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MCR Sports 2000 Performance Analysis

Group Design Project – ACHA5014

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1 Introduction - All

This report is a performance analysis of an MCR competition vehicle competing in SRCC Sports 2000. An MCR is a Mid-Engine, Rear Wheel Drive Sports Prototype shown below in Figure 1. This vehicle utilises double a-arm suspension front and rear, has sensor-guided MoTeC dataloggers and is powered by a 2.0l inline-4 Ford Duratec engine. This report mainly focuses of #40 shown in Figure 1 in yellow.



Figure 1 - Picture of two MCR Sports 2000 Cars

The report is split into three sections, Powertrain (Dylan, Ella), Suspension (Jake, Nathaniel), Data Acquisition (Liam). Powertrain is split into Simulation and Physical testing where Dylan is responsible for Simulation and Ella is responsible for Physical testing (due to outside factors Ella was unable to complete the physical testing, as a result critical testing was reallocated to Dylan). Suspension is split into Front Analysis, Rear Analysis and VI Grade Analysis. All areas of Suspension were worked on by Jake and Nathaniel.

The aim of the report is to gain a better understanding of the vehicle and improve performance within the Sports2000 Regulations.

2 Suspension - NF/JL

2.1 Suspension Introduction

This section of the report focuses on the vehicle's suspension kinematics and handling characteristics. The aim of this section is to create and validate a WinGeo model of the suspension systems on two MCR Sports 2000 chassis, #40 and #44. #40 is a newer specification chassis when compared to #44. #40 is the yellow car and #44 is the black car, shown in Figure 1. The main difference between these chassis from a vehicle dynamics perspective is a difference in location of the a-arm chassis hardpoints. The Kinematics investigation was carried out on car #40 and the coordinates for the suspensions systems of cars #40 and #44 were measured and put into WinGeo. The results from the kinematic test will be compared against the WinGeo model of #40 to validate the accuracy of WinGeo. These results will also be compared in WinGeo against suspension coordinates, from the teams current VI Grade model, to evaluate the accuracy of this model against the new measured coordinates to assess the most accurate model for future development of the suspension system and wheel alignment set-up. Once validated these models can be used to compare differences in the chassis suspension kinematics between the front and rear of both chassis. Additionally, these suspension models can be input into VI Grade, a lap simulation software, in order to analysis the suspension geometry's effect on the vehicle's performance over the course of a lap. The accuracy of the VI Grade can be increased by gathering more accurate data about the vehicle, as such further tests were carried out to gather this data. These tests are a spring stiffness and preload, centre of gravity (COG), damper curve, and tyre stiffness investigation.

Initially mass reduction of suspension components was considered through the development of titanium suspension arms, however as the SRCC Sports 2000 regulations (SRCC, 2022) state that Titanium is prohibited (5.8.6a) and all other suspension components must be steel or ferrous (5.8.2) focus was placed on understanding the current system and facilitating future development through the creating of an accurate simulation model.

2.2 Suspension Kinematics

2.2.1 Instant Centres and Roll Centres

A vehicle's Instant Centre or Instantaneous Centre (IC), as described by Kowalik (2013), is a theoretical point at which the wheel of a vehicle will travel in an arc when moving through bump or droop and is traditionally used for Camber Compensation (CC). As described by Minaker and Nantais (2007) the IC for a double a-arm suspension system can be found by drawing a line parallel to the upper and lower a-arms, when viewing the vehicle from the front, the point at which these lines meet is the IC. A vehicle's Roll Centre (RC) is, as described by Bucchi and Lenzo (2022), the point at which a vehicle's suspension system will rotate for a given axis. The RC can be found by drawing a line from the IC through to the opposite wheels contact patch on a given axle, the point at which these lines cross is the RC as (Mitchell, 1998) further described in his paper and shown in Figure 2.

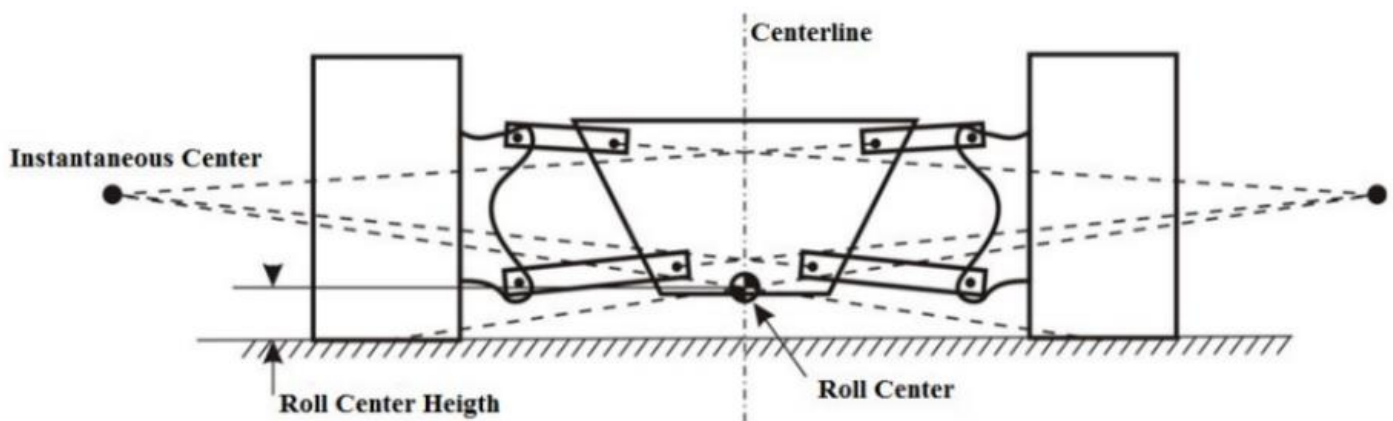


Figure 2 - Instant Centre and Roll Centre (Andrade, Nunes and Silva, 2012).

2.2.2 Wheel Alignment

Camber Angle refers to the angle at which the wheel will lean relative to the chassis when viewed from the front. When the top of the wheel leans towards the chassis this is negative Camber Angle and vice-versa. As described by Uzunov (2021) when a vehicle is cornering the outside wheel leans into positive camber and as such static camber can be used to maintain contact patch during cornering. As described by Tambe (2017) Camber Angle also creates an effect, known as Camber Thrust, which creates a stabilising force as each wheel pushes towards the centreline of the chassis. However, too much static camber causes increased tyre wear. Therefore, ideally minimum static camber is preferred to minimise this, and it would be beneficial if the position of the instant would cause a camber change as the vehicle rolls, this effect is known as Camber Compensation.

Toe Angle refers to the angle of the wheel when viewed from above, when the front edge of the tyre is further from the chassis this is known as Toe-out and is a positive angle value and vice-versa. Scrub, or Track Width Variation, refers to the lateral movement of a tyre along the ground perpendicular to the direction of travel. A vehicle's Motion Ratio (MR) is the ratio of movement between the wheel and the spring and is calculated using the equation below.

$$\text{Motion Ratio} = \frac{\text{Spring Displacement}}{\text{Wheel Displacement}}$$

2.3 Software

2.3.1 WinGeo3

WinGeo3 Suspension Geometry, referred to as WinGeo throughout this report, provides a comprehensive set of tools for use in the design of suspension systems, including suspension geometry design, kinematic and compliance analysis, and optimization. It has the ability to input the vehicle specifications, such as wheelbase, track width, and suspension type and then it generates a 3D model of the suspension system as suggested by Allen (2009). The software also includes a range of tools for the analysis of suspension systems, such as camber and toe analysis, and roll centre calculation. These tools give the engineer the ability to evaluate the performance characteristics of different suspension designs and identify potential areas for improvement. This is why this software has been used to model the front and rear suspension systems as it supplies the tools required to carry out a complete investigation into the suspension systems and the performances linked to them. Then it allowed the comparison between the two vehicles, #40 and #44, suspension systems to investigate which one is potentially better in terms of on-track performance.

2.3.2 VI Grade

VI Grade is a very powerful tool that can be used for the optimisation of the MCR Race cars due to the massive output of data that can be achieved from a single simulation run using VI Grade software. Due to this fact, it seemed pertinent to update the old MCR vehicle suspension system to gain a more accurate simulation of the Vehicle. Going forward it has many benefits when using an accurate model as it allows the MCR Race Team to use it to better decide on vehicle setups before arriving at different rounds, it also allows for potential development to be investigated before production begins to save time and resources, as suggested by Ferraris *et al.* (2019). The usefulness of the simulation continues in the fact that it can aid in the decision on where resources should be allocated by the team in potential

upgrades to the vehicles. This allows an increase in efficiency as it allows the team to avoid working on things that would be less beneficial to the team.

2.4 Experimental Design

For COG, Kinematic and Geometry Measurement investigations, the vehicle set up was checked on the setup floor to ensure consistency between tests, the setup floor was checked and positioned as close to true flat as possible, seen in Figure 8. For the Kinematic and Geometry Measurement investigations it was required that the wheel be removed. As such the vehicle was placed on blocks that matched the static ride height of the vehicle in order to ensure accurate testing.

Table 1 - Suspension Investigations Experimental Design 1

Investigation	Equipment	Methodology
Centre of Gravity	- Gantry - Set-up Floor - Inclinometer	- Suspended the left side of the car with the gantry, seen in Figure 3. - Pushed the car until it reached its balance point, where the COG is directly above the tyre contact to the floor. - Read off the car's angle of incline. This, using trigonometry, specifically tangent, allows the COG height to be calculated using half the track width. - Test is repeated for car #44
Spring Stiffness	- Spring Stiffness Test Rig - DTI - Corner Weight Scale	- Front and Rear springs removed from car #40 - Front spring fitted to the test rig as seen in Figure 4. - Compress the spring by 0.1 of an Inch, one revolution on the DTI and read off the weight measurement. - Repeat all the way to 1 Inch of compression - Repeat for the rear spring.
Tyre Stiffness	- Tyre Stiffness Test Rig - Corner Weight Scale - Air Pump - Pressure Gauge - Thermometer - DTI	- Ambient room temperature taken - Front tyre pressure set to 24 psi - Tyre was fitted to the test rig as seen in Figure 5. - Deflect the tyre by 2mm and read off the weight measurement. - Repeat this through a range of 12-24 psi. - Then repeat this for the rear tyre.
Damper Curves	- Damper Dynamometer	- A front and rear damper were removed from car #40 - Dampers were placed on the damper dynamometer and ran as seen in Figure 6. - The data sets were taken from the dynamometer - The dampers we put back on the car.

Table 2 - Suspension Investigations Experimental Design 2

Investigation	Equipment	Methodology
Front / Rear Kinematic	- Set-up Floor - Toe Plate - Two DTIs - Inclinometer - Height Gauge - Steel Rule - Clamp - Turnbuckles	- Replace the front and rear dampers with turnbuckles seen in Figure 12. - Clamp the brakes on to stop wheel rotation. - Attach the toe plate to the wheel hub. And the inclinometer to the brake disk to measure camber change. - Align the DTIs against the toe plate as seen in Figure 11 and zero them. - Measure the height of the bottom of the toe plate. - Measure the length of the turnbuckle and remain consistent throughout with how you measure which can be seen in Figure 12. - Wind the Turnbuckle so it reaches the end of its travel and where the wheel is in full bump. - Rotate the turnbuckle two times lowering the wheel and take measurements from the two DTIs the height gauge, the damper length, and the inclinometer. - Repeat this until the end of travel is reached on the turnbuckle.
Front / Rear Suspension Geometry Hardpoint 40	- Set-up Floor - Cross Line Lasers - Height Gauge - Steel Rule -Engineers Square	- Replace Front and Rear dampers with turnbuckles. - Set up laser facing down the center line of the chassis and across the relevant axle based on known points to establish x/y axis. Shown in Figures 10 & 11. - X coordinates were measured from the axle line, y coordinate from the chassis center line and z coordinates from the floor, which can be seen in Figures 10 & 11. - Process repeated for front and rear of car 40.
Front / Rear Suspension Geometry Hardpoint 44	- Set-up Floor - Height Gauge - Steel Rule	- Due to time constraints the same process used to measure 40 could not be completed on car 44. - Vehicle was set up to match 40. - Difference in suspension coordinates were measured between 40 and 44 by eye using a steel rule and a height gauge. - These differences are inputted into WinGeo to create 44 suspension model.

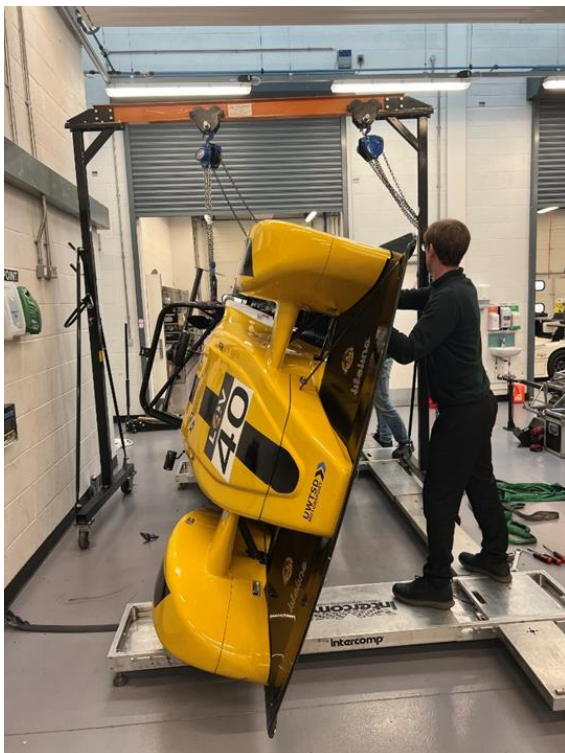


Figure 3 - COG Investigation



Figure 4 - Spring Stiffness Investigation



Figure 5 - Tyre Stiffness Investigation



Figure 6 - Damper Curve Investigation



Figure 7 - Kinematic Investigation Rear



Figure 8 - Set-up Floor

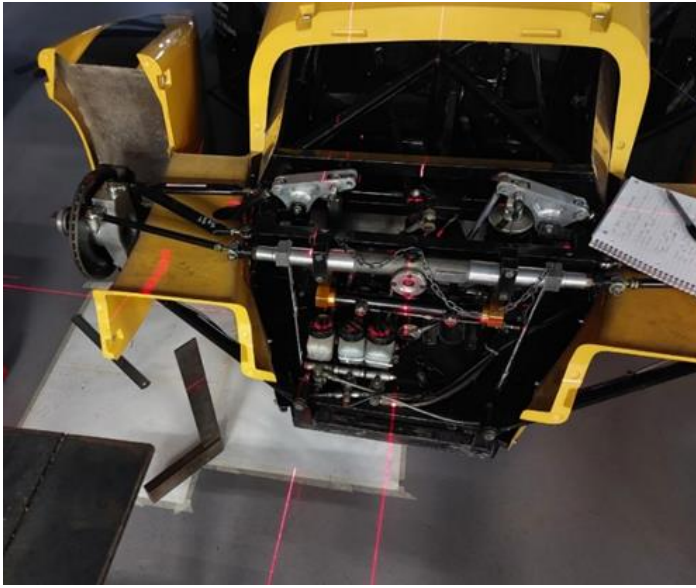


Figure 11 - Suspension Geometry Measurements 1

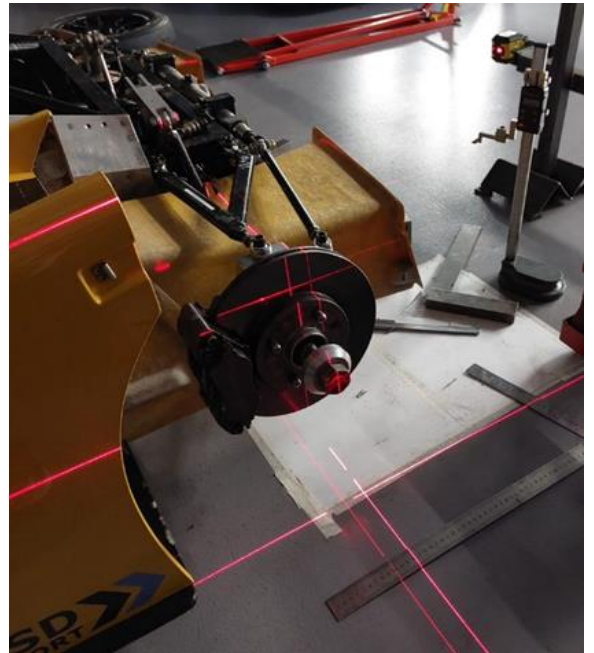


Figure 10 - Suspension Geometry Measurements 2



Figure 9 - Kinematic Investigation Front

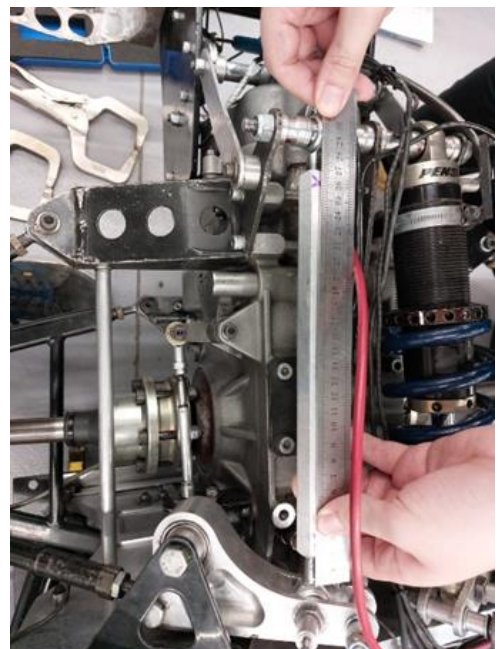


Figure 12 - Turnbuckle and Spring Change Measurement

2.5 Model Validation and Analysis

2.5.1 Front Suspension Analysis

Overlaying the measured Kinematic Investigation of the front suspension data against the WinGeo data for the front suspension allows us to validate the accuracy of the WinGeo model created. This is important because if the models are not accurate then the data gotten from WinGeo will not be useful as it will not match what the actual vehicle is doing, rendering the model useless. Therefore, it was crucial that the models reflected the real-world data, which can be seen in Figures 13, 14, 15, 16 and 17 below, where the data does correlate well and suggests that the model is accurate. This allows further work to continue using the model as it has been validated. For the following figures, every time measured is referenced in this section it is referring to the Front Suspension Kinematic Investigation data for car #40. A full table of the front measured data can be located in the appendix in Tables 6 & 7.

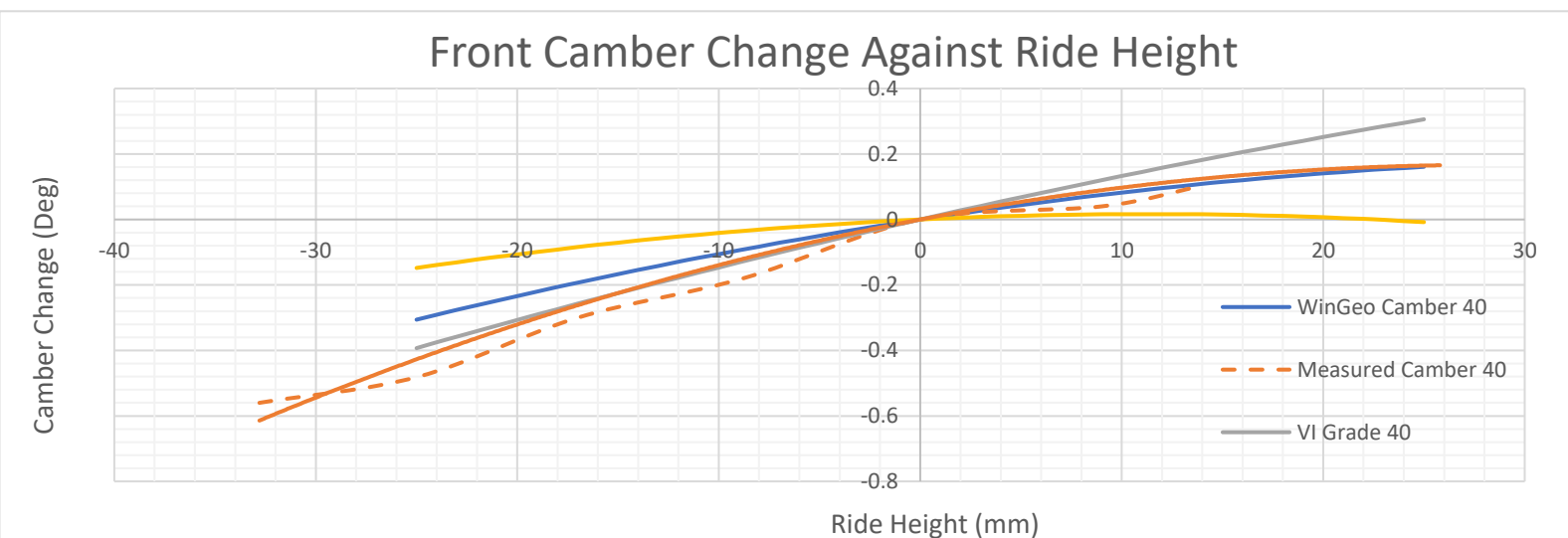


Figure 13 - Front Camber Change Against Ride Height

From Figure 13 it can be seen that the measured camber change curve correlates well with the WinGeo camber change #40 curve, seeing this it is clear that the front suspension model is valid in the sense that it accurately displays how the cars camber would act dynamically. The Figure also shows that vehicle #44 seems to have less camber change through bump and droop. This could be due to the angle of the a-arms on #44 being close to being parallel to the ground as seen in the Figure 62, as suggested by Gaffney and Salinas (1997). While this does support the theory that there isn't much camber in bump due to the angles of the A-Arms we would want to carry out another Kinematic Investigation on #44 to say for certain but due to the scope of the project were unable to include it. Moving onto the VI Grade camber curve it can be seen that the curve is less accurate than the WinGeo #40 one suggesting that the VI Grade model needs to be updated.

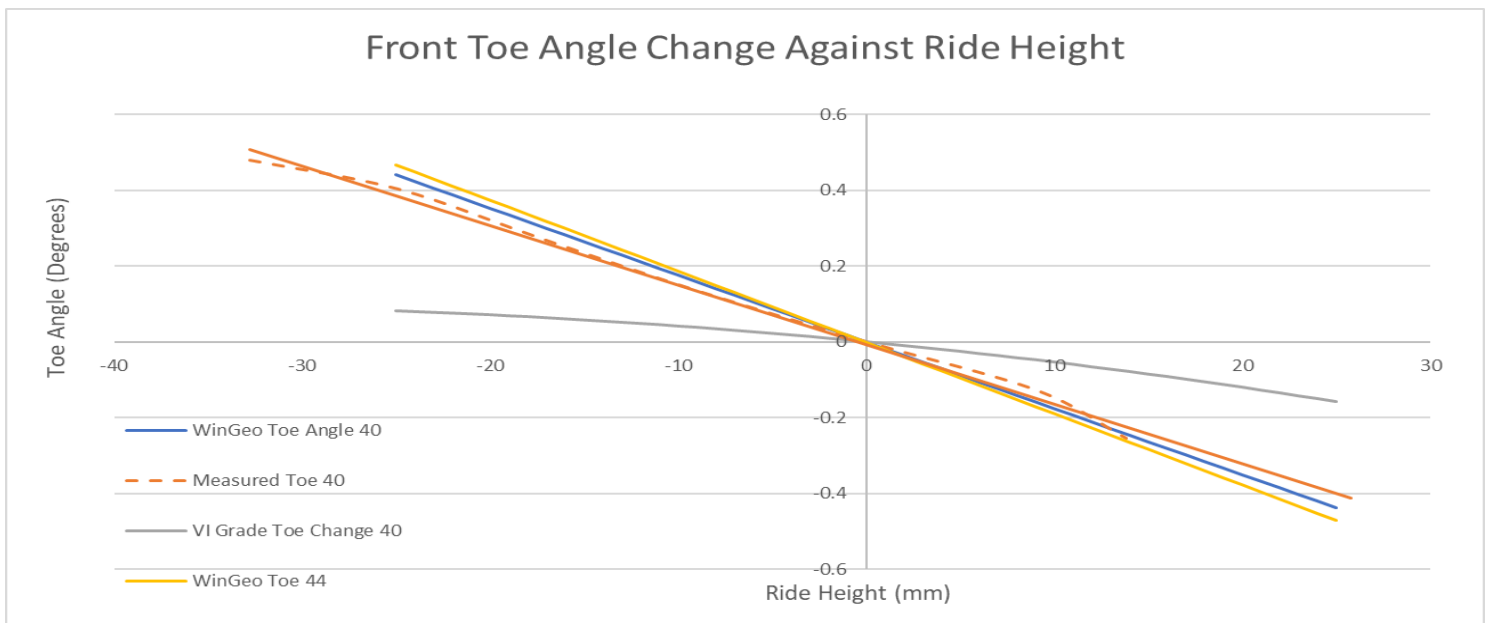


Figure 14 - Front Toe Against Ride Height

Analysing Figure 14 as seen above, it can be stated that the data for the measured toe angle change on car #40 and the data for the WinGeo toe angle change for car #40 demonstrate a strong correlation implying that the model of the front suspension system is accurate to how the #40 behaves. Looking at the #44 toe angle curve now, it is clear to see that there isn't much difference in the way car #44 toes through the chassis' range of motion. This could be due to the fact that the only major difference between the front suspension systems is where the lower a-arm mounts to the chassis on #44, which mounts lower than #40's does by 14.37mm. As described by Sancibrian *et al.* (2010), Due to the fact that the steering arm didn't change positions between the vehicles, it would be expected that there wouldn't be much of a change to the amount of toe angle change. It is clear that the current VI Grade models' toe is not accurate to what is being seen with the measured data. Therefore, this is further justification for why the model should be updated.

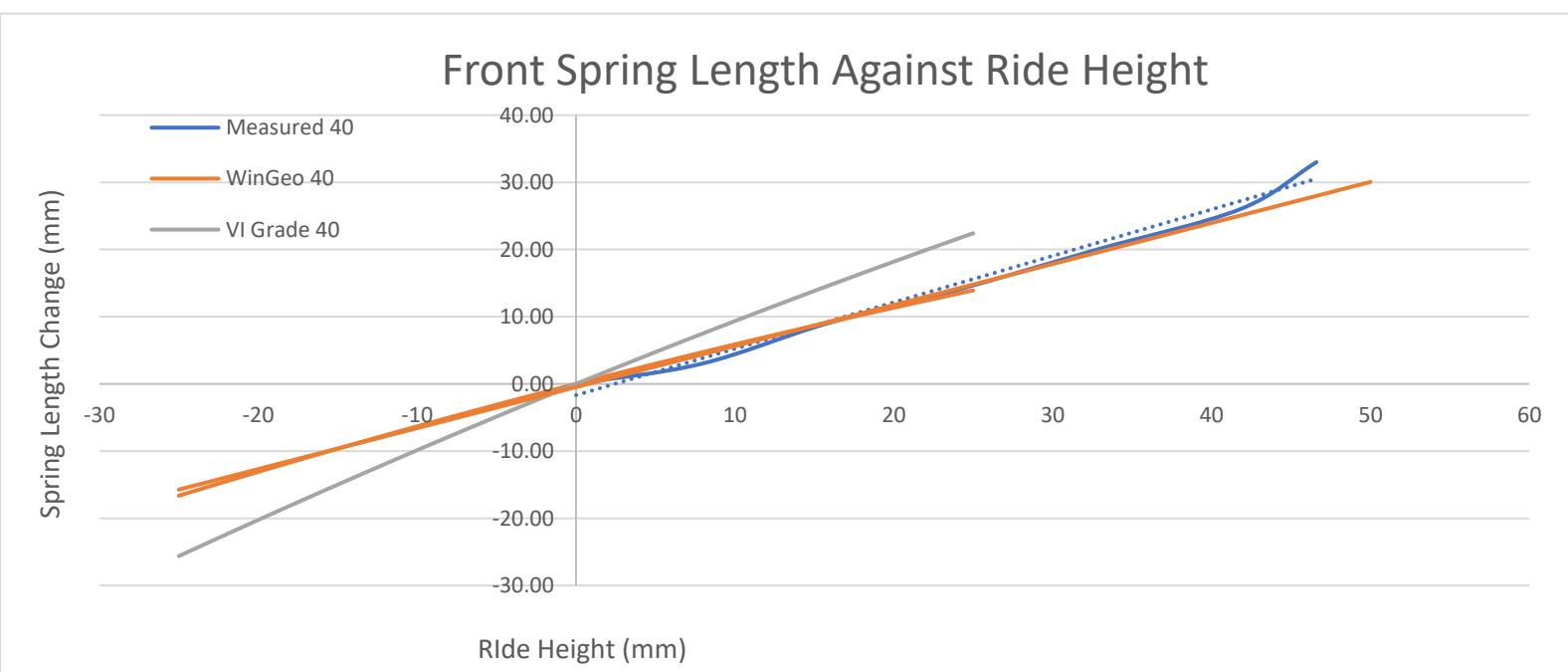


Figure 15 - Front Spring Length Against Ride Height

As seen in Figure 15, which depicts the change in the length of the vehicles spring as the ride height changes. It can clearly be seen that the two data sets, measured #40 and WinGeo #40, display a high degree of correlation. Where small deviations in the measured curve can be seen are most likely due to the fact that during the measurement of the spring length, a steel rule was used and a judgement was made on the length, so this could result in a slight error, as is seen Figure 15, in the data set. The graph is very useful in the validation of the front suspension system model as the gradient of these lines gives the motion ratio for the Vehicle. For ease in showing that the model is valid, only #40s data is on this graph, the full graph for the front system is located in the appendix in Figure 61. Considering the VI Grade data now it is visible that it is less accurate than the WinGeo model created of the front suspension, thus sustaining the idea that the current model used by the MCR Race Team needs reviewing to make it more accurate.

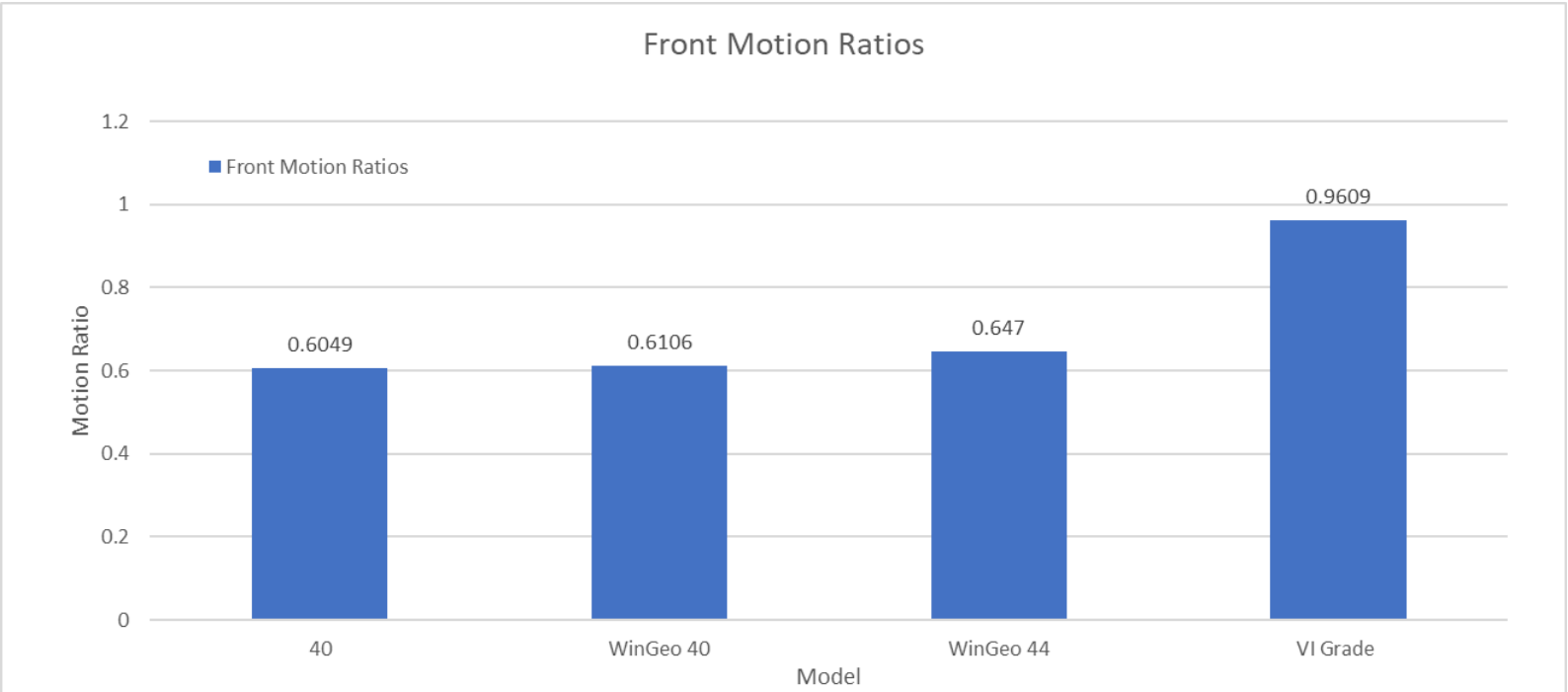


Figure 16 - Front Motion Ratios

From Figure 16, it can clearly be seen that the measured #40 data and the WinGeo data for #40 correlate well with each other. This justifies the thought that the created model is accurate, as it is within 0.9% of error. It also depicts #44’s suspension system as having a slightly higher motion ratio at 1:0.647, whereas #40 only has a 1:0.6101 ratio. Therefore, this shows that for every 1mm of wheel travel, there is 0.647mm of spring compression on car #44 and 0.6101mm of spring compression on car #40. While this figure helps validate the new created front suspension model it also helps disprove the current VI Grade front models validation, which has an error of 57.4%, giving reason to update the model to be more respective of how the car behaves.

Front Scrub Against Ride Height

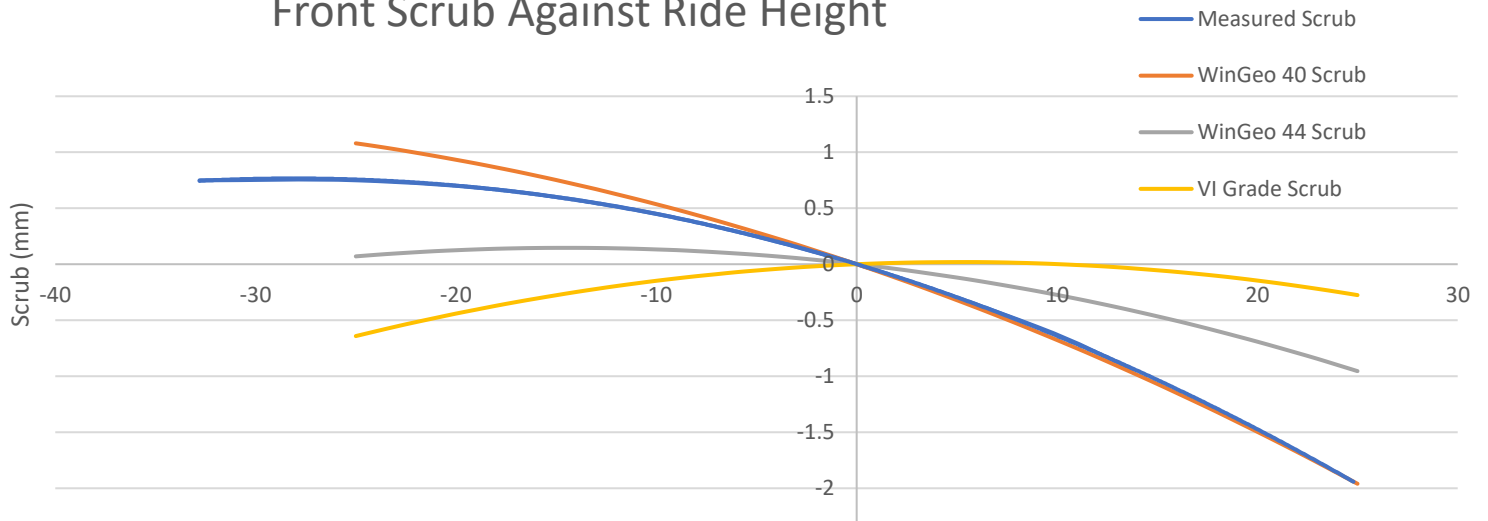


Figure 17 - Front Scrub Against Ride Height

Looking at Figure 17 it is clear that the measured scrub movement of car #40 has a strong correlation with that of the data of the WinGeo scrub movement for car #40. This only strengthens the postulation that the model of the front suspension system for #40 is valid as it shows the curves match, while only slightly wandering towards the lower side of the vehicles ride height, and while this is the case the vehicle would not likely reach that low. Looking at all the curves on the figure it is clear that the VI data is the furthest out and hence is another reason why the current VI Grade model needs to be reviewed to update it to a more representative geometry to allow for more accurate predictions when simulating. Analysing the data for #44 we can see that it has a fair amount less lateral tyre movement with a change in ride height.

Front Camber Change Against Roll Angle

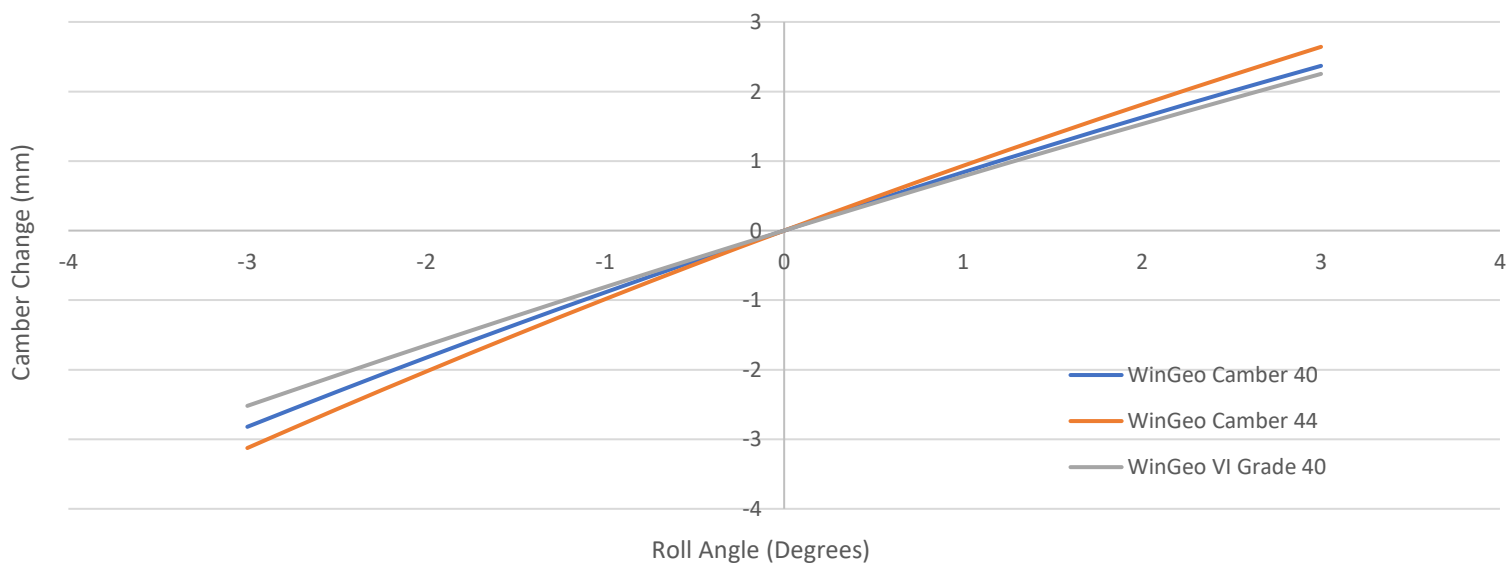


Figure 18 - Front Camber Change Against Roll Angle

From Figure 18 it can clearly be seen that vehicle’s #40 and #44 have a very similar reaction to the chassis rolling, which represents the vehicle cornering, and even the current VI Grade model suggests it does a similar thing. While this graph is useful at understanding how the camber acts as the vehicle goes around a corner. It has a greater use, if you look at the gradients of the lines it will give you the camber compensation if you take the gradient of the lines away from 1. The camber compensation can be seen on the graph below, Figure 19, where it can be seen that between cars #40 and #44, #40 has the most camber compensation, this could be due to the fact that angle or length of the suspension arms differ as suggested by De Paula Eduardo (2005). However, out of all three, the graph suggests that the VI Grade vehicle model has the most camber compensation. While it looks like vehicle #44 has a lot less camber compensation that #40 the difference is only 0.0961, which isn’t a lot.

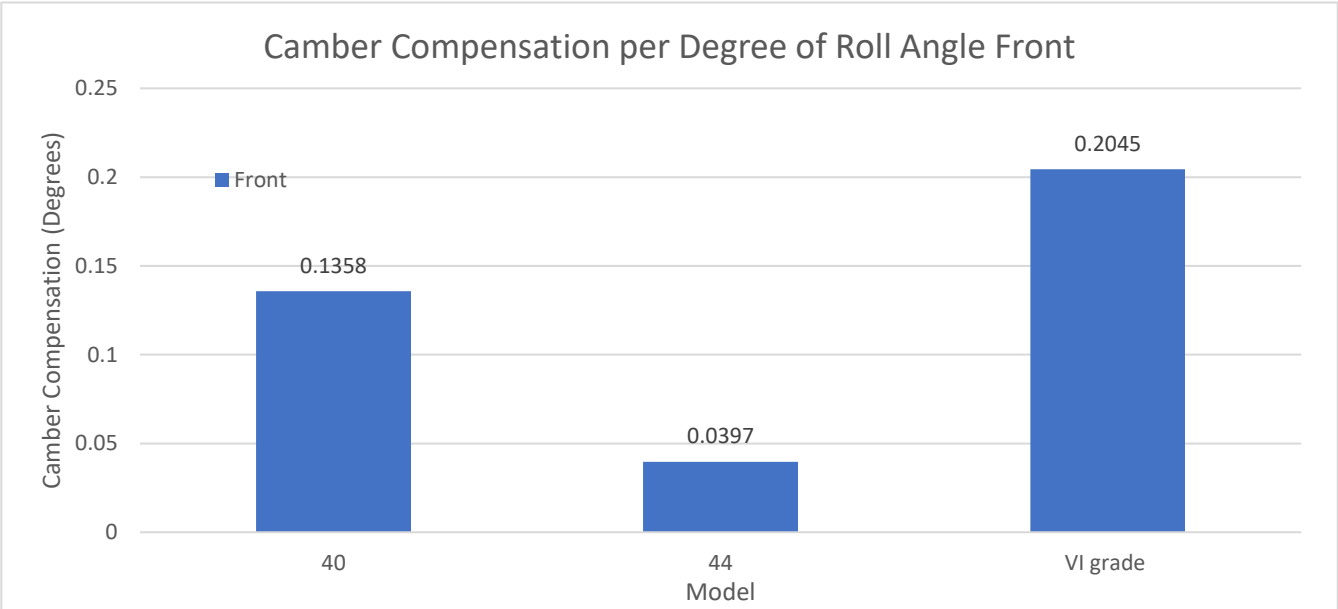


Figure 19 - Camber Compensation per Degree of Roll Angle Front

2.5.2 Rear Suspension Analysis

Results from the rear kinematics investigation are represented as the measured values on the figures in this section, a full table of results is shown in appendices (Tables 8 and 9).

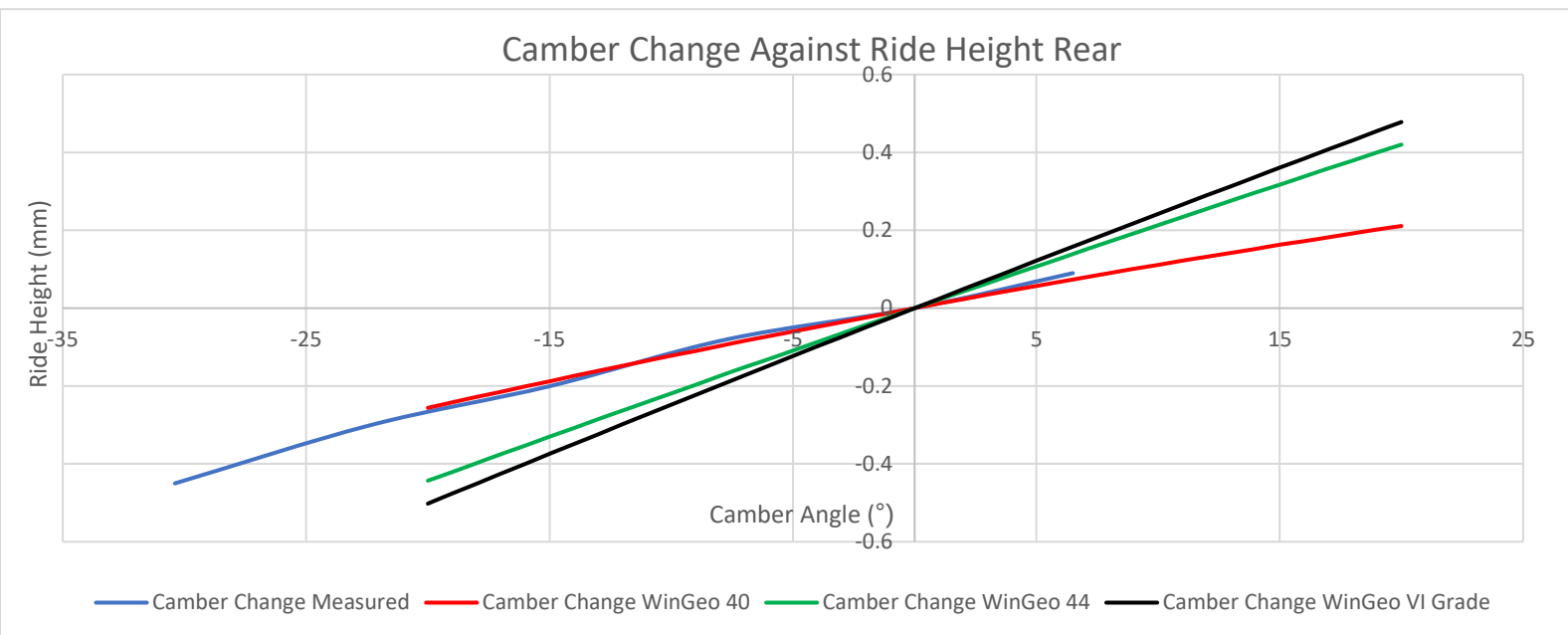


Figure 20 - Camber Change Against Ride Height Rear

From Figure 20 it can be observed that the measured camber change correlates well with the camber change observed in WinGeo for the #40 model. This change in camber is due to the wheel moving through its arc as defined by the IC as further described by Balike, Rakheja and Stiharu (2011). The camber change of #40 can be observed to be non-linear, this is due to the change in suspension geometry as the systems bumps or droops resulting in a different in the rate of change of IC height when compared against ride height as shown in IC Height figure as further described by Theander (2004). This suggests that the WinGeo model is accurate. When comparing #40 and #44 it can be observed that #44 has a greater degree of camber change for the same ride height change suggesting that the older chassis, #44, has more camber compensation than #40. It can also be observed that the teams current VI Grade model is showing camber change similar to #44 suggesting that the current suspension model is a more accurate representation of #44 than #40, however the gradient of the line differs suggesting that the VI Grade model is slightly inaccurate.

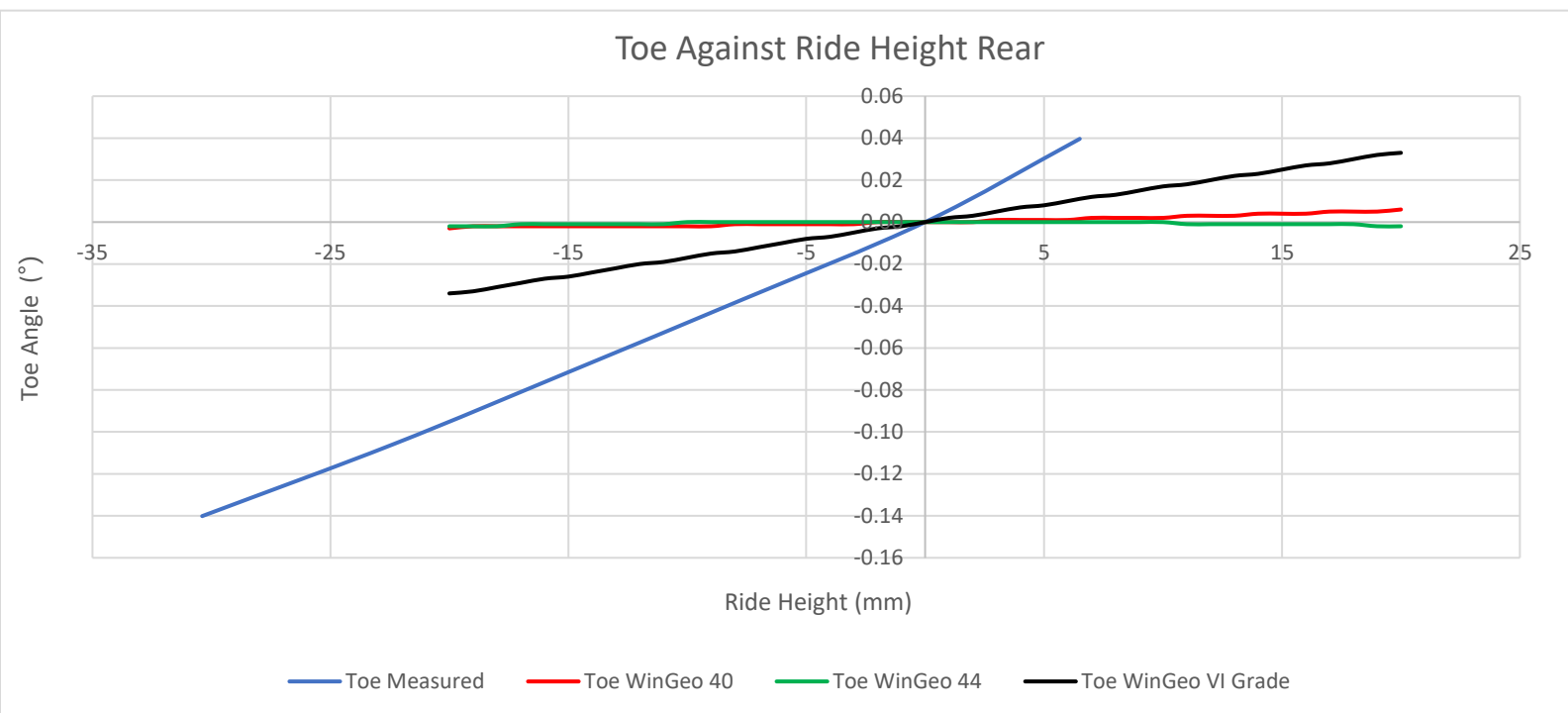


Figure 21 - Toe Against Ride Height Rear

From Figure 21 it can be observed that the measured toe change is drastically different than can be seen in WinGeo. However due to the models for #40 and #44 show very little change in toe angle and that very little if any toe change should be expected as the suspension member that controls wheel steer, for rear suspension the toe arm, is connected directly to the upper a-arm. This is supported to a lesser extent by the VI Grade Model which, despite showing a greater toe change, still shows a trend closer to that shown by the #40 and #44 models. These factors suggest that the experimental data may be incorrect. This may be a result of using an old Toe plate, which shows many scratches and indentations, that may have led to error when recording small changes as the greatest change in the test was 0.3mm.

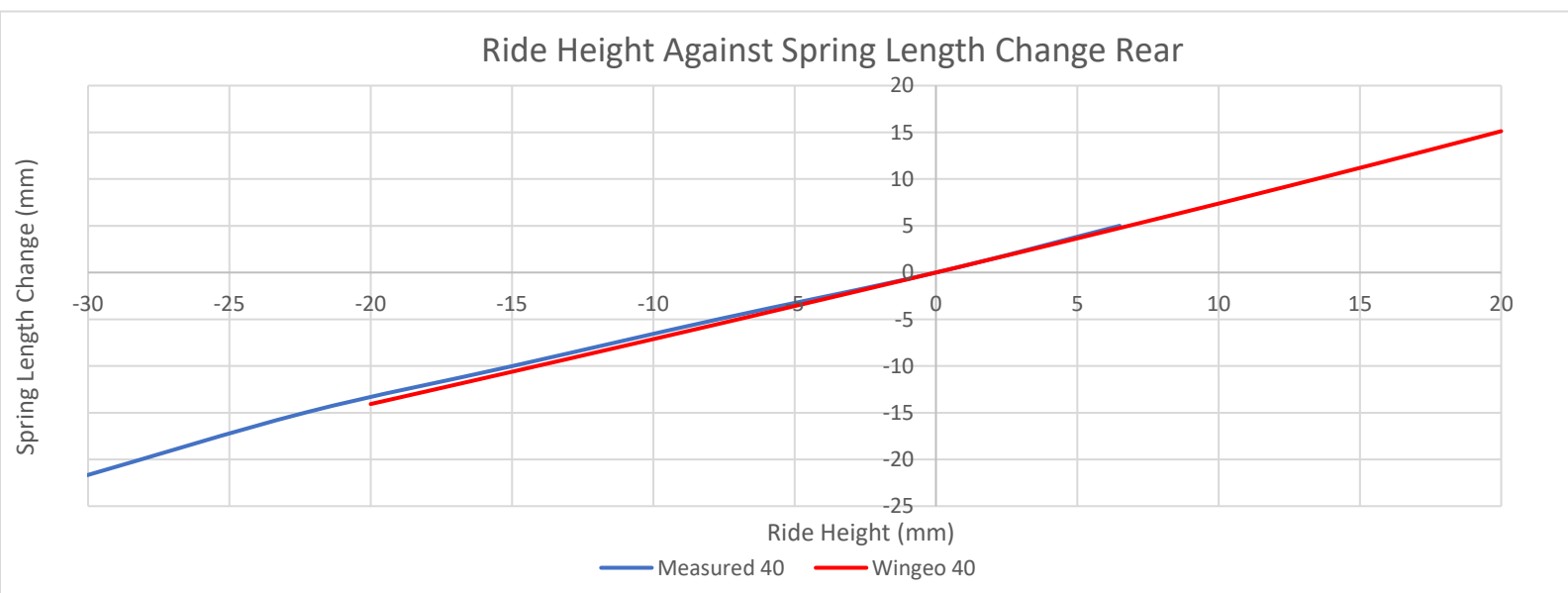


Figure 22 - Ride Height Against Spring Length Change Rear

Figure 22 shows spring length change against ride height change, the gradient of this line is the suspensions systems motion ratio (Figure 65). It can be observed that the WinGeo model correlates well with the measured data, the slight deviation shown at ride height values below -10 may be cause by experimental procedure as spring length values were recorded by eye using a steel rule to the nearest mm, this may explain the variation between the measured motion ratio and WinGeo motion ratio, 0.71 and 0.73 respectively. The Motion Ratios for the rear suspension is shown in Figure 23. From this graph it can be observed that the measured data and the WinGeo models of #40 and #44 show a similar motion ratio while the VI Grade Model is over predicting.

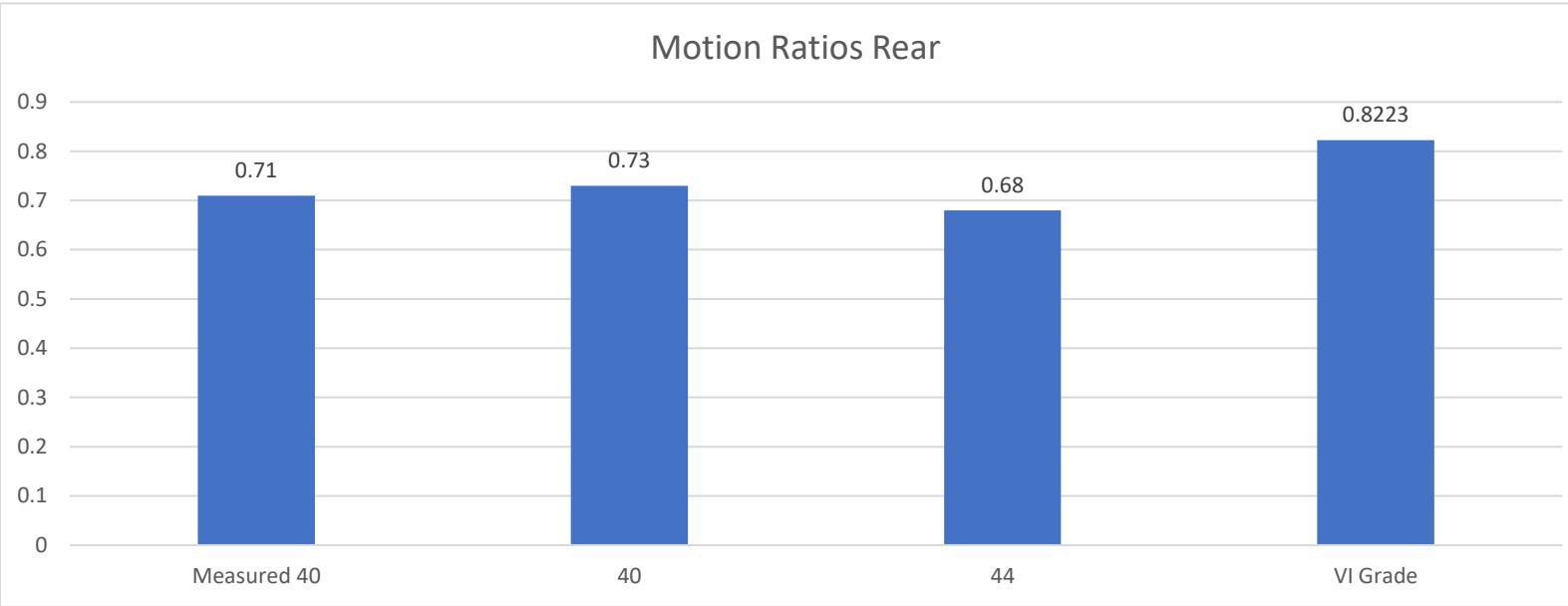


Figure 23 - Motion Ratios Rear

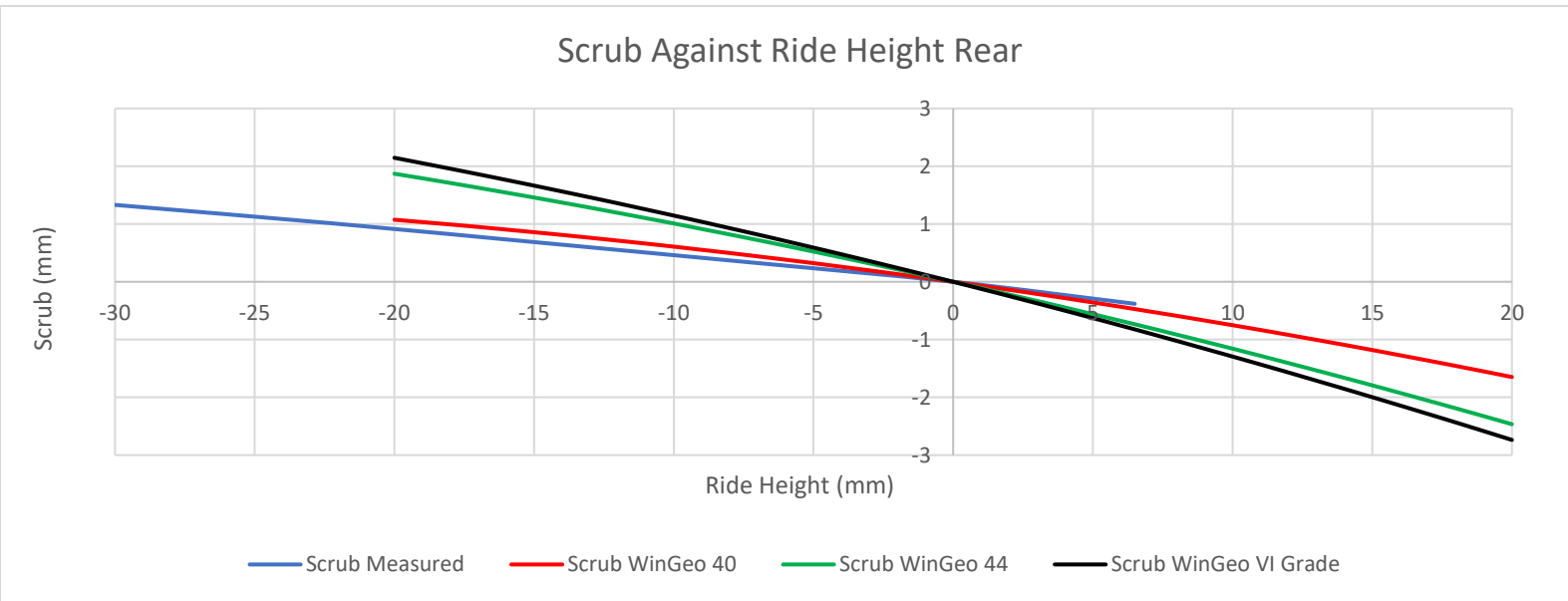


Figure 24 - Scrub Against Ride Height Rear

From Figure 24 it can be observed that the measured data correlates well with WinGeo #40 model, however there is a slight deviation under -10 ride height. This may be caused as the data recorded to calculate scrub was gathered using the same DTI and Toe Plate setup previously described. From this figure it can be further observed that #44 has a greater magnitude of scrub than #40, this may be due to the #44 chassis being designed when the SRCC Sports 2000 series used cross-ply tyres as opposed to the radial slicks that #40 was designed for. The reduction in scrub may be an attempt to reduce heat generation in the rear tyres as the current radial slicks are softer compound of rubber than the cross-ply's previously used, however tyre temperature data would need to be gathered to confirm this. The reduction in scrub could also be caused as a side effect of the reduction in camber change between chassis #40 and #44. It can also be observed that the VI Grade model is more accurate to #44 than #40 however the VI Grade model is overpredicting scrub.

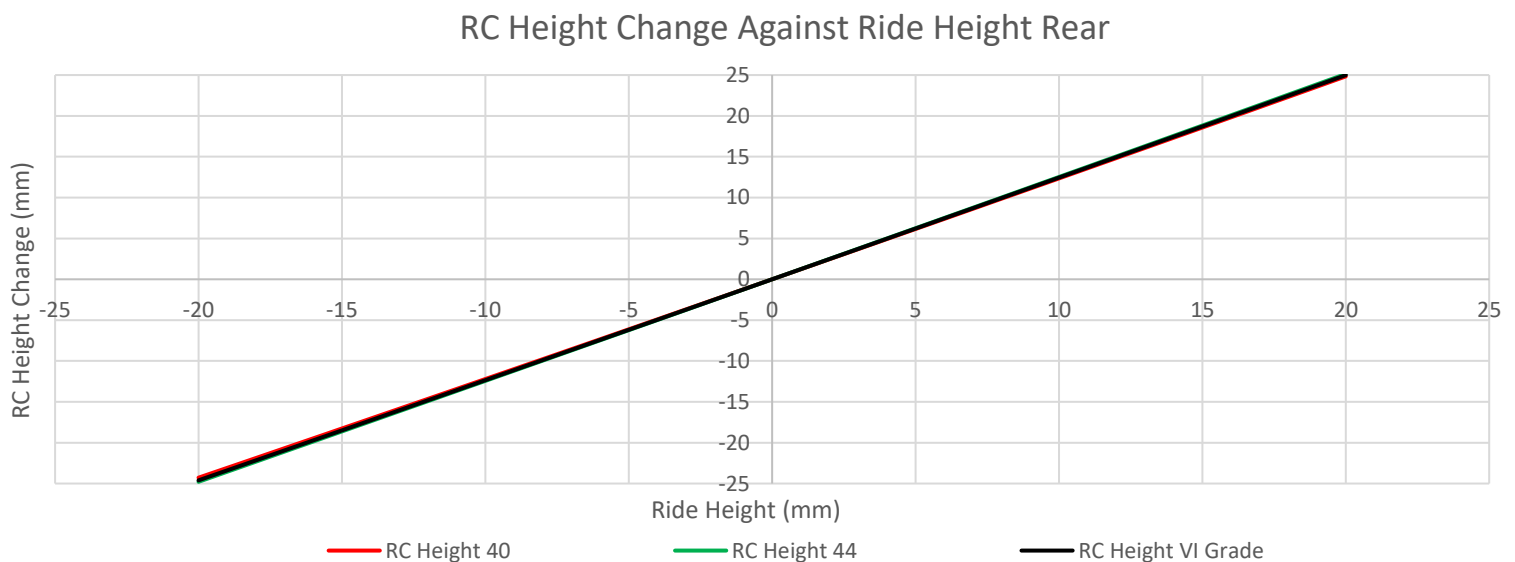


Figure 25 - RC Height Change Against Ride Height Rear

The graph shown in Figure 25 represents the change in RC height compared with change in ride height. From the graph it can be observed that all three models are indistinguishable from one another, as RC lateral change was zero throughout the ride height range it can be assumed that the difference in change of RC position between the models is negligible. RC height can be observed to be changing at a faster rate than that of ride height. This is due to the IC height changing during the ride height sweep, as shown in Figure 26. As the RC is dependent on IC position this change in IC position causes the greater rate of change of RC height when compared to ride height.

RC Height Change and IC Height Change Against Ride Height

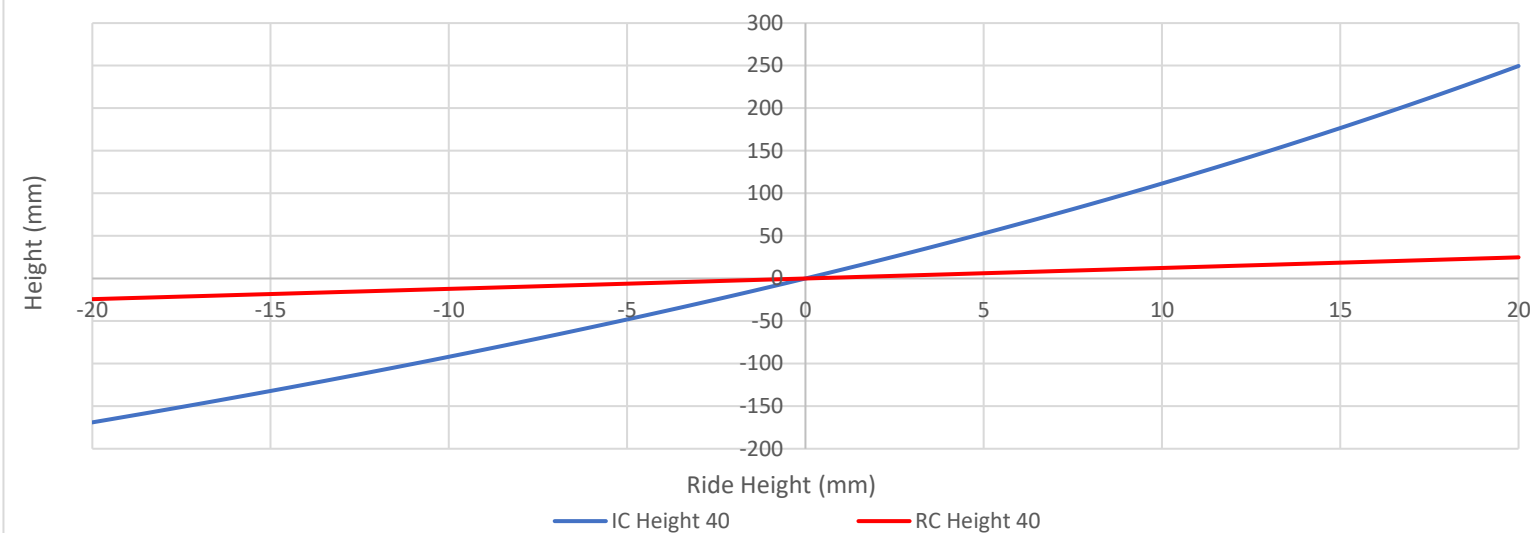


Figure 26 - RC Height Change and IC Height Change Against Ride Height

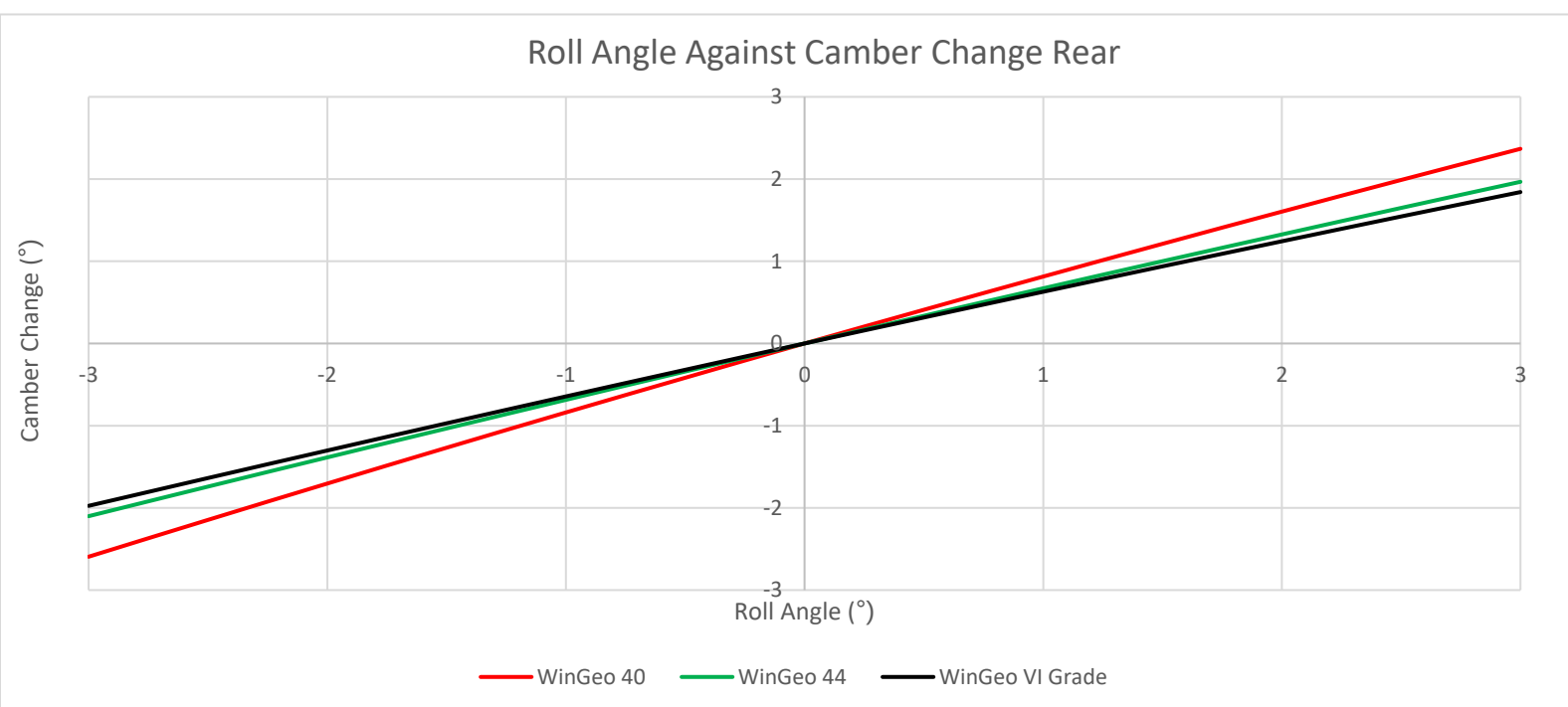


Figure 27 - Roll Angle Against Camber Change Rear

Figure 27 can be used to calculate CC using the gradient of the line shown for each model using the equation below.

$$\text{Camber Compensation} = 1 - \text{Gradient}$$

The results of this are shown in Figure 28. From Figure 28 it can be observed that #44 has greater camber compensation than #40, whereas the VI Grade model can be observed to be a better representation of #44 than #40 but over predicting CC. The difference in CC between #40 and #44 maybe a result of the different tyres the cars were designed to use as cross-ply tyre deform more during roll, due to their construction, and therefore require more camber.

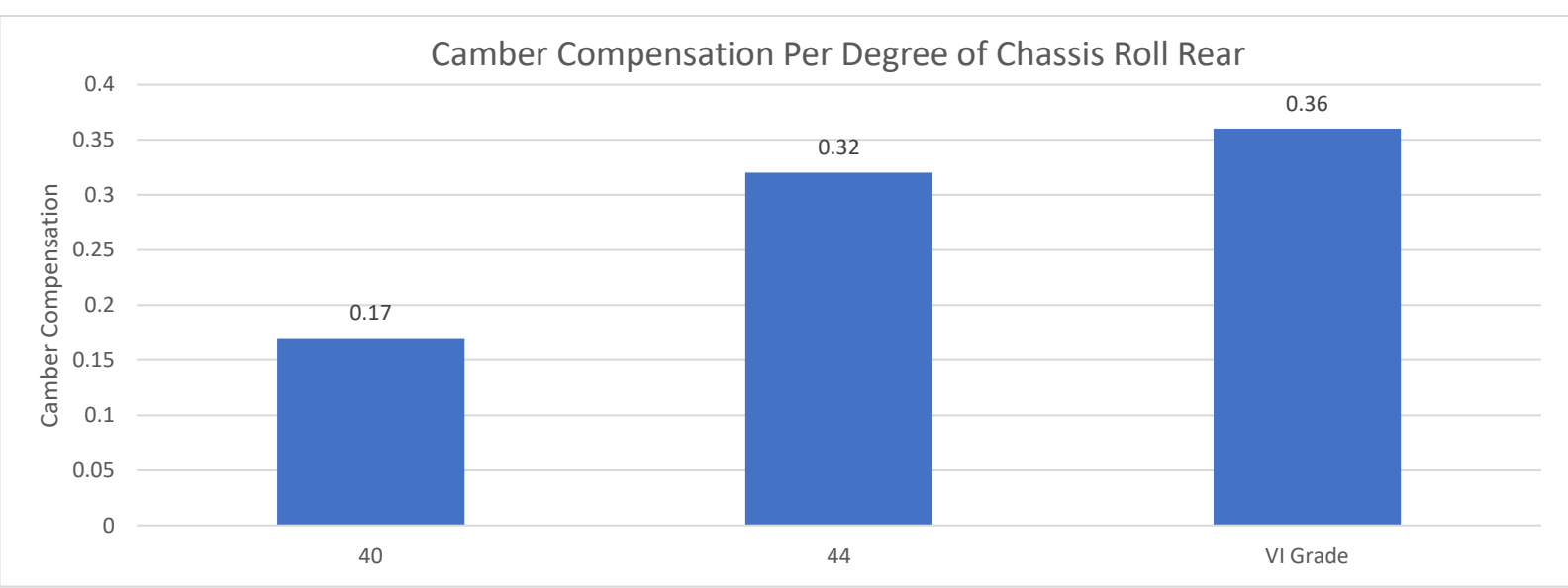


Figure 28 - Camber Compensation Per Degree of Chassis Roll Rear

2.5.3 Front to Rear Analysis

This section compares the WinGeo models of the front and rear suspension systems of cars #40 and #44.

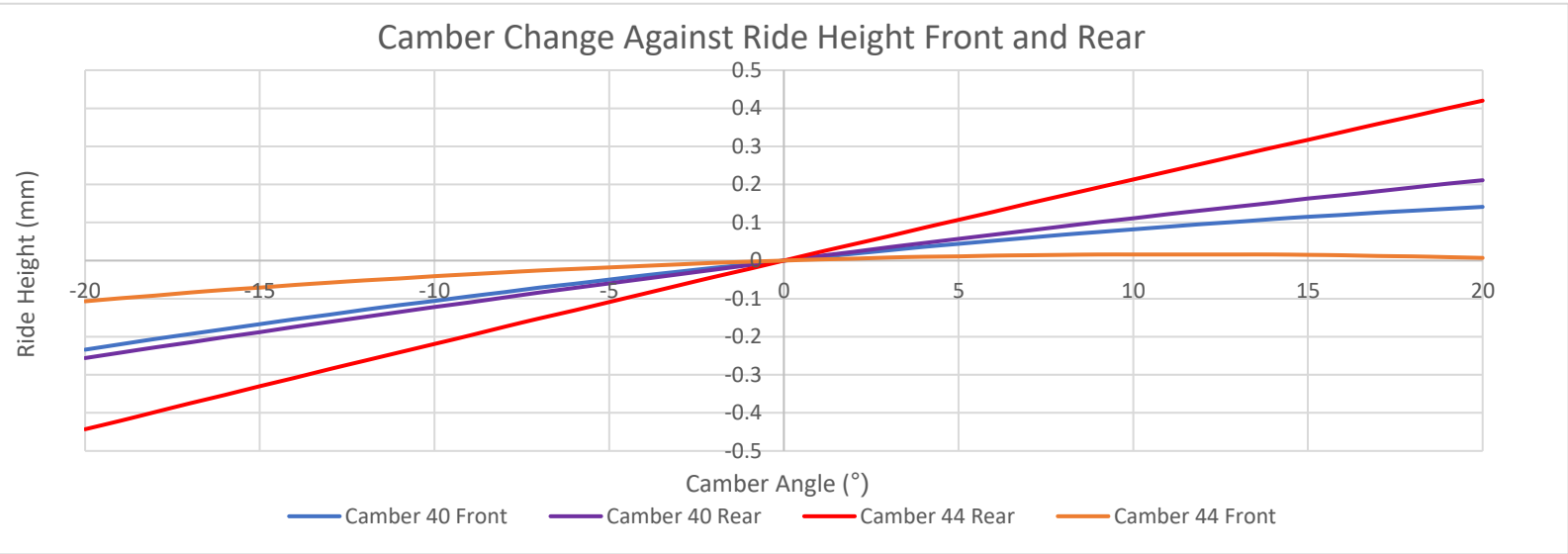


Figure 29 - Camber Change Against Ride Height Front and Rear

From Figure 29 it can be observed that the front and rear camber change on #40 is follows the same trend and is similar throughout its travel, this would make the car more predictable as the size of the contact patch on the front and rear wheels would change at a similar rate, minimally effecting handling characteristics. This further supported by Figure 30 where it can be observed that #40 has similar CC per degree of chassis roll between the front and the rear. It can be observed on both Figures 29 &30 that #44 has a much greater change of camber and CC at the rear when compared to the front, this would give the car a slight understeer balance. Due to the rears compensating more for roll and maintaining a better contact patch when compared to the front, as a result the front would lose more grip than the rear during roll.

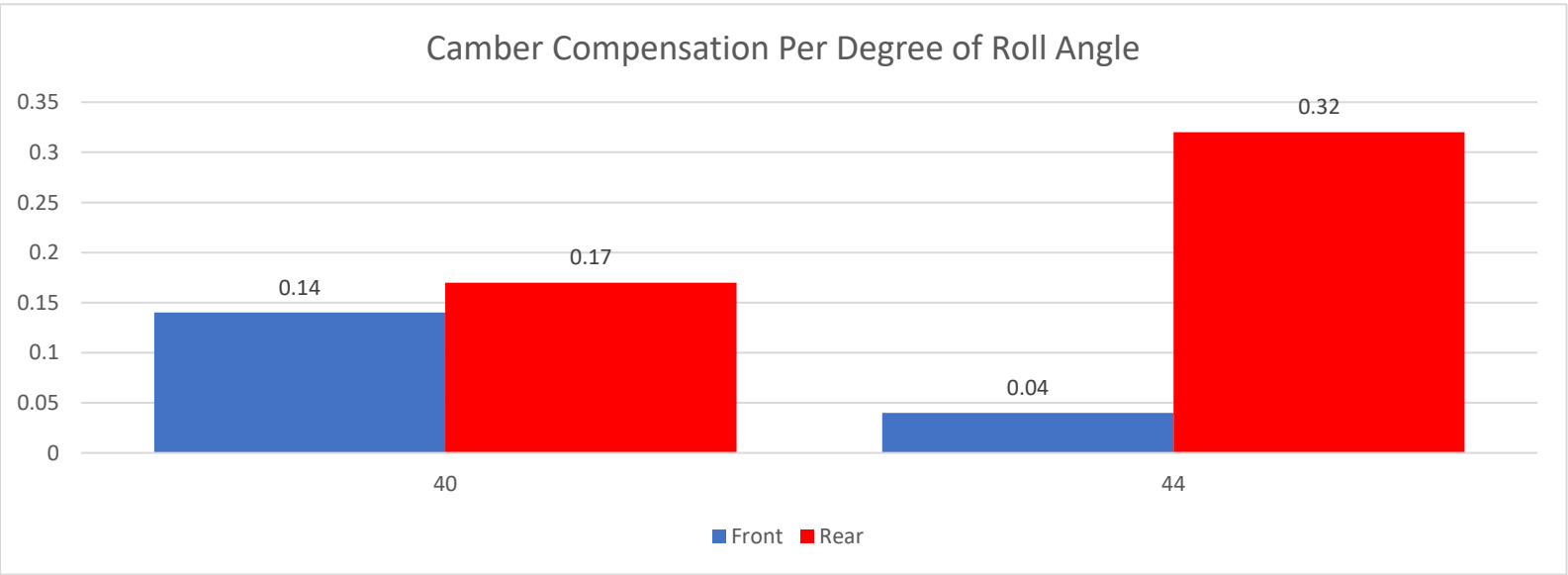


Figure 30 - Camber Compensation Per Degree of Roll Angle

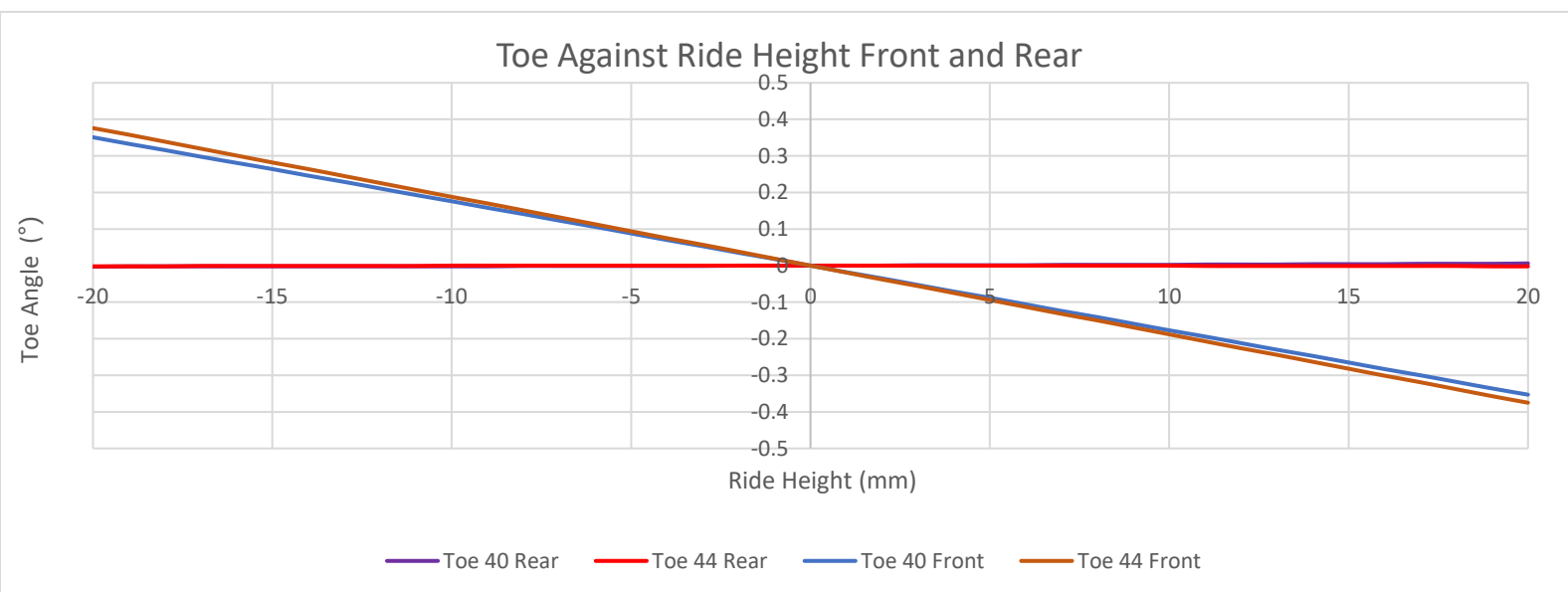


Figure 31 - Toe Against Ride Height Front and Rear

From the graph in Figure 31 it can be observed that #40 and #44 have similar Toe change at the front and rear, the rear Toe change between chassis is nearly indistinguishable on the graph. Both chassis can be observed to have the same characteristics, where front Toe increases (Toe-out) under bump and decreases (Toe-in) under droop. As described by Vijayakumar and Ganesh (2013) Toe-out can increase cornering performance and Toe-in can increase straight line stability. Therefore, the front Toe characteristics may be beneficial as during braking and corner entry, when the front is in bump, the wheels will toe out giving a better turn in response. Additionally, as the front tyres gain Toe-in as they droop this may be beneficial as it would give the car stability on under acceleration on corner exit. Theander (2004) explains how static Toe should be kept at minimum in order to minimise rolling resistance, therefore the dynamic toe change shown in Figure 31 provides the benefits of toe-in and toe-out as described during cornering and return to their original position on the straights to minimise rolling resistance.

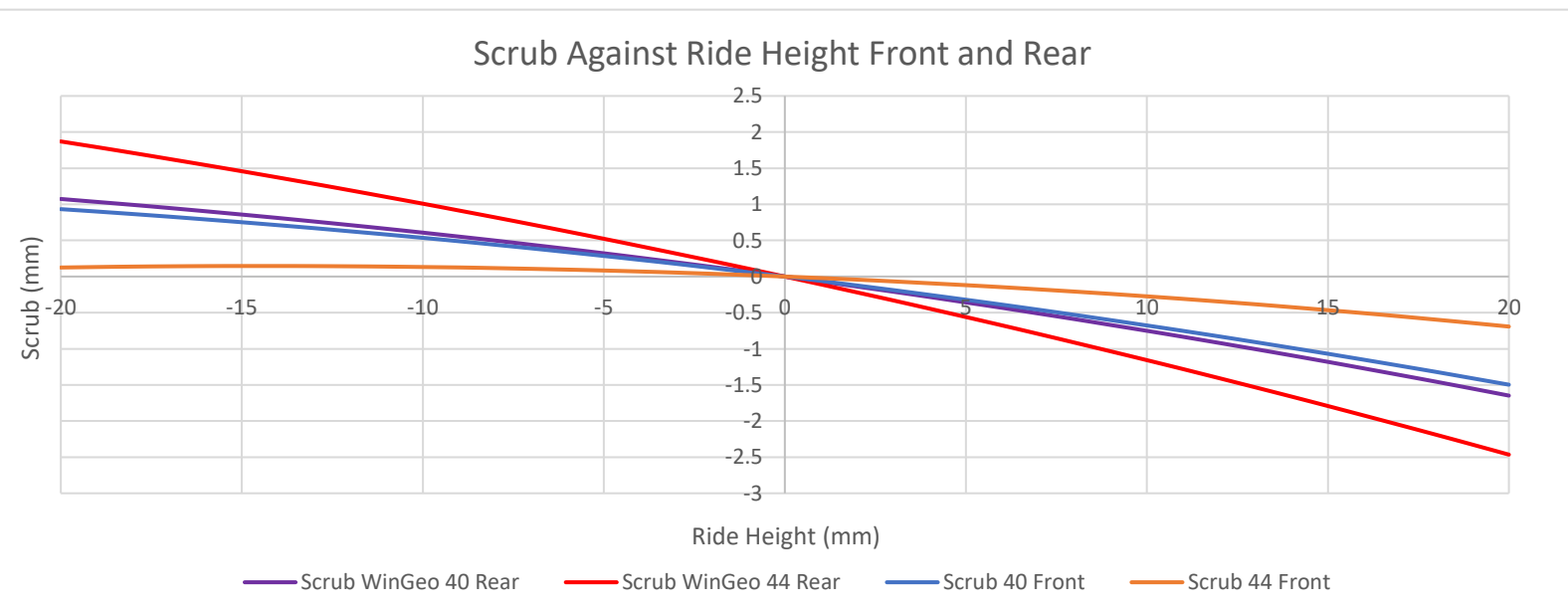


Figure 32 - Scrub Against Ride Height Front and Rear

From Figure 32 it can be observed that the scrub between the front and the rear of #40 follows the same trend and is of a similar value. In comparison #44 has greater scrub at the rear and less scrub at the front. This trend between cars matches that shown in Figure 29, suggesting that the difference in scrub between #40 and #44 may be a side effect of the change made to the geometry to reduce the camber change. This is supported by Chen, Tsai and Lai (2012) who describe how scrub is a result if a slight change in the lateral position of the wheel as it moves through the arc defined by the instant centre.

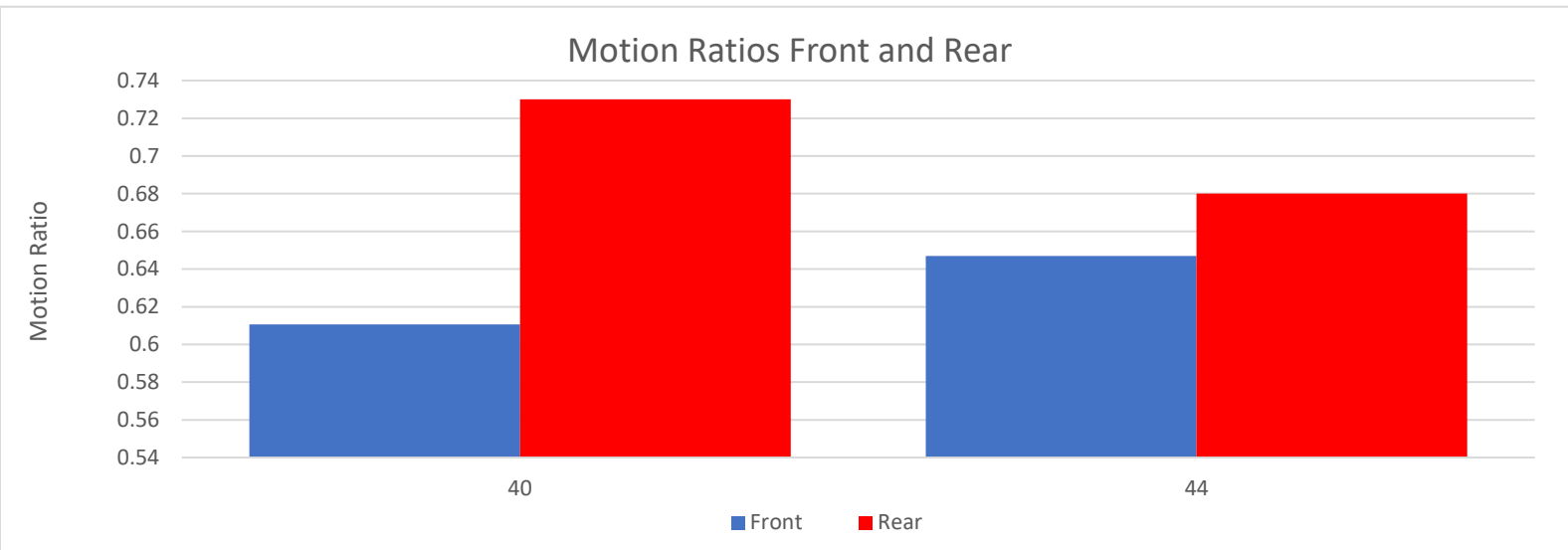


Figure 33 - Motion Ratios Front and Rear

From the graph shown in Figure 33 it can be observed that both chassis show the trend of a higher rear MR compared to front. Given that a lower motion ratio will reduce wheel rate, as shown in the equation below, this would give #40 a more oversteer characteristic, given equal front and rear spring rates, as the motion ratio will reduce the wheel rate to a greater degree in the front than the rear. However, it can also be observed that the difference in MR between the front and the rear is greater on #40 than #44, suggesting that #40 would have a greater resultant oversteer characteristic.

$$Wheel\ Rate = Spring\ Rate \times Motion\ Ratio^2$$

The finding of the comparison between the front to rear characteristics between the chassis suggests that, while #40 will have a greater oversteer characteristic than #44 due to the MR, it will have a more linear 'predictable' behaviour during cornering due to the similarity in the rate of change of camber between the front and the rear. Both cars will possess the same dynamic Toe change on corner entry and exit. #40 is likely to have better rear and worse front tyre life as a result of the differences in scrub.

2.6 VI Grade

After verifying that the new suspension models created were more accurate to real-world data, I updated the current VI Grade model to reflect those changes seen with the more accurate models for car #40, these changes can be seen in Figures 63 & 64, and added them to the overall VI Car Real Time model. Next using the validated suspension models that were adapted from the current model, using the differences seen between the suspension systems of cars #40 and #44, a suitable model for #44 was created. This allows the suspension systems to be compared when simulated around a track to give a better idea of how they would act on a track.

Before being able to run the simulation some other data was required to be input into the models, this was data that was gathered from the other tests that have been described in the experimental design.

Everything that was added to the models was the average COG heights, and the unsprung mass of both cars, of which was not changed until after some of the runs had been completed, the damper curves were plotted in VI Car Real Time as can be seen in the Figures 34 & 35 below, the current spring rates and preload.

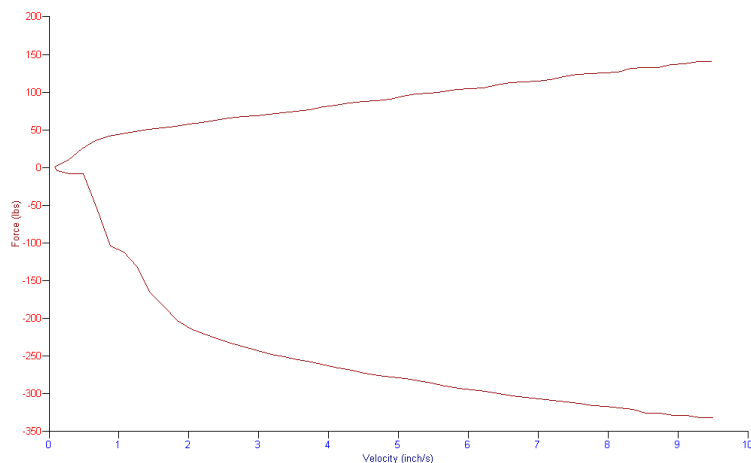


Figure 34 - Front Damper Curve

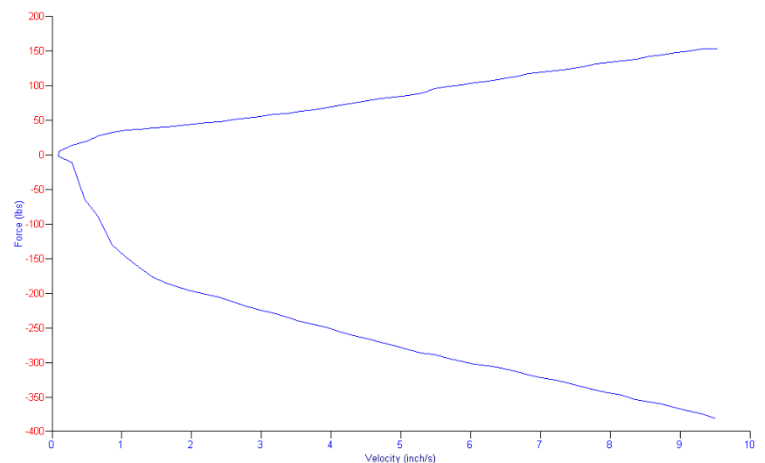


Figure 35 - Rear Damper Curve

The Damper Curves that were originally in VI Grade Car Real Time were not accurate to the cars so imputing the actual damper curves will have improved the accuracy of the model.

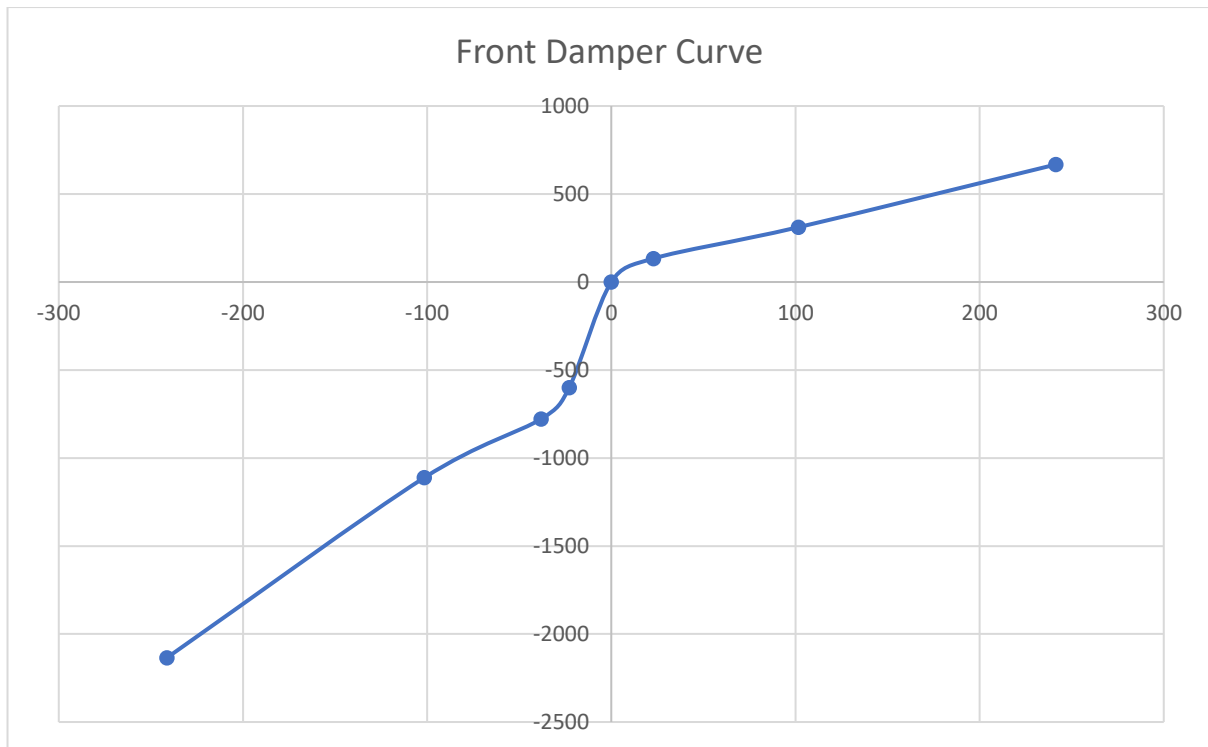


Figure 36 - Front Damper Curve in Excel

Here can be seen, in Figure 36, the front damper curve that was plotted in VI Grade to update the model, in which the lower half of the curve has been flipped to the negative side as that is how the graph was required to be rather than the ones seen in Figures 34 & 35 so it could be inputted to VI Grade.

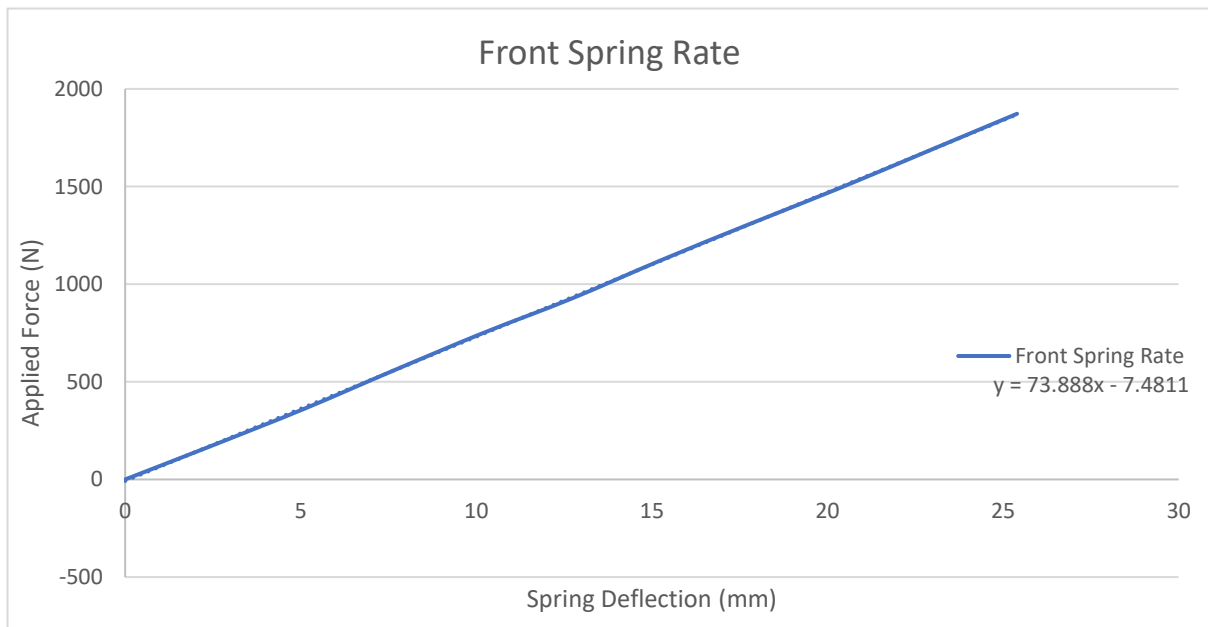


Figure 37 - Front Spring Rate

The Spring Stiffness Investigation described in the experimental design allowed us to get the spring rates for the VI Grade model. As seen in Figure 37 above, the gradient of the line gives the spring rate of the tested spring. This will only improve the accuracy of the simulator.

The earlier mentioned COG investigations were used to calculate both cars #40 and #44's average centre of gravity height. Due to the fact that we are investigating the suspension geometry and how it affects the performance of the cars it was decided that the centre of gravity height would remain constant between the two models, and so #40's COG height was used. This was done so that we could see the effects just the suspension geometry had on performance.

Running the simulations allow the ability to gain a better understanding of how the cars act and perform rather than solely doing static investigations like the Kinematic Investigation that was carried out. After the simulations complete running it outputs the quickest theoretical lap time see Table 3 showing the cars' lap times. Another run was completed as the tyre model used was just a generic road car model, for which the lateral and longitudinal rolling resistances had been adjusted to try and match empirical data, so the simulation was run using slick racing tyres to see how much of an effect the tyre model had on the simulation. The results show that the tyre model has a massive impact on the simulation as a whole and changed the lap times by almost ten seconds, which can be seen in the Table 3, also to be noted is that it swapped which car was simulated to be faster initially it predicted that #44 was quicker, however on the slick racing tires it thought #40 would be the fastest. So due to this outcome, it cannot be definitively stated which vehicle's suspension system obtains the best performance around a hot lap. Something to mention is that all the previous runs have been carried out with the same setups used and the same vehicle masses, this is done so that we can see how the differences in the suspension systems affect overall vehicle performance simulated by VI Grade, which is supported by Avi, Carboni and Stella Costa (2021).

The final thing to do was to input the accurate masses for the vehicles and to run them again to see a more representative lap time for both cars.

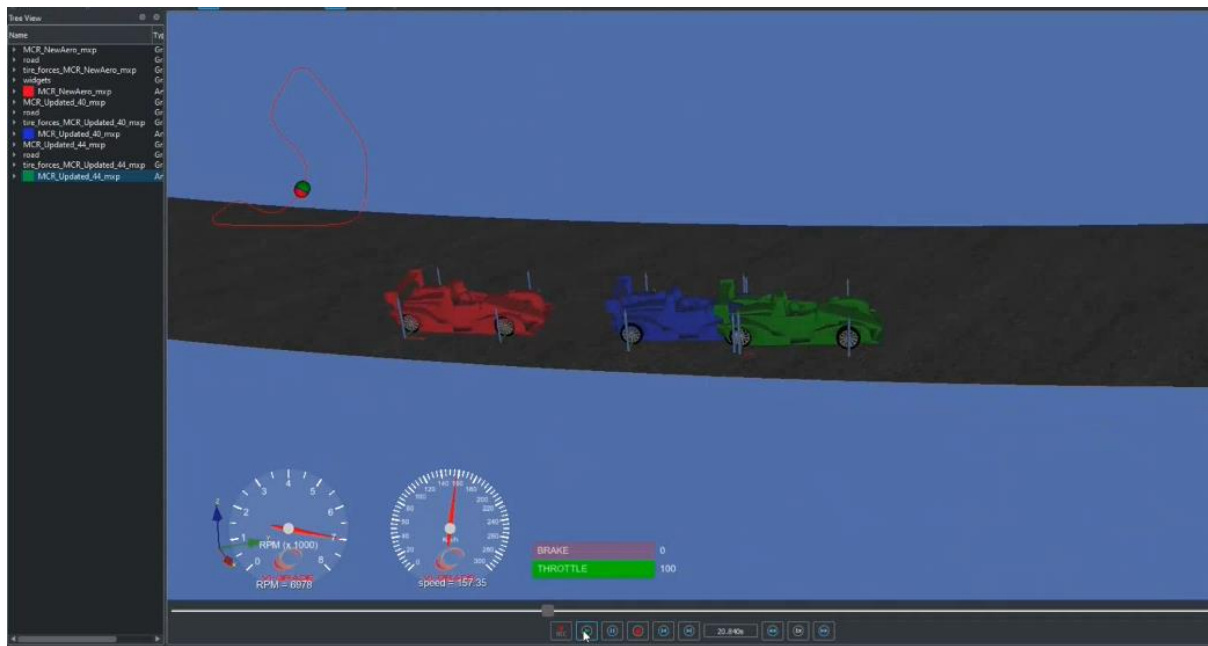


Figure 38 - VI Car Real Time Post Processing

As seen in Figure 38 above, the post-processing can be seen from VI Grade Car Real Time, this allows the entire simulated lap to be viewed to see where the different cars were quicker, where the cars struggled and so on. Something that was clear in the post-processing for the accurate weight run was car #44 was significantly slower on the straights than car #40.

Another thing obtainable from the post-processing is the entire raw data sheet of each model, so you can go in and see which vehicle is performing best in different areas. VI Grade also outputs the quickest lap times from the runs, these can be seen in the Table 3 below.

Table 3 - Lap Times for different VI Car Real Time runs.

Original Tyre Data				Different Tyre Data				Difference in Lap Time				Updated Masses			
Car	40	44	original	Sim	Car	40	44	original	40	44	original	Car	40	44	
lap time	57.541	57.491	57.791	55.938	lap time	48.641	48.671	48.241	8.9	8.820	9.55 (s)	lap time	58.981	59.531	
Cam F	-3.0	-3.0	-3.0		Cam F	-3.0	-3.0	-3.0	15.467	15.342	16.525 (%)	Cam F	-3.0	-3.0	
Cam R	-2.0	-2.0	-2.0		Cam R	-2.0	-2.0	-2.0				Cam R	-2.0	-2.0	
Toe F	0.6	0.6	0.6		Toe F	0.6	0.6	0.6				Toe F	0.6	0.6	
Toe R	-0.3	-0.3	-0.3		Toe R	-0.3	-0.3	-0.3				Toe R	-0.3	-0.3	
delta (s)	0.050	-0.050	0.3	-1.603	delta (s)	-0.03	0.03	-0.4				delta (s)	-0.55	0.55	
pct	0.1	-0.1	0.4	2.78584	pct	-0.06168	0.061676	-0.82235				pct	-0.9325	0.932504	

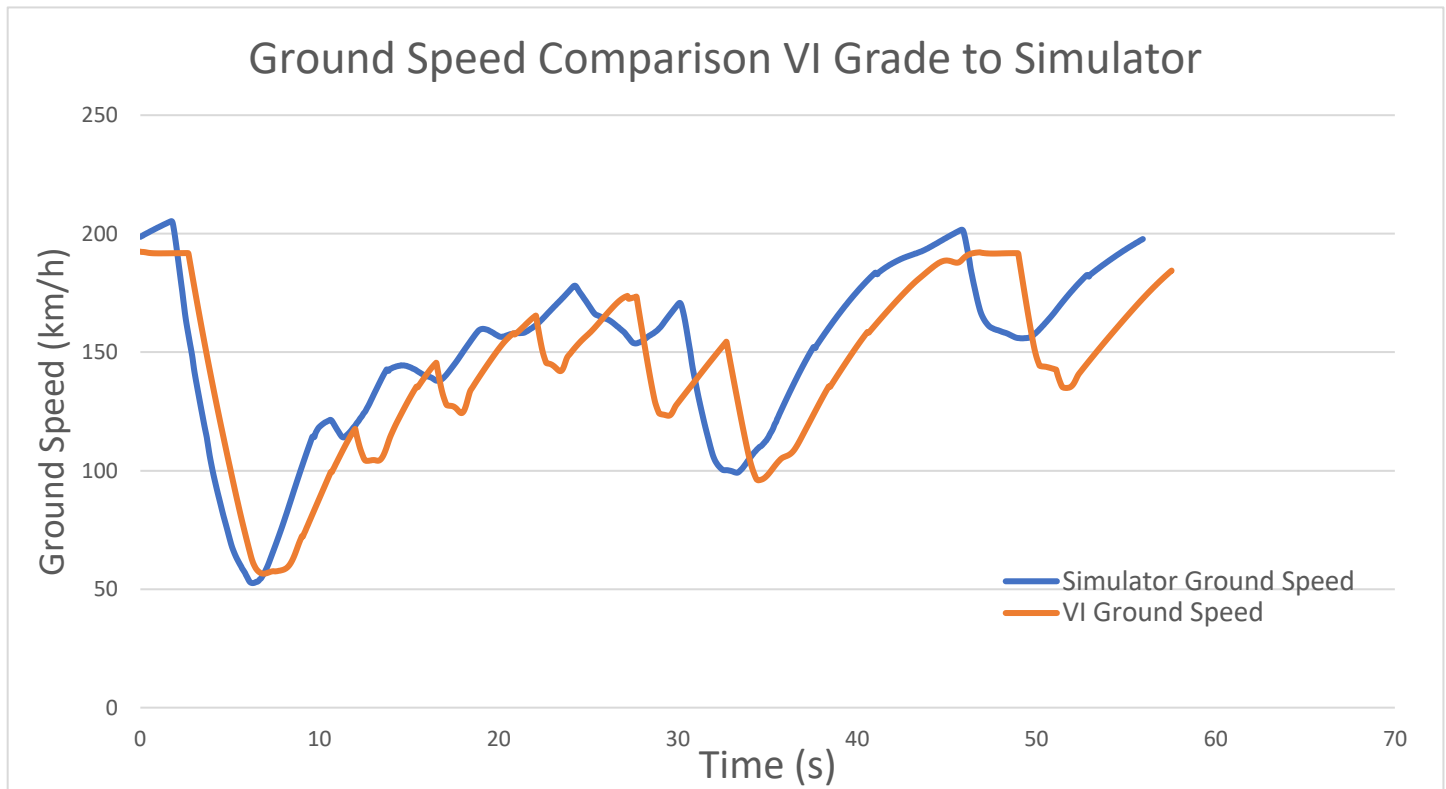


Figure 39 - Ground Speed Comparison VI Grade to Simulator

The Figure above, Figure 39 displays the ground speed for model #40 as it goes around Pembrey and the speed trace for the ground speed of the driver of car #40 doing completing their fastest lap around Pembrey in the simulator. This is useful as it is clear that the speeds being predicted in the lap simulation, produced in VI Grade, are accurate to the speeds that the driver is producing in the simulator. This graph can be used to help validate the VI Grade model to allow it to be effectively used in the future by the MCR Race Team.

2.7 Suspension Conclusion

To conclude the suspension investigation into the effect of suspension geometry on vehicle performance the overall aims of the area have been met, which has been shown previously throughout this report as the created front and rear WinGeo models have been validated through empirical testing of the Kinematics of the vehicles. The use of the created front and rear WinGeo models for vehicles #40 and #44 to update the current MCR model in VI Grade has been justified through the disproving of the current VI Grade MCR model validation against the empirical test data. This improvement to the accuracy of the VI Grade allows the possibility of future investigations into the suspension geometry changes with the aim of improving on track performance, and possible other areas of vehicle development.

2.8 Further work

Due to the scope of the project, we were not able to get accurate tyre models for the vehicles. This would be something that we would want to do to further the accuracy of the simulation and be able to justify which suspension system is better in terms of performance. The tyre stiffness investigation we carried out, which was covered in the experimental design previously, would be helpful for this as it would give us the tyre rates required for the tyre models.

Another thing that would be very beneficial to the model would be to put in the accurate engine data Dylan has gathered in his Engine Performance investigation. And finally, to use the improved data acquisition Liam has been working on to validate the models.

These tasks would only further the accuracy of the simulation allowing better more accurate data to be gotten from running the simulation and hence improving its usefulness in future investigations into the vehicles' performance and possible improvements.

Something that would need to be done would be to validate the model against live track data, for example looking at the lap times and seeing how they match, along with live track data, such as speed traces, corner speeds to check that the model is predicting the correct data.

Once a fully accurate model is created then the simulator could be used for full race simulations to look at how the vehicle acts over an entire race rather than just one lap. This can provide valuable insight into the performance of the vehicle over the full race distance and help to test new components to see how they may fair at a longer distance than just one lap.

Further use for an accurate model of the Race car in VI Car Real Time is that it can look at the performance of different types of tyres. In the situation where different tyres will be used in the future of the SRCC Sports 2000 Championship, as is being considered due to supply issues, VI Grade could be a very powerful tool in adapting to this change. VI Grade can be used in many other applications, including investigating the Aerodynamics of a vehicle and driver training.

3 Powertrain - DA

3.1 Introduction

To develop a competitive car in any race series it is vital to extract the maximum performance from the powertrain within regulations, and in the case of the Sports 2000 series the regulations are very limiting, this creates a very close grid and therefore any small power improvement can make a very large impact making extracting all available power from the engine is crucial to make the car as competitive as possible.

This section of the project investigated this area using both empirical and simulative testing to gain a greater understanding into how the engine functions, and therefore which factors free to be altered within regulations affect the power output by the greatest margin. The methods used to complete this testing involved the use of Ricardo WAVE software to create a representative engine model for future testing and validation purposes, as well as extensive use of a MAHA msr 500 chassis dynamometer to gather real world empirical data from various tests with the aim to improve power and torque. The use of both empirical dynamometers testing to measure torque and power curve changes, and a simulated engine model within Ricardo WAVE to validate any real data collected provide a valuable insight into the powertrain performance.



Figure 40 - MCR Sports 2000 car #40 on a chassis dynamometer

3.2 Section Aims

The investigation into the powertrain of the MCR Sports 2000 car begun by identifying key areas of potential gain within regulations, as any direct engine modifications are prohibited the focus shifts primarily to intake, exhaust and fuelling as all of these areas have some level of flexibility that is yet to be investigated for potential gains. The specific areas planned to be looked into, and the relevant regulations for those are as follows.

3.2.1 Intake

Areas of interest - Airbox shape, throttle body trumpet spacers, pulse tuning.

Notable Regulations – 5.7.1 (d) “It is permissible to mount an airbox base plate between the throttle body and the air horn. Any such base plate shall not exceed 5mm in thickness.”

3.2.2 Exhaust

Areas of interest – Header and exhaust lengths and diameters, pulse tuning.

Notable Regulations – 5.7.2 “Exhaust systems and manifold are unrestricted but must comply with Motorsports UK regulations J5.16.1 to J5.16.6.”

3.2.3 Fuelling

Areas of Interest – Fuel pressure, lambda.

Notable Regulations – 5.7.13 “The fuel pressure in the engine fuel injector rail is free.”

(SRCC 2022)

Due to the nature of these areas both fuelling, and intake were able to be tested empirically because of the lower resource and material requirements than constructing exhausts in order to test different configurations. Because of this pulse tuning calculations were chosen to investigate exhaust lengths if the scope of the project allowed, as the results can be validated using the simulated engine model in Ricardo WAVE in the same way the real-world dynamometer data can be validated using this model.

3.3 Ricardo WAVE Engine Model

WAVE is an engine modelling software developed by Ricardo, it is used for simulating and optimising performance of internal combustion engines, allowing for analysis of virtually any intake, combustion and exhaust system configuration (Nain, A. and Nene, D. 2019). It was chosen to be used for this project as it allows for highly time effective testing which means that any real-world tests can be very quickly reproduced within WAVE and therefore any results can be validated, proving the performance improvement, and providing an extra level of confirmation before any costly modifications are to be made to the cars.

3.3.1 Model creation

The first step to create a working engine model was to build a single cylinder engine model and ensure it is running, this was made by following information in the Ricardo WAVE manual (Ricardo 2019), shown in figure 41.

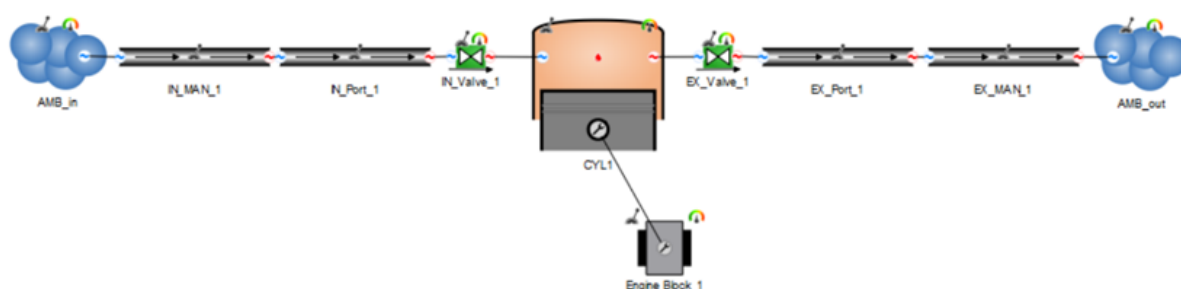


Figure 41 - Single cylinder engine model in Ricardo WAVE

This template was then used to create a basic 4-cylinder model that ran and outputted data, but used default values that could be replaced with measured values from the Duratec 2.0L engine and configuration as used in the MCR.

With a basic engine template created, the required measurements were then collected through research and measuring intake, engine and exhaust components. These and the methods used to collect them can be seen in table 4.

Table 4 - Measured Ford Duratec engine components

Component	Measurement/collection method
Cylinder bore, stroke, clearance volume, compression ratio	Taken from Duratec engine specification online.
Valves	Vernier calliper on spare valves.
Intake and exhaust ports	Vernier calliper on spare Duratec head.
Exhaust manifold and pipe	Removed from car, tape measure and vernier calliper used.
Intake manifold, throttle bodies and trumpets	Vernier calliper on spare manifold, throttle bodies and trumpets.
Cam and valve lifts	Cam lifts measured using optical dividing head, valve lifts determined using these values within MATLAB.
Valve flow coefficients	Spare Duratec head measured on Superflow SF-600 flowbench.

(Hoffman, D 2022)

With the required data collected and inputted into the 4-cylinder model, it could then be run to test how close it is to real engine data as the Duratec 2 Litre used in Sports2000 makes around 200hp at the engine at 7000rpm (SRCC, n.d.). Once the chassis dynamometer testing begun the model could also be compared to a baseline reading in standard configuration before any changes or modifications have been made. This updated model can be seen in figure 42 below.

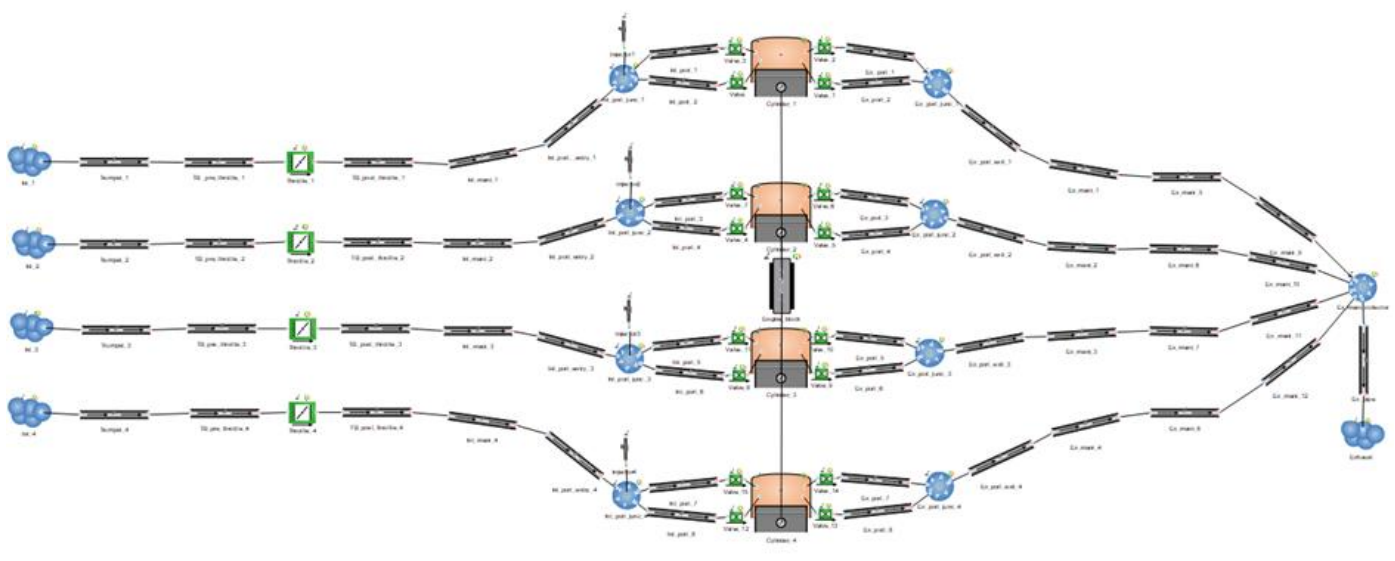


Figure 42 - Final engine model in Ricardo WAVE

3.3.2 Model Refinement

This iteration of the model was then run multiple times, aiming to get as close to the real-life engine data as possible in order to validate it for further testing and comparison. One issue encountered that was preventing the model being accurate is that the measured valve flow coefficients seemed much higher than expected for a stock road car cylinder head. An investigation of a similar 4 valve per cylinder head with intake valve diameters of 36mm compared to the Duratec's 35mm and an intake valve stem diameter of 5.5mm vs the Duratec's 5.47mm resulted in a much lower flow coefficient (M. El-Adawy et al 2017). This is shown in figures 43 and 44 below.

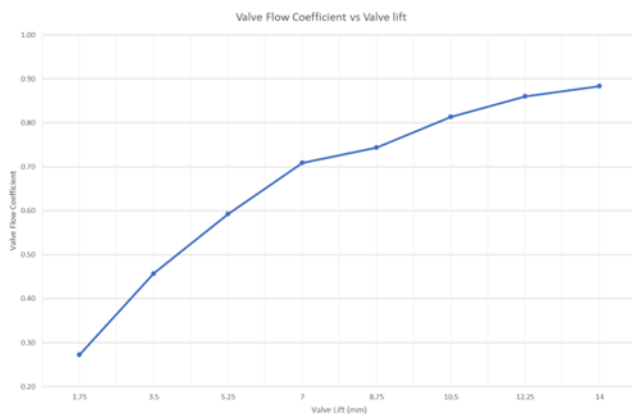


Figure 44 - Valve Flow Coefficient vs Valve Lift

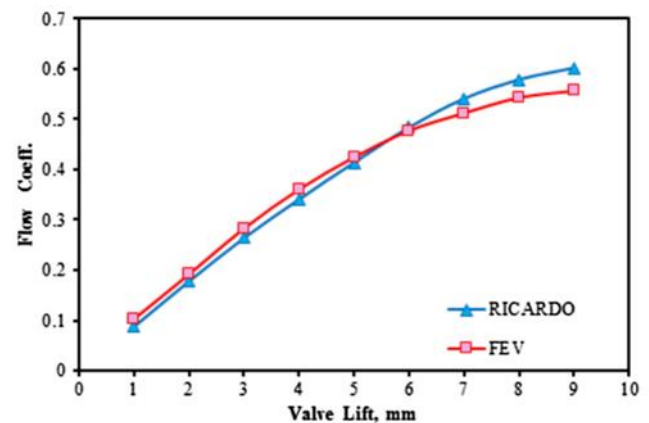


Figure 43 - Flow Coefficient against valve lift (El-Adawy et al 2017)

The measured flow coefficients in figure 4 show a coefficient of 0.75 at 9mm lift, compared to the other investigation which only reached 0.6 at 9mm lift. This inaccuracy in the measured values resulted in the model making peak power at over 9000rpm vs the real engine which makes peak power just after 7000 rpm with a redline of 7250 rpm. With all other inputted values known to be correct the flow coefficient was adjusted so that the model made peak power closer to the real engine, from there even though the torque and power curves are not identical and peak power was slightly low the model could be used for testing comparing instead the percentage power change of the curve to the percentage for real world testing which is most representative. To improve this a repeat cylinder head flow test would have been completed however doing this would have compromised the testing to follow, and as a larger amount of work had to be covered for due to unforeseen circumstances this wasn't possible.

3.3.3 Final Model

With the flow coefficient adjusted, the peak power was very close to that of the real engine data however there was a very large discrepancy in mid-range power, in particular from 4500 to 6500 rpm, with this highest discrepancy being at 5250 rpm where the Ricardo WAVE curve is 22% lower than the real-world data, while only being 2.5% lower than real world data at 6750 rpm shown in figure 45.

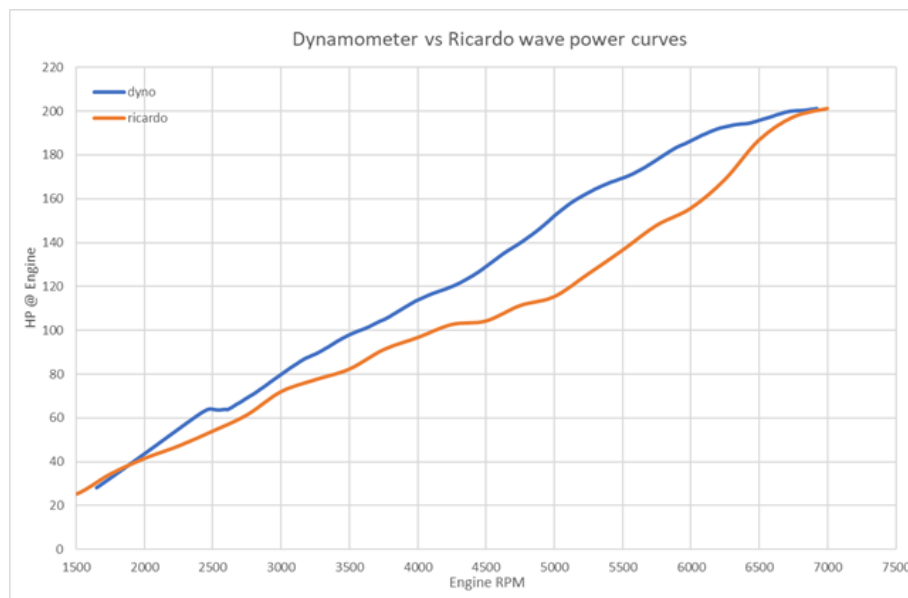


Figure 45 - Dynamometer vs Ricardo WAVE power curves

Unfortunately due to the time and to not compromise the empirical testing that had to be completed, the model was unable to be made any closer to the real data. Although this level of inaccuracy is unacceptable and higher than was planned, the model was chosen to still be used however to compare percentage increase or decrease in power instead of straight curve comparisons as it is still representative of how the engine may react to these changes. Because of that the focus could then be moved towards the empirical testing using the dynamometer.

3.4 Empirical Engine Testing

A chassis dynamometer, also known as a rolling road, is a piece of equipment that measures the tractive force generated at the tire/drum interface simulating the conditions of driving on the road (Martyr A. J et al 2021). This is achieved by running the vehicle's wheels on rollers that are connected to the dynamometer, which in turn measures the output of the engine. Chassis dynamometer testing is useful for improving engine performance in motorsport because it allows for precise and controlled testing of engine performance under a wide range of scenarios. This includes testing the engine's power output at different speeds and loads, as well as air/fuel ratio and other important parameters. The use of dynamometer testing in this project allows for many different setups to be tested in fast succession and any power increase or change in the curve is clear and definitive with it being measured straight from the vehicle itself.

With the key areas to be investigated identified in section 2, various tests could be planned. These can be seen below.

Intake: Test multiple spacers ranging from 1mm-5mm placed between the airbox and throttle body, which increases the overall effective length of the trumpet and throttle body velocity stack assembly. This could potentially give a power increase due to pulse tuning effect.

Fuelling: Test engine performance at various fuel pressures by completing a sweep from 48psi through to 56psi, to find the optimal pressure that the car should be running at over the season.

3.4.1 Throttle Body trumpet spacer testing

To complete the spacer tests the first step was to warm the car up on the chassis dynamometer so that the engine and gearbox oils as well as water were up to operating temperature to begin collecting data, once this had been completed the first test was a baseline run, testing the cars power output in standard configuration to be able to see any improvement. This was completed with the standard airbox assembly which results in around a 1-1.5mm 'spacer' between the throttle bodies and trumpets due to the slight inconsistency of the fiberglass airbox thickness. With the data collected from this run the trumpets and airbox assembly were removed to insert 2mm spacers in between the airbox and throttle bodies, which when added to the airbox resulted in a total spacing of 3-3.5mm between the airbox and throttle bodies.

The car was again run with this setup, and then this procedure was repeated once more with 4mm spacers to give a 5-5.5mm total spacer. To finalise these tests and prove validity of the results a bookend run was completed with all spacers removed and the car being put back to baseline setup, as it was in the initial baseline. This showed that there was no drift in the results and helped to confirm that any change in power curve was due to the spacers and not any inconsistency in dynamometer readings.

3.4.2 Throttle body trumpet spacer results

These tests provided some very positive and insightful information, with the 2mm spacer improving upon baseline by an average of 0.79% across the entire rev range with the largest improvement being around 6500rpm where it peaks at a 1.47% improvement on power. This is improved upon even further with the 4mm spacer which gave an average power gain across the entire rev range of a huge 2.69% which equates to 3.13 wheel hp average gain. Figures 46 and 47 show the power and torque changes for all three tests.

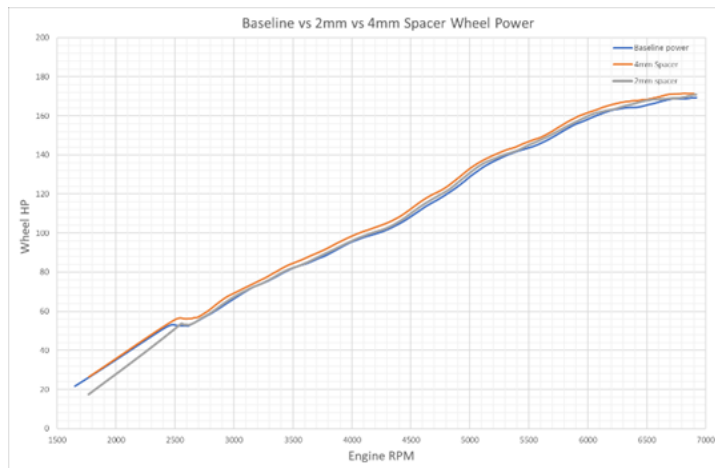


Figure 46 - Baseline vs 2mm vs 4mm Spacer Wheel Power

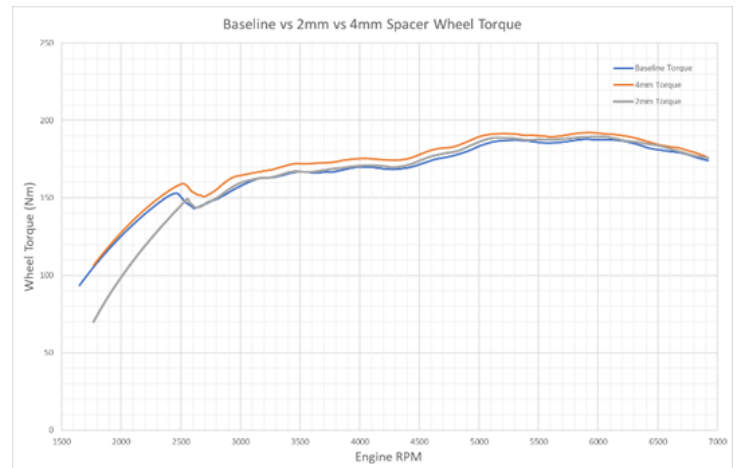


Figure 47 - Baseline vs 2mm vs 4mm Spacer Wheel Torque

These results clearly show that the car should be running at the maximum of 5mm total airbox backplate thickness (spacer) to maximise power and torque with both showing a strong improvement at all engine speeds, this increase is very valuable and provides an excellent price to performance ratio also with a small spacer showing such promising gains. An investigation made into Formula SAE intake showed a significant increase in torque from 5000 rpm up to 7000 rpm for a longer runner length, and a shorter runner length peaking at higher rpm which matches trends shown in the results (Vaz J et al 2017).

3.4.3 Fuel Pressure testing

To complete the fuel pressure sweep test the same start of test procedure as the spacer tests was undertaken, however before this an inline analog fuel gauge was fitted to the fuel rail so that the fuel pressure could be monitored and accurately adjusted using the adjustable fuel pressure regulator fitted to the car. The 4mm trumpet spacers were also installed as they were proven in previous testing to be most effective which may have resulted in better airflow and therefore a higher fuel pressure requirement, so the fuel pressure tests were to prove the optimal setup for the car between both intake and fuelling. To begin the tests the car was then warmed up to operating temperature, and the starting fuel pressure noted down so that a baseline run could be completed, the fuel pressure at baseline was measured to be 52psi.

The chosen sweep was from 48psi through to 56psi in 1psi increments, representing 4psi either way of baseline which gave a comprehensive range of data to view and select the optimum pressure from. After the baseline run had been completed the fuel pressure was dropped down to 48psi to begin the sweep, a power run was completed, and the run data was saved. This process was repeated moving the fuel pressure up by 1psi each time up to 56psi, and then a bookend baseline test was completed after moving the pressure back down to 52psi to test for any drift in the dynamometer readings and ensure validity of the data. The inclusion of 52psi in the sweep essentially provided 3 baselines, which allowed for a baseline reading to be gathered at the beginning, during, and at the end of the test procedure further enhancing the reliability of the test.

3.4.4 Fuel Pressure testing results

As the car had previously been setup to race it was to be expected that the current running fuel pressure was close to or at the optimal value already, and the results showed this with 52psi providing the highest power and torque throughout the majority of the rev range, in particular at high rpms above 6000 this can be seen in figures 48 and 49 on the following page.

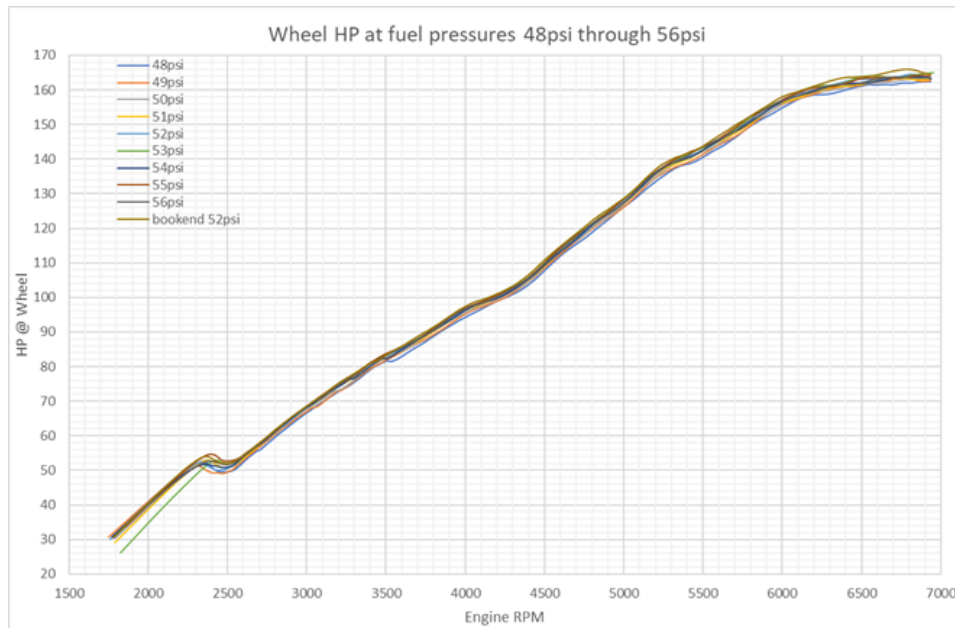


Figure 48 - Wheel HP at Fuel Pressures 48psi through 56psi

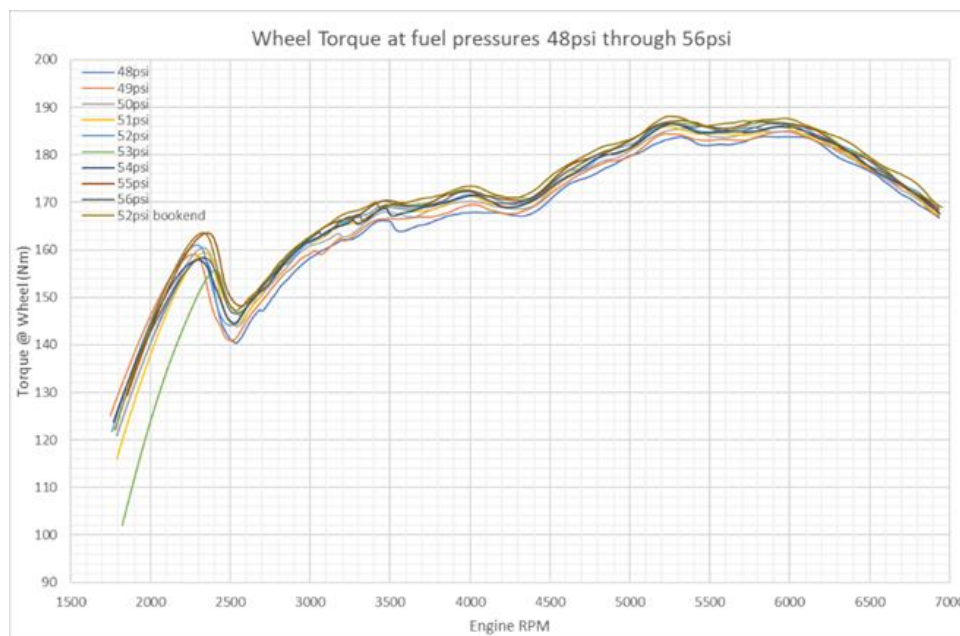


Figure 49 - Wheel Torque at Fuel Pressures 48psi through 56psi

One issue that was encountered is that the 52psi results showed some variance, in particular between the initial baseline compared to the other two 52psi tests. As the middle and final tests at this psi were much closer the first test can be counted as an anomaly and has not been included in the above graphs. Potential causes of this could be if the car hadn't been sufficiently warmed up meaning the engine oil isn't lubricating as effectively or getting the opportunity to boil off contaminants which can result in small power losses (Quinn-Walshe T. n.d.), the pressure may have not been set accurately enough resulting in a discrepancy also. Figure 50 shows this variance between readings of the same pressure.

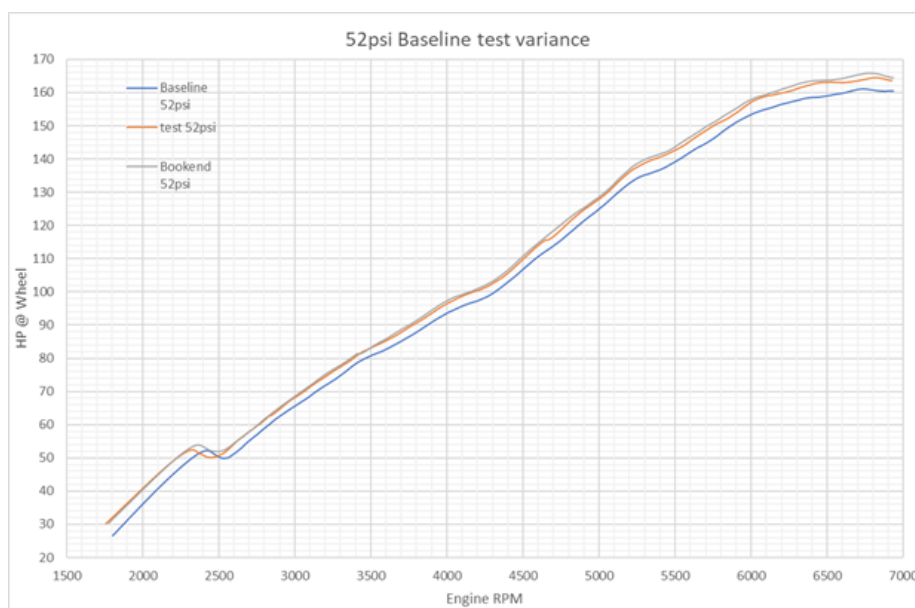


Figure 50 - 52psi Baseline Test Variance

3.5 Powertrain testing results

Looking at the results from the Empirical testing it can be gathered that the optimal setup when tuning the intake trumpet lengths and fuel pressure within regulations is to use a 5mm airbox backplate (spacer) which is the maximal allowable thickness and a fuel pressure of 52psi. As the optimum fuel pressure is what was already being used on the car the total measurable power increase is the gain from adding the 4mm spacer which as stated before equated to an average of 2.69% power increase across the rev range, which is 3.13 wheel hp, and peak improvement of 4.3 wheel hp at just under 5000rpm.

With use of the OptimumLap software this total power increase could be used to estimate how much time would be saved across each circuit that is going to be raced over the 2023 Sports 2000 season and the average over the season, which gives an excellent insight to see which circuits it would have the largest impact on lap time. These values are shown below in Table 5.

Table 5 - Lap time reduction from power improvement using OptimumLap

Run	Season Avg	Silverstone National	Brands Hatch GP	Castle Combe	Donington National	Oulton Park Int	Donington Park
% Reduction	0.374%	0.087%	0.319%	0.530%	0.437%	0.428%	0.445%
Est Laptime reduction (s)	0.314	0.051	0.287	0.363	0.305	0.443	0.436
Baseline (Airbox)		57.240	87.990	66.440	68.990	100.820	97.010
4mm Spacer + Airbox		57.190	87.710	66.090	68.690	100.390	96.580

Looking at table 5 all circuits show promising lap time improvement, in particular Donington GP and Oulton Park International both estimating a saving of over 4 tenths of a second per lap. With the largest percentage saving being Castle Combe being 3.6 tenths faster over a 66 second lap which is very impressive.

3.6 Ricardo WAVE model test comparison

In order to further validate the findings from the empirical testing the same tests can be easily replicated within Ricardo WAVE, and even though the model's power and torque curves do not match up particularly well with the real collected data the percentage increase for the same change can be gathered to see if the engine will respond as expected and is shown from dynamometer results. The spacer tests were replicated within WAVE for engine speeds of 4000-7000rpm which is the prime zone that the engine will be operating at while on track, and figure 51 shows the comparison between the standard and 5mm spacer simulation.

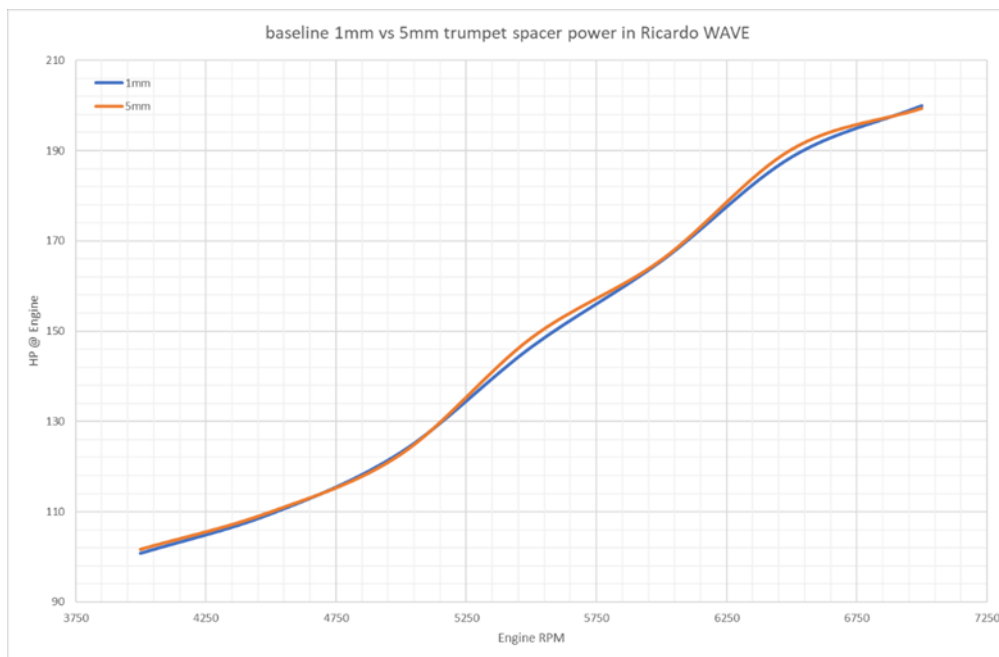


Figure 51 - Baseline 1mm vs 5mm Trumpet Spacer Power in Ricardo WAVE

The 5mm spacer can be shown to increase power however in a different manner and not to the same extent as shown through the real-world testing, only increasing power by a maximum of 1.34% at 5500 rpm and not increasing power through the entire rev range. One limiting factor with this is that the WAVE model is only outputting data in 500 rpm steps whereas the dynamometer data is outputted in intervals of 10 rpm each time, this results in the real-world data giving a much clearer and more accurate representation of any trends shown. It is positive that the model shows an overall increase in power with the larger spacers however due to the inaccuracy of the model discussed in section 3.3 it is difficult to use for validation purposes for the measured data.

3.7 Powertrain Conclusion

To conclude the powertrain section overall the testing did provide some intriguing and useful information, which not only allowed for more power to be extracted from the engine but also a wider understanding was gained over how it can be optimised. The power gains displayed through the addition of the throttle body trumpet spacers has potential to make a large impact in such an even and close grid such as Sports 2000 which is a large success and the fuel pressure tests provided valuable knowledge around how the pressure effects the power and torque through a wide sweep and confirms that the fuel pressure currently being utilised is in fact the optimal pressure for highest performance.

It is unfortunate that the Ricardo WAVE model was unable to be used as effectively for validation purposes of the real world data, and that the power curve was not consistent with that of the measured dynamometer curves, however due to an increase in workload during the project due to unforeseen circumstances the choice to ensure that all empirical testing could be completed to a thorough and good standard was made which proved to be successful with an instant power increase made to the car.

3.7.1 Future Powertrain work

The most important aspect of this project that would be focused upon if completed again, or in future would be to ensure the WAVE model is able to have more time spent refining and testing against dynamometer data so that it could be used to accurately validate any real-world findings without the uncertainty which it currently possesses due to inaccuracy of the power and torque curves. With this model fully working, validated, and proven more accurate it would allow for a wide variety of future testing to be completed and any engine tests completed to be checked to see if they are worthwhile and then validated afterwards. This would begin with completing a repeat cylinder head flow bench test to ensure the valve flow coefficients used in the model are accurate. Also with the help of this model the exhaust lengths could be investigated, and the pulse tuning effect created, as a validated model and data to optimise rpm using hand calculations would provide sufficient information to aid development of a new exhaust system, possibly including the use of titanium to decrease mass as well as increase torque curve with a new configuration.

4 Data Acquisition - LC

Datalogging has become an integral part of modern motorsport, providing teams with valuable insights into the performance of their vehicles and drivers. Data is created/collected from a variety of sensors, including accelerometers, movement potentiometers, GPS systems, and strain gauges. This information can be analysed to understand vehicle behaviour, where areas for improvement can be identified, vehicle setups optimised, and inform strategic decisions throughout and not limited to motorsport. The use of data acquisition technology has revolutionized competitive vehicle racing, allowing teams to discover advantages and push the limits of their vehicles.

Aims and Objectives of data acquisition include understanding the signals within a control system, identifying the present logging equipment and its capability, install proposed upgrades, reverse-engineering sensor ancillaries, assessing results in software and building the CAN-bus loom for the new on-board tyre temperature monitoring system (TTMS).

4.1 Research

4.1.1 Data signal formats

Most sensors have three input wires: power in (DC, usually 5V), a safety ground, and the signal wire; analogue signals use voltage variation to communicate how the parameter in measurement is moving. Not only the value bears importance, the movement pattern and rate of change gives solid representation of specific loads within a system. Analogue voltage signalling is preferred in certain

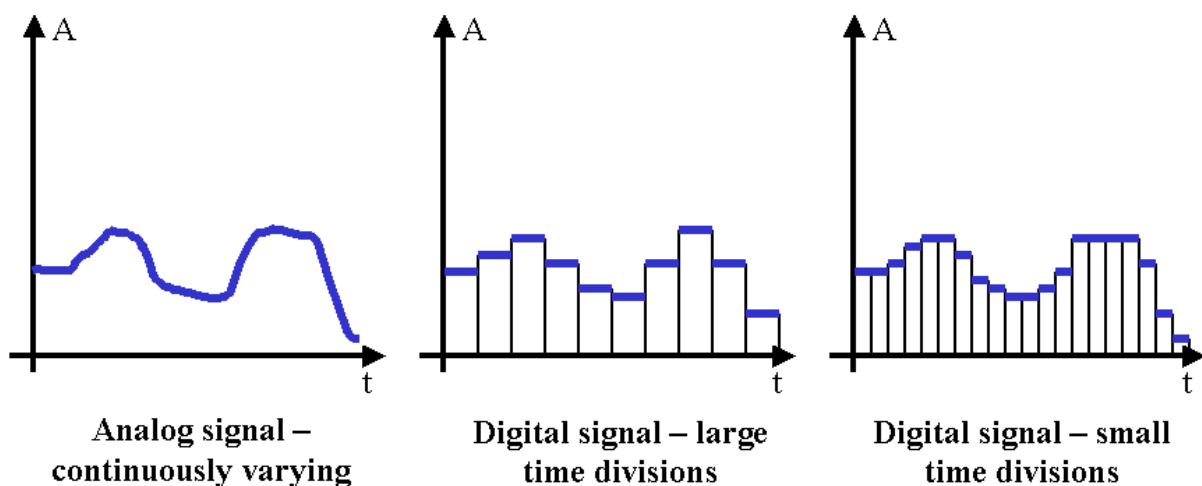


Figure 52 - Analogue vs. Digital signalling & sample rates

applications for its simplicity, as troubleshooting is possible with a multimeter unlike in digital signalling where an expensive oscilloscope is needed to read values and check the signal shapes. The sample rate is the frequency of data collection measured in Hertz, sensors are specified to run anywhere between 30Hz and 5megahertz; where comprehensive detail is required to understand

component behaviour, example being damper behaviour where a high-speed processor will be needed to collate the values quickly. Digital communications have advantages over analogue twinned with growing versatility, as CAN-bus setups have soared in popularity in motorsport and automotive applications over recent decades. Controller Area Network ISO-11898 is a high-speed protocol used in high-speed digital communications, at peak transfer rate of 1megabit/sec (1,000,000bits per second). Figure 53 shows a visualised data packet spanning 96 bits, mostly data but interesting to note is the

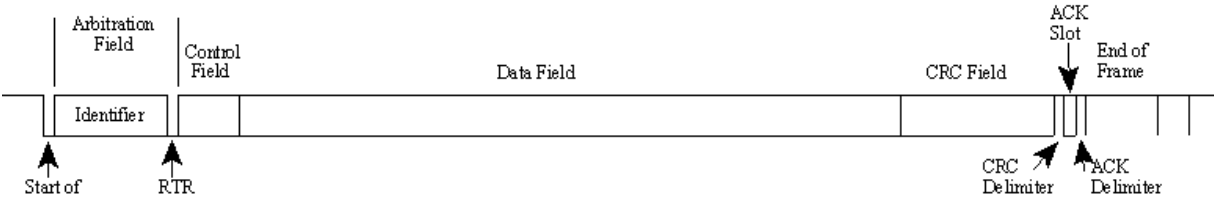


Figure 53 - CAN-bus data packet

identifier at the beginning; use of unique IDs allow for multiple sensors of different types recording simultaneously on the same circuit. If the peak speed of 1Mb/s (125MB/s) is not exceeded by all sensor traffic combined (causing bottleneck), this means sensors of all types all over the vehicle can be linked to a single loom with messages crossing but not interfering with each other, impossible with analogue signalling. Therefore, digital signals offer greater versatility with linking components, and potential for reduced need for wiring, reducing complexity, cost, and vehicle mass. Although, not every application will benefit as its expensive architecture and unguaranteed compatibility renders digital signalling useful but not revolutionary to all mechanical principles.

4.1.2 On-board logging system

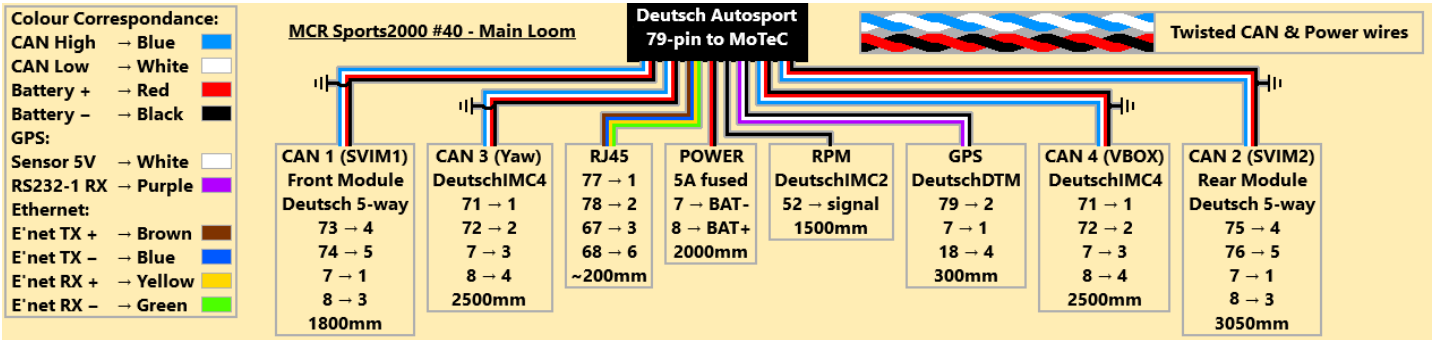


Figure 54 - MoTeC main Inputs

Sports2000 car #40 uses MoTeC logging equipment; the main logger, doubling as the driver’s screen, is the C185 with a 1KHz sample rate and 250MB memory. The acquisitor is assisted by two SVIMs, located at the front and rear of the vehicle, to which the sensors link to them, reducing the need for processing/memory on the main logger. This resource delegation improves the C185’s reliability while operating, as a bottleneck causes lag and in close racing, delayed values in instrumentation causes driver discomfort or worse, therefore efficient electrical control system design is beneficial.

4.1.3 Tyres & Thermal Influence

Sports2000 use Yokohama A005 slicks. They are described as highly forgiving compared to other racing tyres, with a broad competitive lifespan and predictable behaviour. High component stresses in high-speed cars builds heat in the tyres; hysteresis from deforming and reforming longitudinally and latitudinally, as well as shear from unsmooth/jagged road surface causing wear. The tyre's intended temperature range or 'sweet spot' of peak grip can be hard to determine but degradation is anticipated and balanced. Pushing the limits of grip and rising thermal energy within the tyre will cause unpredictable wear of critical effect, risking safety of the driver and other drivers in close racing.

Prematurely shredding tyres is costly, and rules often limit how many sets can be used in one season/championship, so acknowledging the tyre's limits help them last longer while maintaining good competitiveness. Changing inflation pressure has strong influence on how the wheels behave; lower pressure will lower the reaction forces inside the tyre, softening and worsening performance, meanwhile over-inflation causes uneven wear from air forcing a central bulge into the contact patch. (Richardson, 2017) Therefore, correct/optimal inflation pressure often takes experimentation to discover, harder with external influences such as track temperature, heat from braking, ambient air temperature, and road surface.

4.2 Experiment

4.2.1 Redesign Case

The protective cases included with IZZE Racing's infrared tyre sensors were not the aluminium enclosures as seen online, instead a simple plastic case, prompting a revamp. The sensor was measured precisely with a vernier calliper, values used to draw the cocoon in Solidworks. Fixing to the car; adhesives like epoxy would bond firmly but imposes challenge in loom modularity, therefore securing by larger M5 bolts is preferred, giving the sensor case its top-down hexagon footprint. Prototypes (Figure 55) were produced to compare and decide which two-piece lock-together solution is best. Two material choices: PLA; a versatile polylactic acid polymer, commonly used in 3D printing, or SLS; selective laser sintering powdered Nylon 12, offering a high UTS of 50 megapascals. Both are suitable, but the PLA's max UTS is noticeably lower at 33MPa (Subramaniam, et al., 2019) therefore SLS is the chosen material for this application.



Figure 55 - Prototype Casings

4.2.2 Loom Strategy

IZZE Racing's TTMS sensors are linked in one CAN-bus circuit connected to one of the C185's four digital inputs, harness construction following IZZE's specification. The wires, V.IN (red), GND (black), CAN+ (blue) and CAN- (white), are twisted together in pairs (V.IN&GND and CAN+&CAN-) spanning the entirety of the loom with Deutsch 4-pin DTMs connecting and ensuring modularity. The wire's thickness varies between 26gauge (0.4mm) and 22gauge (0.6mm) thicker wires being spinal, and they're twisted to cancel electromagnetic interference (Baltag & Rosu, 2017) preventing skewed values. 120Ω resistors are needed at both axes of the loom to reduce signal bouncing, also invaliding results otherwise. Continuity is important in good design.

4.3 Development

4.3.1 Loom Construction

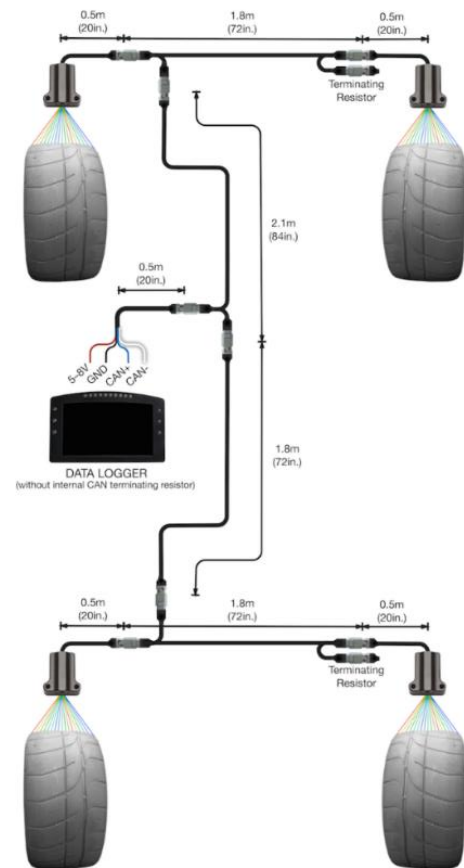


Figure 56 - IZZE's Specification



Figure 57 - The CAN-bus Wiring Harness

Adopting standard rotation and connector arrangements, once car #40's loom residing areas are measured, the wires are cut to length x 1.5 to allow for the twisting plus a safety margin to ensure connectivity. The loom is fully modular bar the bulkhead link, as the narrow and congested wiring crevice makes regular installation there uneasy. The loom consists of a spine section, front & rear axle sections, said bulkhead wire and four infrared sensors to be positioned in front of the tyres. Tools used are a wire crimper, wire stripper, wire cutter, long-nosed pliers, a multimeter and a soldering iron.

4.3.2 RPM Trace

For vehicles without an OBD port, obtaining an engine speed signal is often via clamping onto a HT lead to track ignition pulses. In practice with car #40 on the rolling road, it commonly glitches, so if the MoTeC loom could connect and input RPM to the dynamometer, improved accuracy in power curves is probable, assisting powertrain research. Linking requirements include the analogue signal wire, 12V power and the safety ground from the car to the roller controller. This commenced soldering a 1-pin connector on the engine speed wire next to the footwell, between the crank position sensor and the front module (pictured).



Figure 58 - RPM signal plug-in lead

Testing it, car #40 was cranked with no fuel to confirm the logger shows >0RPM. Sourcing 12volts from the loom is unstraightforward as most onboard sensors use 5volts, so power and ground are linked directly to the vehicle's battery with hooped crimps and a 2-pin connector which can be installed prior to dynamometer testing, then removed after.

4.3.3 Additional Sensors

Vacancies on SVIM inputs allow for more sensors to be recognised by and use the logging network; aside from the TTMS, throttle position can be attained from a linear potentiometer at the pedal, or, like the RPM trace procedure, wire can be soldered onto the throttle potentiometer to ECU signal wire, leading to the SVIM. Other sensors include front and rear brake pressure sensors, which can determine brake force distribution, and a steering potentiometer. Different drivers could post similar lap times but how they interact with their vehicle's controls vary, progressive or aggressive, and highlighting potential gain in data comparison might improve driver skill.

4.4 Validation

4.4.1 Connecting to Vehicle

MoTeC digital input three on Sports2000 car #40 is chosen to link the loom; the IMC connector is removed, stored and a DTM is installed to connect the TTMS harness and sensors to the datalogger. With the loom joined, the laptop is linked to the RJ45 (ethernet) port, and the program connects with the logger. IZZE's standard .dbc configuration loads the sensor IDs and channels into Dash Manager, where it's applied and the MoTeC is rehashed with updated changes.

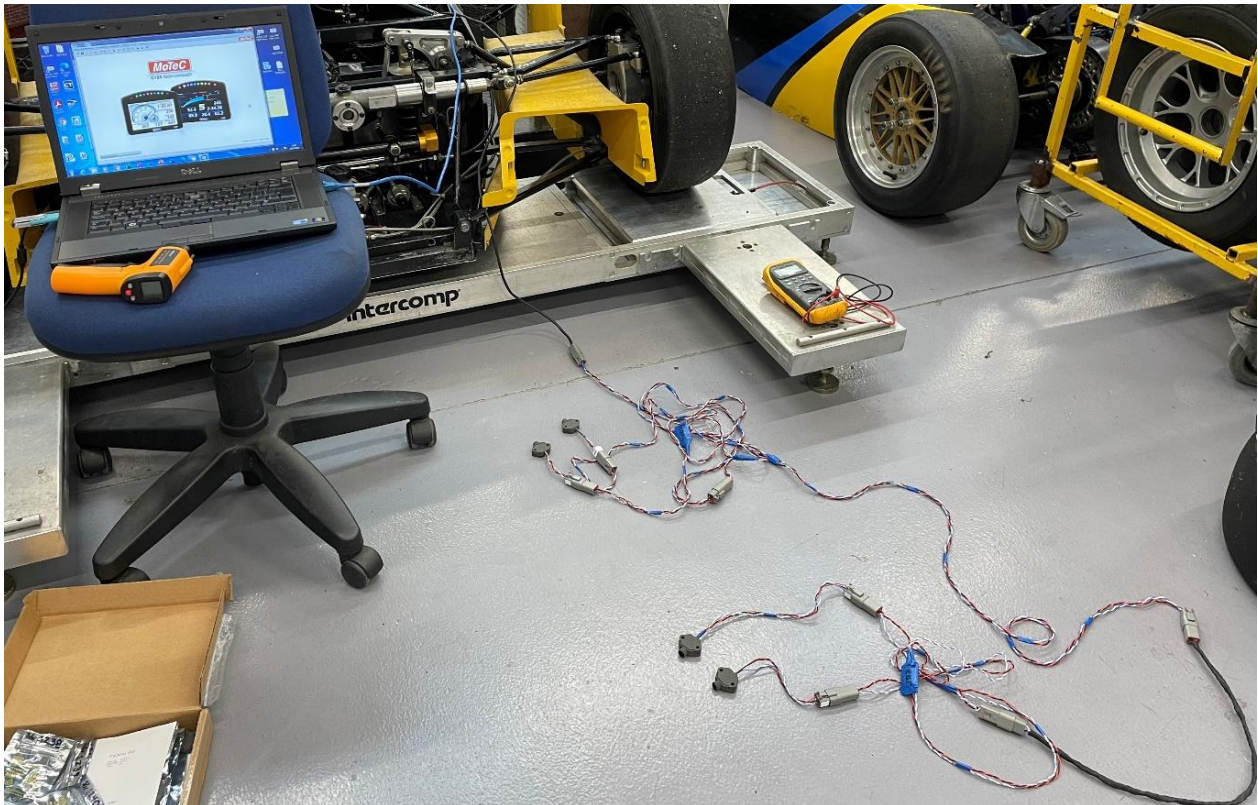


Figure 59 - Testing the Loom

The first run unfortunately yielded no tyre signals, and an unusual response from the rear SVIM. Prompting inspection, it was then discovered the module was defective, likely due to years of vibration and heat fatigue cycling from being located near the engine. The TTMS harness was fully resistance tested again and pins wedge-locked, then rehashed on Data Manager and progress: two sensors reading live. This fault was a 'mirror error'; the decision of organising the DTM pins power/ground top and CAN+/- bottom in the connectors was wise, as on the rear axle's loom voltage was sent to ground and the CAN-bus wires mismatch caused no damage, whereas 8V into a signal output could kill the sensor(s). Teething issues sorted, the MoTeC was rehashed and all four TTMS sensors were recognised and recording temperature data in real time.

4.4.2 Logging Software



Figure 60 - MoTeC i2 GUI & logged data

MoTeC i2 Pro displays the log files as interpretable strings of information; telemetry recorded includes engine RPM, GPS speed/wheel speed, lateral/longitudinal G-forces, and more parameters taken from practice sessions along with races. These values can be mathematized with known vehicle parameters in Math Channels; users can create complex formulae that provide insight into a vehicle's performance, such as aerodynamic forces, tyre slip angles, fuel efficiency, optimal gear ratios and suspension behaviour. These calculations can predict vehicle behaviour as accurately as the operator's calculus, to which can influence motorsport decisions in optimizing vehicle setup, improving driver performance, and ultimately gain a competitive edge on the track. (McBeath, 2002)

In addition to standard mathematical functions like addition, subtraction, and multiplication, i2 supports advanced operations like integration, differentiation, and trigonometric functions. Users can also create conditional statements and use logical operators to create complex logic-based calculations. Overall, MoTeC i2 math channels provide motorsport teams with a powerful tool for analysing vehicle data and gaining a deeper understanding of their performance.

4.4.3 Data Integrity

With the TTMS sensors outputting values, they ought to be correct; the high sampling rate outputs smooth temperature transition when waving hands in front of the sensors, as they detect the change in thermal energy from ambient air to skin temperature. The handheld infrared thermometer (seen in figure x) verified the room air temp as 13.1°C with the sensors reading within the range of 12.5° and 13.5°C, and skin temp of 21°C with sensor readings fluctuating only slightly, around ~21°C.

This verification, although is basic proof of system functionality, isn't representative of the high temperatures withstood by the tyres at race speed, nor entirely accurate. Further verification in the workshop should involve heating a wheel's contact patch artificially, positioning the sensor the same distance away as when installed in the car, and record the infrared energy values from both heat sensors.

4.5 Datalogging Conclusions

In summary, the advent of advanced sensors, telemetry systems, and data analysis tools, teams can gather and analyse vast amounts of information about the performance of their vehicles and drivers. This data allows teams to make informed decisions about vehicle setup, strategy, and driver coaching, ultimately leading to improved performance on the track.

As motorsport continues evolution, data acquisition technology will play an even greater role in the sport's success. Growingly accurate simulation software, high-capacity logging in shakedown sessions and increasingly expensive setup costs is becoming essential in professional vehicle racing.

4.6 Further Work

Future development will include further time investment into Math Channels, iteratively writing and testing specific formula, mathematically representing specific areas of MCR Sports2000 car #40 with values for splitter height, chassis roll angle, dynamic suspension heights, dynamic rake angle and more. Another repentance is failing to discover the damaged SVIM sooner, as more time before the championship would've prompted linking priority sensors to the front module.

5 Conclusion

From the report it can be seen that the aim of the report has been met in all areas, over the course of the investigation knowledge and understanding have been gained. A greater understanding of the suspension kinematics of the two chassis has been achieved and an accurate suspension model of both chassis has been produced. With regards to powertrain a greater understand of how the powertrain responds to changes in intake runner length and fuel pressure. An increase in power has been found through an increase in runner length. In data acquisition a greater understanding of the capabilities of the current MoTeC logging system and potential for future expansion of the sensor suite to aid in development of the vehicle. A TTMS loom has been designed and manufactured to gain tyre temperate data during cornering to aid in alignment optimisation.

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7 Appendices

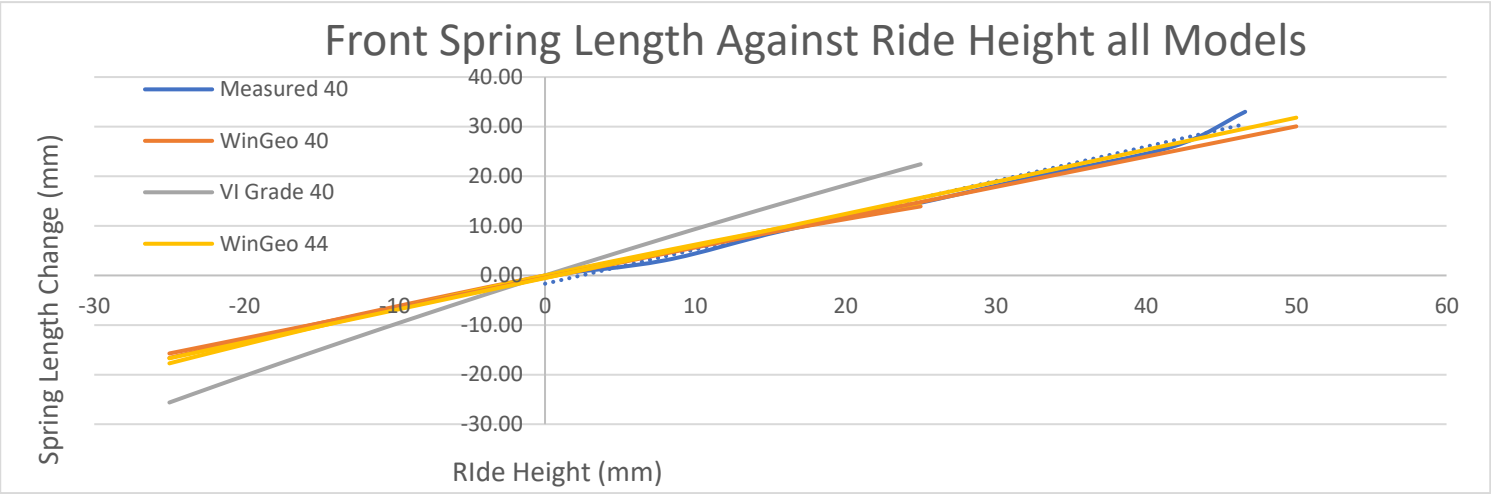


Figure 61 - Front Spring Length Against Ride Height all Models



Figure 62 - Front Suspension of car #44

Left Hardpoints				Left General Data				Left Setup Data			
Symmetric Geometry											
uca front		k	kt	orientation							
location											

wheel center	▼ 0.000 ▲	▼ -806.613 ▲	▼ 241.261 ▲
uca outer	▼ 11.500 /27.0/15.5 ▲	▼ -661.310 /-656.0/5.31 ▲	▼ 362.060 /357.85/4.21 ▲
lca outer	▼ -12.200 /-3.0/9.2 ▲	▼ -682.950 /-694.0/11.05 ▲	▼ 142.315 /117.0/25.315 ▲
tierod outer	▼ -71.120 /-60.0/11.12 ▲	▼ -678.740 /-684.0/5.26 ▲	▼ 371.500 /346.23/25.27 ▲
pushrod outer	▼ -8.620 /-8.0/0.62 ▲	▼ -618.380 /-622.0/3.62 ▲	▼ 175.915 /149.52/30.395 ▲
pushrod inner	▼ 7.900 /-16.5/24.4 ▲	▼ -267.200 /-276.0/8.8 ▲	▼ 402.430 /421.53/19.1 ▲
rocker pivot	▼ 10.000 ▲	▼ -237.895 ▲	▼ 365.655 ▲
rocker pivot axis	▼ -8.100 ▲	▼ -237.895 ▲	▼ 365.655 ▲
rocker link	▼ -67.894 ▲	▼ -35.712 ▲	▼ 507.721 ▲
spring upper mount	▼ 3.190 /-37.0/40.19 ▲	▼ -165.955 /-185.0/19.045 ▲	▼ 119.400 /139.050/19.65 ▲
spring lower mount	▼ 7.900 /-37.0/44.9 ▲	▼ -152.900 /-188.0/35.1 ▲	▼ 415.020 /424.3/9.28 ▲
damper upper mount	▼ 3.190 ▲	▼ -165.955 ▲	▼ 119.400 ▲
damper lower mount	▼ 7.900 ▲	▼ -152.900 ▲	▼ 415.020 ▲

Figure 64 - Current VI Grade Hardpoints compared to Measured Hardpoint locations 2

Table 6 - Front Kinematic Investigation Data 1

Wheel Position (mm)	Ride Height (mm)	Measured Camber (degrees)	Zeroed Camber (Degrees)	DTI A (inches)	DTI B (inches)	DTI A (mm)	DTI B (mm)	Calculated Toe (Degrees)
33.16	-32.8	-3.18	-0.56	0	0	0	0	0.000
25.26	-24.9	-3.1	-0.48	0.015	-0.015	0.381	-0.381	0.127
17.36	-17	-2.92	-0.3	0.026	-0.034	0.6604	-0.8636	0.255
9.16	-8.8	-2.8	-0.18	0.032	-0.059	0.8128	-1.4986	0.386
0.36	0	-2.62	0	0.032	-0.091	0.8128	-2.3114	0.522
-8.64	9	-2.58	0.04	0.026	-0.129	0.6604	-3.2766	0.658
-13.44	13.8	-2.52	0.1	0.02	-0.153	0.508	-3.8862	0.734

Table 7 - Front Kinematic Investigation Data 2

Flipped Toe (Degrees)	Zeroed Toe (Degrees)	Spring Length (mm)		Delta Spring Length from Start (mm)	Motion Ratio	Scrub Measured	Zeroed Scrub
0.734	0.480	240.00	-24.00	0.00	0.731707317	0	0.7493
0.658	0.403	247.00	-17.00	3.00	0.682730924	0	0.7493
0.522	0.267	253.00	-11.00	9.00	0.647058824	0.1016	0.6477
0.386	0.132	259.00	-5.00	14.00	0.568181818	0.3429	0.4064
0.255	0.000	264.00	0.00	20.00	#DIV/0!	0.7493	0
0.127	-0.127	270.00	6.00	26.00	0.666666667	1.3081	-0.5588
0.000	-0.255	273.00	9.00	33.00	0.652173913	1.6891	-0.9398

Table 8 - Rear Kinematics Investigation Data 1

Wheel Position	Wheel Position Norm	Ride Height	Measured Camber	Camber Change Measured	DTI A (Thou)	DTI B (Thou)
276.49	30.39	-30.39	-2.64	-0.45	0	0
268.98	22.88	-22.33	-2.5	-0.3	-8	-21
261.37	15.27	-14.97	-2.36	-0.2	-20	-46
253.8	7.7	-7.7	-2.24	-0.08	-35	-74
246.1	0	0	-2.16	0	-52	-105
239.6	-6.5	6.5	-2.07	0.09	-71	-139

Table 9- Rear Kinematics Investigation Data 2

Calculated Toe	Toe Measured	Spring length	Spring Change	Scrub Measured	Scrub Measured Norm
0	- 0.233590486	267	-22	0	1.3462
0.055166561	- 0.176294706	274	-15	0.3302	1.016
0.110333121	- 0.118998927	279	-10	0.6604	0.6858
0.165499682	- 0.061703147	284	-5	0.9906	0.3556
0.224909824	0	289	0	1.3462	0
0.288563548	0.066110515	294	5	1.7272	-0.381

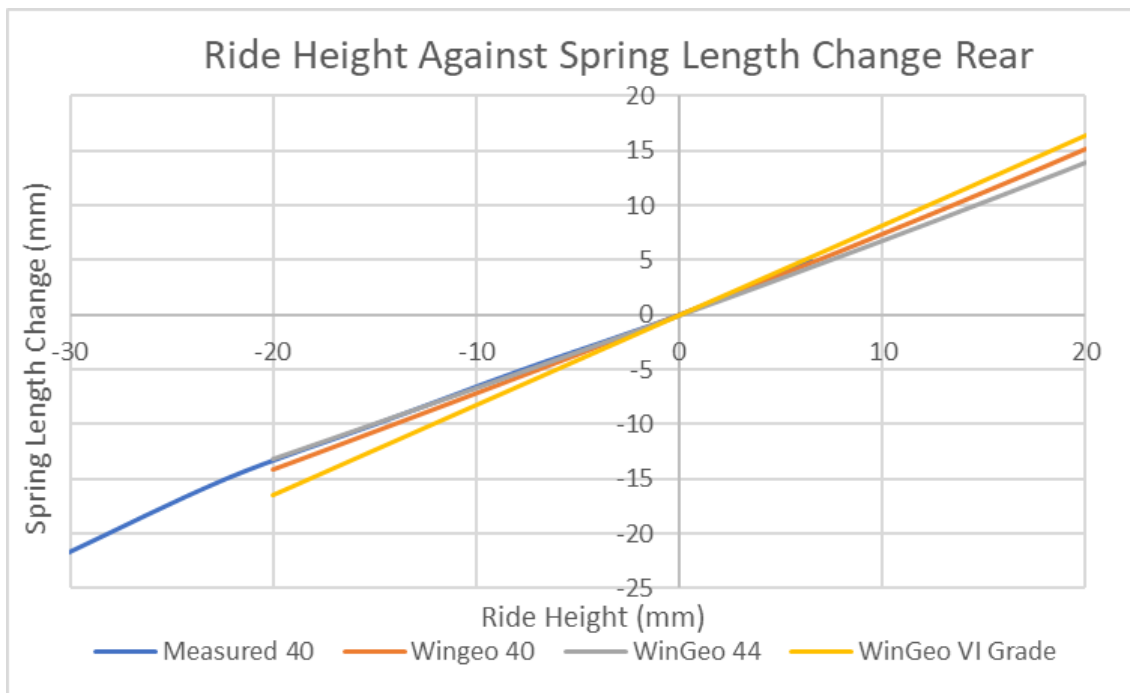


Figure 65 - Ride Height Against Spring Length Change Rear All

7.1 Minutes

26th Oct '22 IQ

- Discussion for Engineering Management assignment; our background information must correspond with group project's info as this will be cross verified. Detailing on an individual basis but sharing knowledge with each other on a linear basis should be unaffected.
- Uncovered a timetable discrepancy affecting those in Motorsport Eng; Cattani rolling road session clashes with classroom practical session. Inform Tim and propose swapping students.
- Brief states to compare individual strengths to decide which areas we will all champion for improvement. Fingers crossed the self-assessment forms from Mark's class will correlate with techniques used within our chosen areas.
- Gantt chart needs creation, a large chart with everyone's developments should be displayed on one page/slide for ease of view. Must be clear, easy to interpret and showing regular amendment.
- Health and Safety should be mentioned regarding practical work, risk assessments should contain everything to mitigate danger from workshop activities. Wear steel-cap boots.
- Lambda sensor for exhaust flow research, find one compatible with #40's ECU (Liam)
- Bill of materials; easy for shopping lists but for kinematics research? How extensive should costing be considered; equipment used, power requirements, technician time?
- Remember the big picture! Eye on milestones and log progress, balance objective scope and project scope with time in mind. Balance vehicle development and presentation preparation fairly so Tim is satisfied, and we get marked highly. Remember group project's brief technically doesn't account for the Sports2000 car, so over-tasking for vehicle development could cost us dearly.
- Reference Belbin's model regarding group decisions influenced by us member's strengths.

29th Oct '22 18:30 online

- Creating the Gantt chart, recollection from previous meetings regarding organisation and areas of interest on the vehicle.
- Frustrated by template Gantt charts, we have decided to make our own and manually colourise elements and date tasks.
- Research into lambda sensors; MoTeC recommends Bosch or NTK sensors with an M18 thread, wideband temperature support and does not plug into the OMEX ECU. The Bosch LSU 4.9 appears to tick all the boxes, retailing £70-£110. Welding in a mounting bung for the sensor, at correct location and angle on the exhaust, will also need doing.

30th Oct '22 17:00 online

- Engineering Management report layout discussion: Belbin's theory discussed and applied so team roles are cross verified with each other's strengths. On the surface it seems credible however some papers suggest otherwise.
- Installed Mendeley Desktop reference manager and collated sources regarding Belbin's model, as well as grouped previous sources from MS Word into an archive folder for later.

- Discussed some CAD regarding the temp. sensors and suitable glue/adhesives.
- Written each of our roles relative to Belbin's model, compliant as far. Unrelated to academia but Meredith Belbin is a bloke.

31st Oct '22 IQ with supervisor Tim

- CAD sensors feedback, both designs to be 3D printed as they each have their pros and cons.
- Talk about Faro arms and measuring them for WinGeo, 8hrs front + rear. Week commencing 7th November is lecture free bar engine flow stuff.
- Tim draws guide for pivots in the rockers for data collection, look for how they knock on each other and how #40 & #44 are different, thus how they behave differently.
- Mass reduction #40 is 548kg w/o Tim #44 is 562kg without John + airbox. Titanium is light but cost is significant. #44 has a big sheet of Aluminium glued in its nose but cutting ribs out of it would risk damage, take lots of time and may not yield much.
- Optimum Lap software good for c_d validation and works out how fast the car is around a track, within 10% of reality. OptimumG.com
- Measure engine head to find coefficient for Ricardo wave model and see what could improve. Filter, box off run tests then gauge resistances.
- MoTeC needs an adapter to read lambda signal, see crib sheet. Use the Bosch LSU4.9 and draw diagram using the open MCR frame as a sketch platform.
- Independent from the temp sensors on the loom is internal pressure monitoring. Work out bandwidth to see how many sensors can run simultaneously to the MoTeC.

7th November '22 IQ

- Updating Gantt chart with recent progress and improved its ease of use by formatting dates and progress bars more clearly.
- Engineering Management report discussion; Belbin's model was published in 1981, to which as a basic introduction to organisation, yes it's ok. But discrepancies have been found within the suspension kinematics team members, this paired with previous reports doubting the model on an advanced level, shows it may not be reliable.

14th November '22 IQ

- On-board Lambda spec complete, bung measured and compatible for sensor so ready to install: decide if track testing is necessary.
- Sensor casing new prototype send to Tim for printing, with a new material: PLA, print one and another in standard SLS and compare which is best. Sensors costed \$300 each (late 2021)
- Ricardo wave single cylinder model working, but overrevving >7,250RPM. Look at varying fuel pressures 40PSI minimum.
- Plan testing with testing to see what can change, test w/o exhausts and compare to reality. Find flow coefficient for engine head. Consider airbox indent to sculp airflow into trumpet.
- Time constraints means measuring one car (#40) is viable. Near complete, consider damper changes for dyno. Wheel rate testing 12-24PSI @+1 increments has good graph potential.

- Suspension evaluations between the two cars, #44 has a lower roll radius so tyres scrub harder and warm-up faster but by how much, and/or does it? Do graphs.
- Battery box design for the new LiPo, check MSUK regulations.

21st November '22 IQ

- Minute taker (Liam) not present due to one-off meeting clash with Learning Support.
- Through peers, no major objectives or conclusions discussed, merely progress updates and advice from the supervisor.

28th November '22 IQ

- Discussions into time management and workflow over the Christmas period; take a break but external reading and some analysis is expected, as well as finish practical reassembly of vehicle #40 before the workshops close.
- Kinematics measuring for WinGeo to be completed before end of term and compare results before/during the holiday. Highlight adversities and link to physical vehicle differences.
- TTMS sensor housings; consider a rear design optimized for fitting in the rear pontoons on the vertical concave surface.
- New LiPo battery coming, dimensions 87x150x105mm w/d/h make a battery box. 3D printed or aluminium fabrication?

5th December '22 IQ

- Module effort allocated to Thermodynamics due to a submission the night before the meeting, hence minimal progress however this was informally discussed.
- Finalize Christmas holiday plans and expectations.
- Dyno data compare with Cattani's runs, flow bench configurations, test plans.
- Finalizing measurements for models, WinGeo validity, damper curves, #40 complete reassembly.
- Critical review of the datalogger's inputs and paths from sensor to SVIM to MoTeC.
- LiPo battery case CAD design perceived as rushed and rubbish, try again with compatibility in floor mounting as well as low mass and room for foam cushioning.
- Improve the rolling road's non-OBD engine speed signal; find its operating voltages for signal and power lines and create an adapter. See Mark Fall for advice.
- Rolling road P-drag values not favoured. Consider calculating rotational inertias, tyre hysteresis, transmission losses etc. into a nominal flywheel-to-wheels inefficiency percentage.

Christmas Holiday – independent research and analysis

9th January '23 IQ

- Not a formal meeting due to supervisor not present, but informal chat between project members to how everyone's doing. No breakthroughs or shortfalls to report.

15th January '23 IQ with supervisor Tim

- Welcome back and recap progress.

22nd January '23 IQ

- Theory work

13th February '23 IQ

- Sensor justification; it's not we have these what can we do with it. Think what we want to measure and propose new sensors to fill the vacant spots. Full justification is needed to purchase new sensors then install, so continue research to clearly understand what i2 Pro is representing.
- Suspension; anomaly in camber/toe change relative to model predictions (WinGeo), consider Tim's older models from VIgrade to validate measurements (in case they're wrong, unlikely but possible). He explains it could be one value throwing the whole model out of expectation so precision inspection recommended. See effects of how position changes affect it all.
- Engine; inquire Owen about Ricardo Wave query, and chatting airbox strategy. Come up with a design and justify how it's better and why.

20th February '23 IQ

- Model tweaks for comparison with VIGrade and WinGeo; understanding where #44 differs kinematically when also compared to #40 (suspension arm mount locations slightly offset changing instant centres etc). Dynamic analysis to broaden the investigation scope.
- Source damper lecture information, calculate front and rear damping ratios, ride frequencies – where are the S2000 cars in this region?
- Compare inlet spacing model with different sizes, evaluate impact.
- Decide whether to use CNC'd inlet spacer or the four smaller 3D-printed trumpet spacers.
- Engine test plan b2b approach, power curve analysis and link engine development to attributes in recorded results on the rolling road.
- Mathchannel lists

27th February '23 IQ

- Minute taker (Liam) not present due to being unwell.