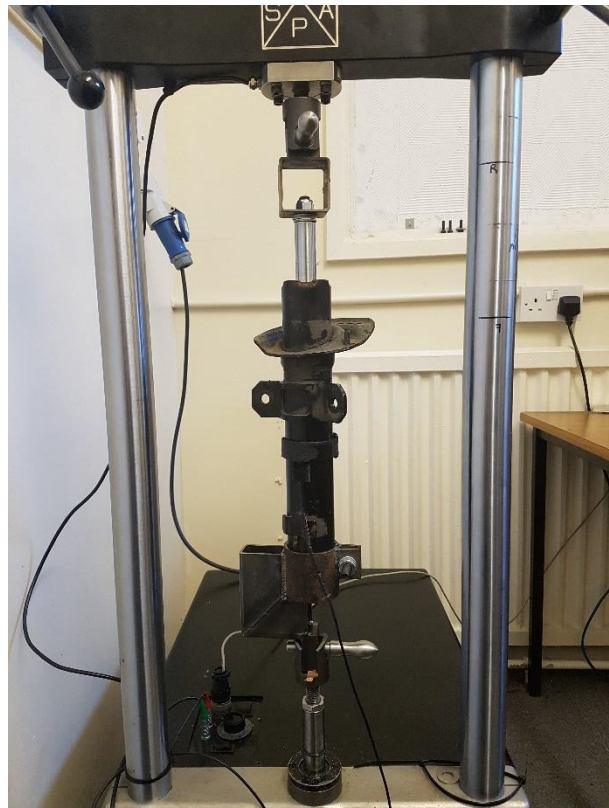


Figure 8 -Standard Mondeo Damper in Dynamometer

3.2.2 MR Damper Testing

In order to accurately simulate the damping levels of the MR damper, it would be beneficial to obtain an MR damper to test on the damper dynamometer. This was not possible due to the high cost of purchasing this technology. Therefore data obtained from tests carried out by other individuals had to be used.

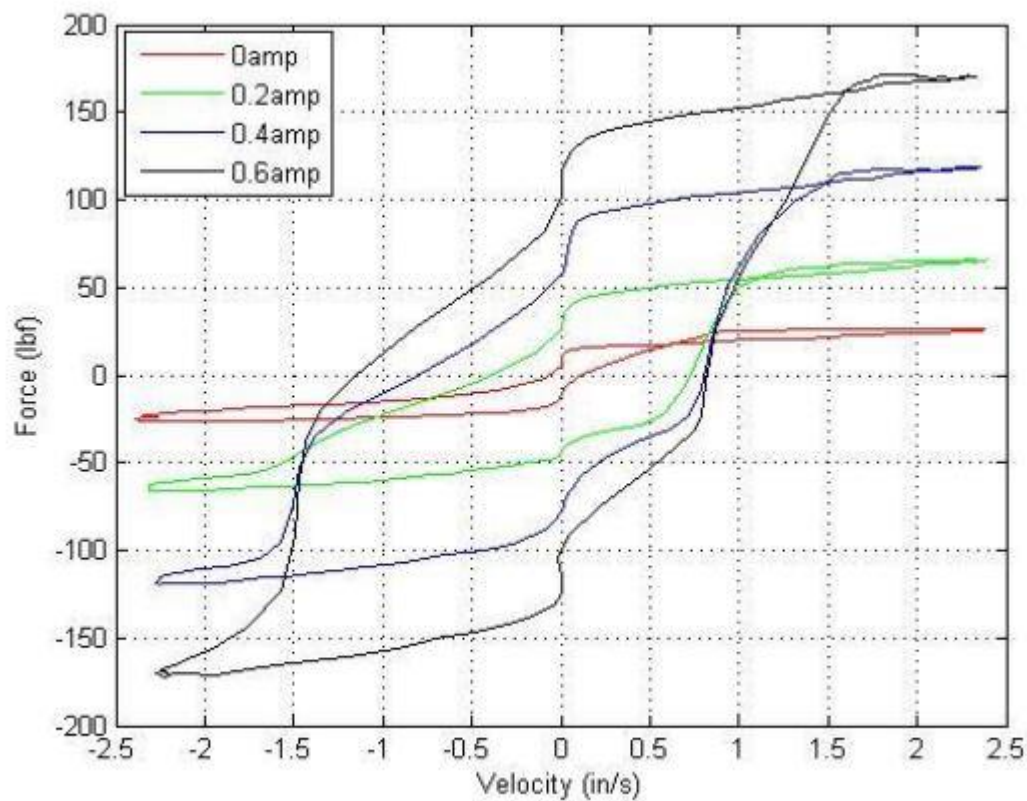


Figure 9 - Force / Velocity Graph of an MR Damper with Varying Electrical Current (Suresh, 2015)

The data collected in Figure 9 shows the performance of the damper under varying electrical current input. It shows the damper is reasonably predictable and linear in the response to progressing current levels, with the exception of no current being applied. It also behaves the same regardless of whether it is in compression or rebound. This allows a pattern to be identified and then predicted through the entire MR damper range. This was done by converting the units and inputting the data into Microsoft Excel, this can be seen in Appendix D. To get the force values from pound force (lbf) to Newtons, multiply the value by 4.44822.

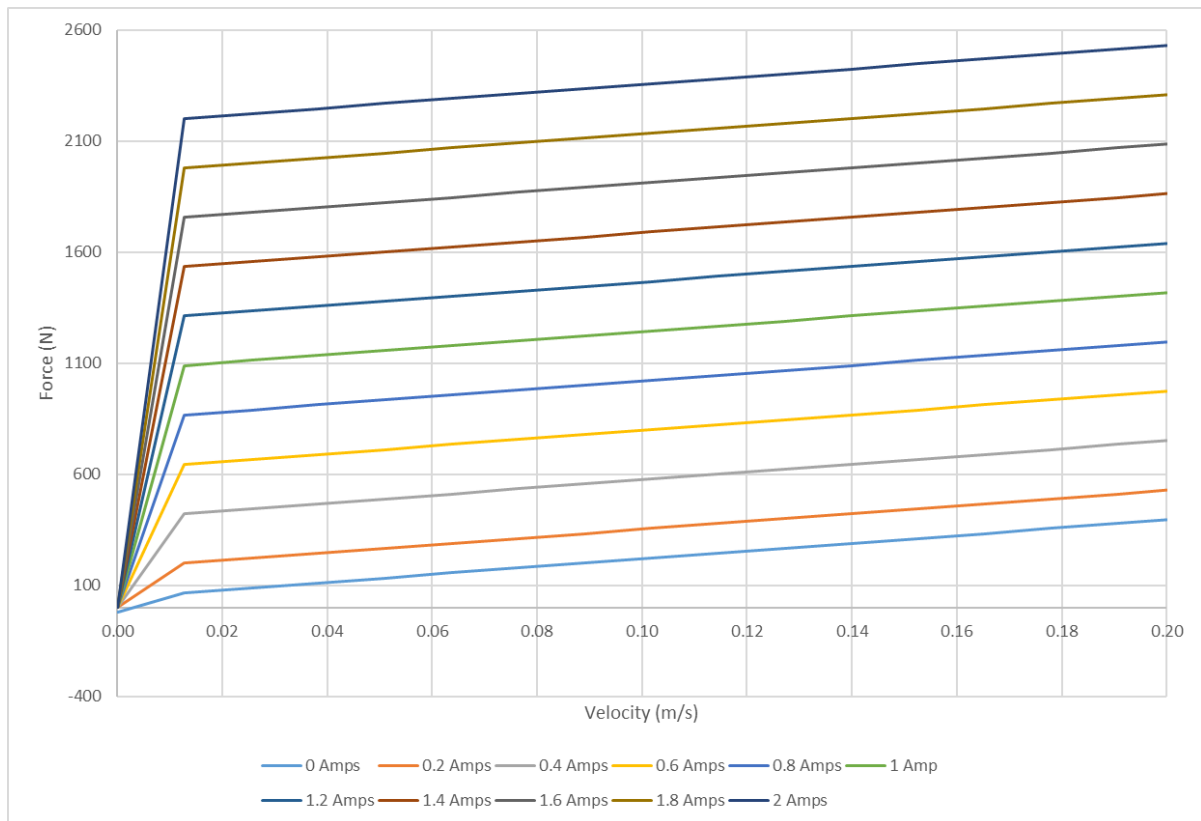


Figure 10 - Predicted Force / Velocity Graph for MR Damper with Varying Current

Having real results obtained under specific test conditions designs for this application would be ideal, however given the circumstances, these results will be used. The Lord Corporation MR damper states a maximum damping force of 2447 N at 0.05 m/s, similar to the predicted maximum. Then a minimum force of 667 N at 0.2 m/s, not a long way off the predicted minimum. This is within the operating range of 0-2 Amps. So this gives confidence in any references made from the predicted results. (Lord Corporation, 2015)

3.2.3 Aftermarket Damper Testing

As stated, the data for the aftermarket dampers was provided by the manufacturer, therefore there was less need to test them on the dynamometer. However it was intended to test the vehicle over a series of manoeuvres on a race circuit at different damper settings. The benefits of this would be to create a plan of repeatable tests (such as a moose test) with varying damper settings, to give a quantifiable results for handling ability. Which would also test and prove the research carried out in section 2.2. However due to lack of budget and circuit availability this could not be completed.

3.3 VEHICLE CONTROL

3.3.1 Microsoft Excel Spring-Mass Damper System Simulation

In order to judge vehicle control, and spring-mass damper system simulator needed to be created in Microsoft Excel. By using values for mass, damping coefficient and spring rate, and inputting a starting velocity (impact velocity), the velocity (ω) and acceleration (α) of the system can be calculated by

$$\alpha = \frac{C}{2m} \quad (6)$$

And

$$\omega = \frac{\sqrt{4mk - C^2}}{2m} \quad (7)$$

Where

C = Damping coefficient in N/m/s

m = Sprung mass in kg

k = Spring rate in N/m

(Oliveira, 2010)

Table 3.1 shows calculated results using example values, for demonstrative purposes. With the time discretisation to be used for the time (t) column in Table 3.2.

Table 3.1 - Example Results for Spring Mass Damper Simulation

Parameters			Units
Mass	m	8	kg
Damping Coefficient	C	2	N/m/s
Spring Rate	k	50	N/m
Velocity	ω	2.496873	Rad/s
Acceleration	α	0.125	Rad/s ²

Initial Conditions		
Starting Position	u_0	0
Starting Velocity	v_0	1
Time Discretisation		
Δt	0.04	seconds

Once these values have been determined, they can be input into a table to plot the damper position with respect to time ($u(t)$) on a graph. $u(t)$ can be calculated by

$$u(t) = \left[u_0 \cos(\omega t) + \frac{u_0 \alpha + v_0}{\omega} \sin(\omega t) \right] e^{-\alpha t} \quad (8)$$

(Oliveira, 2010)

Sign definitions for equation (8) can be found in Table 3.1.

Table 3.2 shows example displacement given from the results of equation (8) when the data used from Table 3.1. To plot a graph to show the displacements of the damper against time, more rows will be required, but Table 3.2 is to give an example.

Table 3.2 - Example Displacement

t (seconds)	u(t) (m)
0	0.00
0.04	0.04
0.08	0.08
0.12	0.12
0.16	0.15
0.2	0.19
0.24	0.22

To view this in graphical form, a scatter plot is used to follow the data trace. With time values along the x-axis and the displacement on the y-axis, this allows the viewer to analyse the amplitude of the oscillations reducing as the motion is damped.

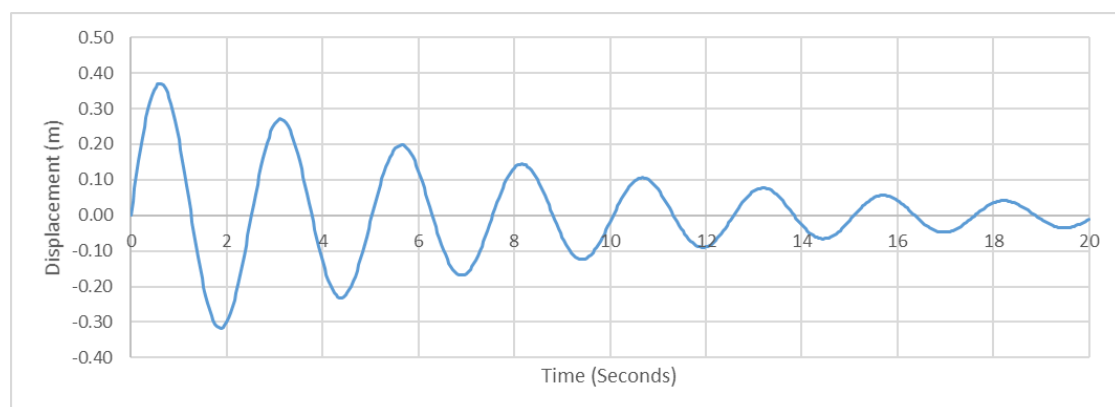


Figure 11 - Example Damped Spring-Mass System Displacement/Time Curve

Once the damping simulator has been built, it can be developed further, to incorporate the measured parameters of the real vehicle in use. Using the equations (2), (3) and (4), the parameters can be set to the measured data, then from this the natural frequency of the system can be calculated, the critical damping coefficient determined. Then by a transposition of equation (4), a damping ratio can be inputted to give the damping coefficient required to achieve the desired damping behaviour. These can be seen in Table 3.3.

If a value of 1 or more is entered into the calculator for damping ratio, the maths encounters an error, so values do not display, however it is not appropriate for this application to have a critically or overdamped system, so this is not necessary. If the user requires a critically damped simulation, they would have to input a number as close to 1 as possible, 0.999 for example. From a visual perspective, this will display the same curve as a critically damped system to the human eye.

Table 3.3 - Example Values Including Damping Equations

Parameters			Unit
Desired Damping Ratio		0.05	-
Sprung Mass	m	8.00	kg
Damping Coefficient	C	2.00	N-s/m
Spring Rate	k	50.00	n/m
Velocity	ω	2.50	m/s
Acceleration	α	0.13	m/s ²
Natural Frequency	f	12.58	Hz
Critical Damping	C_{cr}	40	N-s/m

3.3.2 Simulink Modelling

The configuration of a system that individually alters the current given to each damper separately, according to a desired handling characteristic requires a controller, and a mathematical process to occur within that controller. That mathematical process can be managed in Simulink. The system can read data provided by various sensors on the vehicle and depending on those readings, can alter the damping coefficient and handling in response to the input. The simplicity of the MR damper actuation means that the controller could select a desired damping ratio, and then as the damper will have set currents for damping coefficients, the controller could have these currents associated with a damping ratio. This means the controller just has to have a value of 0-1 to output to the damper control.

Unfortunately, due to a lack of user experience and time to gain the required experience, this Simulink modelling could not be completed for this project. Only the use for the system determined.

3.4 DATA

3.4.1 Ford Mondeo ST220 Test Results

As per the tests described in Section 3.2 and 3.2.1. The static data for the vehicle can be seen in Table 3.4.

Table 3.4 - Measured Ford Mondeo Static Data

Mondeo Specification	Value	Units
Spring Rate Front	27.14	N/mm
Spring Rate Rear	21.90	N/mm
Aftermarket Spring Rate Front	78.45	N/mm
Aftermarket Spring Rate Rear	39.23	N/mm
Total Mass	1514.50	Kg
Front Left Mass	472.00	Kg
Front Right Mass	463.50	Kg
Rear Left Mass	279.50	Kg
Rear Right Mass	299.50	Kg
Front Unsprung Mass	42.60	Kg
Rear Unsprung Mass	34.00	Kg

Table 3.5 - Calculated Mondeo Data from Tests

	Sign	Value	Unit
Desired Damping Ratio		0.5	-
Mass	m	429.40	kg
Damping Coefficient	C	5804.12	N-s/m
Spring Rate	k	78.45	N/mm
Velocity	ω	11.71	m/s
Acceleration	α	6.76	m/s ²
Natural Frequency	f	2.15	Hz
Critical Damping	C_{cr}	11608.24	N-s/m

As can be seen in Table 3.5, the damping coefficient changes in relation to the damping ratio to the critical damping of the system, where 0.5 ratio will give half the value of critical damping.

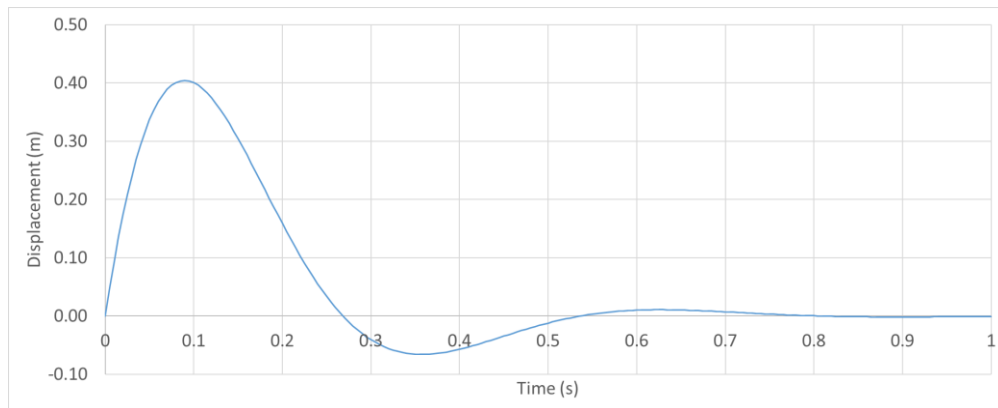


Figure 12 - Calculated Front Damper Displacement/Time Curve with 0.5 Damping Ratio

The values stated in Table 3.5 and Figure 12 are based on information from the front left corner of the car. This can be repeated to simulate each corner of the vehicle individually.

3.4.2 Damping Comparisons between Standard and Aftermarket

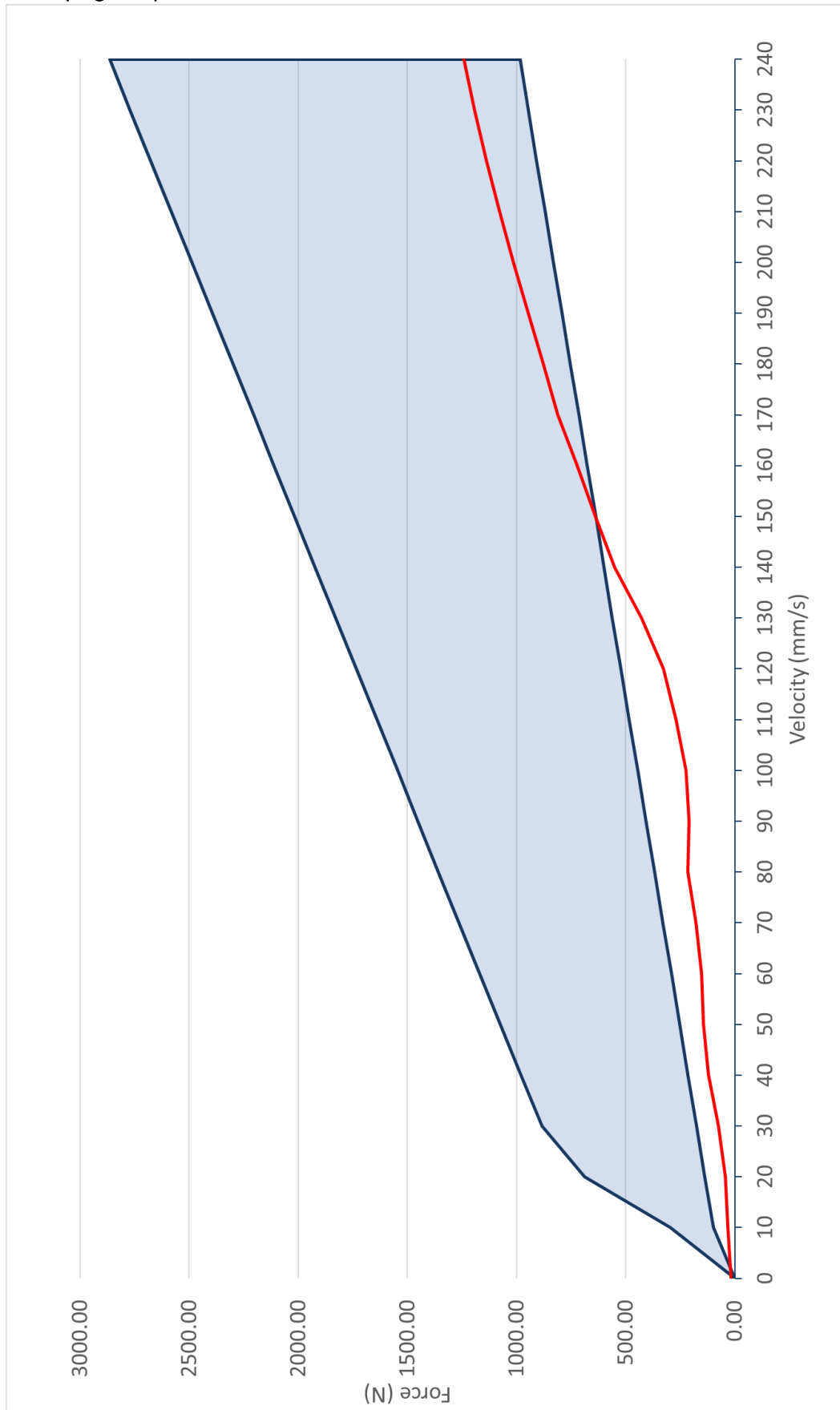


Figure 13 - Comparison between Standard and Aftermarket Front Compression Damping

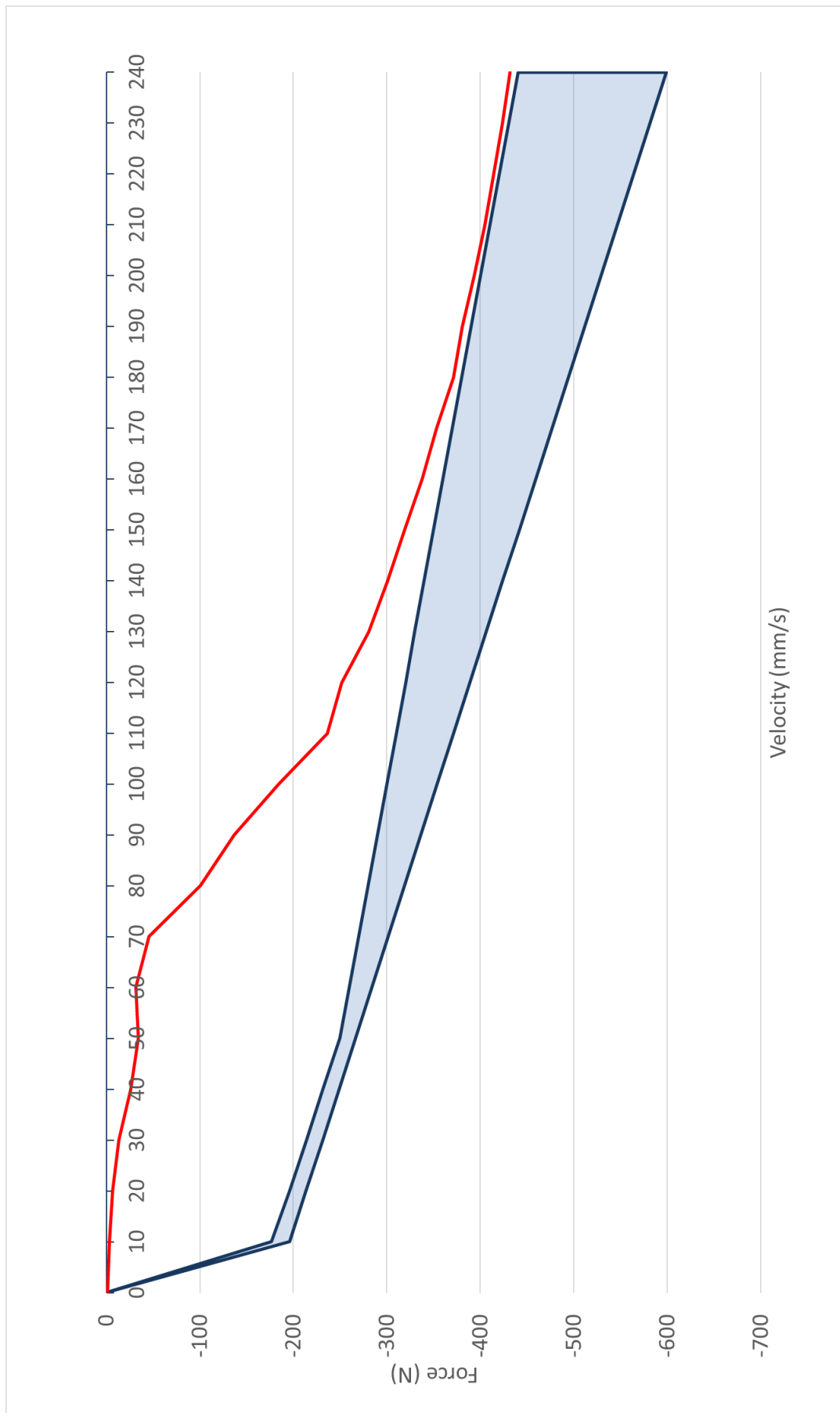


Figure 14- Comparison between Standard and Aftermarket Front Rebound Damping

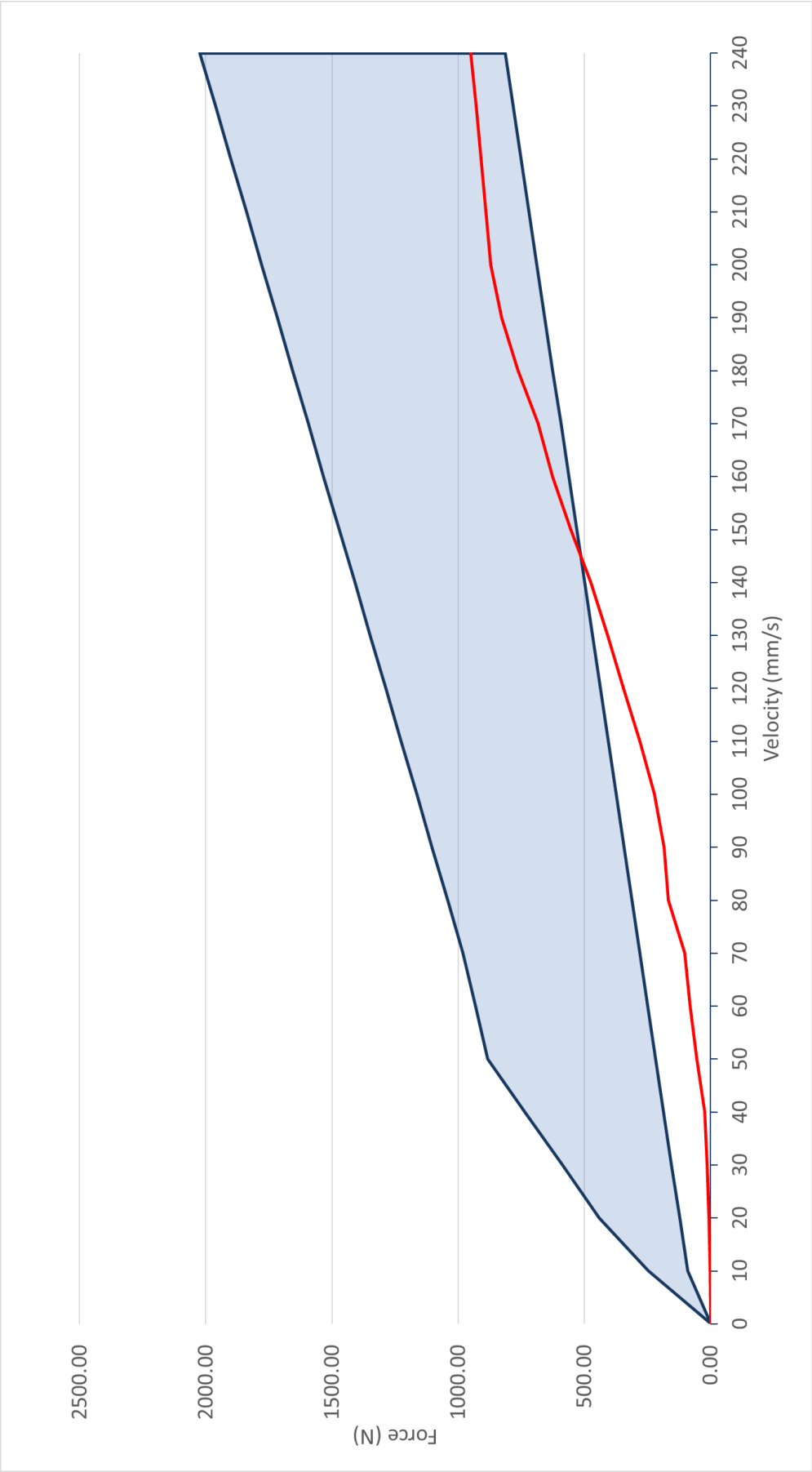


Figure 15 - Comparison between Standard and Aftermarket Rear Compression Damping

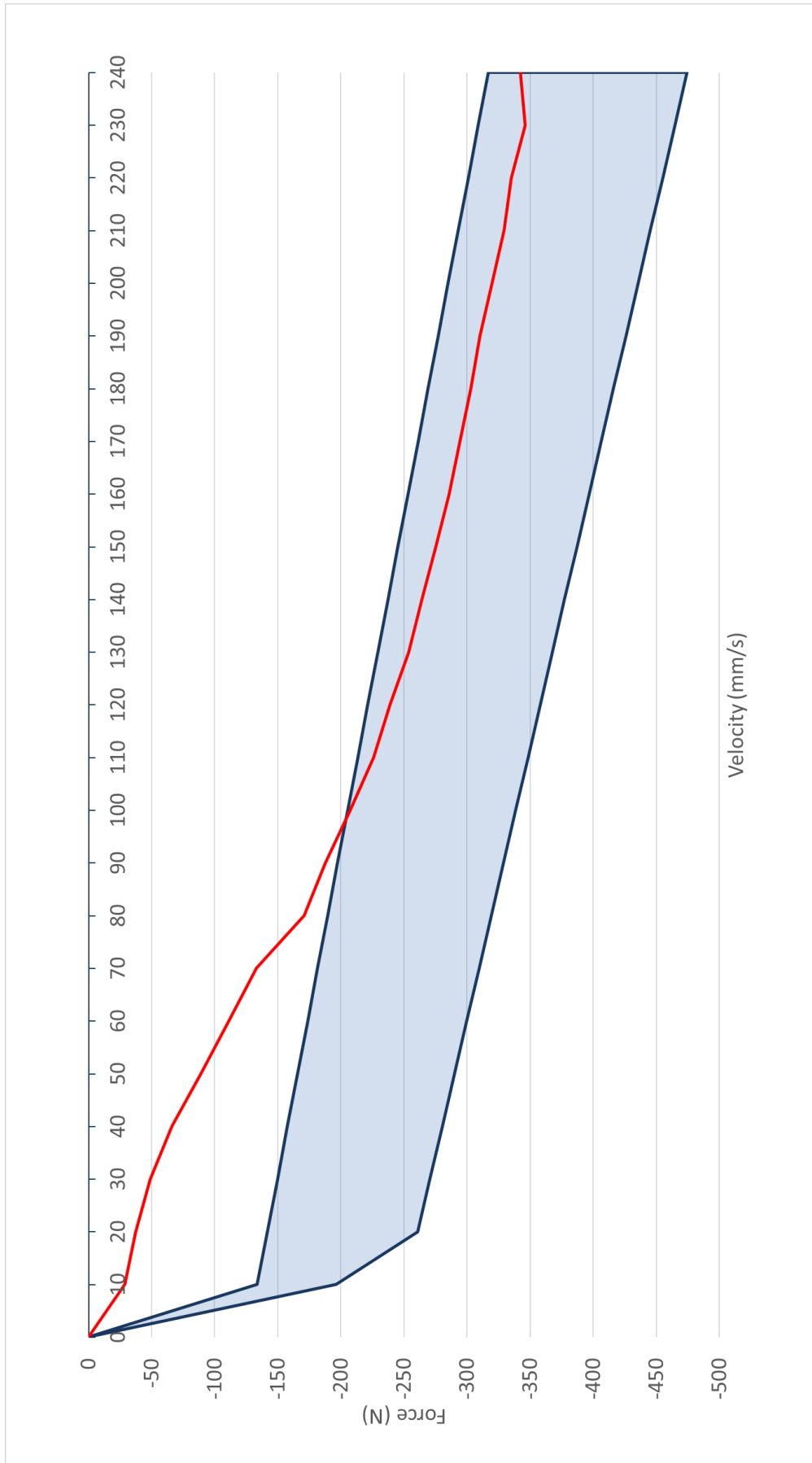


Figure 16 - Comparison between Standard and Aftermarket Rear Rebound Damping

Raw test result data for the standard dampers from the damper dynamometer can be seen in appendix F. The data provided by BC Racing can be seen in appendix G. Due to the force values given from BC racing being in kg force (kgf), these needed to be converted to Newton force, to do this the kgf values were multiplied by gravitational acceleration constant (9.81 m/s^2).

The areas shaded in blue indicate the range of the aftermarket adjustable damper from minimum to maximum. The aftermarket dampers have 30 stages (clicks) of damping between minimum and maximum damping, providing a wide range of adjustability. The damping adjustments can only be done one way, in compression damping, hence the wider range of damping force in the compression damping. The rebound is still affected by these adjustments, but the range is less, this is due to the fluid flowing through the piston valves in the opposing direction still being affected by the adjusted valve. The red line is the measured data from the damper dynamometer on the standard Ford dampers.

The compression stroke of the standard dampers is relatively progressive, but the rebound is very unpredictable, the test was repeated several times, yet the curve remained similar throughout. The damper used was 13 years old, during its life has been driven hard around various circuits, and had covered in excess of 100,000 miles. After testing it became apparent the dampers had also developed a leak, this could be causing pockets of air to form within the cylinder, which could contribute to the low forces seen at low rebound velocities. The cause of this could be investigated further, but was not considered relevant for the progression of this project.

This data shows a little of the design intentions BC Racing had when engineering the dampers. The data shows the minimum damping setting on the aftermarket dampers is similar to that of the standard dampers at high velocities, and increasing the stiffness under low speed damping, taking more force to overcome the low velocities and enter the high speed damping range.

Part of understanding how dampers perform, especially when comparing two damping systems directly, is to find out how the dampers feel from a drivers perspective. So to determine this, an interview with the driver, Richard Stevens, was carried out. A transcript of this can be found in Appendix E. During the interview, Richard expresses that he was not looking to alter the road damping characteristics, but to have the option to set up the suspension to better handle fast circuit driving. The data on the force/velocity curves prove that these dampers are giving the desired effect, with the only notable exception being the “crashy” feel of the car caused by pot-holes. They offer the option to stiffen the low speed damping, keeping the car more balanced and stable through transient cornering.

4 DESIGN

4.1 SENSORS

4.1.1 Steered Angle Sensor

A damper will have optimum operating ranges for transient cornering. Depending on the lateral acceleration of the vehicle. Therefore a steered angle sensor needs to be able to measure the rate of change of steered angle with immediate effect. To measure this, the Bosch LWS3 steering-angle sensor is appropriate. This uses Anisotropic Magnetoresistive Sensors (AMR) for its operation. The electrical resistance varies according to an external magnetic fields direction. The sensor has a range of four complete rotations to



Figure 17 - Bosch LWS3 AMR Steered Angle Sensor (Bosch, 2007)

accurately measure the angle on, making it suitable for a road application, where the steering wheel rotates up to approximately 720° (4 rotations). This sensor uses measured angles of two gears which are rotated by a third gear attached to the steering shaft. The first 2 gears teeth differ by one tooth, allowing an accurate angle reading for all steering positions. The AMR method is suitable to have an accurate angle calculated in a microprocessor, making it suitable for use with the CAN bus. (Bosch, 2007)

4.1.2 Yaw-Rate Sensor

In early navigation systems, vehicles would use a magnetic field sensor which would read the magnetic field of the earth to determine the vehicles direction. However this would be prone to interference from other aspects of the vehicle producing magnetic fields. As an alternative, Bosch produce a piezoelectric “tuning fork” yaw-rate sensor, which incorporates four piezo elements, two above and two below. As voltage is applied, the bottom elements begin oscillating, therefore exciting the “tuning fork” along with the upper piezo elements, which in turn begin counter-phase oscillations. This has a frequency of approximately 2 kHz. (Bosch, 2007)

When the vehicle is being driven in a straight line, there are no Coriolis forces acting upon the fork. The upper piezo elements only experience counter phase oscillations and are only sensitive perpendicularly to oscillations. Therefore they do not produce a voltage. (Bosch, 2007)

As the vehicle is cornering, the rotation around the vertical axis causes the upper section of the fork to leave the oscillatory plane, thus an alternating voltage is produced by the upper piezo elements and sent to the navigation computer. Voltage signal amplitude is a function of oscillation velocity

and yaw rate. The direction of the curve (left or right) determines the sign of the voltage signal, making it easy for the computer to determine yaw position and rate. (Bosch, 2007)

A yaw-rate sensor would help the system determine when the vehicle is oversteering or understeering and the active damper system can therefore adjust itself accordingly.

4.1.3 Potentiometers

Measuring throttle position and braking position with the use of potentiometers allows the active damping system to read the type of ride the driver desires from the car, for example using 100% throttle would suggest the driver would like the car to prioritise speed over comfort, therefore the damping could be adjusted to maximise grip rather than to provide a smooth ride. Under heavy braking the ride could be stiffened to prevent sudden weight transfer which could lead to lift off oversteer.

Potentiometers on the dampers could measure the acceleration of the damper to ensure the damper is behaving as the system intends. Therefore diagnosing potential faults in the damper. By knowing the acceleration of the damper, it is possible to identify the force being applied to the wheel vertically. This could help detect wheel lift for the system to counter this.

4.1.4 Lidar Sensors

Lidar sensors work in a similar way to a radar, but instead of omitting radio waves, it sends infrared lasers invisible to the human eye, and measures the time it takes to return. Therefore providing a digital image of objects/road layouts. This could be used to identify potholes and react to them. If a hole in the road is large enough the damper could become very over damped to prevent the tyre from dropping into the hole, reducing the size of the impact on the other side of the bump. Or to identify bumps in the road, and soften the damping to allow the spring to compress without causing the vehicle to bounce. Combined with a vehicle speed sensor, this could identify the radius of a corner and the velocity of the vehicle, to predict the likely weight transfer and therefore dampen the weight transfer accordingly.

Lidar technology at the time of this report is very expensive, so to implement this technology on a large scale, especially as it could require several of the sensors per application, may not be a realistic option.

4.1.5 Vehicle Velocity Sensors

It would be beneficial to the system to have both wheel speed sensors and vehicle over ground sensors, to calculate the difference between the two would indicate when the vehicle is losing traction.

4.2 CONTROL

Using the Microsoft Excel damping simulator described in section 3.3.1, the damping coefficient can be altered by inputting a desired damping ratio. The damping is constant whether it is in bump or rebound stroke, however due to the responsiveness of the MR fluid the damping coefficient for bump into rebound could be changed during the oscillation.

Having several sensors inputting information, great challenge will come when looking to prioritise certain responses over others. The vehicle safety must always come first where necessary. However when considering corner entry where the system may be trying to adjust to optimum braking, then as steering input increases the system has to integrate the optimum cornering responses. Inevitably there will be a lot of compromise. The ideal way to test which compromises are more effective than others would be through testing the systems on the car, with a series of different set ups for identical tests to gain quantifiable results. This could potentially also be done via computer simulation. However, active damping is banned from the majority of motorsport events, so the available data to verify computer simulation is slim.

Due to the simplicity of altering damper characteristics on the MR dampers, the biggest challenge of the project is to understand what settings work best, and as stated the best way to do this would be with testing. There is very little information regarding handling response to asymmetric damper settings, as it is unnecessary to set up dampers asymmetrically on a passive damper unless it is for oval racing. This makes it difficult to find test data for the effects on handling this type of set up would have. The possibilities of differing configurations for an asymmetric damper are huge, therefore the time and resources to test these set-ups are also huge. It would require a very thorough and strategically thought out test plan like a design of experiments in order to obtain this data.

Figure 18 shows an example of a configuration that could be tested for a slow right hand corner entry, such as a hairpin. The front left is stiff to counter the high levels of force this tyre is likely to experience. The front right and rear left are the same as they are the axis of which the body roll will take place. The front left has been softened up considerably to counter the likelihood that this wheel could lift off the ground, the low damping here is to give the spring the best chance of keeping the tyre on the ground, offering as much grip as possible. The benefits of this type of set up would need to be verified by testing.

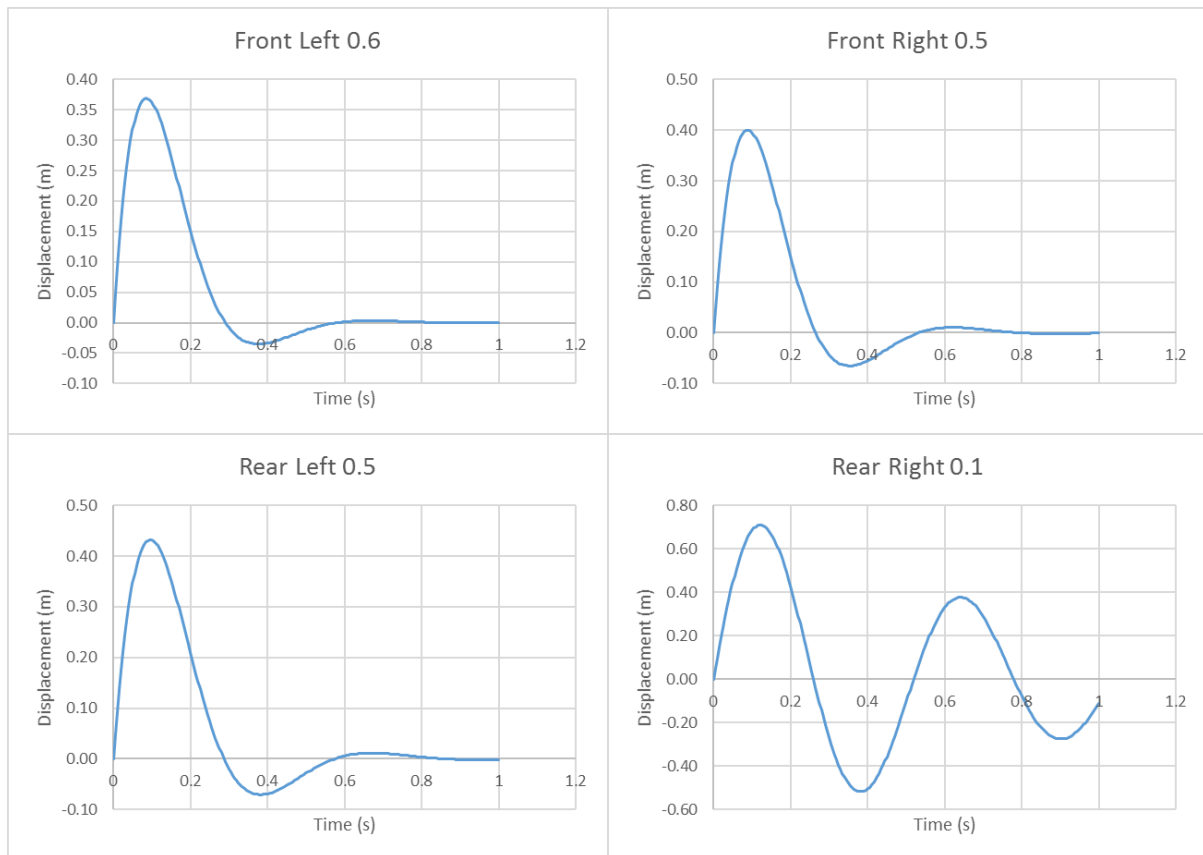


Figure 18 - Example Damping Configuration for a Slow Right Hand Corner

The Mondeo is a front wheel drive vehicle, for a rear wheel drive vehicle, the rears could be made to soften under acceleration to maximise available grip in the driven tyres.

The yaw sensor could be used in conjunction with the steered wheel position sensor to indicate when the car is breaking away from neutral steer during a corner. If the car is beginning to oversteer, the rears could be softened in order to allow the weight to transfer to the rear as quickly as possible and getting traction back, along with slightly stiffening the front to prevent the car from suddenly becoming unbalanced or overshooting the oversteer correction. The opposite could be done for when the vehicle is experiencing understeer. (Alexander, 2015)

5 CONCLUSIONS

Spring mass dampers and their applications have been discussed. The mathematical formulas for characterising simple damper behaviour have been provided and explained. The testing of dampers and vehicle measurements have been described with the data laid out in the report. The factory dampers and aftermarket damper designs have been analysed and compared, this provided an insight into the type of damping outcome the manufacturers, Ford and BC Racing had intended for the vehicle. A method for simulating damper behaviour in Microsoft Excel has been detailed to enable the reader to apply this to further studies and damping system designs. Components that make up the damping system have been discussed enabling the reader to begin constructing and integrating an active damping system to a vehicle. The handling effects of dampers and how they could be manipulated by an active damper system to maximise performance has been discussed.

6 FURTHER WORKS

As stated in section 3.2.2, it would have been beneficial to obtain an MR damper had it been possible. This would have enabled the range to be tested with varying current inputs, and the control of the damper could be further understood, which inevitably would have helped with the project. Being able to test the aftermarket dampers performance at varying settings as mentioned in section 3.2.3 would have provided a deeper understanding of how vehicle handling is effected by damper adjustments.

Given more time, the use of Simulink to construct a circuit to control the system would have been the next logical step in the successful development of an active damping system. A more experienced reader may be able to take the information in this report and implement it within Simulink. However at the point of publishing, the Author would need more time to gain the adequate experience required to construct a reliable Simulink model.

Further research into asymmetric damping could be carried out, this could be an appropriate future project for any engineer. For passive damping, asymmetric damping would be inappropriate unless for oval racing, which only accounts for one type of corner. So to test this on various types of corners would aid the development of active damper systems in the future. This research could be carried out initially with the use of a simulator.

7 REFERENCES

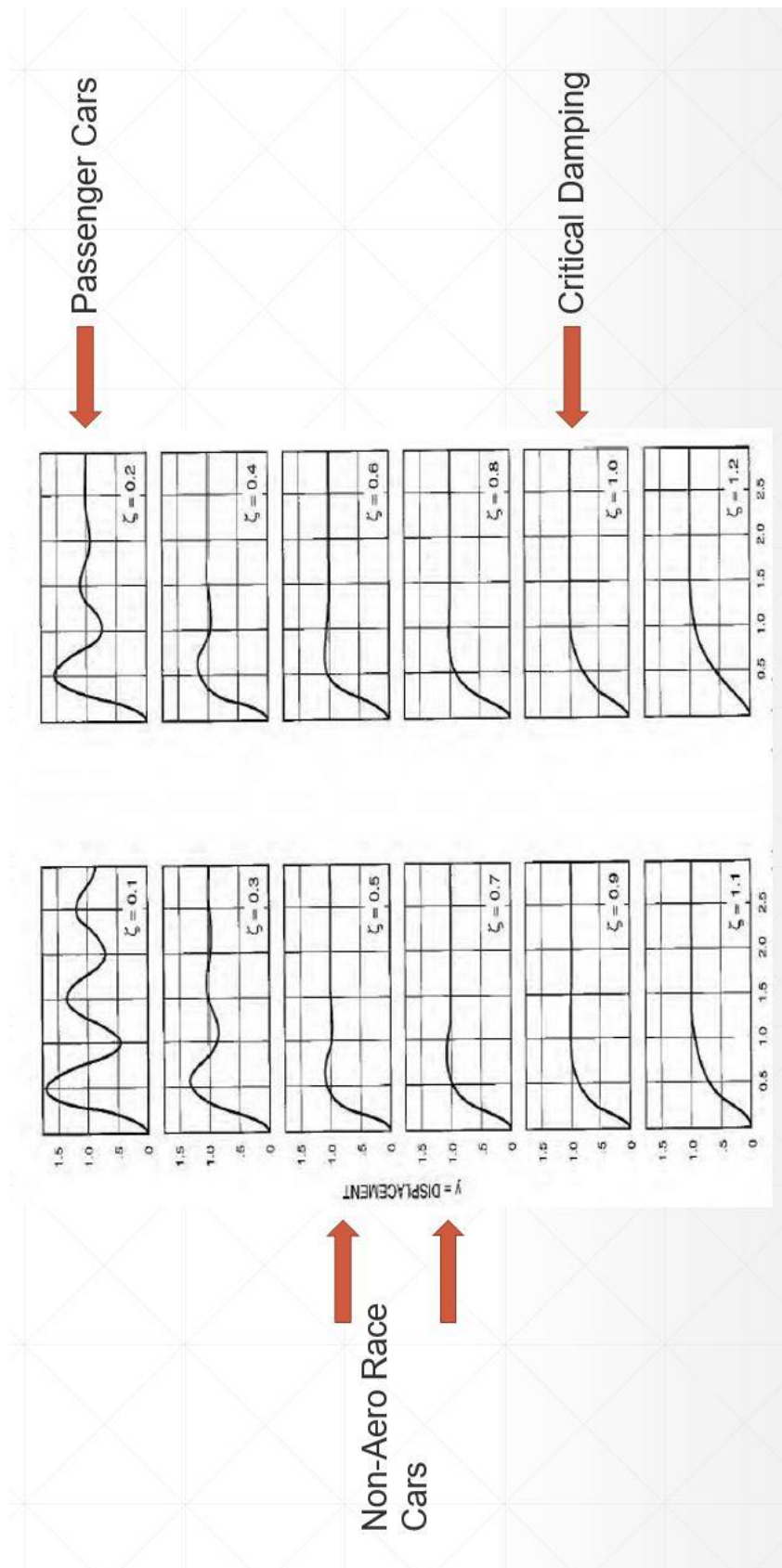
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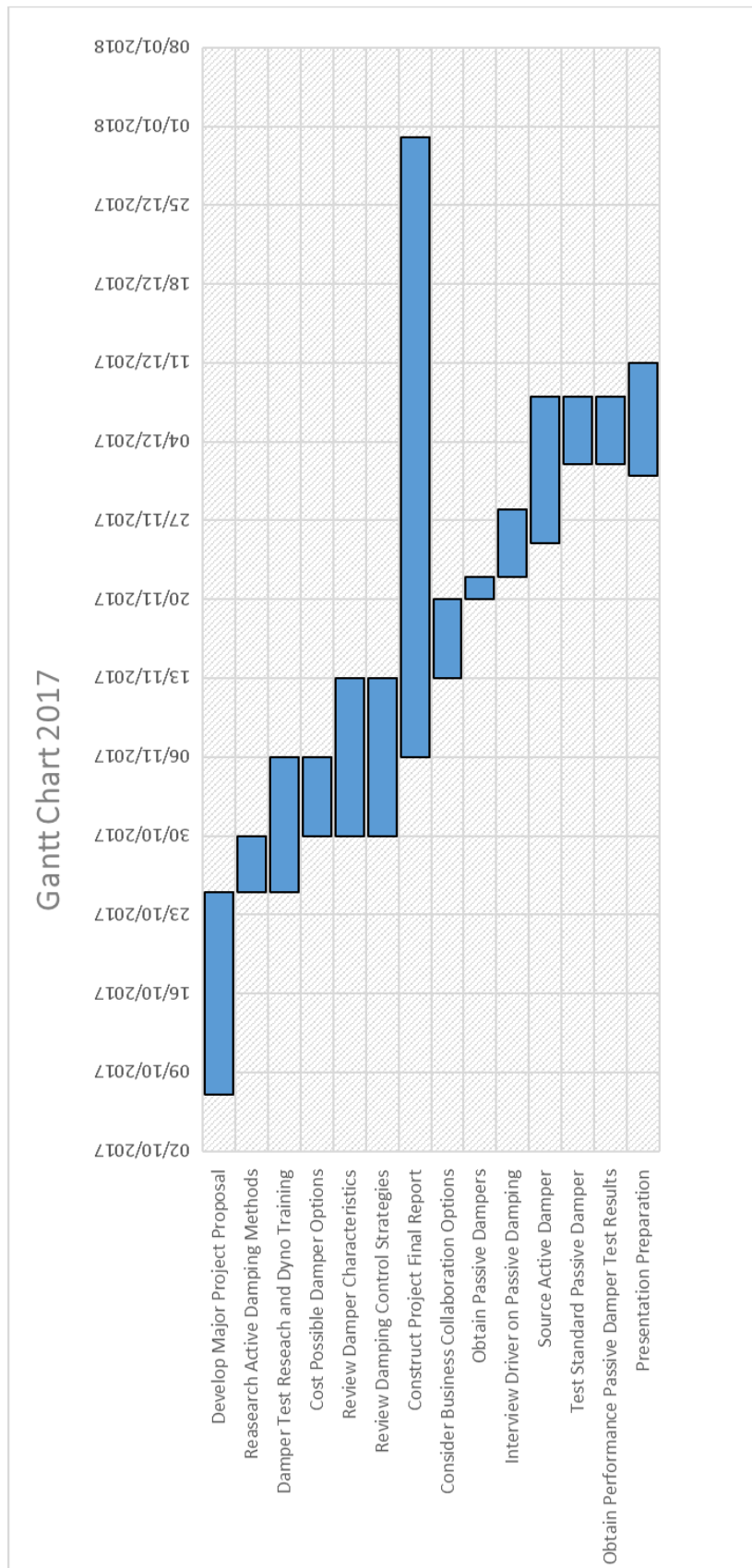
All images, graphs and tables are property of Joel Thomas unless stated otherwise.

8 Appendices

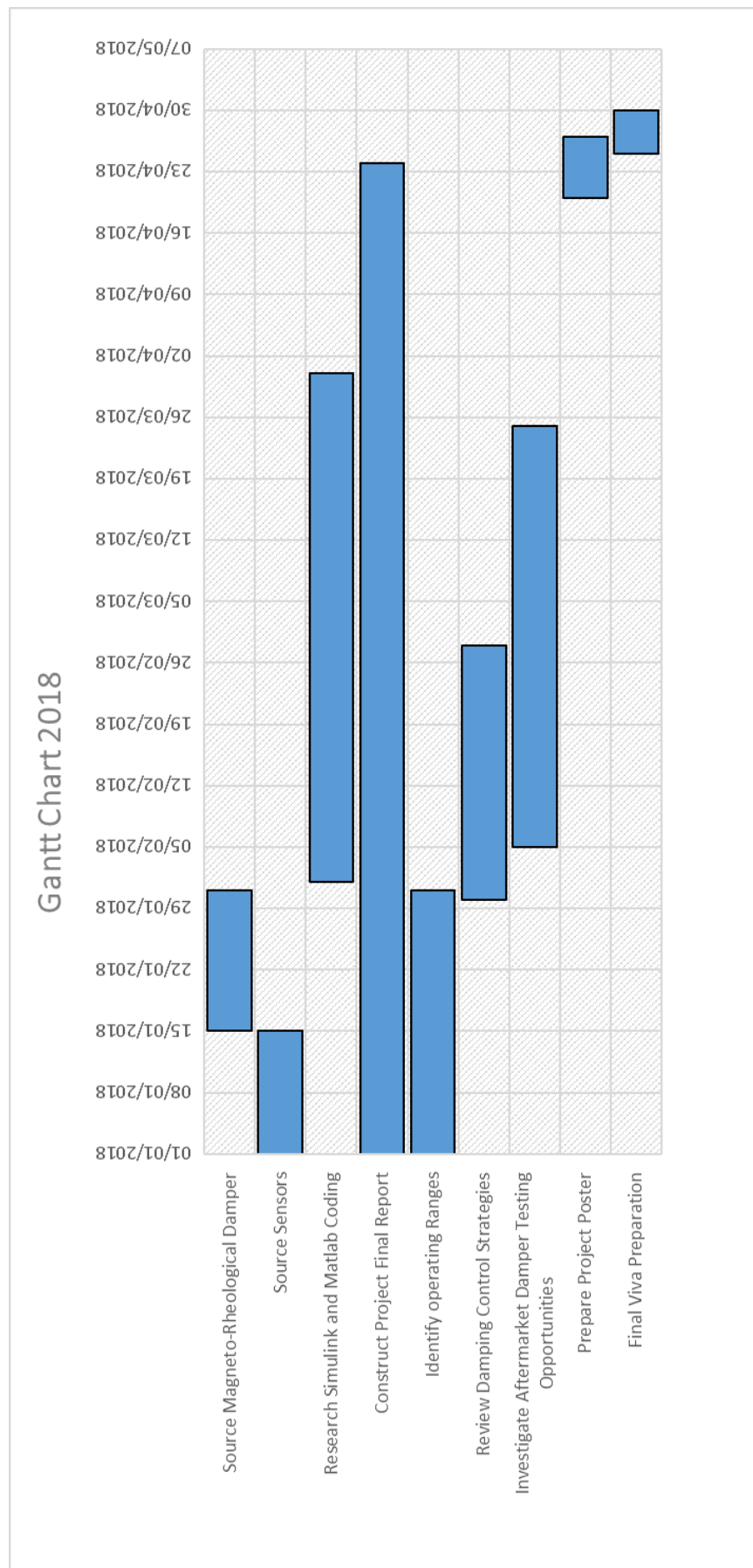
A - Damping Ratios Effect on Ride



B – Gantt Chart 2017



C – Gantt Chart 2018



D - Predicted MR Damping Forces

Current	0 Amps	0.2 Amps	0.4 Amps	0.6 Amps	0.8 Amps	1 Amp	1.2 Amps	1.4 Amps	1.6 Amps	1.8 Amps	2 Amps
Velocity (m/s)	Newton Force (N)										
0.00	-22	0	0	0	0	0	0	0	0	0	0
0.01	67	200	423	645	867	1090	1312	1535	1757	1979	2202
0.03	89	222	445	667	890	1112	1334	1557	1779	2002	2224
0.04	111	245	467	689	912	1134	1357	1579	1802	2024	2246
0.05	133	267	489	712	934	1157	1379	1601	1824	2046	2269
0.06	156	289	512	734	956	1179	1401	1624	1846	2068	2291
0.08	178	311	534	756	979	1201	1423	1646	1868	2091	2313
0.09	200	334	556	778	1001	1223	1446	1668	1890	2113	2335
0.10	222	356	578	801	1023	1246	1468	1690	1913	2135	2358
0.11	245	378	601	823	1045	1268	1490	1713	1935	2157	2380
0.13	267	400	623	845	1068	1290	1512	1735	1957	2180	2402
0.14	289	423	645	867	1090	1312	1535	1757	1979	2202	2424
0.15	311	445	667	890	1112	1334	1557	1779	2002	2224	2447
0.17	334	467	689	912	1134	1357	1579	1802	2024	2246	2469
0.18	356	489	712	934	1157	1379	1601	1824	2046	2269	2491
0.19	378	512	734	956	1179	1401	1624	1846	2068	2291	2513
0.20	400	534	756	979	1201	1423	1646	1868	2091	2313	2535

E - Driver Interview Transcript

The following is a transcript of an interview conducted on the 12th December 2017 with the driver of the Ford Mondeo ST220 used to obtain data for this project.

Joel: Why did you install aftermarket coilovers?

Richard: I was looking to not only improve the aesthetics of the car by lowering it by 30-40mm but I also wanted to have some control over the damping of the car because I enjoy taking it on circuit, meaning it was more logical to fit coilovers rather than just lowering springs.

Joel: Why did you choose those ones?

Richard: I chose the coilovers on the car as they allowed for adjustable damping where as others on the market didn't. Meaning I can stiffen the car for track days and soften it for on the road.

Joel: Did it have the desired effect?

Richard: I have yet to take it on circuit however I like the look of the car lowered and have adjusted the damping for the roads

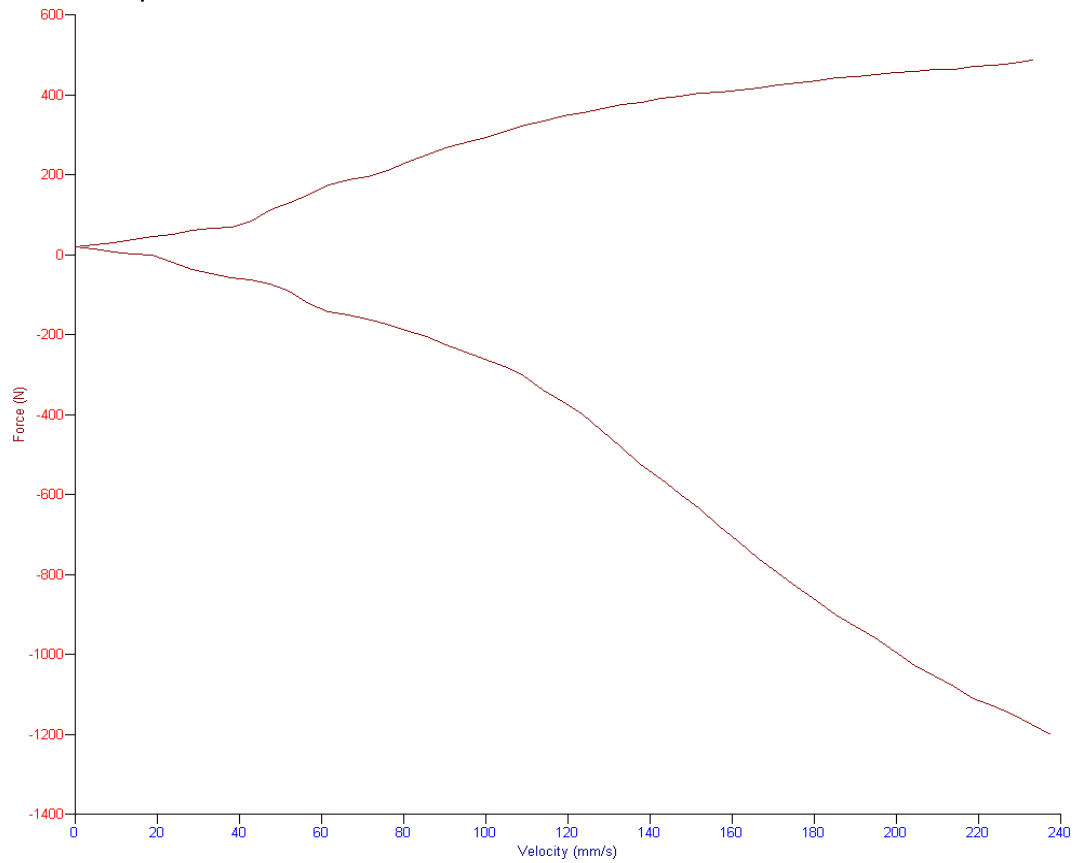
Joel: What is better about them?

Richard: They ride really well, no one has complained about the ride of them and I have completed several long distance journeys with them, the handling of the vehicle has also improved however I am yet to test it on a circuit.

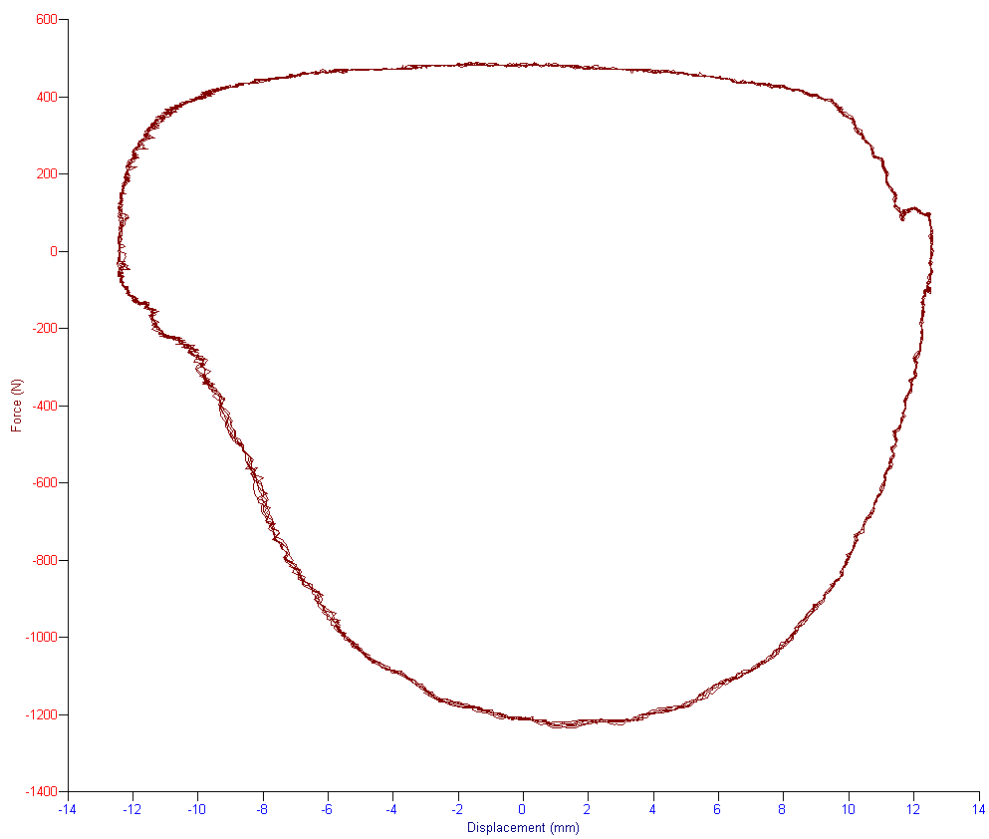
Joel: What do you miss about the old ones?

Richard: They can be very crashy when going over a pothole where as the original suspension would do a better job of softening the impacts.

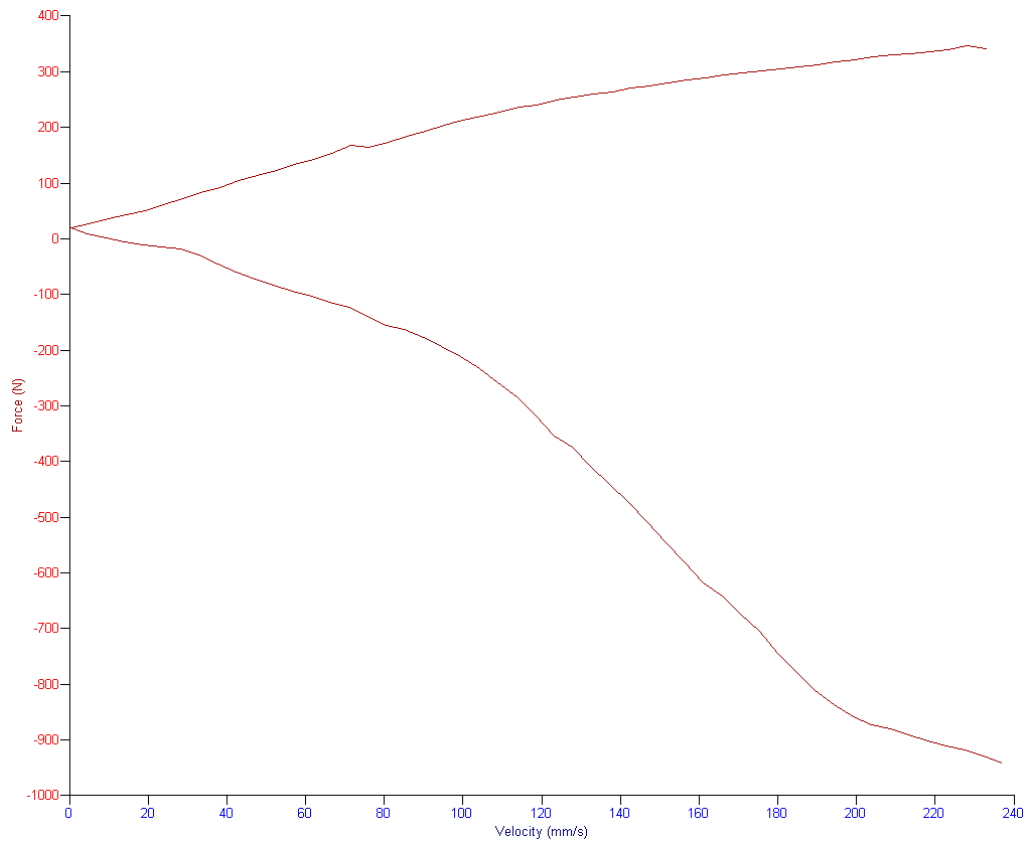
F - Standard Damper Results



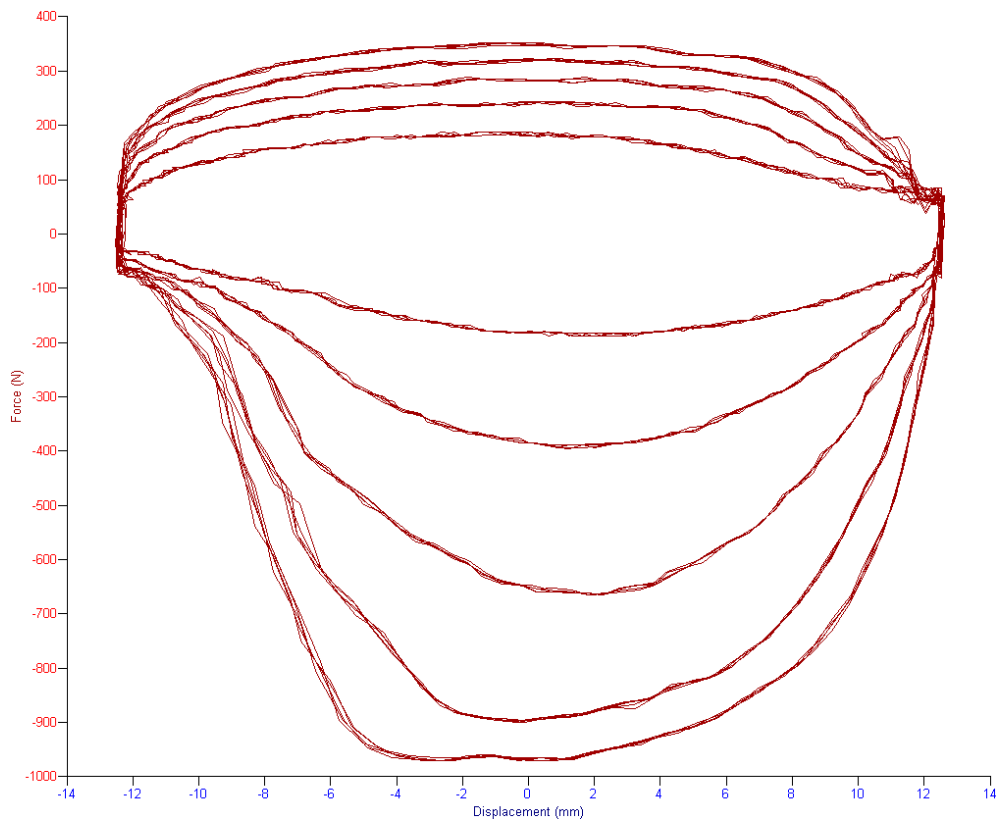
i. Front Standard Damper F/V Graph



ii. Front Standard F/D Graph

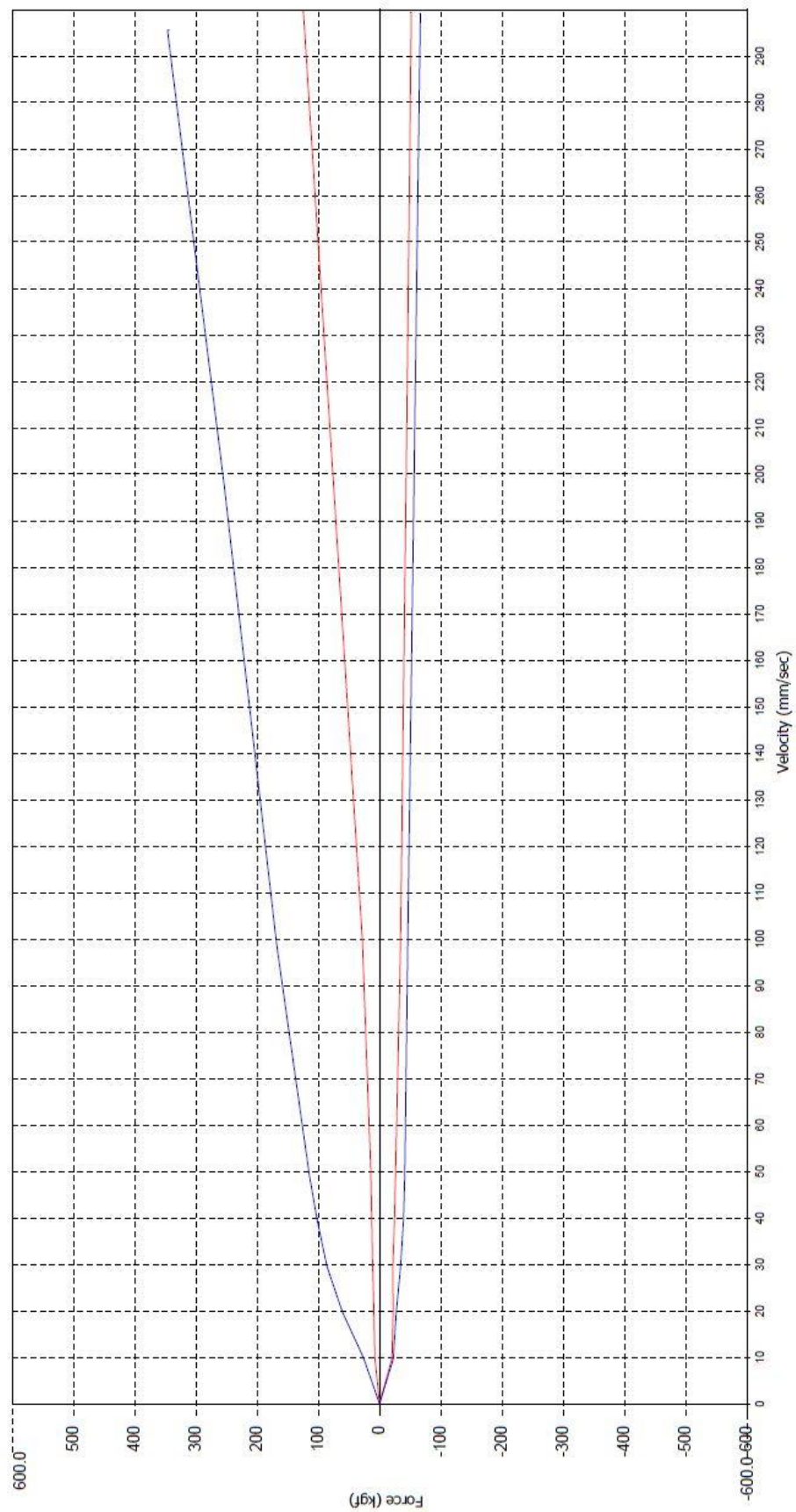


iii. Rear Standard Damper F/V Graph

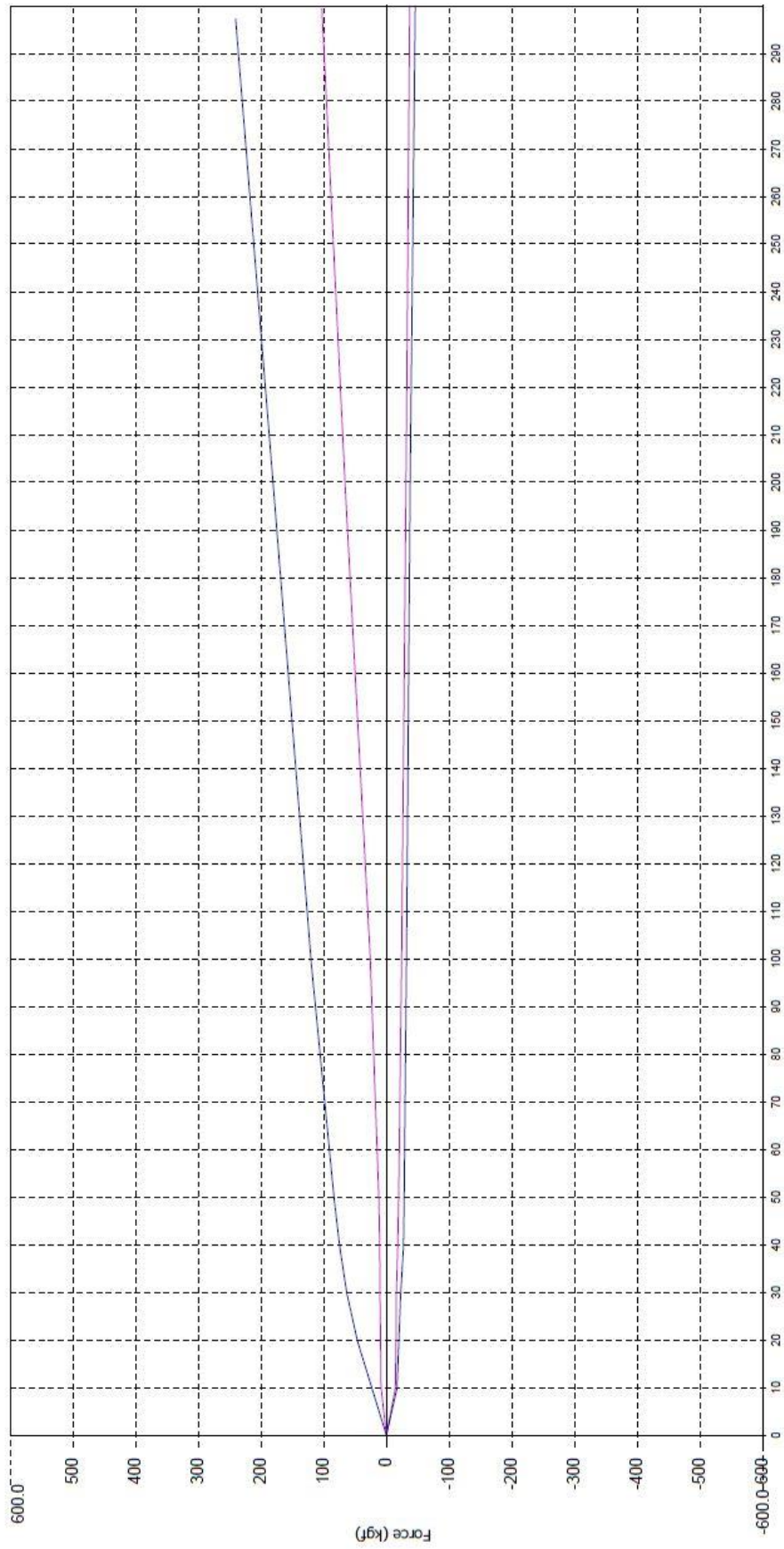


iv. Rear Standard Damper F/D Graph

G - Aftermarket Damper Data

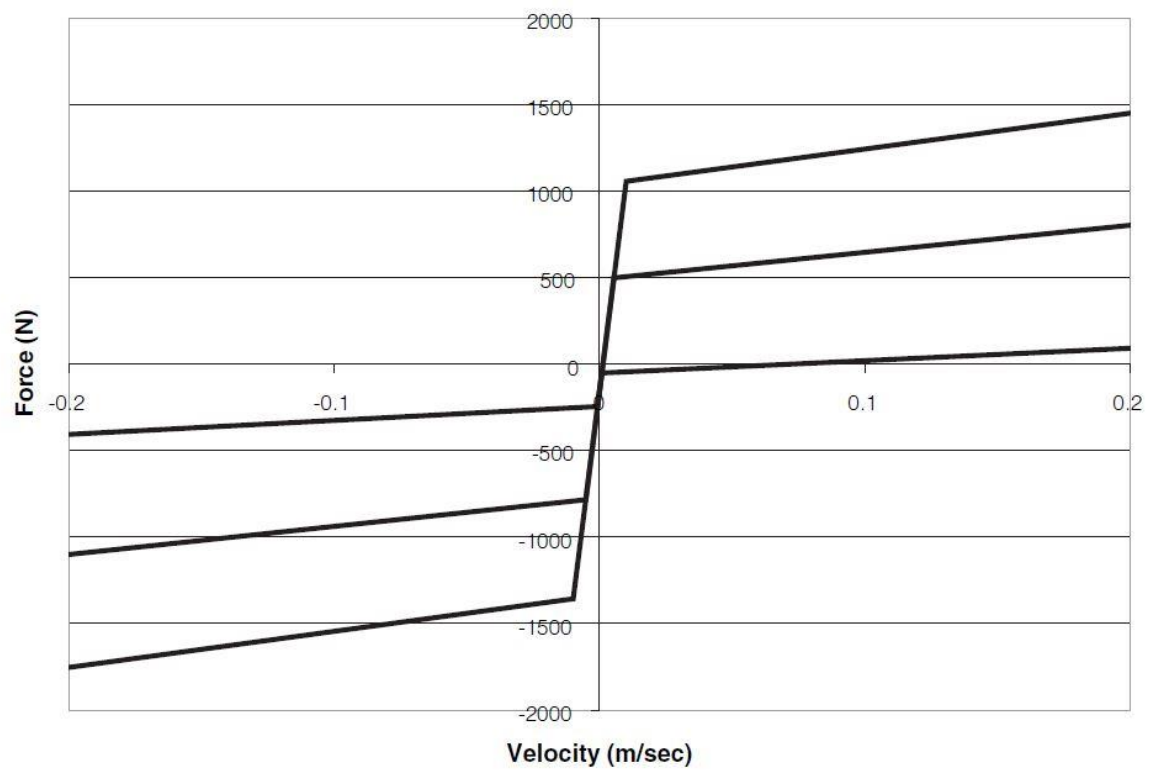


v. Front F/V Graph of Aftermarket Damper (BC Racing, 2017)



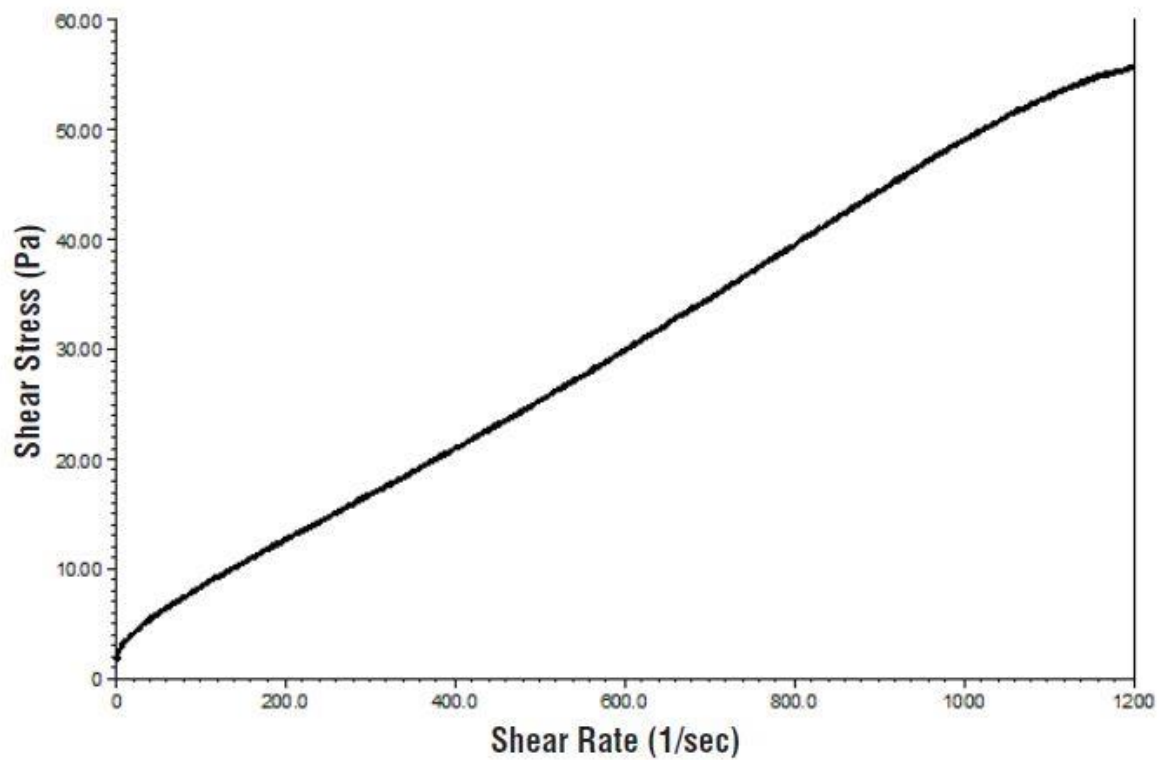
vi. Rear F/V Graph of Aftermarket Damper (BC Racing, 2017)

H – LORD Corp MR Damper Force Velocity Graph

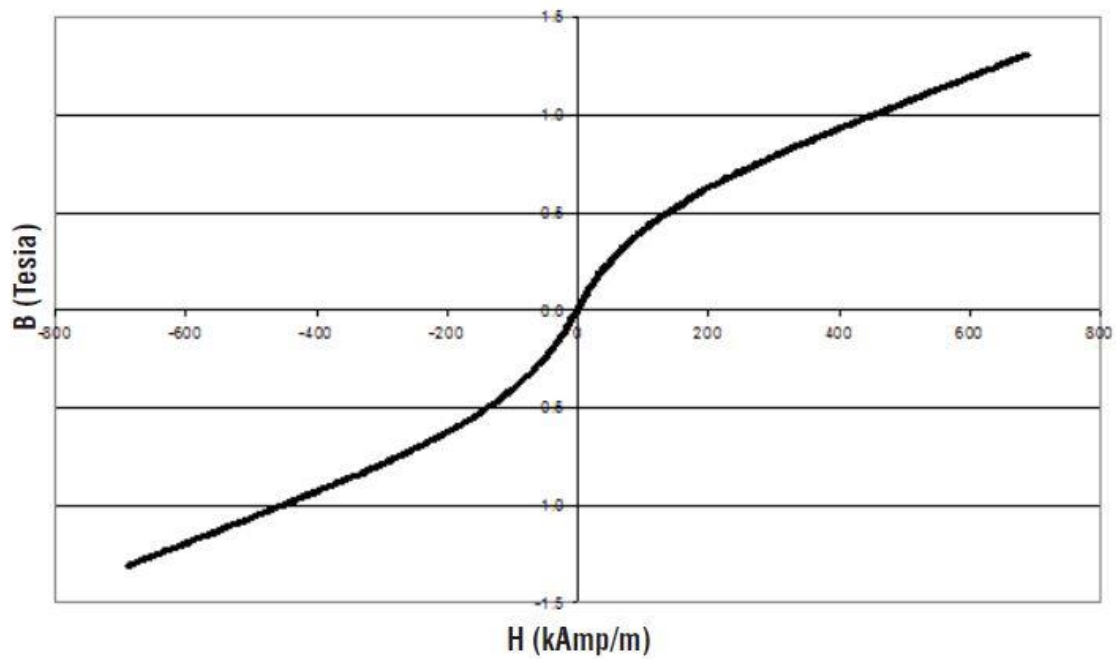


vii. (Lord Corporation, 2015)

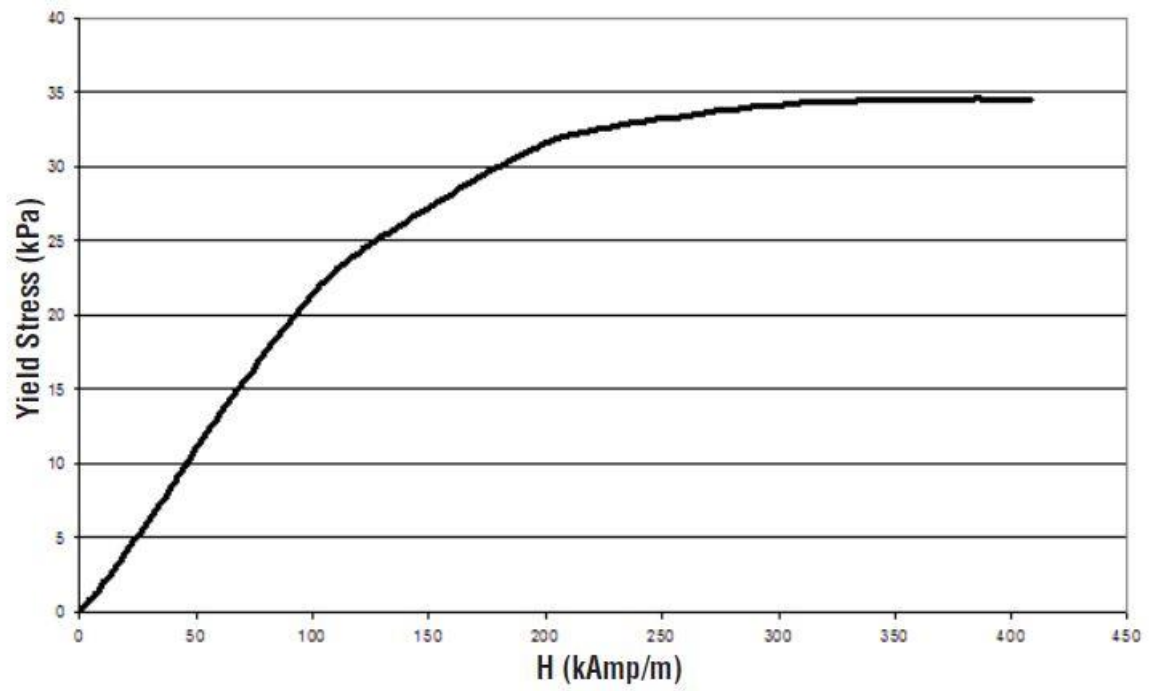
I – LORD Corp MR Fluid Property Graphs



viii. *Shear Stress as a Function of Shear Rate with no Magnetic Field Applied at 40°C (Lord Corporation, 2015)*

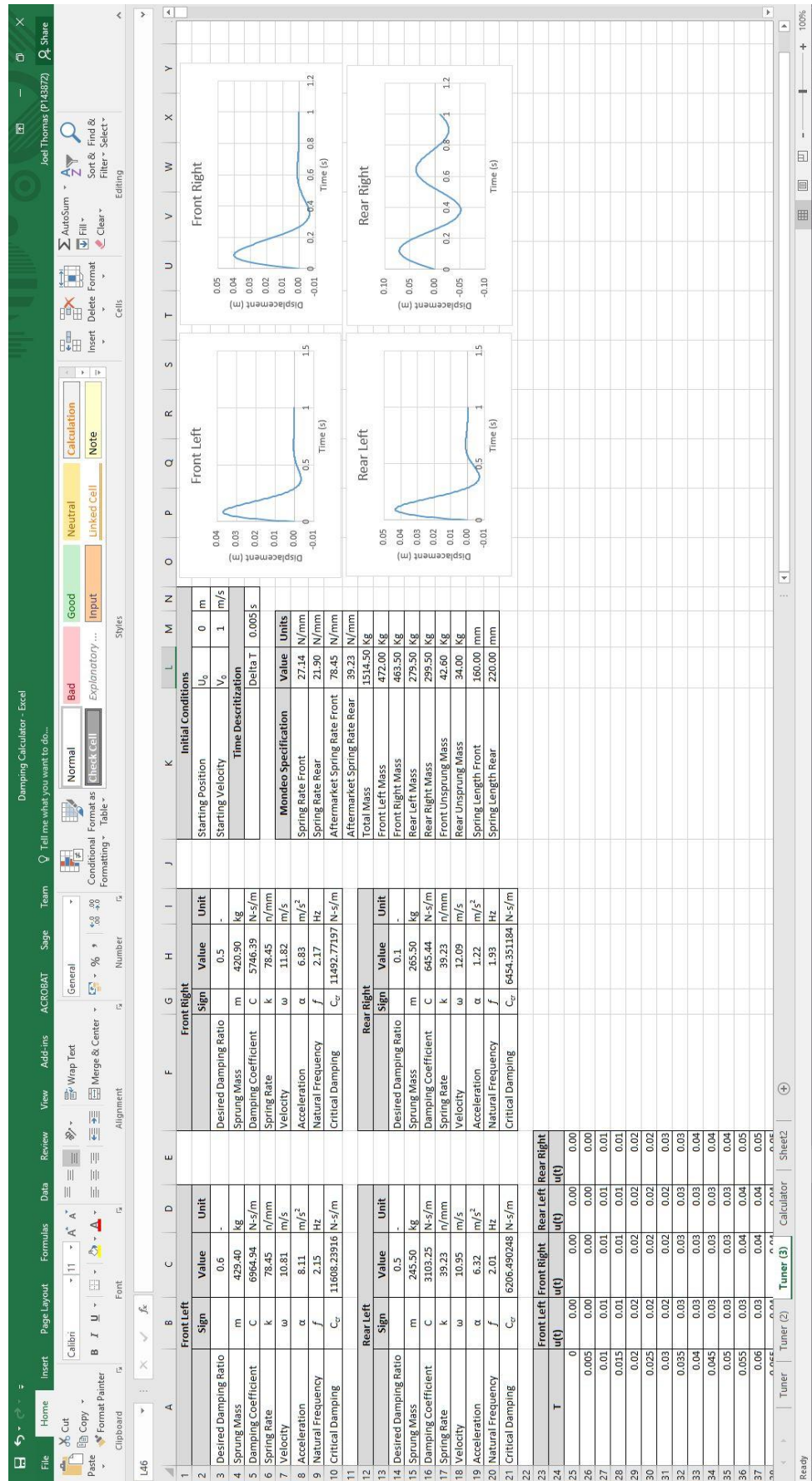


ix. *Typical Magnetic Properties (Lord Corporation, 2015)*



x. Yield Strength vs Magnetic Field Strength (Lord Corporation, 2015)

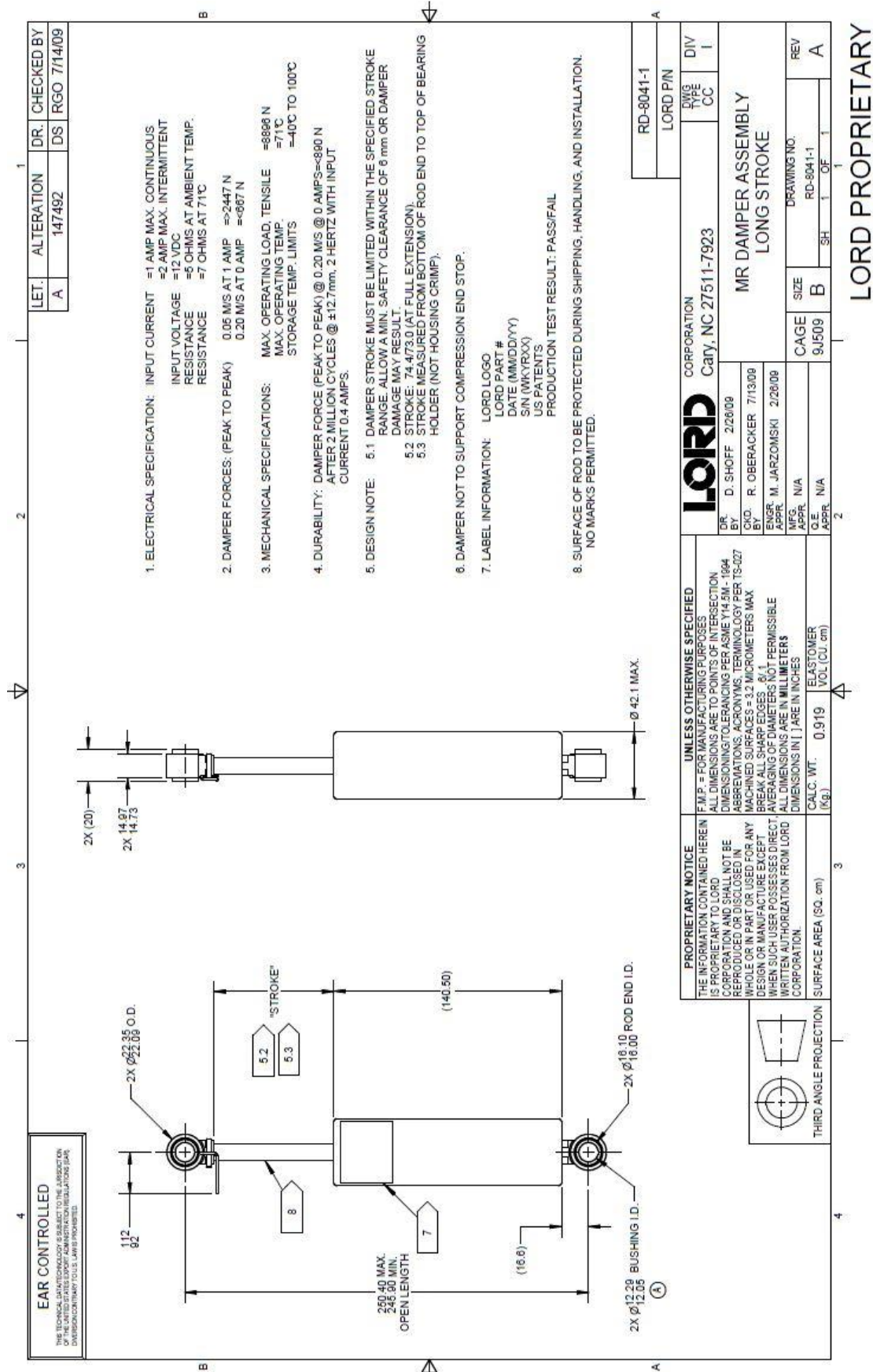
J – Damping Calculator in Excel



K – Damper Test Results and Aftermarket Damper Data



L – LORD RD-8041-1 Technical Drawing



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