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An MCR Race Car DOL Lap-Time Model in VI-Grade: Box–Behnken DOE for Multivariable Sensitivity and Optimization

Master's Thesis in Motorsport Engineering
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Abstract

This thesis develops and validates a Driver-Out-of-the-Loop (DOL) simulation model of the MCR race car using VI-Grade / VI-CarRealTime. The aim was to replicate real-world behaviour through the integration of measured data, including suspension geometry, spring and damper rates, tyre characteristics, aerodynamic maps, and powertrain parameters. Correlation with experimental data ensured that the model could be reliably used for sensitivity studies and optimisation.

A series of validation tests were performed, covering tyres, camber, toe, springs, dampers, anti-roll bars, and aerodynamics. These confirmed that the model reproduced expected dynamic behaviour, demonstrating how even small changes in parameters affect handling, stability, and lap time.

Optimisation was carried out using a Box-Behnken Design of Experiments in MATLAB, focusing on front damper scaling, rear damper scaling, and camber angle. Results showed that camber had the strongest influence on lap time, while dampers primarily affected handling balance and steering feedback. The optimised setup achieved measurable lap-time improvements, though performance gains were found to depend on racing line and operating conditions.

The research demonstrates the value of simulation-driven optimisation in motorsport, providing a framework for future studies that incorporate additional variables, multi-objective optimisation, and robustness testing.

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1. Introduction

1.1 Historical Context

Vehicle simulation has evolved significantly from its early reliance on mechanical testing to advanced computational modelling. In the mid-20th century, vehicle testing primarily depended on physical prototypes and experimental methods. However, the introduction of computer-aided designs (CAD), finite element analysis (FEA) or finite element method (FEM) in the 1950s and 1970s revolutionized the field, enabling engineers to simulate car behaviour under different conditions. Knowing that the first article on FEM was published in 1956 (Turner et al., 1956).

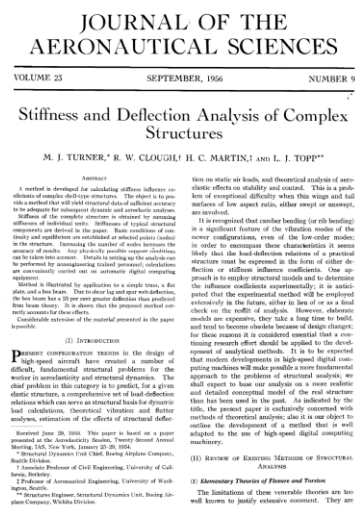


Figure 1 First FEM article published in 1956 (Turner et al., 1956)

By the 1980s and 1990s, real-time simulations and driver-in-the-loop systems became standard, enhancing the accuracy of vehicle dynamics representation. But even before they became the standard in the automotive or motorsport industry the DIL simulators for racing applications have been around for quite some time. Colin Chapman Lotus founder by 1966, already experienced the pinnacles of motor racing success, having won the Formula One world championship. Ever-eager to promote Lotus's engineering capabilities, Chapman decided to take a swing at manufacturing driving simulators and in 1965 and 1966 it was reported that around 20 of Lotus simulators were produced and sold (Haigh, 2024).

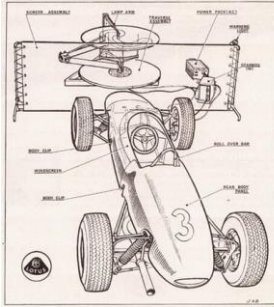


Figure 2 Lotus Driving Simulator - Technical Bulletin (1966) (Haigh, 2024)

In recent years, significant advancements have been made in driving simulators, particularly in Driver-in-the-Loop (DIL) simulation. VI-grade has been at the forefront of these developments, creating fully customizable DIL simulators, including the latest DiM400 and higher models. These simulators, part of the DiM family, integrate state-of-the-art technologies to enhance driving simulation fidelity. They retain VI-grade's signature architecture, featuring a lower stage that translates and rotates in a plane—representing chassis and tire-roadway dynamics—coupled with an upper stage equipped with a six-degree-of-freedom mechanism (Vi-grade.com, 2025).

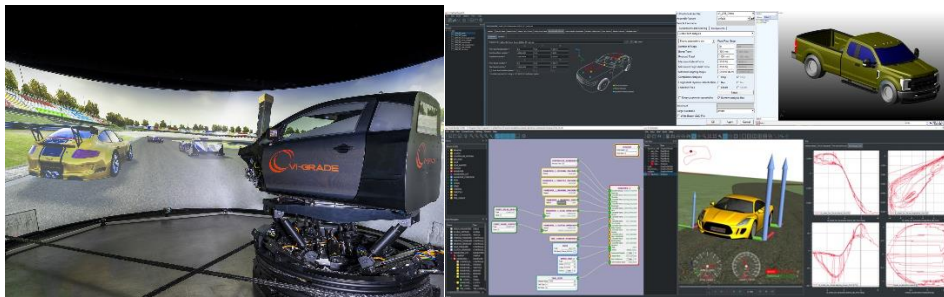


Figure 3 DiM400 Driver-in-the-loop Simulator Rig & VI-CarRealTime Simulator Software (Vi-grade.com, 2025)

Beyond hardware, VI-grade has also developed VI-CarRealTime, a real-time vehicle simulation environment that enables a streamlined development process from concept to sign-off. This software utilizes a simplified vehicle model using multibody simulation, allowing vehicle dynamics engineers to efficiently conduct large-scale Design of Experiments (DOE) and multi-objective optimization studies. The ability to rapidly evaluate various design configurations makes VI-CarRealTime a powerful tool for accelerating vehicle development (Vi-grade.com, 2025).

1.2 Aim and Objectives

1.2.1 Aim

The intended research aims to develop a Driver-Out-of-the-Loop (DOL) simulation model of an MCR race car using the VI-CarRealTime platform within the VI-Grade simulation ecosystem. The objective is to predict the kinematic and dynamic behaviour of the vehicle under various operating conditions, closely replicating its real-world performance.

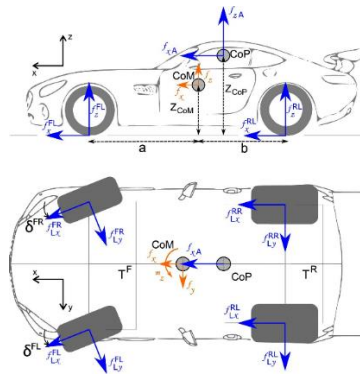


Figure 4 Physical representation of the different forces acting on the car (Tucker et al., 2021)

To achieve this, the study will incorporate real vehicle parameters, including gear ratios, engine power curves, front and rear spring and damper rates, tire characteristics (calibrated by testing various tire models to enhance correlation with actual data), suspension geometry, camber, toe settings, anti-roll bar stiffness and an aerodynamic attitude study (different ride heights)

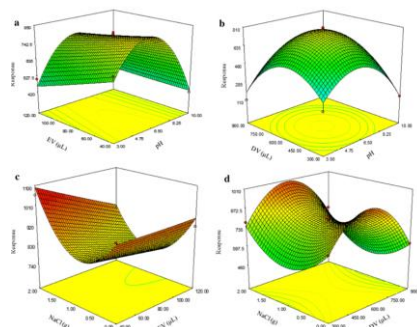


Figure 5 Example Response Surface and contour plots using Box-Behnken design (Habibi, Mohammadi and Kamankesh, 2017)

Once the model accurately reflects the behaviour observed in real test data, a performance optimization phase will follow. This will employ a Design of Experiments (DOE) methodology, specifically the Box-Behnken Design (BBD) within the Response Surface Methodology (RSM)

framework. The optimization will focus on three key variables, analysing their influence on lap time, the chosen performance metric, to identify the optimal setup.

Response Surface Methodology (RSM) is a sequential procedure designed to identify optimal operating conditions efficiently. It begins with linear models, which approximate the system response with minimal curvature, and progresses to quadratic models that capture curvature and locate the true optimum. Central to this approach is the method of steepest ascent, which defines the path of maximum improvement from the initial operating conditions. This path is oriented perpendicular to the contour lines of the response surface, ensuring the greatest possible increase in the performance metric (Rech, 2025).

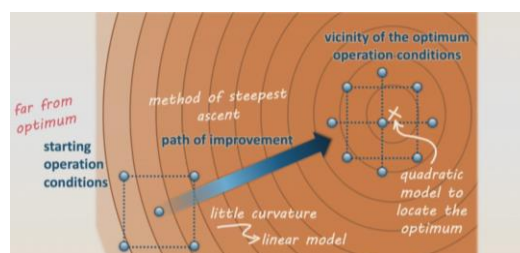


Figure 6 Response Surface Methodology as a Sequential Procedure (Rech, 2025)

When multiple responses are involved, RSM enables simultaneous analysis of factors through multi-response optimization. For example, in chemical processes, a combination of temperature and time may be selected to achieve targets for yield, viscosity, and molecular weight. In this study, the same principle applies to vehicle parameters such as tyres, camber, toe, spring rates, and damper settings. While contour plots can provide an approximate visual solution, more rigorous optimisation can be achieved through desirability functions. This method, introduced by Derringer and Suich (1980), transforms multiple objectives into a single composite function, allowing for systematic identification of the most desirable vehicle setup.

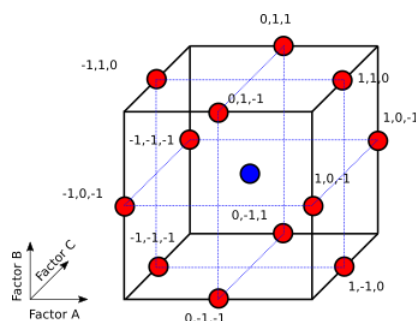


Figure 7 Box-Behnken Design (Develve.net, 2022)

The Box-Behnken Design is a robust multivariable optimization method based on a quadratic model, enabling efficient exploration of factor interactions while minimizing the number of

simulation runs compared to traditional factorial designs (www.itl.nist.gov, n.d.). It is particularly suitable for engineering optimization problems requiring high accuracy with limited experimental resources (Zolgharnein, Shahmoradi and Ghasemi, 2013).

1.2.2 Expected to achieve through the research

The primary expected outcome of this research is to develop one or more optimized race car setups that yield the best lap time for each track tested. Initially, the research aims to establish an optimal baseline setup for a representative track from the racing season. However, with further refinement, the goal is to tailor setups for multiple tracks, allowing for improved adaptability across different circuit conditions.

Additionally, this research aims to produce a fully functional VI-Grade VI-CarRealTime simulation model of the MCR race car, capable of accurately predicting its dynamic behaviour. A successful implementation of this model will not only validate its predictive accuracy but also serve as a tool for real-time setup evaluation. If time permits, an interactive testing framework could be developed, enabling rapid setup validation at the university workstation. This would allow for on-the-fly adjustments to vehicle parameters, facilitating pre-event preparation before race weekends.

As a final step, the Design of Experiments (DOE) approach will be leveraged to refine the lap time optimization process. By analysing key variables such as suspension settings, aerodynamic balance, and tire characteristics, the research aims to determine the most influential parameters affecting lap time. The Box-Behnken statistical method will be used to quantify these effects, systematically identifying the most effective configurations. The goal is to generate data-driven, optimized setups that enhance vehicle performance across different conditions, providing a competitive advantage in real-world racing scenarios.

1.3 Structure of the Thesis

This thesis is structured to reflect the sequential stages of the research, beginning with the creation of the simulation model and advancing through validation, sensitivity analysis, and optimization. The cornerstone of the project is the development of a reliable VI-CarRealTime simulation model, as the validity of the research relies on its accuracy and correlation with real-world data.

The initial phase focuses on building the simulation model by systematically inputting the real car's data, organized folder by folder, corresponding to key vehicle subsystems: Body, Brakes, Front and Rear Suspension, Front and Rear Tires, Powertrain, Steering Gear, Wheels, Driveline, Compliance, and Aerodynamic Forces. This structured data entry ensures each subsystem is correctly represented within the simulation environment, laying the foundation for accurate dynamic behaviour prediction.

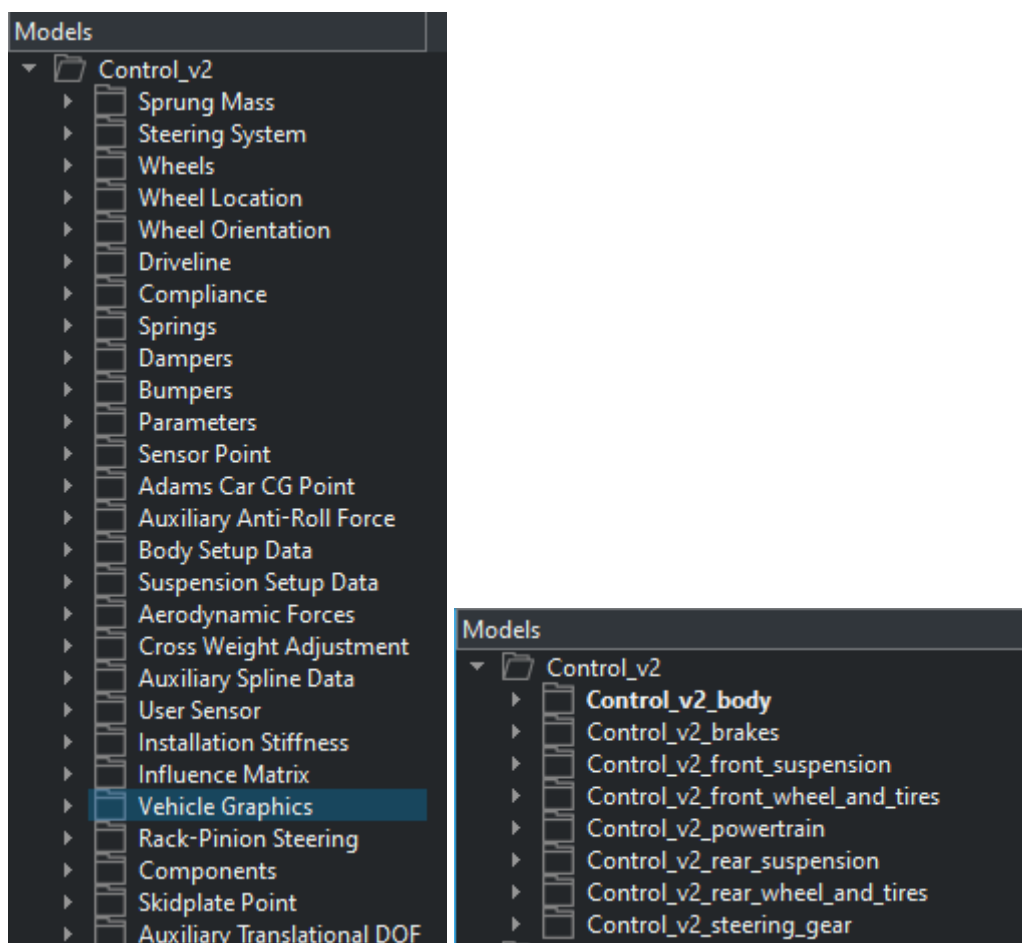


Figure 8 On the left side are the component types and in the right side the Subsystems

Following model creation, the validation process is carried out on two levels. First, a MATLAB-based analysis tool will be developed to directly compare simulation outputs with real-world data, verifying the model's steady-state and transient responses. Second, a sensitivity analysis will be conducted to assess the model's dynamic behaviour under controlled parameter variations. For example, the effect of reducing front and rear camber by 1° on both top speed and cornering performance will be analysed. This sensitivity testing will encompass critical factors such as tire behaviour (tire testing and modelling), spring rate variations, damper rate adjustments, anti-roll bar stiffness changes, and aerodynamic performance, particularly through the attitude study, which evaluates downforce response to varying front and rear ride heights.

To complete the validation stage, a sample of the driver model used within the simulation will be presented, demonstrating how it emulates real driver behaviour during the test sessions. Once the model exhibits consistent and expected responses to subsystem changes, the project will advance to the optimization phase. The insights gained from the sensitivity analysis will guide the selection of key factors to be optimized using the Box-Behnken Design (BBD) methodology, ensuring a focused and data-driven approach to setup optimization.

2. Industrial Relevance

The development of accurate vehicle models has become a necessity across the automotive and motorsport sectors, forming a standard practice for engineering teams worldwide. In both industries, the ability to predict performance, efficiency, and handling through simulation is critical to reducing the risks and costs associated with physical testing. Modern race teams, for example, rely heavily on vehicle models to explore a wide range of setups, evaluate system interactions, and optimise lap time performance before the car ever reaches the track. Similarly, automotive manufacturers employ these models to assess safety, ride comfort, and fuel efficiency during the early stages of design, where changes are easier and less costly to implement.

Vehicle models provide an essential framework for identifying the limitations and potential failures of subsystems under controlled conditions. Faults, errors, or inefficiencies that may not be apparent during initial design stages can be revealed through model correlation and sensitivity studies, ensuring that critical issues are addressed before the vehicle is tested in real-world scenarios. This approach is particularly valuable in motorsport, where budgets are tightly controlled and track testing is both time-limited and financially demanding. Without accurate simulation models, the only alternative would be extensive on-track experimentation, which could involve spending thousands of pounds per day in resources, logistics, and track time, often without guaranteeing meaningful progress.

For these reasons, vehicle modelling is not simply an additional tool but rather a foundational element of modern engineering methodology. It enables data-driven decision making at every stage of the design and development cycle, from the evaluation of conceptual designs to the refinement of detailed components. By creating a reliable model that correlates with real-world data, teams can run predictive analyses, test the sensitivity of performance to various parameters, and optimise their designs within a virtual environment. Ultimately, this approach ensures that once physical testing begins, it is guided by insight rather than trial and error, reducing development time and significantly improving cost-effectiveness.

3. Literature Review

3.1 The Science Behind Anti Roll Bars

The antiroll bar is a circular sectioned torsion bar mounted transversely on spaced out rubber bush swivel bearings to the body and having cranked arms. That is, bent extended ends which are attached to each swing-arm via short vertical link-rods by means of rubber-bush joints. Thus, the function of the antiroll bar to relieve the main suspension springs of some of their load every time the body rolls. An antiroll bar stiffens the suspension springing when the body rolls or one wheel goes over a bump or dip in the road (Gillespie, 2021).

The antiroll bar therefore permits softer suspension springing to be used when the vehicle is moving straight ahead, as this responds far better than hard springing to small ripples or irregularities in the road, hence it improves the ride comfort of the passengers. In the extreme, some antiroll bars are tuned to take as much as 30–40% of the total vertical load imposed on the suspension when subjected to severe body roll (Bayrakceken, Tasgetiren and Aslantas, 2006).

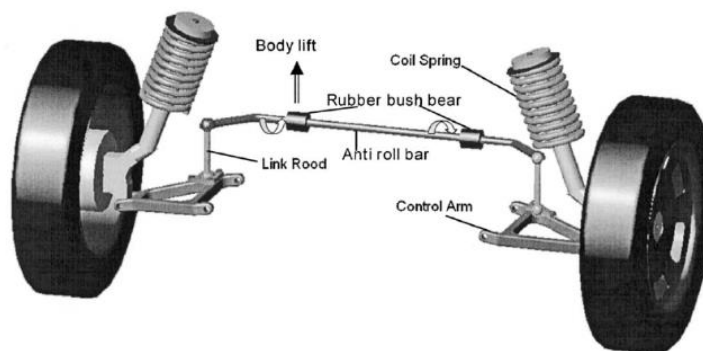


Figure 9 Schematic presentation of an anti-roll in action (Bayrakceken, Tasgetiren and Aslantas, 2006)

The operating principle is that, under torsional loading, the outside wheel gains vertical load while the inside wheel loses an equivalent amount. The torque carried by the bar, however, cannot exceed the combined effect of the inside unsprung weight and the compression load of the suspension spring. In most dynamic scenarios, both left and right suspensions move either in opposite directions or in the same direction with differing amplitudes. To quantify this behaviour, two parameters are central: angular rate, defined as the torque required to twist the bar by a given angle, and linear rate, defined as the vertical displacement of the pickup point on the anti-roll arm

per unit of applied load. Together, these describe the stiffness contribution of the bar and its influence on vehicle dynamics (Cronjé and Els, 2010).

When determining the angular rate of an anti-roll bar, the shear modulus (G) must be considered, as it is significantly lower than the tensile modulus. For example, in E-4130 chromoly steel, the shear modulus ranges between 11.4×10^6 and 11.7×10^6 psi, while the tensile modulus is approximately 30×10^6 psi. This distinction is critical for accurately predicting torsional behaviour, as the bar's angular rate in radians is calculated using the following relationship (Rowley, 2003):

$$K = G * \frac{(OD^4 - ID^4)}{I}$$

Equation 1 Angular Rate for Tube in Radians (Rowley, 2003)

Where K is the bar spring rate in force per radian, the G is the shear modulus, G is the modulus times Pi over 32. The OD is the outside diameter of the tube, the ID is the inside diameter of the tube finally the L is the length of the tube/bar, the end of the bar versus distance between the mounting bushings (Rowley, 2003). In torsional loading, fibre displacement increases with distance from the bar's centre, meaning the outer fibres carry most of the stress while the inner fibres contribute little. As a result, hollow anti-roll bars provide nearly the same stiffness as solid bars but with significantly less weight. Dynamically, during an outside wheel bump, the inside arm of the bar moves minimally until the transmitted forces exceed the inside unsprung load, highlighting that load transfer through the bar is ultimately limited by suspension constraints (IJERA Journal, 2014).

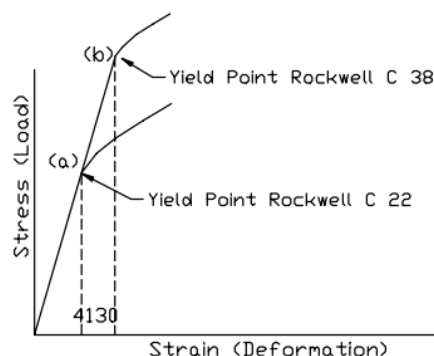


Figure 10 2 Different Anti Roll Bars on a Strain vs Stress Graph (Rowley, 2003)

Anti-roll bars are typically manufactured from steel alloys with a carbon content ranging from 0.28% to over 0.43%. Common materials for torque tubes or torsion bars include 4130, 4140, and 4340 steels. While these alloys exhibit similar shear moduli, their yield strength and elastic limits vary significantly depending on heat treatment and tempering. For example, a torsion bar hardened to Rockwell C22 possesses a much lower yield point than one treated to Rockwell C38. At higher hardness values, the bar not only achieves a greater yield point but also allows more angular deformation within the elastic range. Consequently, the design of torque tubes must carefully balance yield strength with the permissible elastic range (Rowley, 2003).

In motorsport applications, particularly in formula cars, anti-roll bar adjustment is a critical aspect of setup. Many vehicles are equipped with cockpit-adjustable blade mechanisms, enabling the driver to modify stiffness while on track. Typically, the inner lever controls the front bar, while the outer lever adjusts the rear, allowing real-time tuning of the car's balance to suit changing conditions (Rowley, 2003).

Blade Style Cantilever Stiffness

For blade-type anti-roll bars, stiffness is determined differently, as the blades are manufactured from plate or bar stock rather than tubular sections. The most accurate method of evaluating stiffness is through CAD modelling and stress analysis; however, approximate values can also be obtained analytically. To calculate the stiffness of a cantilever blade, the following parameters are required: the Young's modulus of the material, the length of the cantilever (from its mounting point on the torsion bar to the drop link), and the blade's length, width, thickness, and orientation, which define the second moment of area. The governing equation is the same as that used for a tubular section (Shi et al., 2012)

$$\text{Cantilever Stiffness } (k) = \frac{3El}{L^3}$$

Equation 2 Cantilever Stiffness (k) for a blade-type anti-roll bar (Shi et al., 2012)

Where:

- E = Young's Modulus of Rigidity (Pa)
- I = Second Moment of Area (m⁴)
- L = Length of Cantilever (m)

The second moment of area for a bladed anti-roll bar differs from that of a circular section. It is essential to use the blade dimensions corresponding to the direction of bending, as orientation determines stiffness.

In the stiffest configuration, the triangular blade is positioned vertically. Deriving an exact expression for the second moment of inertia of a tapered triangular section would require complex integration, which is impractical for most applications. As many bladed anti-roll bars incorporate a slight taper, the rectangular section equation is commonly used as an approximation. While this provides a close estimate, the most reliable method for determining stiffness remains CAD-based modelling and analysis.

To determine blade stiffness, the second moment of area must first be calculated with the blade in its stiffest orientation, where the triangular section is vertical. In this case, the section can be approximated as a rectangle, and the second moment of area is given by (Suspension Secrets, 2017):

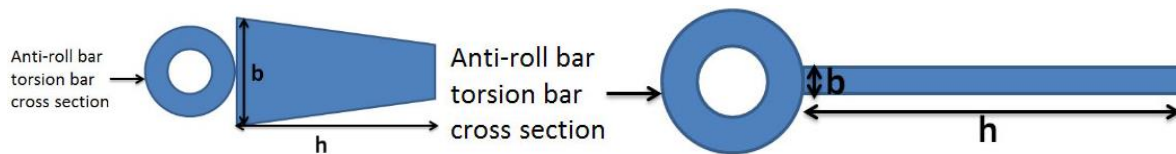


Figure 11 Bladed Cantilever side view (Suspension Secrets, 2017)

The equation is:

$$\text{Second Moment of Area } (I) = \frac{bh^3}{12}$$

Equation 3 Second Moment of Area (I) (Suspension Secrets, 2017)

Where:

- b = Base width of blade section (m) or Thickness of the blade side profile (m)
- h = Total length of blade (m) or Length of the cantilever blade (m)

Substituting the result from equation 3 into the cantilever stiffness equation gives the stiffness of the blade in its softest position. With these two stiffness values established, it is assumed that

intermediate blade positions produce stiffness values that vary linearly between the maximum and minimum, which are separated by 90°. A graph can therefore be constructed to represent the range of stiffness values across the full range of blade rotation. From this graph, the required blade angle for a target stiffness can be interpolated and carried forward into the overall anti-roll bar stiffness calculation.

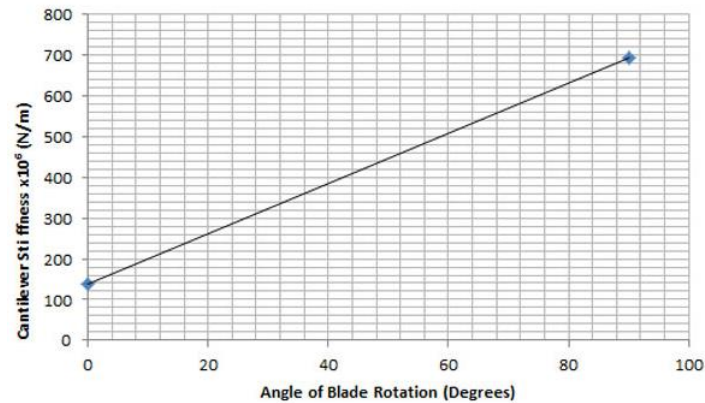


Figure 12 Angle of Blade Rotation (Degrees) vs Cantilever Stiffness (N/m) Graph (Suspension Secrets, 2017)

3.2 The Science Behind Tyre Loads

The kinematic centre represents the theoretical mass centre of a chassis when subjected to dynamic acceleration forces. Unlike the static centre of mass, which remains fixed under inertial acceleration, the kinematic centre shifts as the load distribution changes at the four tyre contact patches. Variations in roll or pitch also alter the vertical position of the kinematic centre, reflecting the influence of suspension geometry and load transfer (Oertel and Wei, 2012). Moreover, tyres exhibit a non-linear relationship between normal load and grip, as efficiency decreases with increasing vertical load. For example, a tyre operating at 500 lb of load may achieve an efficiency of 157%, but at 1250 lb this value drops to 78%. This reduction directly limits the performance envelope of the chassis (Rowley, 2003).

To illustrate, Tyre #1 can theoretically generate 2.54 g of cornering force at 125 lb of load, but only 0.5 g at 2000 lb. This highlights the importance of managing load transfer, as balanced lateral loads generally produce greater combined grip than unbalanced conditions. For instance, a chassis with 250 lb on the left front and 750 lb on the right front generates a combined cornering force of 1443 units, whereas evenly distributing 500 lb across both tyres increases this to 1566 lb an improvement of 123 lb (Rowley, 2003).

$$L_T = \frac{W * g * CG_z}{t}$$

Equation 4 Load Transfer Formula (Rowley, 2003)

Where:

A key principle of chassis dynamics is that certain terms are interdependent: load transfer is influenced by acceleration forces, while acceleration forces themselves are dependent on load transfer. This relationship underpins all dynamic corner setups, regardless of chassis type. Any change that alters the effective wheel rate whether through spring stiffness, anti-roll bars, dampers, or anti-jacking forces will produce comparable effects.

Table 1 The Influence of Spring Rate Adjustment of Dynamic Load Distribution (Rowley, 2003)

The Influence of Spring Rate Adjustment on Dynamic Load Distribution			
Front Left	Front Right	Rear Left	Rear Right
Increasing LF spring rate under braking raises load at LF and RR, while reducing RF and LR.	Increasing RF spring rate under braking raises load at RF and LR, while reducing LF and RR.	Increasing LR spring rate under braking reduces load at RF and LR, while increasing LF and RR.	Increasing RR spring rate under braking reduces load at LF and RR, while increasing RF and LR.
Under left-turn acceleration, load increases at RF and LR, decreasing LF and RR.	Under left-turn acceleration, load increases at RF and LR, decreasing LF and RR.	Under left-turn acceleration, load increases at LF and RR, decreasing RF and LR.	Under left-turn acceleration, load increases at LF and RR, decreasing RF and LR.
Under rearward acceleration, load decreases at LF and RR, increasing RF and LR.	Under rearward acceleration, load decreases at RF and LR, increasing LF and RR.	Under rearward acceleration, load increases at RF and LR, decreasing LF and RR.	Under rearward acceleration, load increases at LF and RR, decreasing RF and LR.

3.3 The Science Behind Brakes

Disc brakes consist of a rotor and a caliper assembly. The caliper houses hydraulic pistons that press the brake pads against the rotating disc, generating a clamping force. This clamping force produces friction, which in turn dissipates kinetic energy as heat. Consequently, disc design is primarily governed by the requirements of friction generation, heat transfer, and dissipation, as well as the magnitude of applied force [12]. Figure 1 illustrates a typical disc brake assembly and its main components up (Naveed, 2019).

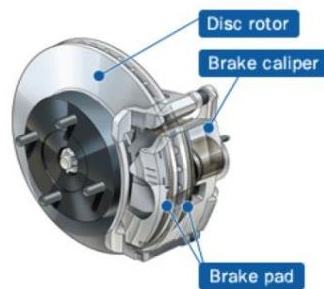


Figure 13 A typical arrangement of a disc brake and its components (Naveed, 2019)

The braking system must be capable of generating sufficient force to consistently exceed the maximum deceleration capacity of the tyres at any vehicle speed and sustain this performance throughout the race. The relationship between pedal pressure and braking force should be direct and linear, ensuring predictable modulation for the driver. Pedal effort must also remain within an appropriate range: excessive force requirements would compromise drivability, while overly light inputs would increase the risk of premature wheel lock-up (Naveed, 2019).

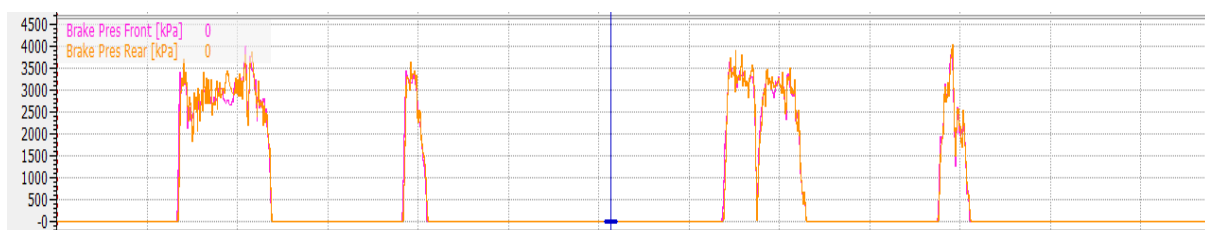


Figure 14 Visual representation of the Brake Pressure over 1 Lap (MOTEC i2 Pro)

To think about the brakes on average road racing circuit, something less than ten percent of the time required to complete a lap is spent braking, and in most cases, we want to reduce that percentage to single digits. Therefore, a five percent improvement in braking performance (not brake efficiency) would net a theoretical improvement in lap time one half of one percent or about one half a second in a 90 second lap (Rowley, 2003).

3.4 The Science Racing Tyres

Racing tyres are compliant, visco-elastic structures that convert normal load into usable lateral and longitudinal forces through shear of the rubber in the contact patch and deformation of the carcass. The performance is influenced by several key factors:

1. Motion Ratio and Wheel Rate
2. Friction Coefficient
3. Vertical Load
4. Tyre Temperature
5. Tyre Pressure
6. Slip angle
7. Slip ratio
8. Camber
9. Toe

Motion Ratio and Wheel Rate

Motion ratio is the difference in motion between two interconnected components due to some type of mechanical advantage or in other words is the ratio between the wheel movement and spring movement, this can be applied to dampers too, (Rouelle, 2020). The wheel rate is the vertical wheel deflection caused by a given acceleration or its resultant load transfer which is determined by the wheel's rate resistance expressed in pounds of force necessary to cause a deflection of one inch and measured at the wheel centreline (Smith, 1999).

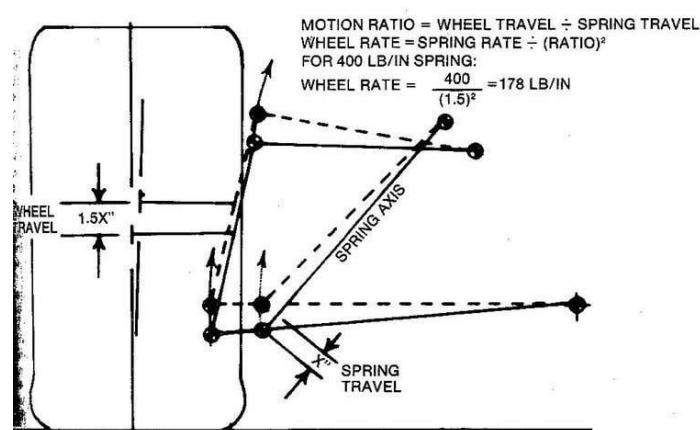


Figure 15 Wheel rate vs spring rate, conventional layout (Smith, 1999)

Vertical (normal) load and load sensitivity

Vertical load is the sum of the vehicle's static weight and aerodynamic downforce, both of which vary dynamically with pitch, roll, and weight transfer. As vertical load increases, both lateral and longitudinal forces also rise but in a sub-linear manner, meaning the coefficient of friction decreases with higher loads. This phenomenon, known as load sensitivity, is characteristic of racing slicks (Smith, 1999).

At the microscopic level, greater contact pressure induces higher shear within the rubber, more localised slip, and an elongation of the footprint. Stress distribution becomes non-uniform, so each additional unit of vertical load produces proportionally less grip. In high-downforce cars, this creates a trade-off: while downforce increases available peak force, it also exacerbates saturation of the tyre. Traction generally improves with vertical loads up to approximately 500–750 lb per tyre; beyond this range, frictional shearing forces diminish, reducing the effective coefficient of friction (Smith, 1999).

$$\frac{F_x}{\mu F_z} + \frac{F_y}{\mu F_z} \leq 1$$

Equation 5 Friction Ellipse Equation (Smith, 1999)

Tyre Coefficient of Friction

For example, the road can have 250 N of Vertical Load (friction) and the tyre can do 400 N of vertical Load dividing this we will obtain. The tyre can generate certain maximum amount of lat or long g-forces but that doesn't mean that the car will be able to do that as it is subject the vehicle dynamics, external factors of the car and the driver (Smith, 1999).

$$\mu = \frac{400}{250} = 1.6 \text{ g}$$

Equation 6 Coefficient of Friction calculation from Vertical Forces (Smith, 1999)

Cornering Power and Traction Power

Tyres generate two primary types of force: cornering capacity and traction capacity. Cornering capacity refers to the lateral force produced during cornering, or the tyre's ability to resist centrifugal acceleration. It is influenced by several factors, including (Smith, 1999):

1. Cornering Capacity of the tyres depends on
 - a. Suspension geometry
 - b. Vehicle load transfer characteristics

- c. Vehicle Downforce
- d. Size and characteristics of the tyres
- e. Vehicle Gross Weight
- f. Height of the Vehicle Centre of Gravity

The traction capacity is the acceleration and braking in the longitudinal sense of the tyre force. Increasing vertical Load on any given tyre will increase the traction capacity of that tyre.

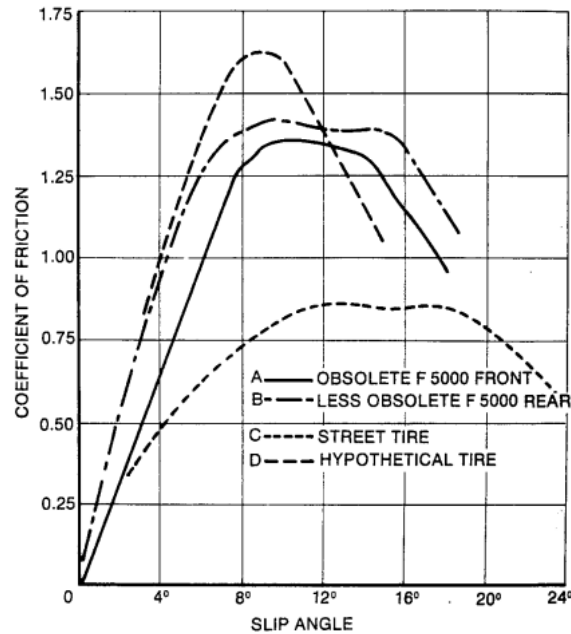


Figure 16 Tyre coefficient of friction vs slip angle curves (Smith, 1999)

Slip

Tyre slip is generally classified into two forms. The first is the slip angle, defined as the angular displacement between the plane of rotation of the wheel (the direction in which the rim is pointing) and the actual path taken by the rolling tyre on the road surface. At low slip angles, the tyre efficiently generates lateral force, but as the angle increases the elastic deformation of the contact patch reaches its limit, at which point sliding occurs. Excessive slip angles result in grip loss, although partial recovery may be achieved through minor steering corrections. Figure 16 illustrates the tyre slip angle as viewed from the road, with successive tread particle paths shown on the tyre surface (Smith, 1999).

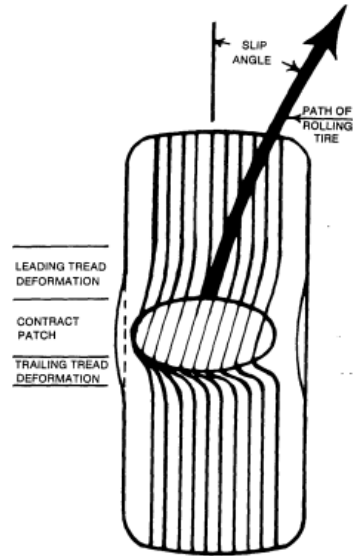


Figure 17 Tyre slip angle viewed from the road with successive tread particle paths depicted on tyre tread surface (Smith, 1999)

The distinction between elastic friction and sliding friction is central to understanding this behaviour. The tyre's contact patch, typically elliptical in shape due to tread and sidewall compression, initially deforms elastically under load. Beyond a critical slip angle, however, sliding friction dominates as parts of the contact patch lose adhesion. Figure 17 depicts this transition, showing tread particle paths projected onto the road surface (Smith, 1999).

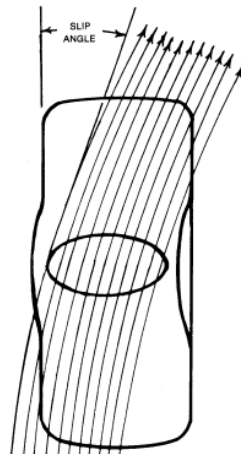


Figure 18 Tyre slip angle with tread particle paths projected on to road surface (Smith, 1999)

Camber and Camber Thrust

Coefficient and cornering power vary with camber angle, relative to the surface of the road, not to the chassis. Variability, maximum cornering force will be realized at some small value of negative camber. This is due to "Camber thrust" caused by the straightening out the arc of the

contact patch as the tread of a cambered tyre rolls over the ground. To have a better understanding, if the tyre is cambered in the negative sense, this force acts in the direction of the centre of curvature and increases cornering power. On the other hand if the tyre is cambered in the positive direction, it acts away from the centre of curvature and decreases cornering power (Smith, 1999).

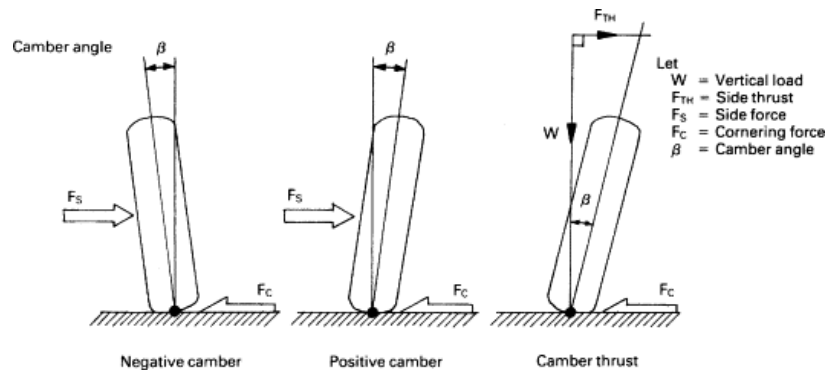


Figure 19 3 types of Camber acting on a tyre (Chang, 2013)

It is important to note that with wide, flat-profile tyres, excessive camber causes the vehicle to ride on one edge of the tread while lifting the opposite edge off the track. This not only reduces the effective contact patch area but also alters the pressure distribution across the footprint. At present, dynamic camber control remains limited, meaning some degree of adverse camber change must be tolerated.

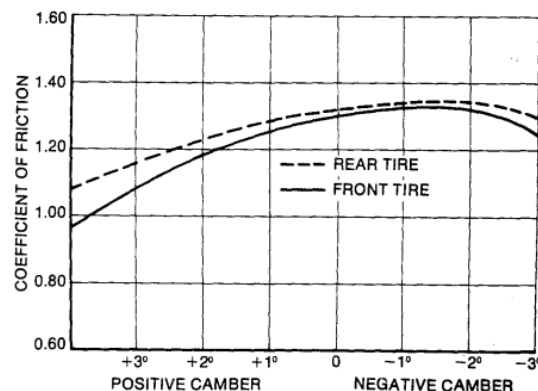


Figure 20 Camber angle vs Coefficient of Friction (Smith, 1999)

Under extreme lateral loads, tyres must dissipate stored energy, sometimes producing oscillations in the carcass. It has been observed that, under very high cornering and acceleration forces, the tyre can deform into a wave-like shape ahead of the contact patch. The sudden release of this energy at the trailing edge generates judder, often severe enough to be felt directly by the driver. Advances in tyre construction have reduced this effect to more manageable levels.

In practice, however, judder can serve as a useful indicator: if the rear tyres are not juddering on corner exit, it may suggest that either the chassis is not fully exploiting tyre performance or that the driver is not pushing the car to its potential (Smith, 1999).

Tyre Temperature

Any process involving friction generates heat, and part of the energy used to compress and distort the tread at the contact patch is dissipated as heat when the tread recovers at the trailing edge. Racing tyres are typically designed to deliver maximum traction within tread temperatures of 190–220 °F (88–104 °C). Rain tyres, made from softer compounds, reach peak performance at 140–160 °F (60–71 °C), while stock car tyres are designed to operate at even higher levels.

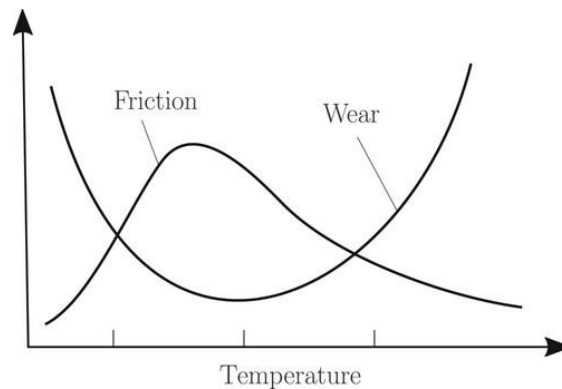


Figure 21 Temperature dependence of rubber friction and wear (West and Limebeer, 2020)

Maintaining tyres within their optimal range is critical. Below the operating window, grip is reduced; above it, tyres risk blistering or chunking due to thermal degradation of the rubber compound.

As a general rule, tread temperatures should remain above 175 °F (79 °C) but not exceed the specified compound limit. Several factors can accelerate local overheating, including:

1. Run too much negative camber and burn out the inside edge of the tyre
2. Run too low an inflation pressure or run with slow a slow leak
3. Run too soft a compound for the track or run rain tyres on dry track – which amounts to the same thing.



Figure 22 Tyre Temperature Monitoring System (Drewitz, 2018)

Temporary increases in tyre temperature can be achieved by lowering inflation pressure to the allowable minimum, applying additional static toe-in within safe limits, or increasing negative camber. These adjustments, however, must be balanced carefully to avoid compromising stability or accelerating uneven wear (Alessandro Bortolin et al., 2021).

Tyre Pressure

An important consideration with racing tyres is their tendency to lose pressure over time. The sidewalls are constructed with minimal rubber sufficient only to bond the cords resulting in higher permeability and increased leakage compared to road tyres (Richardson, 2017).

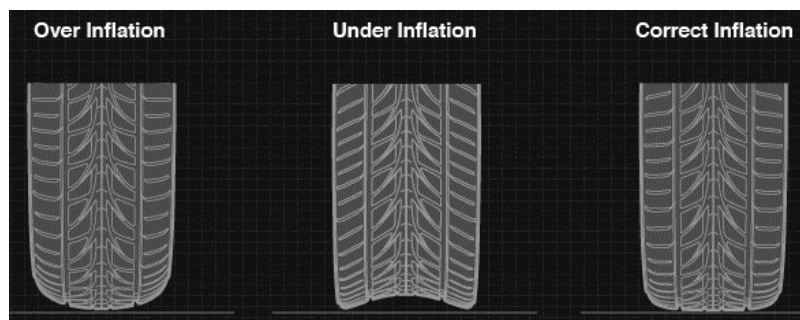


Figure 23 Types of tyre inflation (Bridgestone.co.in, 2025)

Caster

Caster refers to the angle of the steering axis when viewed from the side of the wheel. A steering axis positioned vertically corresponds to zero caster. Tilting the axis rearward by approximately 2–5° creates positive caster, while a forward tilt of the same range produces negative caster. Increasing positive caster generates additional negative camber during steering, enhancing grip in cornering. It also increases trail, which in turn improves steering feedback and stability (Vo, Jazar and Fard, 2017).

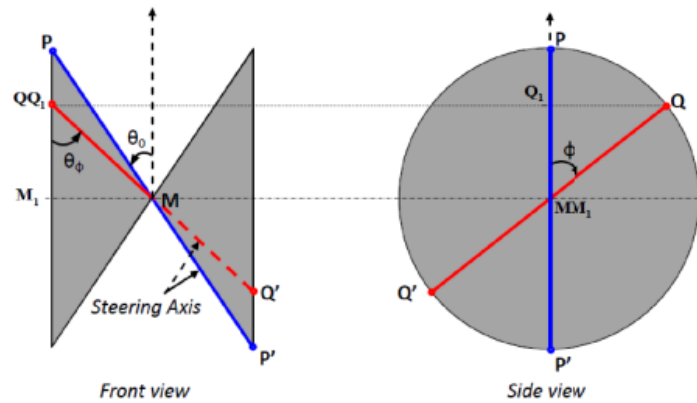


Figure 24 Geometry of the Variable Caster Mechanism (Vo, Jazar and Fard, 2017)

Pneumatic trail arises from the elastic deformation of the tyre under lateral force. When a side force is applied, the effective centre of pressure within the contact patch shifts rearward and outward relative to the geometric centre. This displacement creates a moment arm, producing a self-aligning torque that resists steering input and naturally returns the wheel toward the rolling direction. Together, caster and pneumatic trail define the level of steering feel, stability, and feedback experienced by the driver.

Tyre Appearance with Under or oversteer

Is the front tyre shows more signs of abuse than the rear it is telling us that the car has too much understeer, regardless of what the driver says on the other hand is too much abuse in the rear tyres too much oversteer in the tyres (Smith, 1999).

The Traction Circle

Racing tyres are capable of generating comparable levels of force during acceleration, braking, and cornering. When the maximum force vectors are plotted, two representations can be obtained. The first, shown in Figure 24, illustrates the idealised traction circle, in which the tyre can achieve the same maximum force in any direction. In this example, the tyre is capable of producing either 1.4 g of longitudinal acceleration or 1.4 g of lateral cornering force. However, it cannot deliver both simultaneously at their maxima.

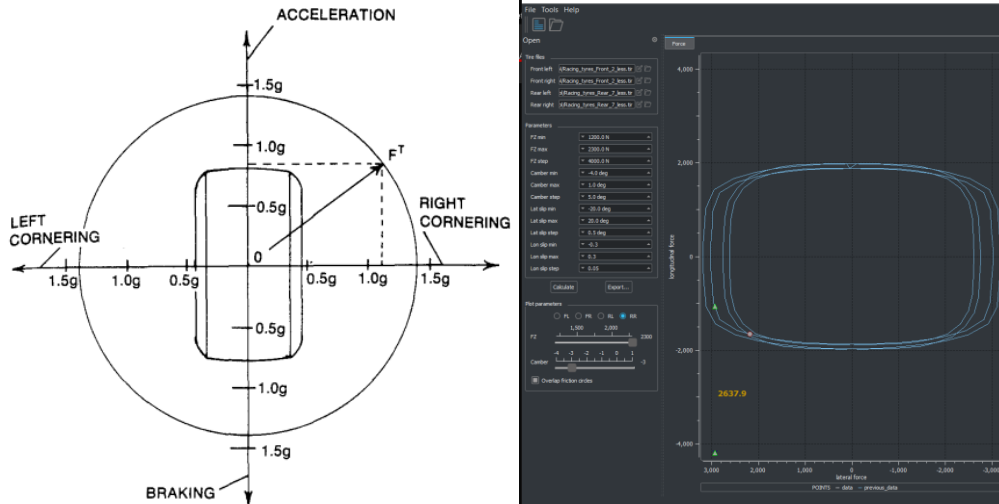


Figure 25 The traction circle (example) showing vehicle acceleration while turning right next to the maximum tyre limit from the model VI-TyreLimit (Smith, 1999)

When a tyre is required to generate both longitudinal and lateral forces, it must deliver a reduced a

Figure 25 presents the traction ellipse generated from VI-Grade's VI-TyreLimits tool, which more accurately reflects tyre behaviour. Unlike the idealised circle, real tyres display anisotropy due to material properties, construction, and load sensitivity. As described by Smith (1999), "the tyre can simultaneously generate an amount of braking thrust and an amount of cornering force which, added together, will total more force than the tyre is capable of developing in any one direction." This illustrates how the vector resolution of forces enables tyres to exceed their nominal single-axis capacity, albeit with trade-offs in efficiency.

"If we are going to utilize all of the performance potential designed and built into our tires, then we must keep the tire operating at very high combined force level at all times while the car is turning. We must ride the rim of the traction circle" by balancing the brakes, cornering force and throttle so as to keep the tire's resultant line of force just inside the boundary of the circle." (Smith, 1999).

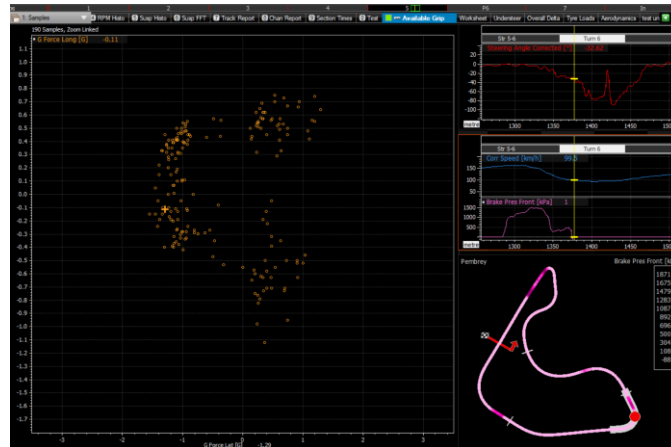


Figure 26 Graphical representation of the real data using MoTeC Pro i2 (corner analysis on Pembrey Circuit)

A graphical representation of tyre force progression through a corner is shown in Figure 27. The inset traction circle in Figure 27 illustrates three distinct approaches to cornering. The line labelled possible represents the ideal trajectory, closely following the rim of the circle, and reflects the performance of elite drivers as Mario, A. J. and Carroll Smith mentions. The probable line corresponds to the behaviour of a skilled club driver, while the classic line predicts the traditional methods of braking, cornering, and accelerating in the three separated phases. The comparison highlights the objective of maximising tyre utilisation by blending these phases into continuous trajectory.

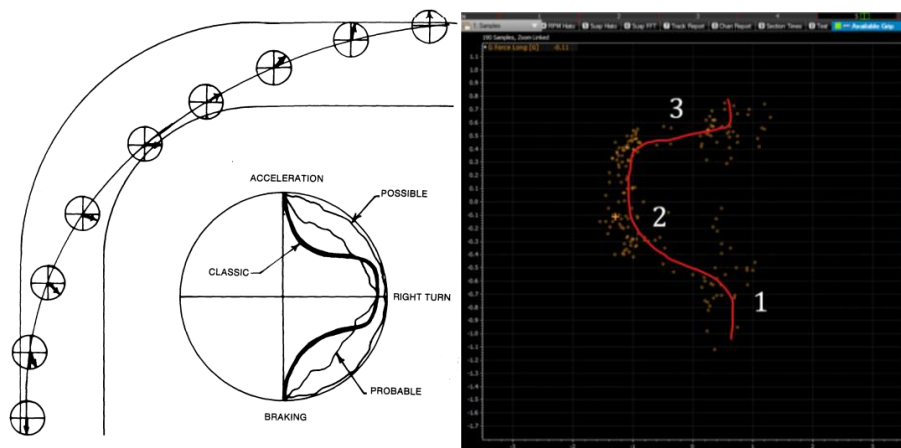


Figure 27 The traction circle and the tyre force vector as the vehicle progresses around a corner (example vs real data)

In this case the driver is braking into the corner sort of in a straight line (1) as the data points shows, and then in an effort to use more of the tyres he pushes the car by accelerating into the corner, this creates the curved lower section of the graph (2) which tend to look like the probable line stated in the book. Further into the corner reaching the apex it reaches peak lateral force around -1.55 g and -0.15 g and then leaving the corner into the corner exit the driver follows the

classical approach just turning the car when possible and not using all the lateral grip possible, showing that the driver is using a mixture of classical approach and probable leaving performance on the table.

3.5 The Science behind shocks (Dampers)

Dampers control the oscillations of springs, and in race cars their adjustment is critical to maintaining both suspension sensitivity and tyre compliance. Excessive damping reduces responsiveness and limits the tyre's ability to follow surface irregularities, while insufficient damping produces a soft, unstable response (Lacroix, Seers and Liu, 2006).

In general, greater damping force is required in rebound than in bump. The bump stroke primarily damps the motion of the unsprung mass, which is relatively small and does not vary significantly under dynamic load. Conversely, the rebound stroke damps the response of the sprung mass following compression during the bump phase and therefore requires more control. Manufacturers typically design dampers with higher rebound than bump forces, and while this ratio is fixed in non-adjustable or single-adjustable units, double-adjustable dampers allow independent tuning of bump and rebound and are considered essential in racing applications (Smith, 1999).



Figure 28 Type of dampers – Fox Vanilla RC - Penske 8760 (Shihady et al., 2006)

An underdamped car exhibits excessive oscillation, feeling soft and unstable. An overdamped car, by contrast, feels harsh, with wheel patter and loss of compliance. The optimal setting provides sufficient damping to allow the sprung mass to settle quickly, with minimal oscillation, while ensuring the wheels return to the road surface promptly without sacrificing sensitivity (Smith, 1999). Modern dampers can be built and adjusted to virtually any velocity versus forces profile, dismantle and re-valved. There are four primary shock/dampers stages thought often only tree are shown in the velocity versus forces dyno graph.

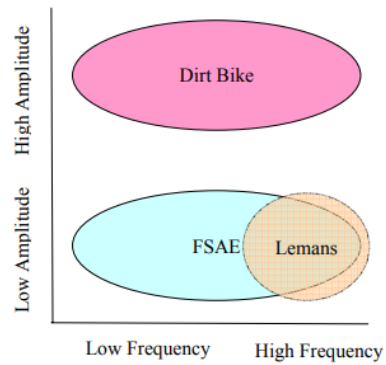


Figure 29 Frequency/Amplitude input ranges for dampers used in Dirt Bikes, FSAE and Lemans respectively (Shihady et al., 2006)

1. Gas Preload/Breakaway:

- a. The initial force required to move the shock shaft, caused by internal gas pressure acting on the shaft area. It increases with gas pressure and is distinct from shim stack preload.

2. Nose (Low Speed or Bleed):

- a. Controlled by the low-speed valve adjuster, which regulates bleed flow and thereby affects the rate of load transfer. Modern piston designs may use multiple shim stacks to refine this region

3. Knee (Transition):

- a. The point where the curve shifts from low-speed to high-speed behaviour, influenced by bleed valve type, piston design, shim stack preload, and adjuster settings.

4. Slope

- a. Mainly determined by piston and shim stack design, representing high-speed damping (up to ~152 mm/s). Modern pistons can deliver digressive, linear, or progressive profiles.

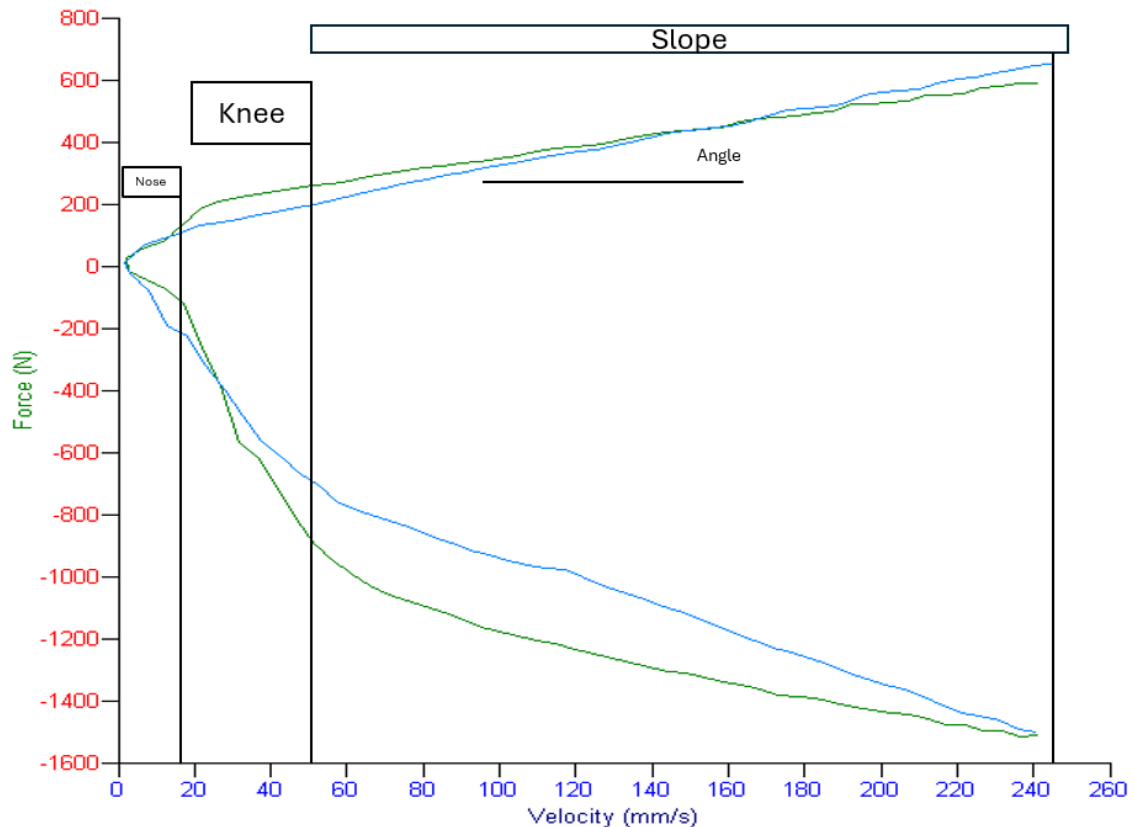


Figure 30 Section of interest from a Force vs Velocity damper Graph from real testing

Dampers fulfil several critical functions in both stock cars and race cars. From a mechanical grip perspective, they manage load transfer by influencing corner load distribution, the rate of transfer, and the rate of balance change. They also control spring motion by regulating compression travel and absorbing stored energy during rebound (Rowley, 2003).

Another major function of dampers is aerodynamic management. By influencing how loads are transferred through the chassis, dampers affect the aerodynamic pressure centre of the vehicle. This, in turn, impacts corner load distribution, the rate of load transfer, and the rate at which aerodynamic balance shifts. Dampers also contribute significantly to driver feedback. The way they are tuned affects how quickly a driver perceives changes in chassis balance, which directly influences confidence and car control at the limit (Rowley, 2003).

It is important to note that dampers work in close conjunction with springs, anti-roll bars, and anti-jacking forces. Although each of these elements has a distinct function, they cannot be considered in isolation. Unlike springs or anti-roll bars, dampers, apart from their gas preload

affect corner loads only when in motion. Their influence is therefore transient, with damper forces capable of changing significantly within a single second. This transient nature is why damper data is typically recorded at high sampling frequencies, often between 25–50 Hz (Rowley, 2003).

At low speeds, damper settings are used primarily to manage transitional conditions such as corner entry and exit. They stabilise spring behaviour by damping oscillations, working in series with the springs during compression and controlling the release of stored energy during rebound. Once the chassis is balanced, stiffer springs usually require softer compression damping but firmer rebound damping to maintain stability and responsiveness (Rowley, 2003).

3.6 The Science behind Spring and Spring Rates

A coil spring, although commonly referred to as a compression spring, is essentially a torsion spring. As the spring compresses or extends, the wire twists, and its resistance to this twisting motion generates the spring force, known as the spring rate. The resistance is primarily determined by the number of active coils, the mean diameter of the coil, the wire diameter, and the material properties. In function, a compression spring behaves similarly to a torsion bar or anti-roll bar, with energy stored through twisting.



Figure 31 Coil Spring Suspension on Supertox Oval Race Car (McGill Motorsport, 2021)

Springs are essential in maintaining tyre contact with the track surface by controlling vertical wheel deflection over bumps and irregularities. If a tyre loses contact and begins to bounce or float, it can no longer sustain steady-state or transient forces effectively. The rebound (extension) phase of a spring is just as important as compression, as it helps control ride height, contributes to roll and pitch resistance, and balances front-to-rear load transfer. Springs absorb and store energy from ride, bump, roll, and pitch inputs, while dampers dissipate this energy without storing it. Together, these components maintain stability and enable fine-tuning of ride and handling characteristics (Rowley, 2003).

Two basic spring configurations are commonly used in motorsport. Closed-end springs, flattened at both ends for smooth seating, are widely applied, while springs closed at one end and open at the other are typically restricted to front suspensions of NASCAR-type stock cars. Quality and consistency are critical; manufacturers such as Hyperco and Eibach are widely used, with distinct epoxy colour finishes to denote brand. Over time, springs may experience sag, a reduction in free height after use, which can alter their effective rate. Only high-quality springs tested for rate consistency should be used to minimise variability when evaluating suspension and chassis compliance, torsional rigidity, or bind effects (Rowley, 2003).

Suspension and chassis deflection can also affect spring performance. For example, in stock car configurations, if the front spring is compressed non-linearly due to misalignment of the upper perch, control arm pocket, or excessive arm arc, the spring may deflect away from its centreline. This results in a non-uniform compression and inconsistent rates. Springs must therefore be compressed squarely to maintain linearity and predictable performance. Even small inconsistencies in spring rate versus wheel rate can significantly affect the car's performance envelope (Rowley, 2003).

Roll resistance under lateral acceleration is determined by the vehicle's roll rate, expressed as the force required to resist one degree of roll. This resistance comes from the compression of the outboard springs and the contribution of anti-roll bars. Selecting suitable springs requires defining parameters such as inside diameter, maximum and minimum free length, loaded height at ride, and the desired rate in pounds per inch (Rowley, 2003).

4. Model Creation and Correlation

Model creation and correlation form the foundation of this study's methodology. The process involves building a virtual representation of the vehicle using data measured directly from the car, combined with component specifications provided by manufacturers. Accurate replication of the physical system is essential, as a model that does not behave in line with the real car cannot be used for further testing.

Correlation is therefore performed in parallel with model development, ensuring that simulated behaviour matches real-world performance at every stage. This iterative process validates the model's physical accuracy by comparing outputs such as vehicle dynamics, suspension responses, and tyre behaviour against experimental data. Without correlation, the model would lack predictive reliability and would be unsuitable for sensitivity studies or optimisation. Establishing this link between simulation and reality ensures that the virtual environment can be used confidently to evaluate setup changes and explore performance improvements.

4.1 Environment (Track and Path)

The environment for the simulation was defined by replicating the circuit and the exact racing line used during the real driving stint. The road geometry was provided and imported into VI-Road, where the path was then constructed to match the driver's actual line. Although it is possible to test alternative trajectories—such as wider or tighter cornering lines—the objective of this study was to reproduce the real-world conditions as closely as possible.

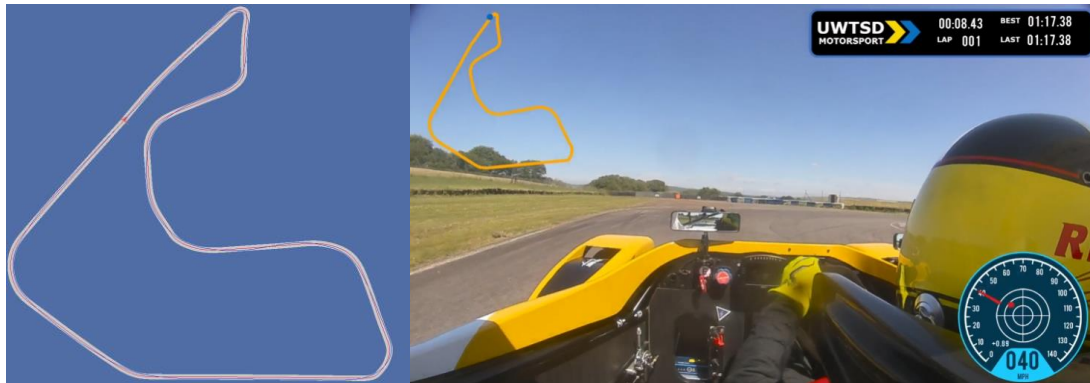


Figure 32 Pembrey Track on VI-Animation and onboard footage from a hot lap

To achieve this, onboard footage from the fastest lap and full stint was analysed, and the driver's line was reconstructed in VI-Road. The racing line generated in the simulation was later reviewed and confirmed by the driver as representative of his actual approach during the test. This step was essential to ensure that the correlation process began with a realistic baseline, reflecting both the circuit layout and driver behaviour.

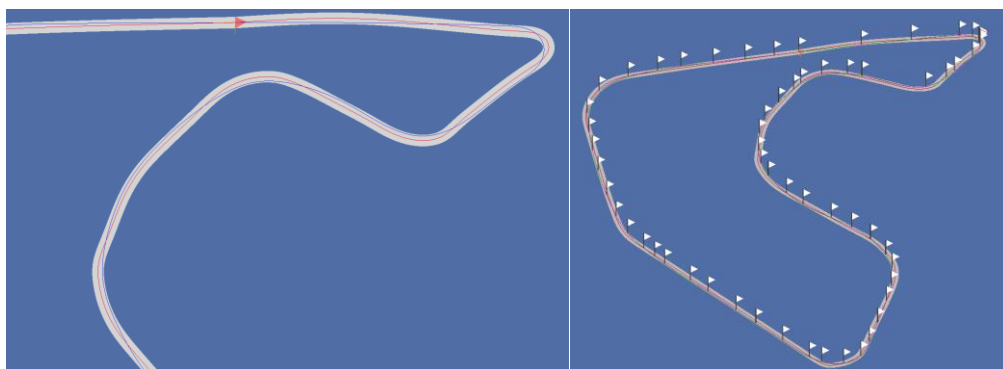


Figure 33 First Corner zoom and flag representation of racing line followed by the real driver

Figures 33 illustrate the reconstructed track, the applied racing line, and path optimisation options within VI-Road. Comparisons of alternative racing lines are also shown for specific corners, highlighting the differences between potential trajectories and the path ultimately selected for the correlation process.

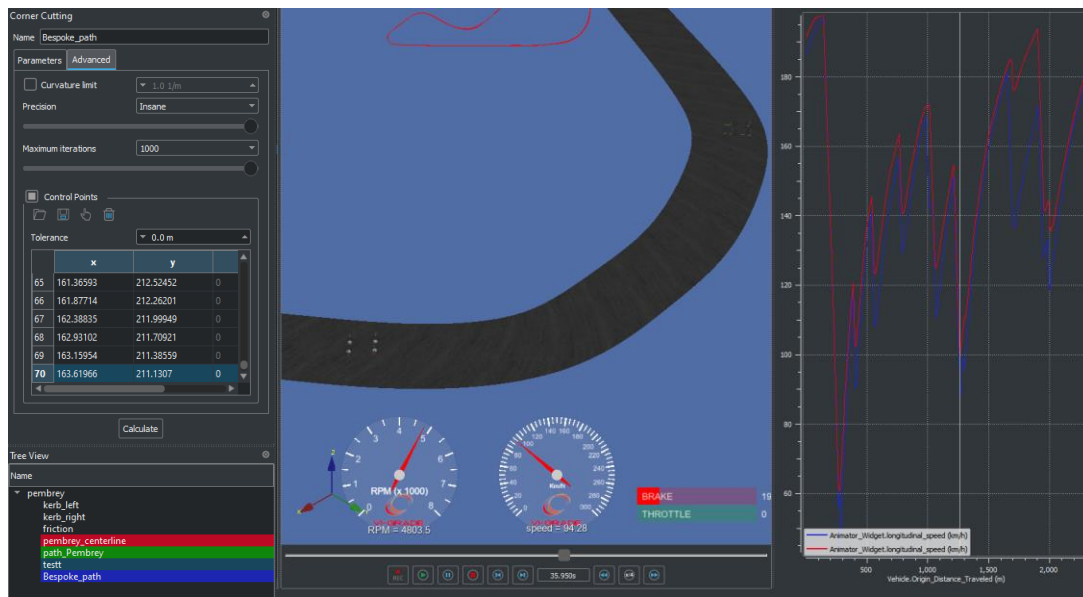


Figure 34 Optimised path speed graph vs a path selected by VI-Animation

With the circuit and path validated, the model correlation could proceed with confidence, ensuring that subsequent comparisons between simulated outputs and experimental data were based on realistic operating conditions.

4.2 Driver Model

The driver model for this study was based on the Max Performance event within VI-CarRealTime. This event is specifically designed to compute the dynamic limit velocity profile of a vehicle on a defined track, effectively determining the fastest possible lap time while respecting the car's physical limits in terms of tyre grip, aerodynamics, braking, and acceleration. It represents the “ultimate lap” mode within the software and is widely used to benchmark maximum performance and to evaluate how setup changes influence lap time.

The process begins with a static solver, VI-SpeedGen, which generates an estimated velocity profile along the chosen path using a simplified vehicle representation without suspension but including aerodynamic properties and ride-height maps. This profile is then tested in the full vehicle simulation by VI-Driver. If infeasible conditions are detected, such as excessive path length or zero velocity, the solver iteratively corrects the profile until a valid dynamic solution is obtained (VI-CarRealTime Documentation).

In the static phase, the model inherits the mass, tyre, and aerodynamic properties of the full car but omits the suspension system, which is accounted for indirectly through ride-height maps. These maps can be updated via coast down runs to capture aero-suspension interactions. Several parameters influence the event setup, including the path file (defining the racing line), initial velocity and gear conditions (disabled in flying lap mode), performance factors for longitudinal acceleration, braking, and lateral grip, as well as correction step size and maximum path distance to guide optimisation (VI-CarRealTime Documentation).

The Max Performance event is typically employed before detailed sensitivity or investigation studies to establish a reliable baseline performance envelope. Once this baseline is created, it can be combined with setup variations such as changes in springs, anti-roll bars, or aerodynamic maps to quantify lap time effects. Additionally, features like Performance Modifier Files (PMF) allow simulation of specific systems such as DRS, scaling drag and downforce over defined track sections.

4.3 Tyre selection

Tyres are the most critical component in determining the performance envelope of a race car, as they are the only point of contact between the vehicle and the track. Their ability to generate grip depends on several interacting factors, primarily vertical load, camber, and slip angle/ratio (Sankar Mohanta and Kumar, 2021).

First, tyres must be operated within the correct vertical load range. If the load is too low, contact patch pressure is insufficient, leading to reduced grip. If the load is too high, the rubber becomes overloaded, and grip response becomes non-linear. Since tyres are load-sensitive, maintaining consistent and balanced loading is essential. In simulation, preload values were set to match real-world conditions (calculated from actual forces on the race car): 1200 N at the front and 1750 N at the rear. Downforce contributions were also included to replicate the dynamic vertical load.

Second, camber is critical in maintaining maximum contact patch area during lateral force generation. Correct camber ensures that the outer shoulder of the tyre remains flat under load. This was validated through VI-TireLimits testing, where peak lateral force was identified across a camber sweep from -5° to $+1^\circ$. For the tyres used, the optimal camber corresponded to negative values in the front to maximise cornering grip.

Third, slip angle and slip ratio determine how efficiently the tyre generates lateral and longitudinal forces. At low slip values, grip is insufficient, while excessive slip leads to saturation and sliding. The balance between these effects was studied through both VI-TireLimits and VI-TestRig, where tyre performance envelopes were compared with driver model behaviour to determine how much potential grip was being exploited.

The car was fitted with Hoosier racing slicks, which are commonly used in Sports 2000 and club-level prototype racing. Typical compounds include the Hoosier R45B (medium-hard, suited for durability), R25B (softer, higher peak grip but lower longevity), and the more recent S1350. While manufacturer-specific tyre data was not available, parameters such as the scaling coefficients

f



```
[DIMENSION]
UNLOADED_RADIUS      = 0.26035          $Free tyre radius
WIDTH                = 0.1778          $Nominal section width of the tyre
ASPECT_RATIO         = 0.65            $Nominal aspect ratio
RIM_RADIUS           = 0.1651          $Nominal rim radius
RIM_WIDTH            = 0.1778          $Rim width

[SCALING_COEFFICIENTS]
LFZO                  = 1.0              $Scale factor of nominal (rated) load
LCX = 0.74            $Scale factor of Fx shape factor
LMUX = 1.176          $Scale factor of Fx peak friction coefficient
LEX = 1.24            $Scale factor of Fx curvature factor
LKX = 2.88            $Scale factor of Fx slip stiffness
LHX = 1.01            $Scale factor of Fx horizontal shift
LVX = 1.10            $Scale factor of Fx vertical shift
LGAX = 0.96           $Scale factor of camber for Fx
LCY = 0.88            $Scale factor of Fy shape factor 1.0
LMUY = 1.7052         $Scale factor of Fy peak friction coefficient
```

Figure 35 Front Tyre Characteristics used for the Vehicle Model

This intentional reduction in rear tyre grip was implemented to mitigate oversteer and induce mild understeer, with the purpose of improving stability. The adjustment also reduced yaw spikes caused by sudden rear end sliding, aligning the simulation more closely with the behaviour of amateur and semi/professional drivers, who typically lift earlier and apply smoother steering inputs.

```
[DIMENSION]
UNLOADED_RADIUS      = 0.26035          $Free tyre radius
WIDTH                = 0.1778          $Nominal section width of the tyre
ASPECT_RATIO         = 0.65            $Nominal aspect ratio
RIM_RADIUS           = 0.1651          $Nominal rim radius
RIM_WIDTH            = 0.1778          $Rim width

[SCALING_COEFFICIENTS]
LFZO                  = 1.0              $Scale factor of nominal (rated) load
LCX = 0.74            $Scale factor of Fx shape factor
LMUX = 1.116          $Scale factor of Fx peak friction coefficient
LEX = 1.24            $Scale factor of Fx curvature factor
LKX = 2.88            $Scale factor of Fx slip stiffness
LHX = 1.01            $Scale factor of Fx horizontal shift
LVX = 1.10            $Scale factor of Fx vertical shift
LGAX = 0.96           $Scale factor of camber for Fx
LCY = 0.88            $Scale factor of Fy shape factor 1.0
LMUY = 1.6182         $Scale factor of Fy peak friction coefficient
```

Figure 36 Rear Tyres Characteristics used for the Vehicle Model

Figures 35 and 36 show examples of the tyre parameter files. The correlation ensured that tyre forces in simulation reflected real-world behaviour, providing a reliable foundation for setup sensitivity studies and optimisation.

4.3.1 Tyre Limit

From a perspective of evaluation, the next step would be to evaluate the operating limits of each tyre using a gg diagram or Traction Circle. This analysis allows verification and correlation of how much load was applied to each tyre under real running conditions and whether the model approached the theoretical grip encloser. To know this the aerodynamic force was measured through the pushrod sensors on the suspension giving the following peak values:

- Front Downforce: 2194.8 *N*
- Rear Downforce: 1726.7 *N*



Figure 37 Real Data from the car shown in MOTEC i2 Pro (Fastest Lap) - Aerodynamic Workbook

Mechanical grip was defined from static preload of 1200 *N* at the front and 1750 *N* at the rear. Combining mechanical and aero contributions it results on the following per-tyre vertical load:

$$F_z(\text{front}) = 1200 + \frac{2194.8}{2} = 2297.4 \text{ N}$$

$$F_z(\text{rear}) = 1750 + \frac{1726.7}{2} = 2613.25 \text{ N}$$

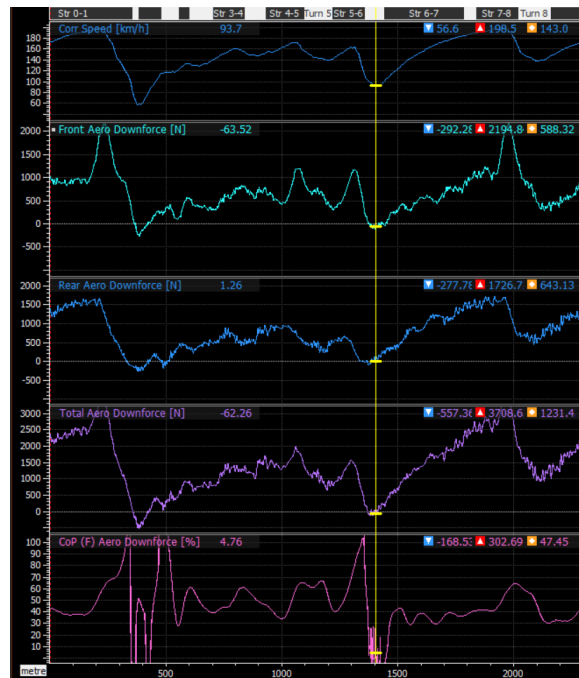


Figure 38 Real Data from the car shown in MOTEC i2 Pro (Fastest Lap) - Aerodynamic Workbook

Therefore, each front tyre operated between 1200 N (minimum) and 2297.4 N (maximum), while the rear tyre operated between 1750 N (minimum) and 2613.25 N (maximum). The load ranges were used to configure VI-TireLimit predictor, enabling evaluation of individual tyre utilisation during a hot lap. Four separate plots generated, one for each tyre (FL, FR, RL, RR). The measured data from the fastest lap was overlaid on these operating clusters to highlight the extent to which each tyre approached its theoretical grip limit.

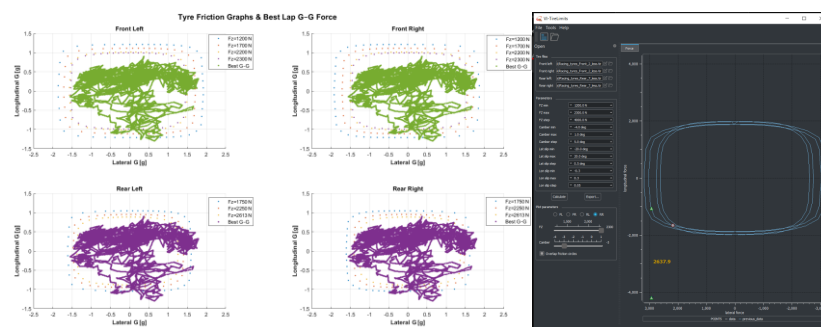


Figure 39 Lateral and Longitudinal Force data points collected from the fastest lap on Pembrey Track and VI-TyreLimits results for each tyre

Figures 40 show the resulting tyre limit plots, demonstrating how the tyres were loaded throughout the lap and identifying where unused performance potential remained.

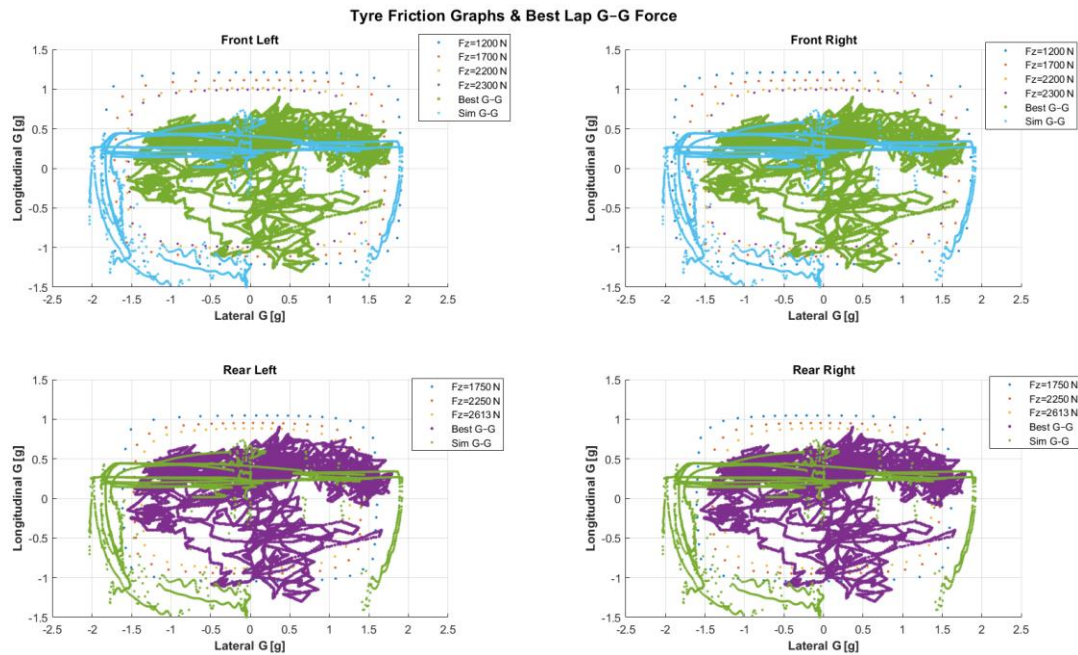


Figure 40 Lat and Long G Forces comparison between the best lap on the real car vs the vehicle simulation results from best lap

The front tyres remain within the predicted envelope, operating close to their theoretical boundaries under combined braking and cornering. This indicates good correlation and efficient use of front grip. By contrast, the rear tyres frequently exceed the tyre limit cluster in high-speed sections. This means the simulated rear axle is operating beyond the available grip envelope, which manifests as slight oversteer when cornering at high speeds. This behaviour is consistent with load transfer and tyre sensitivity: as vertical load increases, the rear tyres reach saturation earlier, causing them to slide beyond their predicted traction limits. While this generates some oversteer, it remains recoverable and contributes to a realistic replication of vehicle dynamics. The exceedance highlights that the rear tyres are the limiting factor in high-speed stability, and any optimisation must focus on balancing grip distribution through spring, damper, anti-roll bar, or aerodynamic adjustments.



Figure 43 Vented Performance Brake Disc

The brake pads used are BP-40 compound, supplied by Wilwood. This pad compound is widely used in club racing due to its balance between friction stability, fade resistance, and wear. Manufacturer data shows a friction coefficient curve ranging from $\mu \approx 0.40$ at 150°F to a peak of $\mu \approx 0.525$ at $750 - 900^\circ\text{F}$, before slightly decreasing to $\mu \approx 0.51$ at 1200°F . This operating window ensures consistent braking performance across the range of racing conditions. The compound is rated highly for modulation, release, and fade resistance, making it suitable for repeated high-load applications (Figure 44).

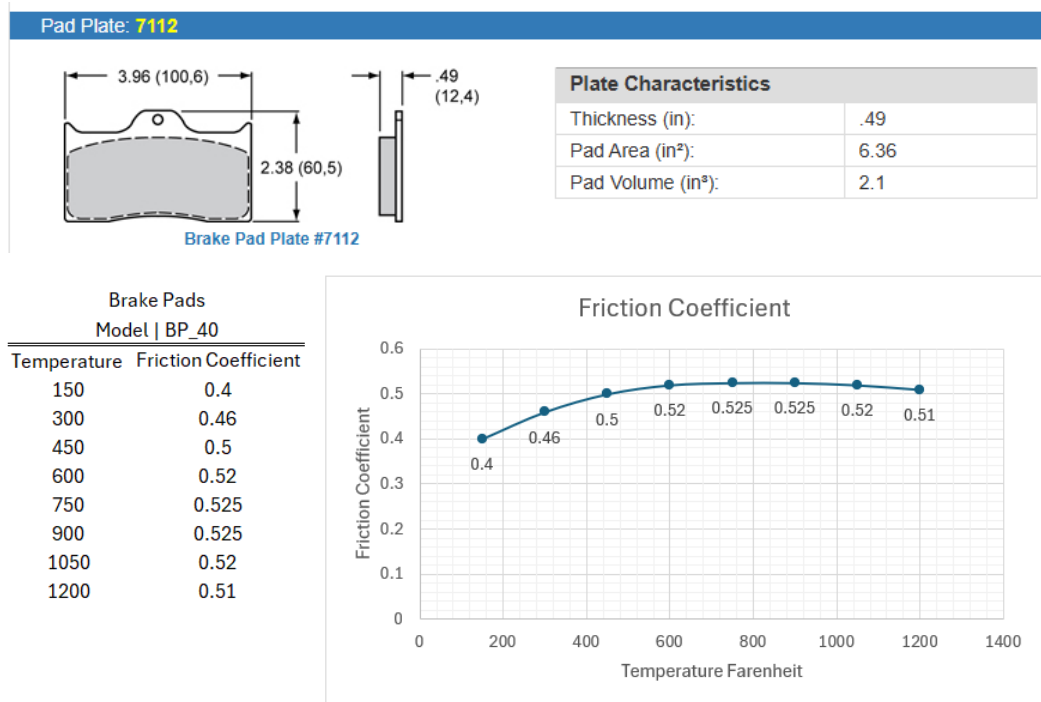


Figure 44 BP-40 Coefficient of Friction vs Temperature graph showing the increase in friction with temperature and the brake pads measurements (Wilwood.com, 2018)

4.5 Powertrain

The powertrain model was configured as a conventional internal combustion engine layout with a clutch and manual gearbox, driving the rear wheels (RWD). This reflects the configuration of the MCR race car, which is powered by a 2.1-litre Ford-tuned RS2000 engine prepared by D'Amico Engines Ltd. (2023).

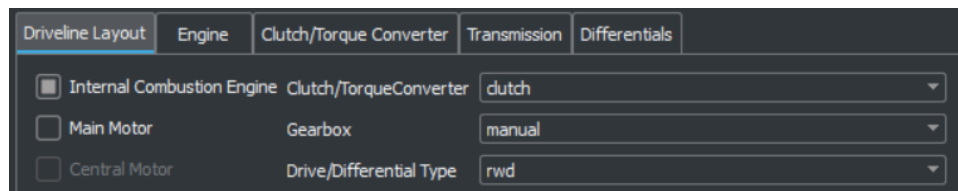


Figure 45 Driveline window for general subsystem selection

In VI-CarRealTime, the mechanical properties of the engine were set according to manufacturer specifications, including a rev limit of 7450 rpm. Although the original engine specification allowed up to 7500 rpm, initial simulations exceeded this value (up to 7550 rpm). To ensure realistic behaviour and avoid surpassing the physical limits of the real engine, the rev limiter was reduced slightly, with the simulation reaching a maximum of 7460–7470 rpm.



Figure 46 Set up configuration for the Rear of the Car

Traction control was disabled in the driver model to replicate real racing conditions, where drivers manage slip and lateral velocity directly without electronic aids. For gear shifting, the transmission was configured to operate in a constant mode, requiring sequential gear changes. Upshifts occur consistently at 7250 rpm, while downshifts are triggered at 4500 rpm, mirroring the operational behaviour of the race car.

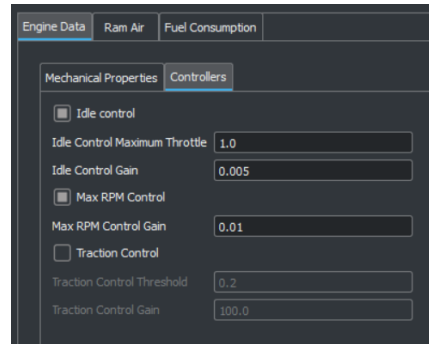


Figure 47 Engine controllers

For the transmission the gear shift method is constant so it need to goes one speed per change like standard race car, the up shift is 7250 rpm constant and the downshift is 4500 constant so every time the car reaches those rpm it either downshift or upshift.

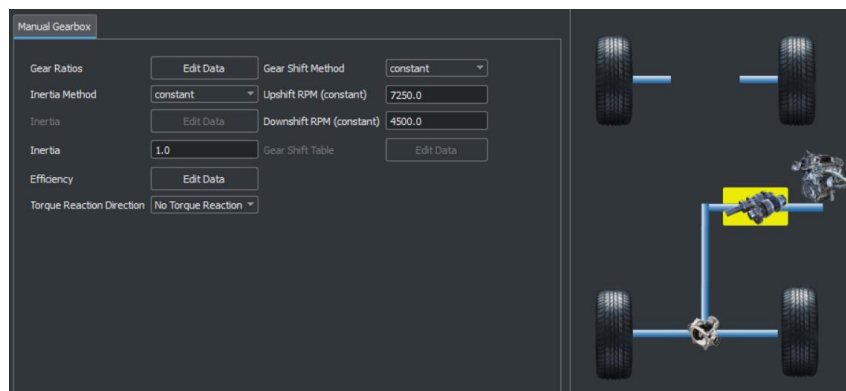


Figure 48 Manual Gearbox shown in the powertrain subsystem

The manual gearbox was parameterised with ratios corresponding to those used during the second session at Pembrey:

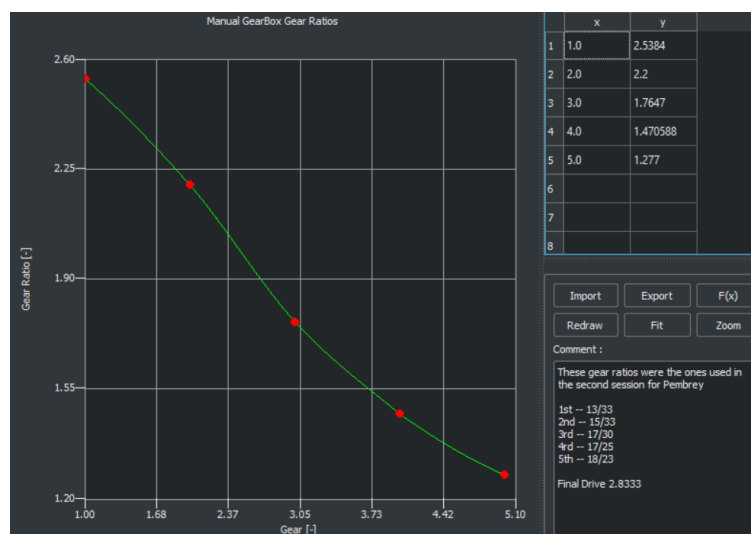


Figure 49 Manual Gear Box Ratios, going from 1st up to 5th speed

The final drive goes into the differential section of the car which by just checking the components of the car the final drive is 2.833.

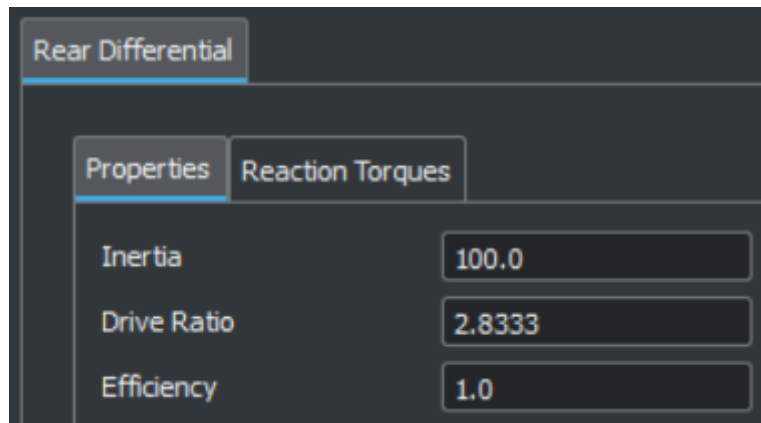


Figure 50 Final Drive value used for the rear differential

This configuration ensured that the simulated vehicle replicated the actual power delivery, gearing, and drivetrain behaviour, providing a realistic foundation for both correlation and performance optimisation studies.

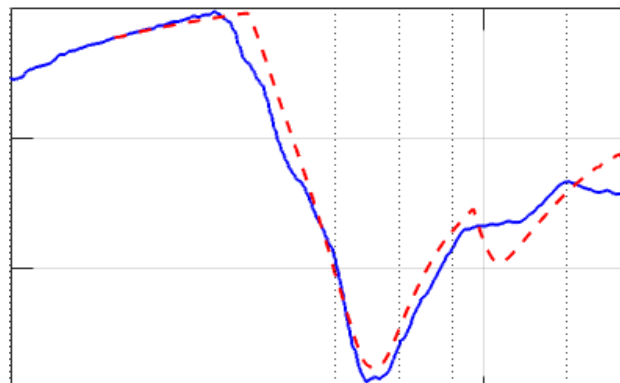


Figure 51 Speed vs Distance Graph of Real Data against the vehicle simulation showing the same rate of increase in speed as it accelerates

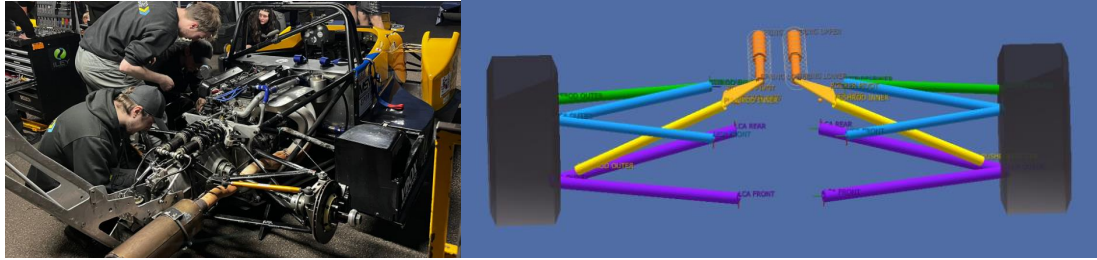


Figure 54 Rear Suspension Setup - Double Wishbone Pushrod system

Springs

For the MCR race car, the baseline spring rates were measured at 750 lbs/in (131.35 N/mm) for the front suspension and 850 lbs/in (148.86 N/mm) for the rear. This arrangement contrasts with the default VI-CRT race car model, which assumes the opposite stiffness distribution, with the front springs stiffer than the rear. The choice of a softer front and stiffer rear in the MCR configuration is deliberate and closely linked to the car's aerodynamic package.

Table 2 Race Car Model Spring Rate

Race Car Model Spring Rates		
Units	N/mm	lbs/in
Front	202.3	1155.163
Rear	100.2	572.1567

The softer front springs provide greater compliance, helping the front tyres maintain contact with the track surface and ensuring that the aerodynamic platform remains within its operating window during braking and corner entry. Conversely, the stiffer rear springs stabilise the rear axle at speed, limiting rear ride height variation and preserving the efficiency of the rear diffuser. Together, these settings maintain the car within the tested aerodynamic operating range of 55 mm front ride height and 65 mm rear ride height, values determined through the attitude test with the latest geometry.

Table 3 MCR Race Car Model Base Line Spring Rates

Spring Rate [lbs/in]	Original Values from the test	Spring Rate [N/mm]	Original Values from the test
Front Suspension	750	Front Suspension	131.3451263
Rear Suspension	850	Rear Suspension	148.8578098

The springs were modelled in VI-CRT with a constant compression ratio of 1.0, corresponding to linear behaviour. As shown in Figures 55, the force–deflection characteristics form straight-line relationships, consistent with the assumption of linear springs. While non-linear or progressive springs can be used to tune compliance across different ranges of travel, the decision to use

linear modelling reflects the goal of simplifying correlation between the physical and simulated car.

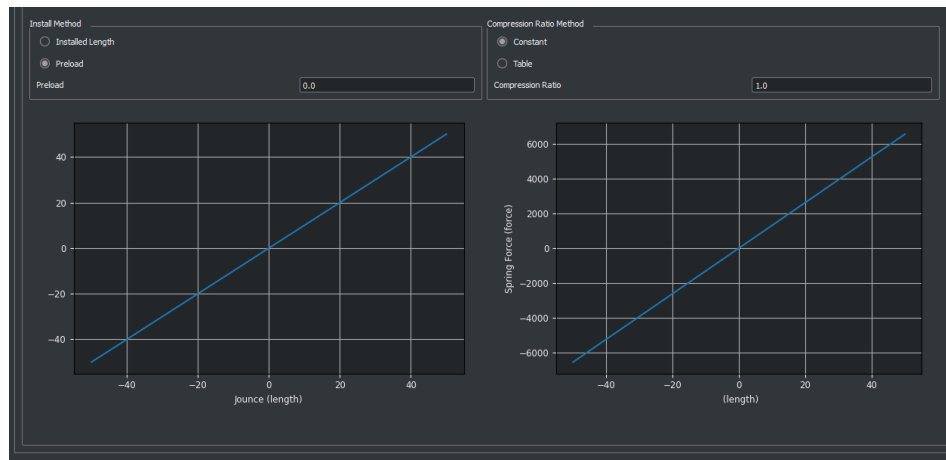


Figure 55 Install Method and Compression Ratio, Front - MCR Race Car

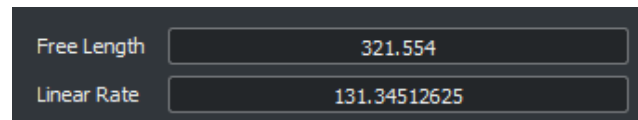


Figure 56 Front Spring Rate - MCR Race Car

In the real car, the springs were installed without preload, meaning no initial compression was applied as part of the installation method. This ensures that the suspension response is dictated solely by dynamic loads rather than any preset tension. The compression ratio was set to a constant value of 1.0, confirming that the springs were treated as linear elements. This assumption simplifies modelling and aligns with the measured characteristics of the hardware, where spring force increases proportionally with deflection.

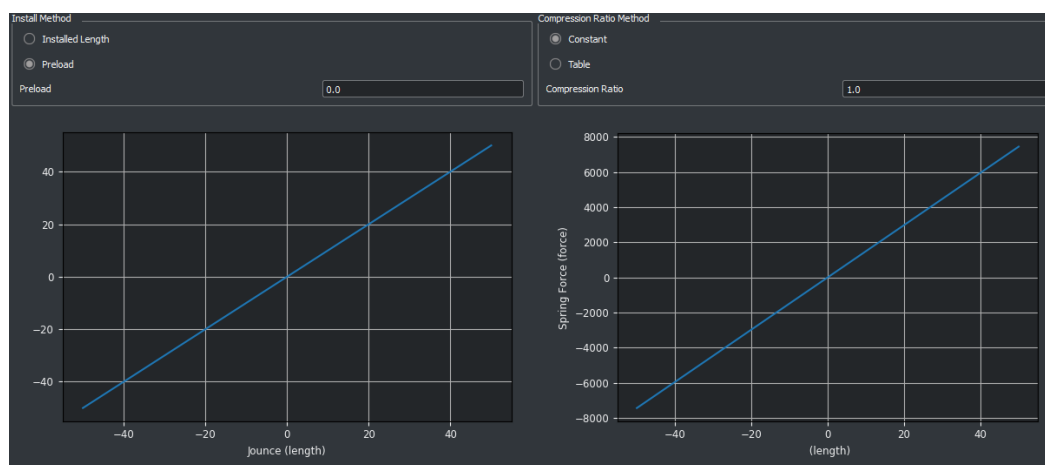


Figure 57 Install Method and Compression Ratio, Rear - MCR Race Car

Both rear springs were specified with a linear rate of 850 lbs/in (148.85 N/mm), providing consistent stiffness throughout their range of travel and ensuring predictable rear axle behaviour under load.

Free Length	311.16
Linear Rate	148.85780975

Figure 58 Rear Spring Rate - MCR Race Car

Dampers

Moving into the damping of the model. The standard vehicle model uses also a simplified version of the model using a non linear damping coefficient (for the VI-CRT Race Car) having a response in low and high speed scenarios. The response of the front damper is shown in 59 this behaviour is shows 2 distinctive cases in which at low speeds bumps the car a high degree of damping and as it goes into higher speed bumps The front damper deflection length is linear as the spring, although the values are different.

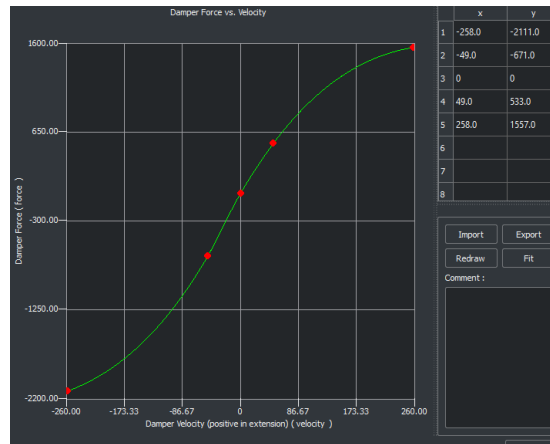


Figure 59 VI-CRT Race Car Model Front Damper Characteristics

The rear dampers in the VI-CRT race car model also exhibit a non-linear response in both bump and rebound, as shown in Figure 60. Compared with the front axle, the damping coefficients at the rear are significantly lower. From a theoretical standpoint, this distribution creates a chassis balance where the front is more heavily damped than the rear.

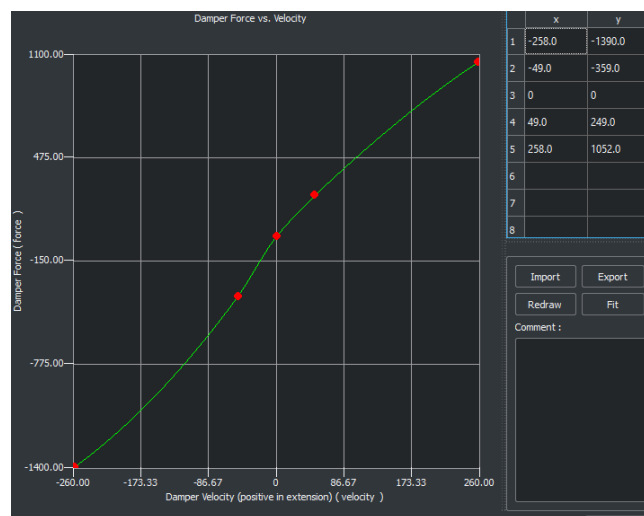


Figure 60 VI-CRT Race Car Rear Damper Characteristics

The actual dampers fitted to the MCR race car were bench-tested to determine their compression (bound) and rebound characteristics. The results, shown in Figure XX, reveal that the front (blue) and rear (green) dampers exhibit broadly similar behaviour in compression, with only minor differences in their force–velocity response. In rebound, however, the rear dampers display a slightly softer characteristic compared with the front, allowing for greater extension compliance.

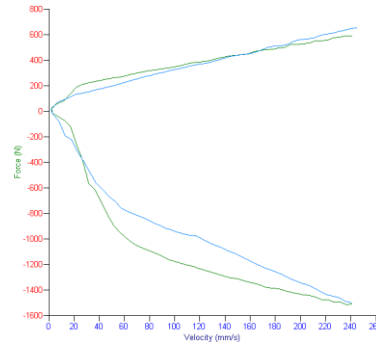


Figure 61 Damper Test Characteristic Force (N) / Velocity (mm/s) - Blue Front - Green Rear

The measured force–velocity curves provided a reliable basis for modelling, as they capture the non-linear behaviour of the dampers across the tested velocity range. These curves were exported into a .CSV file format and directly imported into VI-CRT, replacing the default simplified coefficients. This ensured that the simulation reflected the actual damping hardware installed on the car.

Table 4 Front and Rear Damper Characteristics under test

Front Dampers				Rear Dampers			
Front Bump		Front Rebound		Rear Bump		Rear Rebound	
Velocity (mm/s)	Force (N)	Velocity (mm/s)	Force (N)	Velocity (mm/s)	Force (N)	Velocity (mm/s)	Force (N)
2.2	27.4	2.2	27.4	1.9	14.9	1.9	14.9
7	63.1	-2.9	-17.9	4.5	42.9	-2.5	-2.9
11.9	82.1	-9.2	-57.1	7	71.4	-3.1	-21.3
16.8	132.1	-12.1	-75	13.2	95.2	-5.5	-45.7
21.8	185.7	-17.4	-128.6	18.1	109.5	-8	-72.1
26.6	210.7	-21.3	-246.4	21.6	128.6	-12.9	-190.3
34.3	231	-26.3	-392.9	28.5	147.6	-15.5	-206.2
39.7	241.7	-31.5	-564.3	35.5	161.9	-17.9	-222
47.4	257.1	-36.3	-614.3	50.7	195.2	-20.4	-271.6
57.6	266.7	-41.6	-710.7	73	261.9	-22.8	-318.3
84.2	323.8	-48.4	-850	92.5	304.8	-28	-396.4
102.2	352.4	-56.2	-942.9	110.9	347.6	-37.8	-562.5
114.3	381	-64.4	-1017.9	121	371.4	-45.4	-644.6
123.9	385.7	-71.7	-1064.3	126.2	371.4	-48	-671.4
144.8	433.3	-81.3	-1100	138.3	409.5	-52.7	-707.1
157.4	447.6	-87.2	-1125	146.3	428.6	-57.4	-751.8
169.9	476.2	-94.2	-1160.7	154.3	442.9	-63.3	-783.9
186.9	495.2	-113.8	-1214.3	162	457.1	-72.3	-823.8
192.2	519	-132.8	-1267.9	169.3	476.2	-78.7	-852.4
200.9	523.8	-143.3	-1303.6	175.5	504.8	-82.5	-876.2
207.2	528.6	-148.6	-1310.7	181.8	509.5	-98.5	-938.1
210.6	547.7	-161.2	-1342.9	189.1	519	-109.7	-971.4
221.3	552.4	-166.6	-1353.6	195.7	552.4	-117	-981
227.6	576.2	-173.3	-1382.1	204	561.9	-126.3	-1023.8
236.1	595.4	-185	-1396.4	211.3	576.2	-148.7	-1123.8
241.1	592.3	-191.7	-1421.4	218.3	600	-168.9	-1223.8
		-206.3	-1439.3	225.2	609.5	-181.6	-1266.7
		-212.6	-1464.3	235.7	638.1	-186.1	-1281
		-216.4	-1478.6	244.7	647.6	-190.3	-1309.5
		-221.9	-1472.6			-208.8	-1376.2
		-226.9	-1495.8			-220	-1438.1
		-231.6	-1494.9			-230.3	-1466.7
		-236.5	-1511			-235.9	-1489.2
		-241.2	-1506.7			-240.9	-1499.8

The data input into the VI-CRT model is shown in Figure 62, which illustrates the clear division between slow (1), medium (2), and high-speed (3) rebound regions. The front damper exhibits a higher damping gradient in the medium-speed rebound range compared with the rear damper.

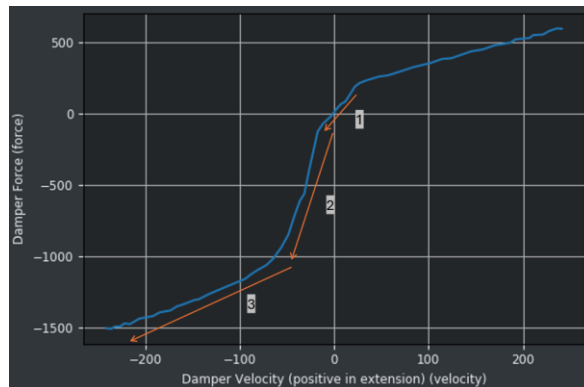


Figure 62 Front Damper Velocity / Force graph

In summary, the front damper response in rebound is highly non-linear, while in bump it is mostly linear. At low bump velocities, the damper shows a steep gradient, providing strong damping, but this effect eases as velocity increases, reducing the rate of force rise. The maximum compression force recorded was 592 N at 241 mm/s, compared with an absolute rebound peak of 1506.7 N at the same velocity.

The rear damper follows a similar trend in the bump (compression of the damper) section, figure 63 shows a the graphical representation of the behaviour of the rear dampers.

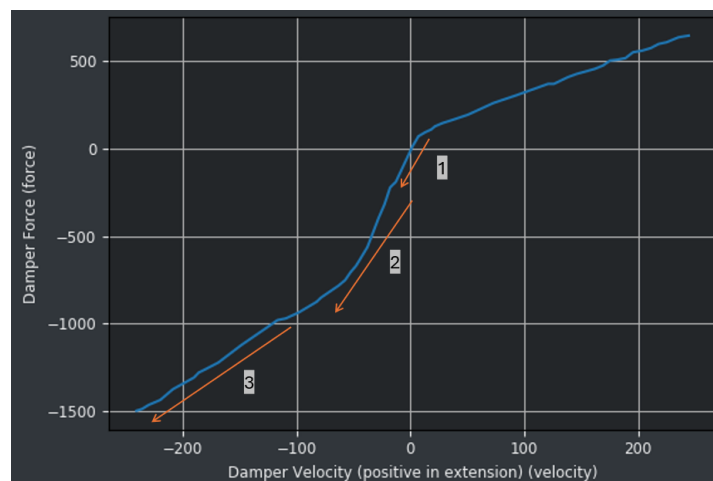


Figure 63 Rear Damper Velocity / Force graph

In this case, the rebound gradient is lower at slow and medium rebound velocities compared with the front dampers. However, as the velocity increases, the rear damper exhibits a steeper

gradient, generating higher rebound forces at high speeds. An interesting observation is that both the front and rear dampers converge to similar force levels at their maximum bump and rebound velocities. Table XX summarises the maximum forces recorded at the highest tested speeds.

Table 5 Front and Rear Dampers Maximum Values at Bump and Rebound Scenarios from Testing

Front and Rear Dampers Maximum Values at Bump and Rebound				
MCR - Race Car				
Side of the car	Bump		Rebound	
	Velocity (mm/s)	Force (N)	Velocity (mm/s)	Force (N)
Front	241.1	592.3	-241.2	-1506.7
Rear	244.7	647.6	-240.9	-1499.8

Table 5 summarises the maximum bump and rebound forces recorded for the front and rear dampers of the MCR race car. In bump, both axles exhibit a similar gradient, with damping force increasing proportionally with velocity. In rebound, however, the behaviour diverges: while the front dampers generate slightly higher peak rebound forces, the rear dampers display a different gradient. This asymmetry could influence the overall balance of the car, particularly in transient conditions, if not carefully managed.

When compared with a control model in VI-CRT—where identical parameters were retained except for the dampers, which were taken from a reference vehicle—the differences become more evident. The reference dampers displayed a more linear response and reached higher maximum forces at comparable velocities. In contrast, the tested MCR dampers exhibited non-linear behaviour with slightly lower peak rebound forces. This distinction highlights the importance of incorporating the measured damper data into the simulation to ensure realistic representation of chassis dynamics.

Table 6 presents the maximum bump and rebound values for the control race car dampers. Compared with the measured MCR dampers, these units reach significantly higher damping forces in both bump and rebound, by almost 2000 N, at similar operating velocities. The control dampers also show a more linear force–velocity relationship, producing greater resistance across the full range of speeds.

Table 6 Front and Rear Damper Maximum Values at Bump and Rebound Control Race Car Model

Front and Rear Dampers Maximum Values at Bump and Rebound VI-
CRT Race Car Model

Side of the car	Bump		Rebound	
	Velocity (mm/s)	Force (N)	Velocity (mm/s)	Force (N)
Front	258	1557	-258	-2111
Rear	258	1052	-258	-1390

This distinction is critical for model validation. While the control dampers provide higher absolute forces, they do not reflect the measured data on the tested dampers. For this reason, both sets of dampers will be analysed in the validation and sensitivity study, where their effect on vehicle behaviour under track conditions can be compared.

4.7 Steering

The steering system was characterised experimentally by measuring steering-wheel angle (SWA) against the road-wheel angle on the shop floor. SWA was acquired from the vehicle electronics and monitored in real time using a MoTeC data interface over Ethernet; the front road-wheel angle was measured incrementally on a compass at symmetric positions about centre. The resulting SWA–wheel-angle pairs were recorded from -180° to $+180^\circ$ in 45° steps, with finer sampling near the on-centre region. Zeros were aligned with the wheels straight-ahead and the dataset was symmetrised to remove small offsets and hysteresis.

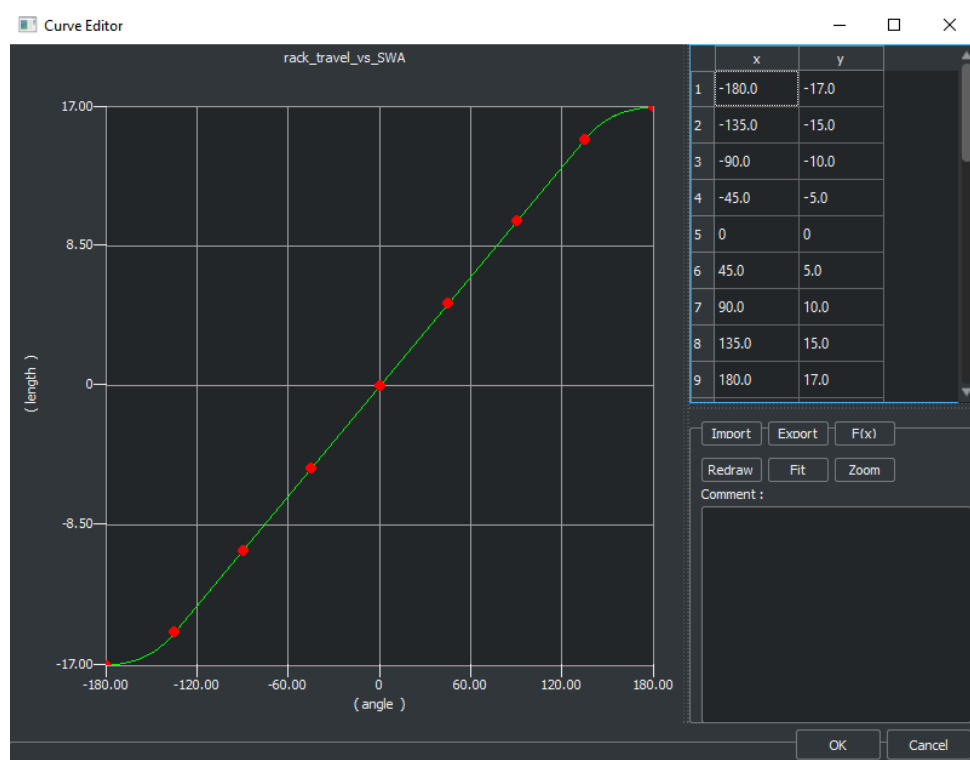


Figure 64 Rack Travel vs SWA Graph from the Steering Subsystem of the Vehicle model

Because VI-CarRealTime drives the rack kinematics from handwheel angle, the measured points were converted into an effective rack travel vs SWA curve. Around centre the relation is close to linear (constant steering ratio), while the outer points exhibit mild non-linearity from steering geometry and internal stops. The final mapping was created by fitting a monotone cubic spline to the processed data and enforcing odd symmetry about SWA = 0. The curve used in the model is shown in Figure 64 [rack_travel_vs_SWA] and spans approximately ± 17 mm of rack displacement over $\pm 180^\circ$ of handwheel rotation.

4.8 Aerodynamics

The aerodynamic model was developed using the actual vehicle geometry, with the objective of performing an attitude study across a range of front and rear ride heights. The results of this study were used to generate baseline aerodynamic maps for the simulation model. These maps provided lift, drag, and aerodynamic balance values as functions of ride height, capturing the influence of rake on aerodynamic performance.

Table 7 Aerodynamic Coefficient Values

Coefficient [N]	Aerodynamic Values				
	Original Values from the test	+5%	+10%	+15%	+20%
Front Downforce	630	661.5	693	724.5	756
Rear Downforce	1330	1396.5	1463	1529.5	1596
Drag	604	-	543.6	-	-

The baseline aerodynamic values, measured at a reference speed of 126 km/h, corresponded to a front ride height of 55 mm and a rear ride height of 65 mm, giving a rake of 10 mm. This configuration was chosen as it matched the ride height settings used on the physical car during testing. The attitude study also explored a wider range of ride heights (e.g., 10–30 mm, 30–50 mm, 35–65 mm, 20–50 mm, 25–60 mm, and 20–30 mm), which demonstrated how variations in rake affected downforce, drag, and aerodynamic efficiency.

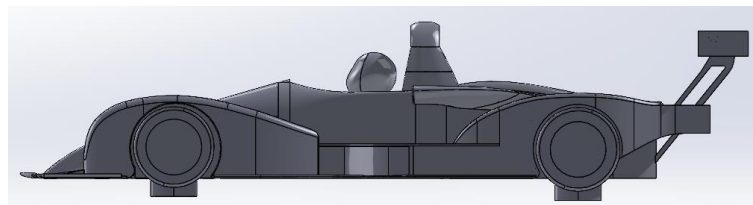


Figure 65 Side view of the Geometry of the Race Car

For correlation, the aerodynamic values implemented in the model were later increased by up to 20% at both the front and rear axles to better reproduce the grip levels observed in specific corners. This adjustment ensured that the simulated aerodynamic forces aligned with the real-world performance envelope of the car. Table 7 summarises the baseline coefficients and the values obtained with incremental increases of +5%, +10%, +15%, and +20%.

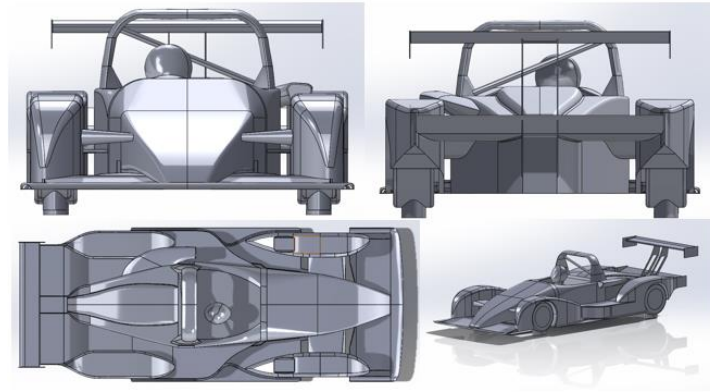


Figure 66 Front, Back, Top and Isometric view of the Race Car

Looking ahead, the aerodynamic maps from the attitude study represent a valuable resource for future optimisation work. In particular, they could be integrated into the Box–Behnken Design methodology, with variables such as drag, downforce, and efficiency forming part of the multi-variable optimisation process.

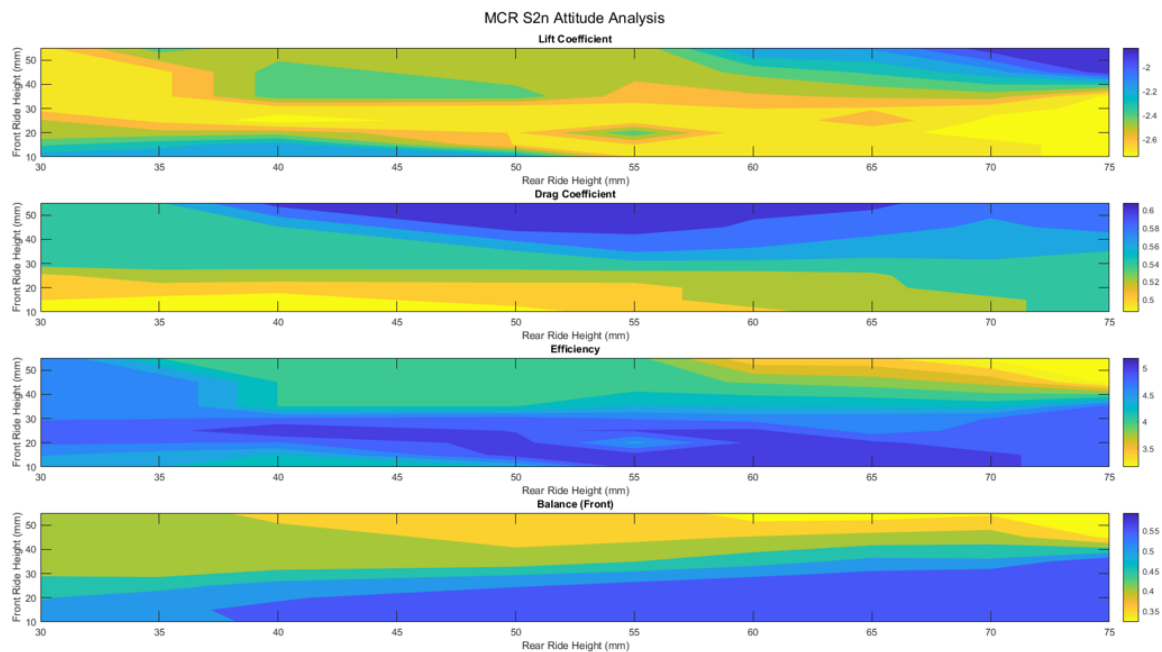


Figure 67 MCR S2n Attitude Analysis - Ride Height Analysis

5. Sensitivity Test – Validation

5.1 Tyre Model and Tyre Testing

The tyre model in VI-CarRealTime is based on the Pacejka “Magic Formula,” which is widely used in motorsport and automotive simulation to describe tyre behaviour. This semi-empirical model expresses longitudinal and lateral forces as non-linear functions of slip ratio, slip angle, vertical load, and camber, with additional factors to capture combined slip conditions. The VI-Grade implementation uses scaling coefficients (e.g., LMU_x and LMU_y) to adapt the model to measured data, enabling correlation with real tyre performance.

The expected outcome is that tyres will generate increased force with greater vertical loading, but in a sub-linear manner, reflecting saturation effects. Behavioural changes are also observed with temperature: within the optimal window, grip increases, but once exceeded, performance deteriorates due to compound degradation.

Preloads were applied to replicate static vertical forces: 1200 N at the front axle and 1750 N at the rear axle. Scaling factors were set within realistic ranges: LMU_x between 1.5–1.8 (nominally 1.6) for braking and traction, and LMU_y between 1.7–2.0 (nominally 1.85–1.9) for lateral grip. These values reflect expected friction levels for racing slicks.

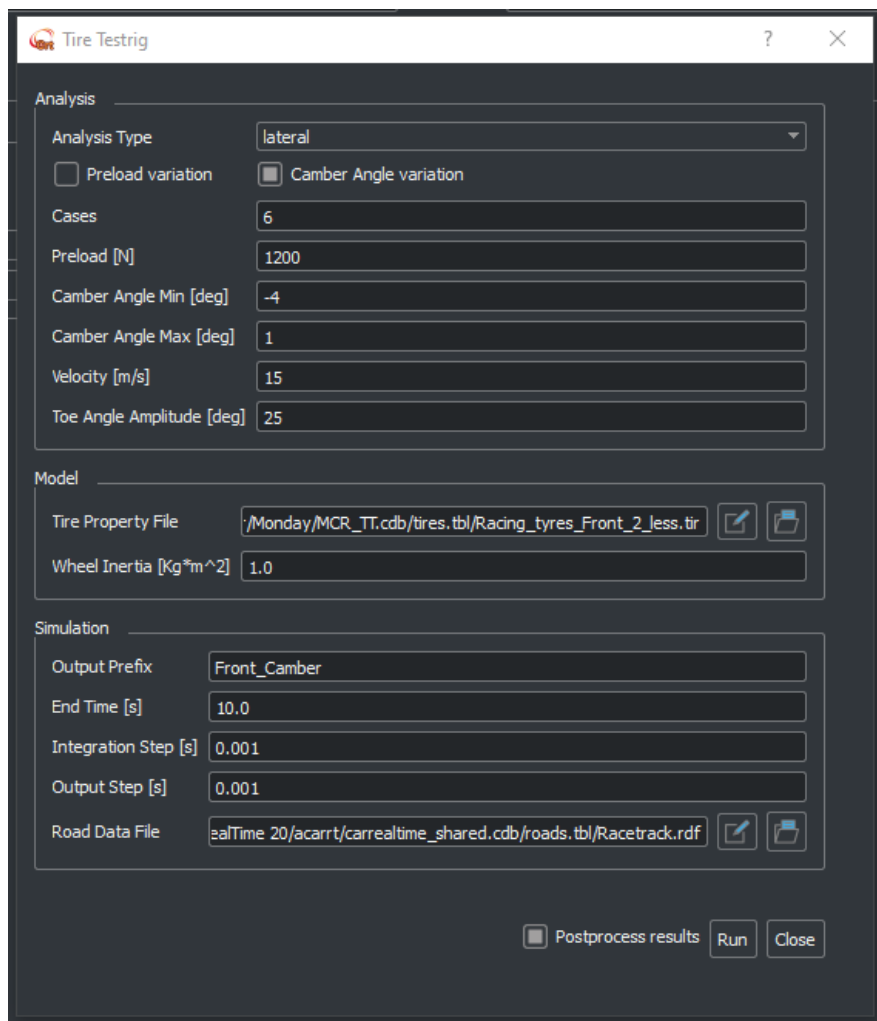
The operational tyre window was defined by an optimal temperature range of 75–90 °C and pressures of 14–15 psi cold, typically rising to 20–22 psi hot after several laps. Within this range, the tyres achieve:

5.1.1 Camber Validation

To validate the tyre model, a series of camber sensitivity studies were carried out on both the front and rear tyres. Although the compound used was the same across axles, the rear tyres feature a wider geometry, providing additional stability and mechanical grip compared with the smaller front tyres. These differences required separate validation for each axle.

The first study focused on the front tyres, where a camber sweep from -3° (negative/outward camber) to $+1^\circ$ (positive/inward camber) was performed in 1° increments. The preload was set at 1200 N, corresponding to the static vertical load without additional aerodynamic contributions. A test velocity of 15 m/s was used to simulate a steady-state cornering condition.

Lateral Study for the Front Tyres



The screenshot shows the 'Tire Testrig' software interface with the following settings:

- Analysis**
 - Analysis Type: lateral
 - ☐ Preload variation
 - ☒ Camber Angle variation
 - Cases: 6
 - Preload [N]: 1200
 - Camber Angle Min [deg]: -4
 - Camber Angle Max [deg]: 1
 - Velocity [m/s]: 15
 - Toe Angle Amplitude [deg]: 25
- Model**
 - Tire Property File: /Monday/MCR_TT.cdb/tires.tbl/Racing_tyres_Front_2_less.tir
 - Wheel Inertia [Kg*m^2]: 1.0
- Simulation**
 - Output Prefix: Front_Camber
 - End Time [s]: 10.0
 - Integration Step [s]: 0.001
 - Output Step [s]: 0.001
 - Road Data File: ealTime 20/acarrt/carrealtime_shared.cdb/roads.tbl/Racetrack.rdf

At the bottom, there is a checkbox for 'Postprocess results' and buttons for 'Run' and 'Close'.

Figure 68 Lateral Study for the Front Tyres

The results, shown in Figures 69, demonstrate the effect of camber variation on lateral force generation. Increasing negative camber improved the maximum lateral force, with peak values rising from approximately 2202 N at +1° to 2273 N at -4°.

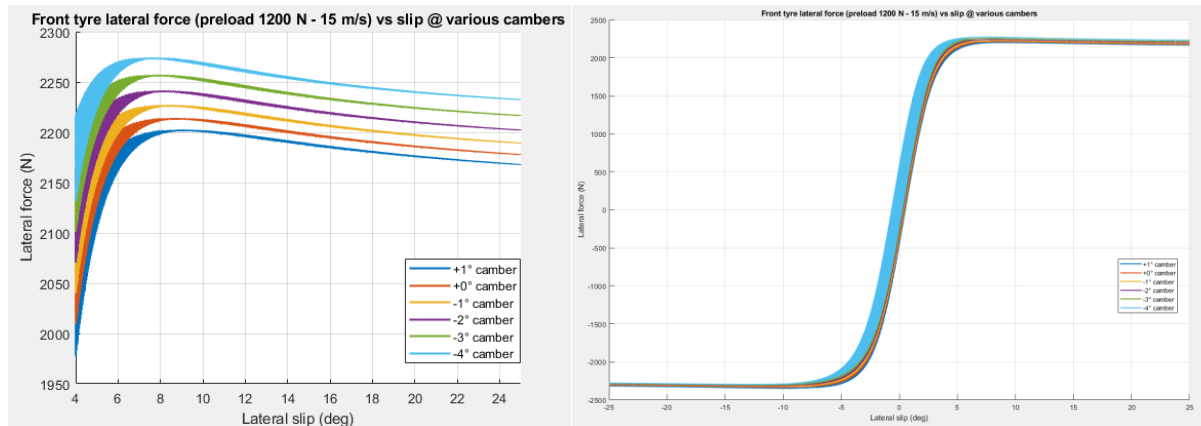


Figure 69 Front Tyre Lateral Force (preload 1200 N) vs slip at various camber angles - Front

This improvement reflects the enlarged contact patch and more uniform pressure distribution under load. At the same time, the slip angle at which peak force occurred decreased progressively, from around 9.5° at +1° to 8.0° at -4°, indicating greater efficiency in generating lateral grip.

Table 8 Camber Sweep comparison Max Lateral Force N and Slip Max Deg - Front

Tyre	Camber_deg	MaxLatForce_N	SlipAtMaxForce_deg
"Front tyre"	1	2202.1	9.4945
"Front tyre"	0	2213.6	9.1447
"Front tyre"	-1	2226.6	8.8222
"Front tyre"	-2	2241	8.5571
"Front tyre"	-3	2256.7	8.2909
"Front tyre"	-4	2273.8	8.0533

Self-aligning torque was also analysed to assess the steering feedback implications. As shown in Figures 70, maximum aligning torque increased with greater negative camber, rising from 42.9 Nm at +1° to 48.1 Nm at -4°. While this confirms enhanced grip, it also implies increased steering effort, which may contribute to higher driver fatigue over a stint.

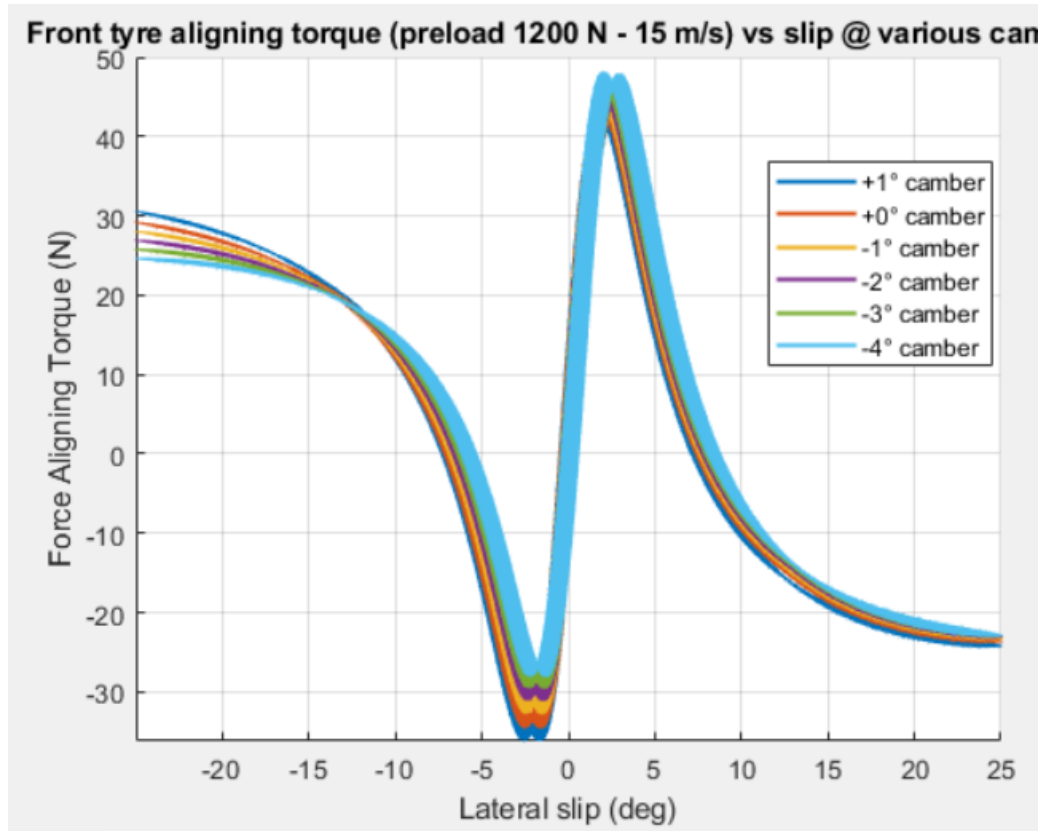


Figure 70 Self aligning Torque force (preload 1200 N) vs slip at various camber angles - Front

Table 9 Self Aligning Torque Values for the Camber Sweep Study - Front

Tyre	Camber_deg	Max Self Aligning Torque Nm	SlipAtMaxForce_deg
"Front tyre"	1	42.944	1.7578
"Front tyre"	0	43.878	1.8205
"Front tyre"	-1	44.687	1.8832
"Front tyre"	-2	45.665	1.9145
"Front tyre"	-3	46.815	1.9771
"Front tyre"	-4	48.136	2.0398

Lateral Study for the Rear Tyres

The same camber sensitivity study was performed on the rear tyres, with a preload of 1750 N and camber sweep from -4° to $+1^\circ$ in 1° increments at a velocity of 15 m/s. Compared with the front tyres, the rear tyres generated higher absolute lateral forces, consistent with their wider geometry and higher static loading.

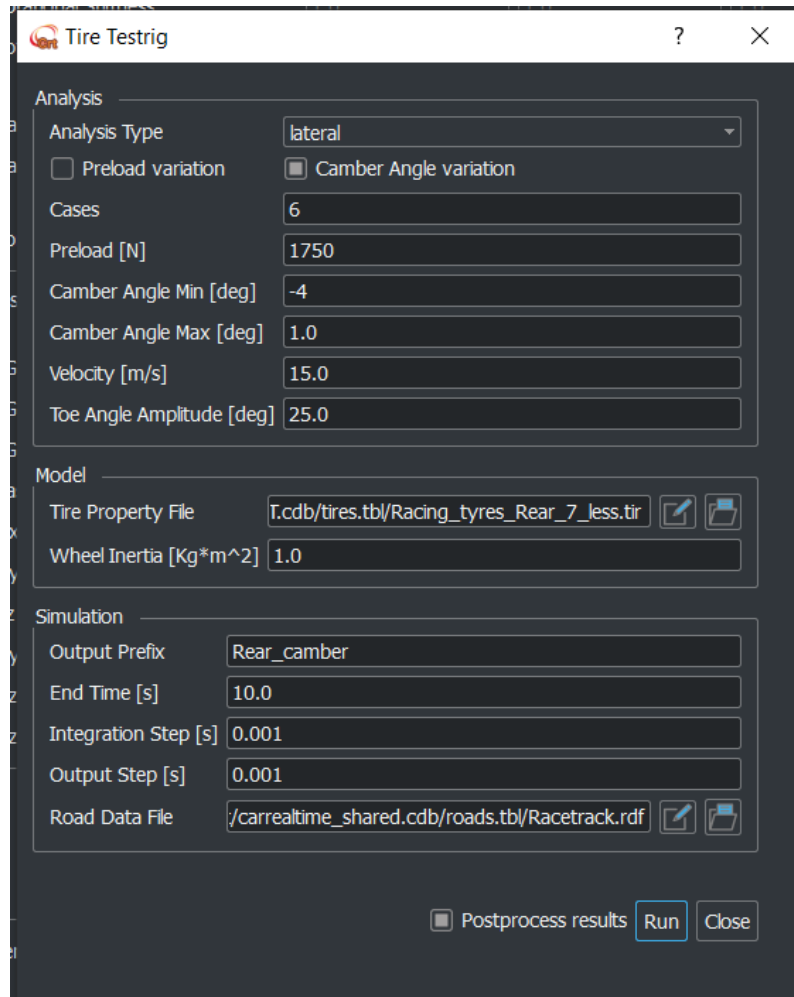


Figure 71 Lateral Study for the Rear Tyres

As shown in Figures 71, maximum lateral force increased from approximately 2729 N at $+1^\circ$ camber to 2891 N at -4° camber, while the slip angle at which peak force occurred decreased from 12.0° to 10.3° .

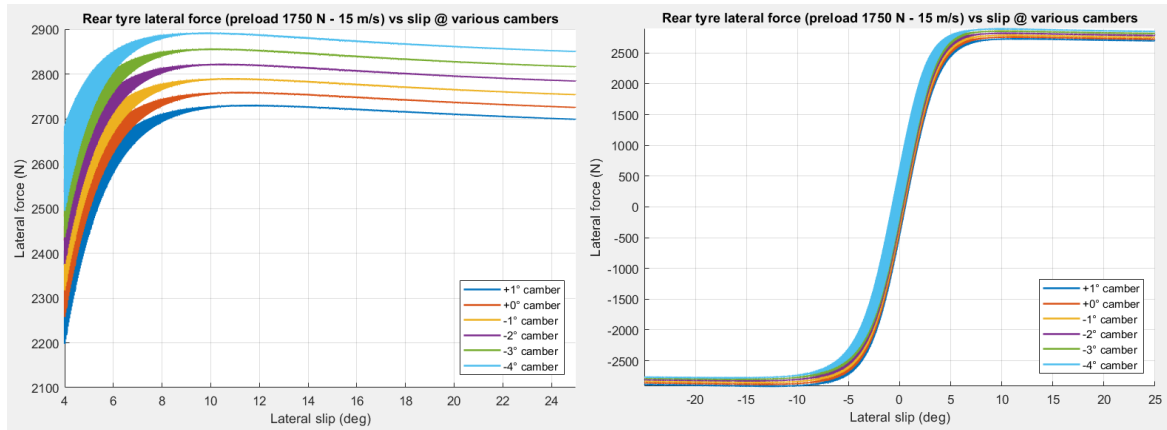


Figure 72 Rear Tyre Lateral Force (preload 1750 N) vs slip at various camber angles

This indicates that additional negative camber improved grip efficiency, though the effect was less pronounced than at the front axle due to the already larger contact patch of the rear tyres.

Table 10 Camber Sweep comparison Max Lateral Force N and Slip Max Deg - Rear

Tyre	Camber_deg	MaxLatForce_N	SlipAtMaxForce_deg
"Rear tyre"	1	2729.6	12.044
"Rear tyre"	0	2758.5	11.629
"Rear tyre"	-1	2789.1	11.238
"Rear tyre"	-2	2821.4	10.871
"Rear tyre"	-3	2855.4	10.559
"Rear tyre"	-4	2891.2	10.274

Self-aligning torque also followed the same trend, with higher negative camber producing stronger restoring moments. Peak values rose from 74 Nm at +1° camber to 84 Nm at -4° camber, with slip angles at maximum torque shifting from 2.2° to 2.7°. While this improves feedback, it also increases the steering effort required to stabilise the car at the limit.

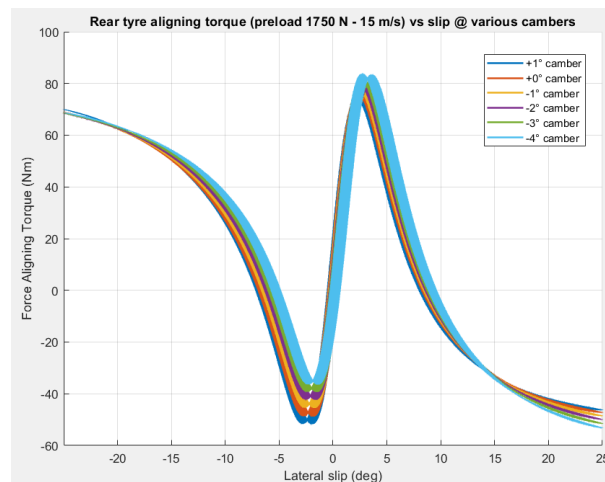


Figure 73 Self aligning Torque force (preload 1200 N) vs slip at various camber angles – Rear

Table 11 Self Aligning Torque Values for the Camber Sweep Study - Rear

Tyre	Camber_deg	Max Self Aligning Torque Nm	SlipAtMaxForce_deg
"Rear tyre"	1	74.104	2.1963
"Rear tyre"	0	75.969	2.2901
"Rear tyre"	-1	77.574	2.384
"Rear tyre"	-2	79.408	2.509
"Rear tyre"	-3	81.468	2.6028
"Rear tyre"	-4	83.756	2.7277

It is important to note that increasing negative camber on the rear tyres also reduces the longitudinal contact patch during straight-line conditions, particularly under braking or acceleration. This introduces a trade-off: greater lateral capability at the expense of reduced longitudinal efficiency. For this reason, camber settings must balance cornering grip with stability under traction and braking, particularly given the higher loads carried by the rear axle.

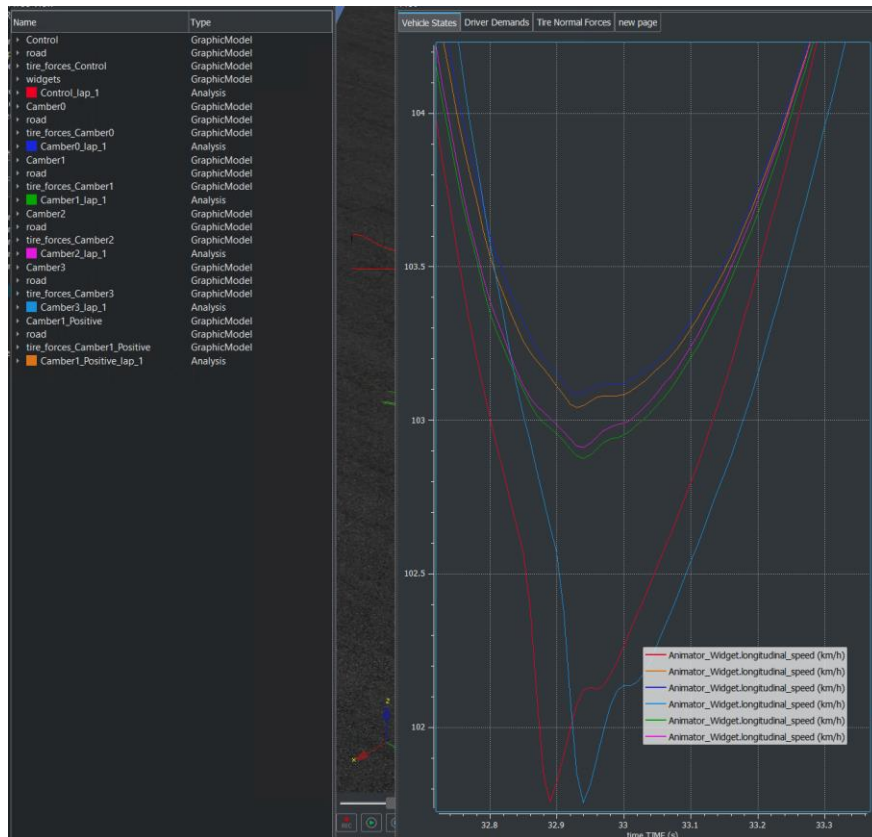


Figure 74 Speed Trace Graph showing the different minimum speeds at one particular corner of the lap for different camber angles

The speed trace result in Figure 74 illustrates the influence of front and rear camber variation on cornering performance, with -3 camber, the minimum cornering speed dropped to 101.76 km/h, compared with 102.91 km/h at -2 camber. The control setup, set at -2.5 on both front and rear wheels, maintained overall good performance.

These results indicate that excessive negative camber can reduce grip and overall speed through the corner. While negative camber typically improves lateral force generation by optimising the contact patch under load, excessive values reduce the effective contact area in straight-line and transitional phases, leading to a measurable loss in cornering performance. In this case, the penalty was approximately 1 km/h at the apex, confirming that camber sensitivity has a clear trade-off between peak grip and stability.

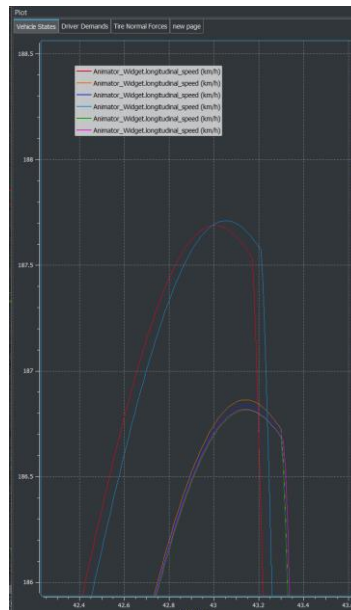


Figure 75 Top Speed of a certain corner through a Camber Sweep

But in contrast in top speed having 0 camber it increases the top speed in comparison to camber -1 and -2 but using camber -2.5 and -3 it goes from 186.85 up to 187.7 I mean it is only 1 km/h of difference.

The MCR suspension model was configured with a static front camber of -2.5° . To evaluate its behaviour under suspension movement, a jounce analysis was performed to examine dynamic camber variation.

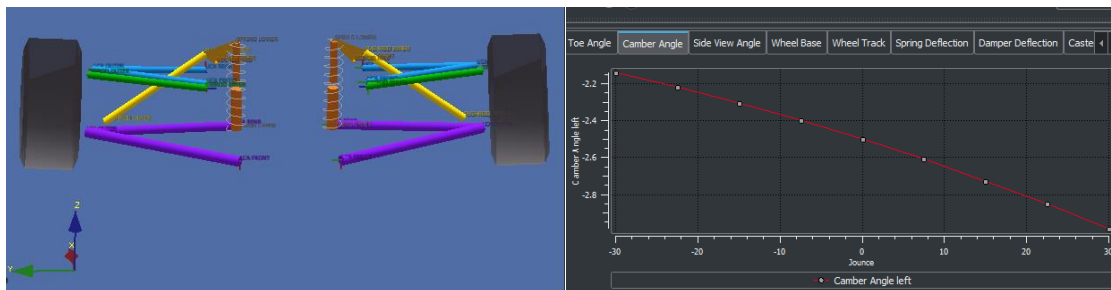


Figure 76 Front Dynamic Camber - Camber angle vs Jounce

As shown in Figure 76, camber becomes progressively more negative as the suspension compresses. Starting from -2.2° at nominal ride height, the camber angle continues to increase with jounce, reaching approximately -2.8° at full compression. This indicates that the suspension geometry is designed to maintain or increase negative camber under load, thereby improving the tyre's lateral grip capability in cornering by keeping the contact patch aligned with the road surface.

This behaviour is consistent with the geometry of a double wishbone suspension system, where the shorter upper arm induces camber gain during compression. While this enhances mid-corner grip, excessive camber gain may accelerate tyre shoulder wear or reduce longitudinal grip in braking zones, highlighting the importance of balancing static camber settings with dynamic characteristics.

5.1.2 Toe

In contrast to camber, toe cannot be directly tested through a dedicated tyre rig analysis in VI-CarRealTime. Instead, evaluation requires creating a new vehicle model and adjusting the suspension geometry parameters within the front and rear suspension setup. This is done by modifying the wheel angle settings in the suspension section of the model, which allows simulation of different toe-in or toe-out configurations. While less extensive than camber testing, this method enables the assessment of toe angle effects on vehicle behaviour, particularly in relation to straight-line stability, corner entry response, and tyre wear.

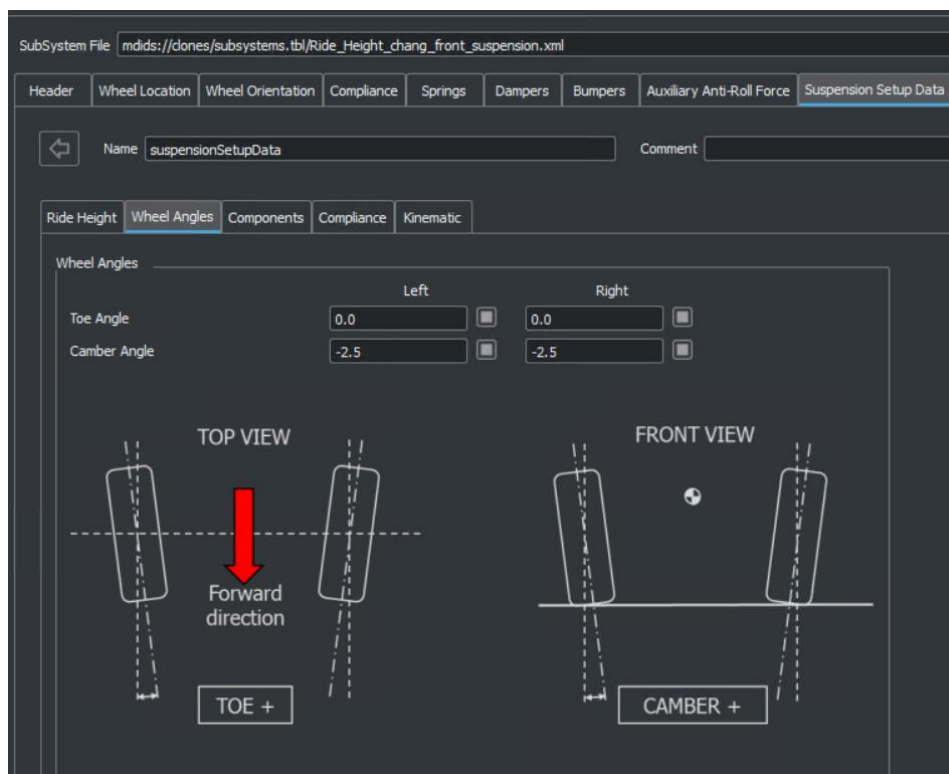


Figure 77 Change in Toe setup representation

As shown in Figure 77, toe settings can be modified individually for the left and right wheels in the front suspension subsystem. Adjustments are made through the wheel angle parameters, where positive values correspond to toe-in and negative values to toe-out. To change the rear toe configuration, the same process must be followed by accessing the rear suspension subsystem.

To investigate the effect of toe on vehicle performance, a series of simulations were carried out in VI-CarRealTime. The control model was configured with -2.5° camber on both axles and 0° toe, representing the baseline condition. Additional models were created with the following configurations:

- Toe 0: 0 Toe and 0 Camber on all wheels
- Toe -2: -2 Toe combined with -2.5 camber
- Toe +2: +2 Toe combines with -2.5 camber
- Multi-toe: Toe out (0.1) combined with rear toe in (+0.1)

The tree view of the simulation setup is shown in Figure 78. The results for Pembrey are presented in Figures 79 and 80.

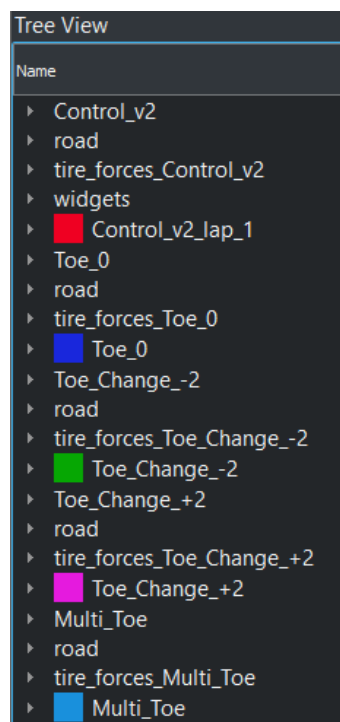


Figure 78 Tree view of the animations and post process data from the vehicle models tested



Figure 79 Graphical Representation of the 5 vehicle models tested on Pembrey using VI-Grade

The lap time comparison revealed negligible performance differences between some configurations. The fastest model, Toe 0, achieved a lap time of 55.91 s, followed by the control model at 55.92 s, and the multi-toe model at 55.93 s. These results confirm that, at the tested settings, small toe variations did not produce a significant impact on overall lap time.

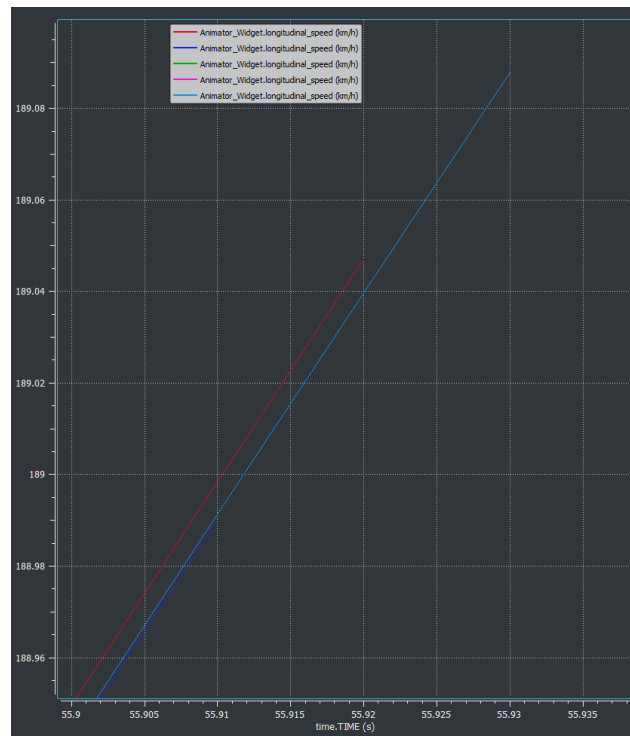


Figure 80 No virtual change between the top 3 models which are Red - Control | Dark Blue - Toe 0 | Light Blue - Multi Toe

This outcome is consistent with expectations: low toe angles primarily affect transient handling characteristics such as initial turn-in response and straight-line stability, rather than producing large differences in steady-state lap time. Within the small range tested, the model did not register performance changes beyond hundredths of a second, suggesting that toe tuning in this context has a secondary influence compared with other suspension parameters such as camber or spring rates.

For visualization purposes in figure 81 it is only going to be presented the control variable and the +2 toe and -2 toe. This is to make a direct comparison between toe -2/0/+2.



Figure 81 Visual representation of the Control - Red | Toe -2 - Green | Toe +2 – Pink

Here it is possible to see that the control variable is getting a faster response from the vehicle but a more approachable way to drive the car is the green version with -2 toe which makes the car more stable when braking into the small-medium fast corners.

To further understand the behaviour of the model with the change of static toe this was the result of the control variable vs a change in -2, -1, 0, 1, 2.

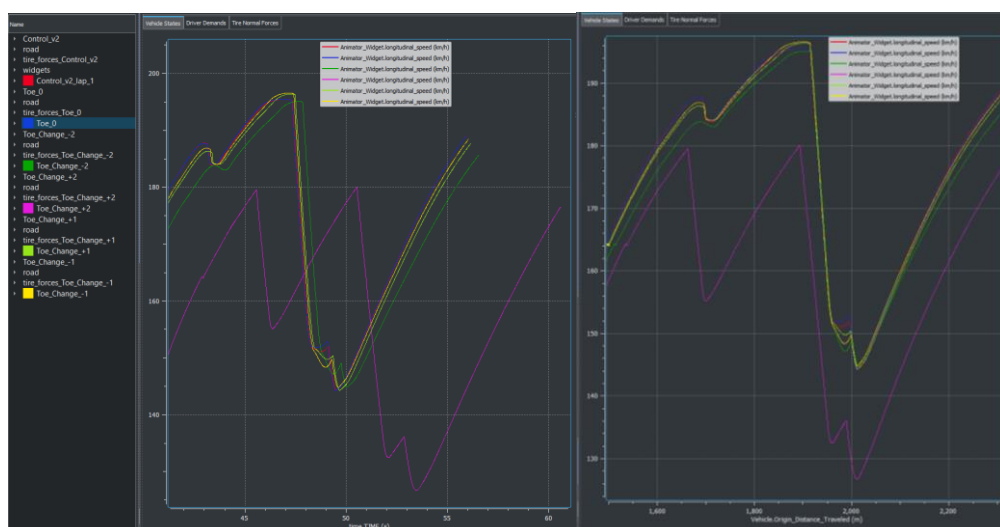


Figure 82 Visual representation of the change in toe on the same vehicle model in time representative | Distance

The change in -1 (out) and +1 toe (in) reflects a small change in performance talking strictly about lap time but the top speed as well as the speeds reached in the high-speed corners also changes according to the theory. Let's take the example the comparison between toe -2 (out) and the toe +2 (in), due to large increment in toe in the car is not able to get to high speeds making the car

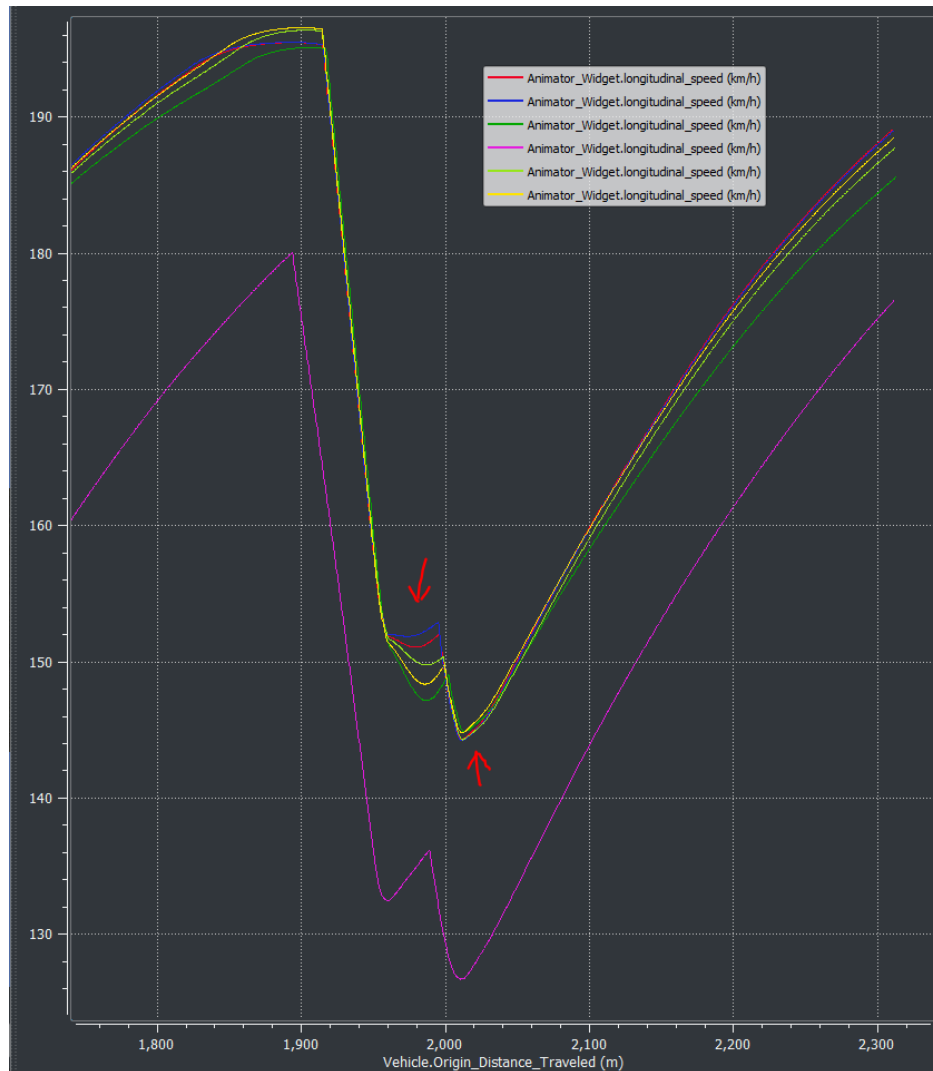


Figure 83 Last high speed corner for different vehicle models using different toe

There is a clear difference between the toe changes and the response in the model but also the same time the only model that deviates from a range of minimum speed in the last high speed corner is the model using toe +2 deviating around 10 km/h and the other models are between 1 km/h up to 2.5 km/h.

5.2 Spring

This subsection establishes whether the VI-CarRealTime model reproduces the expected platform-dynamics trends when coil spring rates are varied about the measured baseline. Increasing spring rate should raise wheel-rate and natural frequencies, reducing heave, pitch, and roll amplitudes, yielding a more stable aero platform.

Softening the springs should produce the opposite behaviour. Because spring changes also alter the roll-stiffness distribution between axles, asymmetric cases are expected to shift balance (under/oversteer) in a predictable direction. Because both axles move together, overall roll gradient should drop or rise without materially shifting balance. Directionality and proportionality of these effects form the core validation checks for the model at this stage.

Table 12 Suspension Spring Rate Values in lbs/in

Spring Rate [lbs/in]	Original Values from the test	Suspension Values					
		-10%	-5%	Equal	+5%	+10%	Hard front Soft rear
Front Suspension	750	675	712.5	750	787.5	825	850
Rear Suspension	850	765	807.5	750	892.5	935	750

Starting from the baseline 750 *lb/in* (front) / 850 *lb/in* (rear), six configurations are evaluated while holding geometry, ARB, damper settings, tyres, and driver model constant. Statics is solved via preload to recover identical target ride heights for each case. The matrix comprises: both axles +10% (825/935), both +5% (787.5/892.5), both -5% (712.5/807.5), both -10% (675/765), an equalised reference of 750/750, and an asymmetric hard-front/soft-rear of 850/750.

These values are summarised in Table 12 (“Suspension Values”). Something to take into consideration is that the values accepted by the model are in [N/mm] therefore Table 13 have the converted values from [lbs/in] to [N/mm].

Table 13 Suspension Spring Rate Values in N/mm

Spring Rate [N/mm]	Original Values from the test	Suspension Values					
		-10%	-5%	Equal	+5%	+10%	Hard front Soft rear
Front Suspension	131.3451263	118.2106	124.7779	750	137.9124	144.4796	148.8578
Rear Suspension	148.8578098	133.972	141.4149	750	156.3007	163.7436	131.3451

The results are the following. The original spring set up was used as a baseline for comparison purposes.

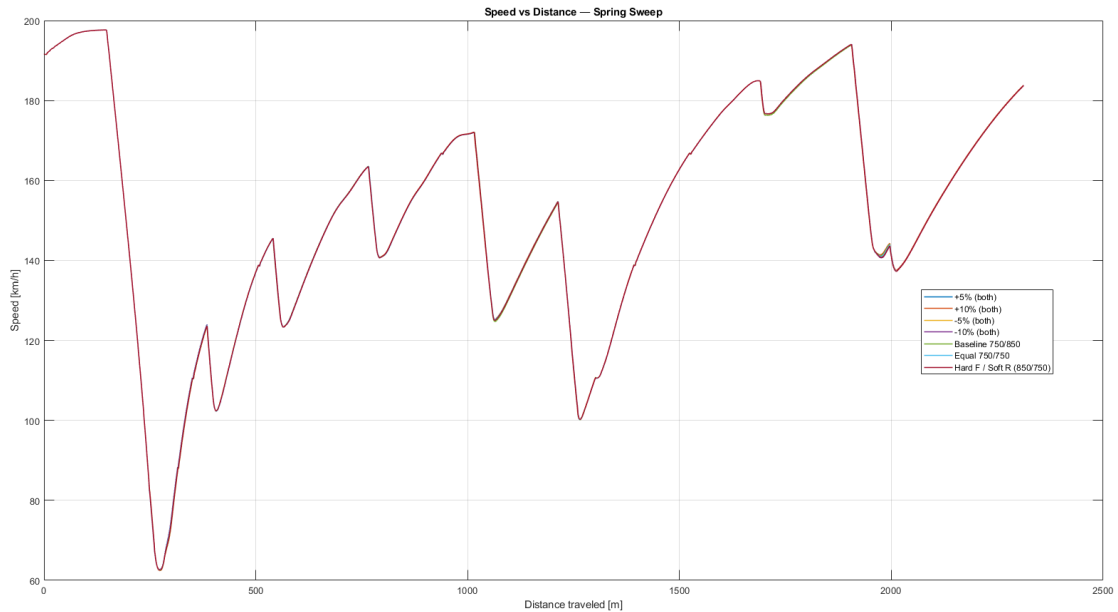


Figure 84 Speed vs Distance - Spring Rate Sweep Front & Rear

From the looks all models behave time wise the same and virtually similar speeds in slow and high speed corners but the important factors to look for is not just speed trace but the forces and the chassis dynamics of the model in the different scenarios. The first 2 corners are displayed in the following graphs showing the difference in driver behaviour on the steering wheel handle, the braking percentage and the chassis pitch on the corner.

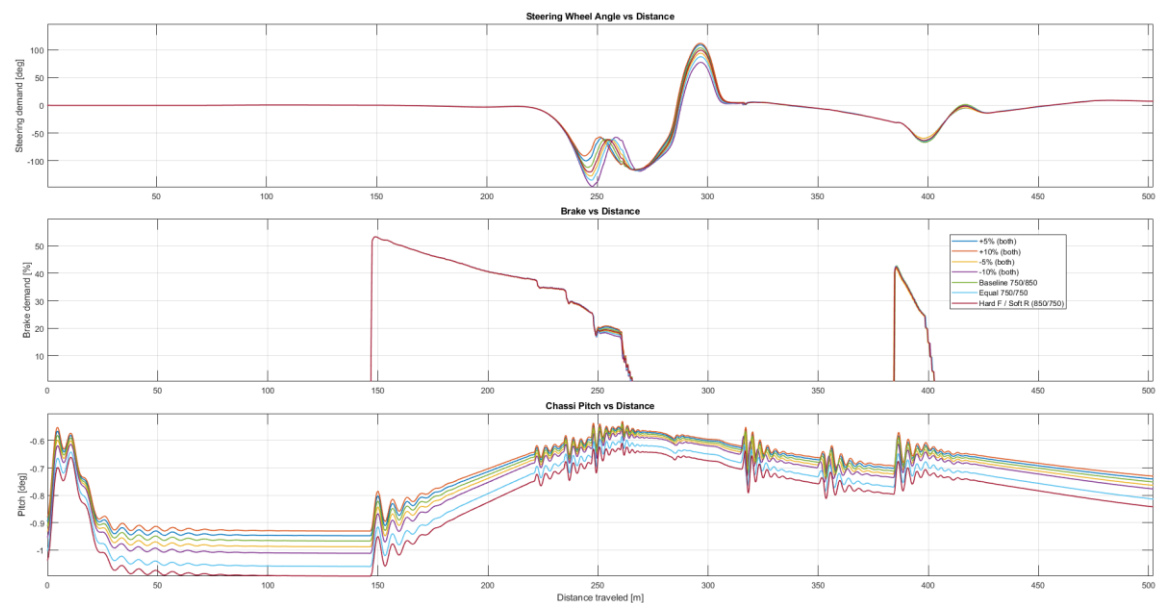


Figure 85 Steering Wheel Ange, Brake % and Chassis Pitch vs Distance - Spring Rate Sweep Front & Rear

When analysing the braking traces in Figure 85, it is evident that the models exhibit nearly identical behaviour during the corner entry and apex phases. The braking effort and vehicle dynamics remain consistent across the different spring rate configurations. However, at the corner exit, small deviations begin to appear between the models. This indicates that while the initial and mid-corner braking response is stable and repeatable, differences in suspension setup affect how each model transitions out of the corner where longitudinal load transfer interacts with chassis pitch and suspension compliance.

Now looking at the whole lap distribution in Figure 86 there is a clear difference in the pitch of the chassis vs the other models throughout the lap.

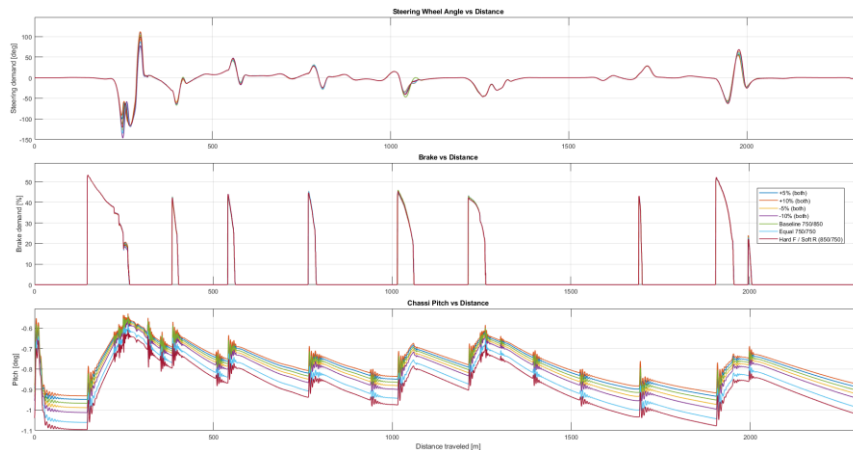


Figure 86 Lap distribution of the different spring rate configurations

Throughout the lap it can be seen that the more stable/less pitch is the suspension system with +10% of stiffness in the springs and then as the % goes lower and then to the negative of the base line it becomes a more pitch sensitive. The case with higher absolute pitch is the change in spring rate swapping and making it hard in the front and soft in the rear 850/750 [lbs/in], getting a maximum of -1.096 in pitch deg while the +10% from the base line is -0.93

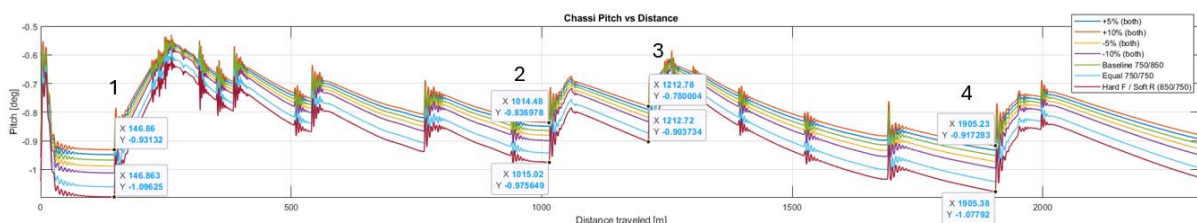


Figure 87 Chassis Pitch vs Distance Graph - Spring Rate Sweep

The difference between the most stable and the least stable

Table 14 Average Pitch Difference throughout the lap – Spring Rate Sweep

Difference in Pitch angle	
1	0.1649
2	0.1387
3	0.1237
4	0.1606

This difference makes the study a more powerful model as it is behaving as expected while with a higher spring rate or stiffness the vehicle will respond by having a less pitch change in comparison to a higher spring rate model. The difference between each other stays roughly the same during high braking points like the first and previous to last corner as throughout the middle of the track in two consecutive laps, which having some similar % of breaking the pitch angle difference between both models are roughly the same too.

Throughout the lap it is shown that the baseline in green is closer to the “optimal case” in quotation marks but the models that start deviate from the actual range or start to deviate at a higher gradient are the models with equal spring rate and the Front Hard and Rear Soft.

Further into the study, the relationship between steering wheel input and the rack forces experienced throughout the lap was analysed. Figure 88 presents the correlation between steering demand (driver input) and rack force (N). At first glance, the graph appears complex due to overlapping traces from the various models. However, focusing on the far left of the plot, corresponding to the largest steering angles, reveals noticeable differences between the suspension setups.

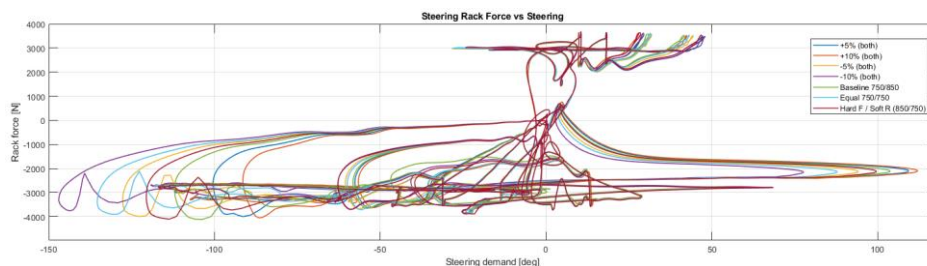


Figure 88 Steering Rack Force applied on the car through the lap - Spring Rate Sweep

These differences indicate that while the steering force characteristics remain broadly similar in moderate steering ranges, the effect of spring rate changes becomes more evident under high steering demand, such as during heavy cornering. In these scenarios, the rack force diverges between models, showing that suspension stiffness influences not only chassis balance but also the steering feedback perceived by the driver.

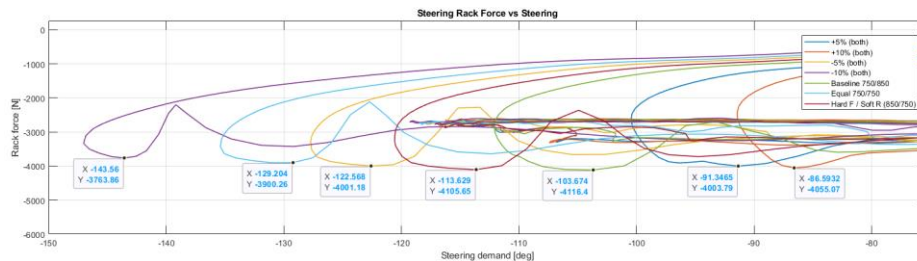


Figure 89 Steering Rack Force applied on the car through the lap, zoom on the left- Spring Rate Sweep

Table 15 Steering Rack Force vs Steering Demand Table - Spring Rate Sweep Models

Model	Steering Demand [deg]	Steering Rack Force [N]
-10% Spring Rate	-143.56	-3763.86
Equal 750/750	-129.204	-3900.26
-5% Spring Rate	-122.568	-4001.18
Hard Front Soft Rear	-113.629	-4105.65
Baseline 750/850	-103.674	-4116.4
+5% Spring Rate	-91.3465	-4003.79
+10% Spring Rate	-86.5932	-4055.07

The results in Figure 90 and Table 15 show that as steering angle increases, the steering rack force opposing the input decreases slightly, meaning the driver experiences less resistance at higher lock. However, the more significant observation is how spring stiffness influences the steering angle required for a given manoeuvre.

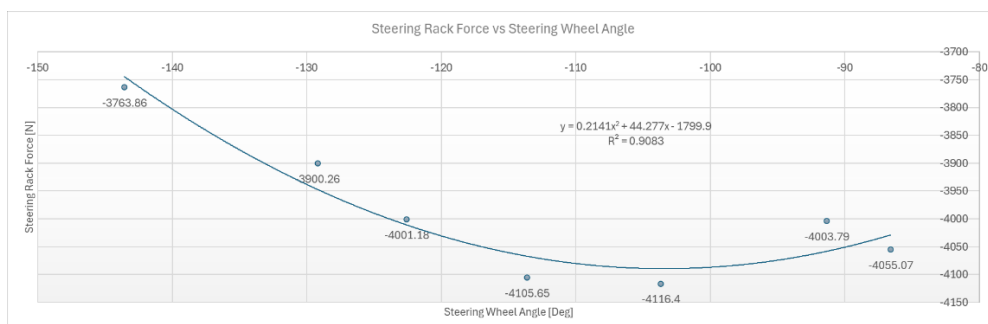


Figure 90 Steering Rack Force vs Steering Wheel Angle - Spring Rate Sweep

With softer spring rates (-10% and -5%), the car requires larger steering angles (-143.6° and -122.6° respectively) to achieve the same cornering response. This reflects a more compliant chassis that generates less lateral load transfer efficiency, demanding greater driver input.

Conversely, with stiffer spring rates (+5% and +10%), the required steering angles are much lower (-91.3° and -86.6°), indicating that the car responds more directly to smaller inputs. From a performance perspective, this means stiffer setups produce a more sensitive and pointy car, turning in with less steering effort but transmitting higher forces through the rack. Softer setups, on the other hand, provide greater steering angles with lighter resistance, which may feel easier to control but can reduce precision and responsiveness.

So, the choice is not simply between “easier to steer” or “harder to steer”. Instead, it is about driver preference and balance:

- Stiffer Spring Rate -> Shaper response, less steering angle, higher feedback
- Softer Spring Rate -> Larger steering angle, lighter feel, less precision

Also, it is possible to observe that the relation between the going from a lower value up to a higher value of spring rate is quadratic showing a value of $R^2=0.908$. This discovery suggests that the relation between steering wheel angle and steering rack force is sensitive to change of spring rate.

Model	Steering Demand [deg]	Left	Steering Demand [deg]	Right
		Steering Rack Peak Force [N]		Steering Rack Peak Force [N]
-10% Spring Rate	-143.56	-3763.86	77.6528	-2149.44
Equal 750/750	-129.204	-3900.26	87.8338	-2140.56
-5% Spring Rate	-122.568	-4001.18	94.1811	-2135.58
Hard Front Soft Rear	-113.629	-4105.65	99.6233	-2123.22
Baseline 750/850	-103.674	-4116.4	103.76	-2119.2
+5% Spring Rate	-91.3465	-4003.79	109.372	-2108.72
+10% Spring Rate	-86.5932	-4055.07	112.101	-2103.1

The steering analysis reveals a directional asymmetry. With stiffer springs (+10%), the car requires smaller steering angles in left turns (-86.59°) than with softer springs (-10% , -143.56°), a difference of 56.97° . In right turns, the trend is weaker and partially reversed: the +10% setup needs 112.10° vs 77.65° for -10% , a difference of 34.45° .

Thus, the left–right disparity in steering demand between the softest and stiffest setups is 22.52° larger for left turns than for right turns. Rack forces on right turns are similar across extremes (-2103 N for +10% vs -2149 N for -10% ; $\Delta \approx 46$ N), indicating that the required steering angle changes more than the steering load.

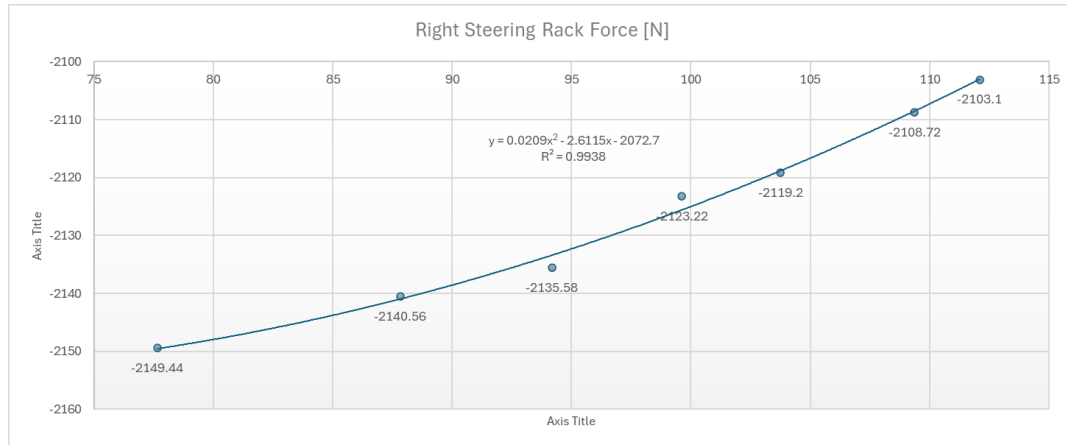


Figure 91 Right side Steering Rack Force vs Steering Wheel Angle - Spring Rate Sweep

The right side also shows a quadratic relation going from a softer choice of spring rates to a stiffer spring rate, now this relation has a value of $R^2=0.993$ which means that the trendline does predict the data presented.

Figure 92 shows the relationship between steering wheel angle and driver feedback force across different spring rate setups. The results indicate that increasing spring stiffness within the $\pm 10\%$ range reduces the steering angle required to achieve peak steering force. In other words, the car becomes more responsive at smaller steering inputs when the suspension is stiffer. Conversely, with softer springs, the car requires greater steering angles to achieve the same response, making the steering feel less direct.

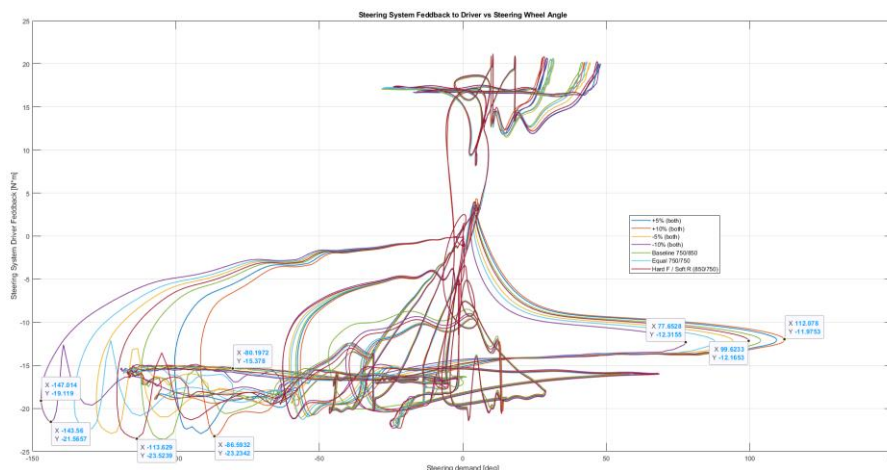


Figure 92 Steering System Feedback to Driver vs Steering Wheel Angle

From a driver's perspective, this translates into a clear difference in handling balance. A stiffer setup provides sharper response and greater sensitivity, but it also increases the feedback felt through the steering system, which may make the car feel nervous at the limit. A softer setup

reduces steering sensitivity, requiring more lock to achieve the same path, but offers smoother feedback and a more progressive response. This trade-off illustrates how suspension stiffness directly affects both driver effort and vehicle agility, influencing the overall understeer/oversteer balance.

Table 16 Steering Feedback Peak Force at different Steering Angles within the Spring Rate Sweep

Model	Left		Right	
	Steering Demand [deg]	Steering Feedback Peak Force [N]	Steering Demand [deg]	Steering Feedback Peak Force [N]
-10% Spring Rate	-143.56	-21.5657	77.6528	-12.3155
Equal 750/750	-129.204	-22.3862	87.8338	-12.2647
-5% Spring Rate	-122.568	-22.9253	94.1811	-12.1628
Hard Front Soft Rear	-113.629	-23.5239	99.6233	-12.1653
Baseline 750/850	-103.674	-23.5855	103.76	-12.0669
+5% Spring Rate	-91.3465	-22.9403	-12.0669	-12.007
+10% Spring Rate	-86.5932	-23.2342	112.101	-11.9753

The spring sensitivity analysis confirms that the model behaves in the expected manner, with changes in stiffness producing the anticipated effects on steering response, handling balance, and overall vehicle dynamics. This validates the spring model as a reliable representation of the real car, ensuring that subsequent optimisation studies are based on accurate and predictable behaviour.

5.3 Damper

There is an interesting discovery is that in terms of experiencing load all 4 tyres using the real values of the dampers that were tested in Figure 93 it is possible to observe that the biggest change are the rear tyres having at 1 point at the fastest section of the track while the front tyres do experience some similar forces at the beginning of the track but as we go further in to the lap those values start to deviate from the results measured.

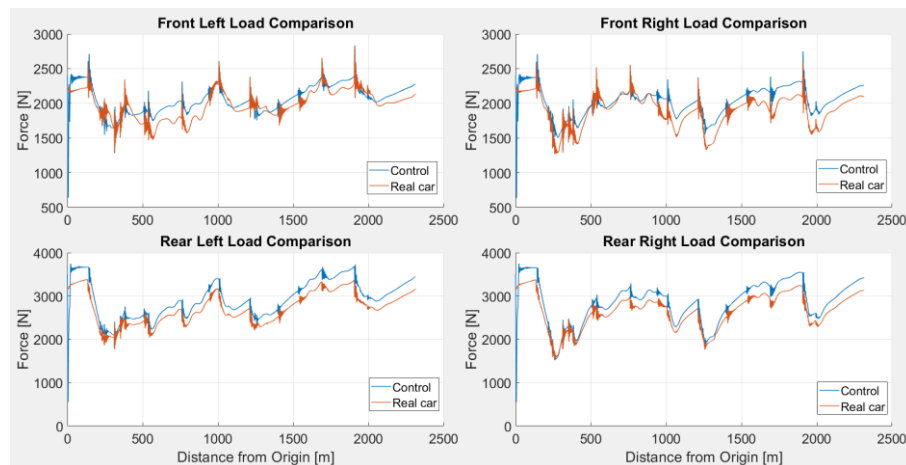


Figure 93 Front and Rear Tyre Loads comparison between Control Model vs Control Model with tested Dampers

There is roughly a mismatch around the 1250 m mark which that would be one of the high speed section of the circuit and there the tyre load for example for the Front Right tyre is experiencing around 350 N less using the tested dampers than the control model using a higher damped dampers. However throughout the lap the car is maintains similar tyre loads however is it not expected for them to have the same tyre loads but a similar trend it is expected, an increase and decrease through the corners as the control model.

Going further into the analysis at lateral forces the model with the tested dampers is capable of getting to the maintained longitudinal forces as the other model and at the braking is similar reaching roughly -1.6 g of braking force, but that's because the tyres are the same as mentioned previously so the simulation driver from the model is capable to doing the same braking at the same moments of the lap.

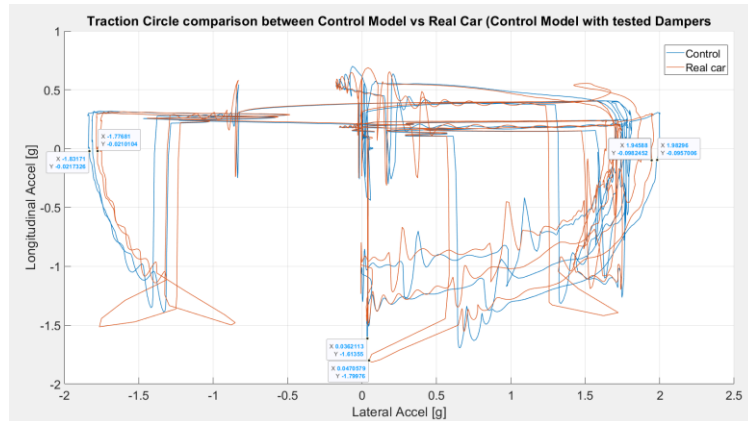


Figure 94 Traction Circle comparison between Control Model vs Control Model with Tested Dampers

The traction-circle comparison between the Control model and the real-car configuration fitted with the tested dampers shows a clear reduction small but reduction in lateral acceleration capability for the real car vehicle model. The longitudinal envelope (braking and drive) remains broadly comparable, indicating similar longitudinal grip; however, the lateral branches of the ellipse for the tested dampers are visibly contracted and more regular as the control model.

This pattern is consistent with an under-damped platform response in the lateral domain: roll/yaw transients disturb tire load distribution and prevent the vehicle from sustaining peak lateral forces however the real car model is up to its limits therefore reaching similar lat forces. In practice, the car loses some stability as it approaches the lateral limit; during tuning the model it repeatedly spun or left the track when pushed to exploit cornering grip, behaviour attributable to insufficient damping of lateral/roll motions rather than any deficit in longitudinal traction but later in the simulation the reason why it was happening it was not due to the underdamped dampers but the line that the model was taken, so an small change in the lap was made to make it possible now instead of spun the car just oversteers enough to keep it on the track.

The speed distance traces indicate a systematic small loss of pace with the tested damper configuration across the whole lap. Relative to the control, the real-car trace shows a similar decelerations, and similar recoveries, features that repeats at low, medium, and high-speed corners. This produces a profile with consistent throttle modulation to manage transient stability. Because exit speed sets the entry speed to the next straight, even with these local deficits around the track the car arrives at subsequent braking zones with a similar terminal velocity even when powertrain performance is unchanged.

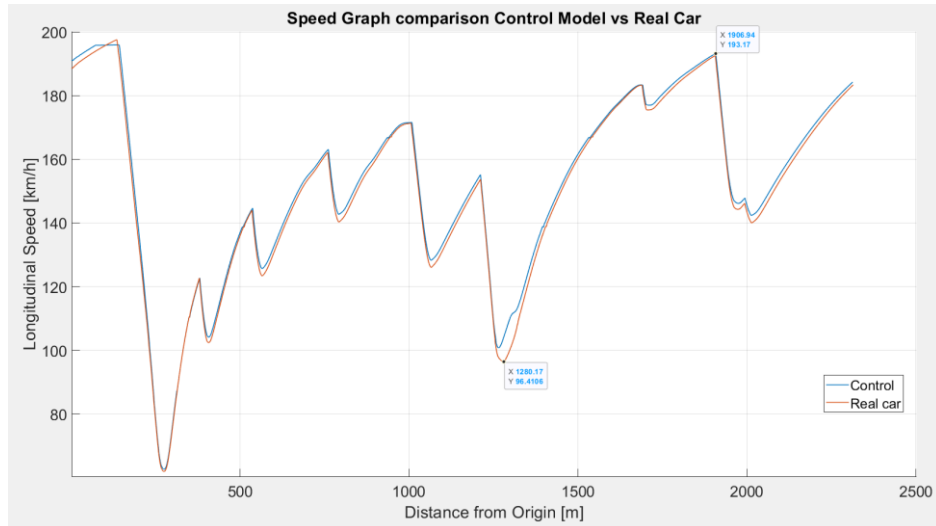


Figure 95 Longitudinal speed comparison graph between the Control vehicle model and the real car (control model with tested dampers)

A clear example occurs at the penultimate corner: the real-car trace drops a couple km/h to re-establish tyre grip before committing to the apex and accelerating, whereas the control sustains a higher minimum and a cleaner rise back to maximum velocity. Similar gaps appear at earlier complexes, where the control model's "textbook" V-shaped corner signatures are preserved and the tested-damper trace shows similar valleys and delayed ramp-up. Taken together, the plot shows that the damping change alters the platform's transient behaviour enough to make the car similar to the real race car which make achievable corner entry, mid-corner, and exit speeds, yielding a slower lap time.

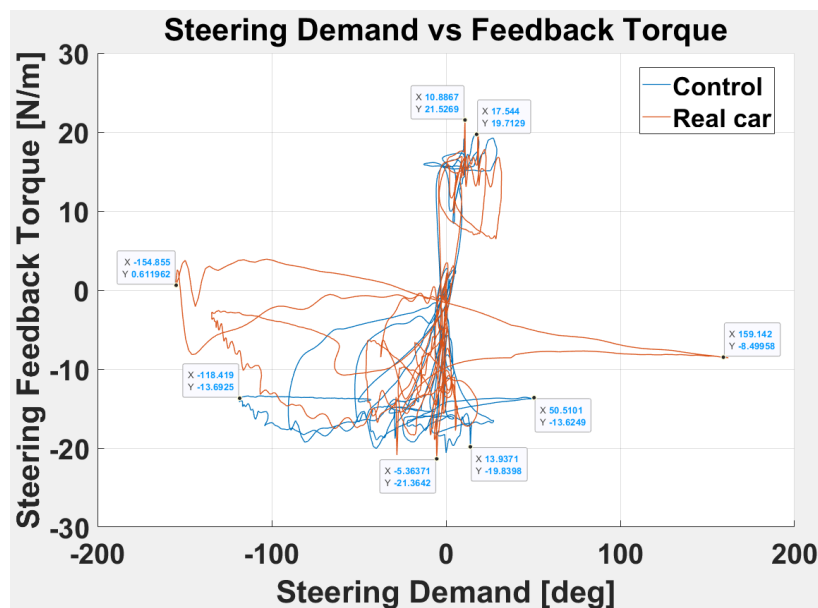


Figure 96 Steering Demand vs Feedback Torque graph comparing control vehicle model vs real car (control with tested dampers)

The steering demand–feedback torque map Figure 96 shows a stark contrast between configurations. The control model produces broad, well-defined curves with clear peaks about 19 N·m at roughly 13.5° of steer and similarly large peaks on opposite lock, with pronounced increase through entry, mid-corner, and exit. The tested-damper configuration remains tightly clustered near the origin and just about the same, even at comparable steering demands, yielding a narrow, low-gradient of points with almost no increase of torque feedback as the control model.



Figure 97 Driver steering demand and braking vs Distance from the origin

The combined steering and brake traces indicate similar stability for the tested-damper configuration as the control model. Just passed the middle point of the track at one of the high speed corners the steering signal shows an abrupt, opposite-sign correction consistent with recovery from incipient oversteer; the wheel angle steps rapidly to correct the yaw and prevent a spin. The control model, in the same segment, follows a smoother profile with modest steering amplitudes and no corrective spike, indicating that it remains within the stable operating window.

Overall, the tested-damper configuration forces the driver to shed same speed and to intervene just once to keep the car balanced, which explains the slight lower sustained speeds observed in the accompanying speed distance and g-g analysis. From the comparison, it is evident that the vehicle fitted with the new dampers exhibits a reduction in pitch motion under braking. The control setup (blue) shows a sharper drop in pitch angle, while the real car with the tested dampers (orange) maintains a more stable pitch profile. This indicates that the revised damping characteristics help manage longitudinal load transfer more effectively, reducing nose dive and improving braking stability.

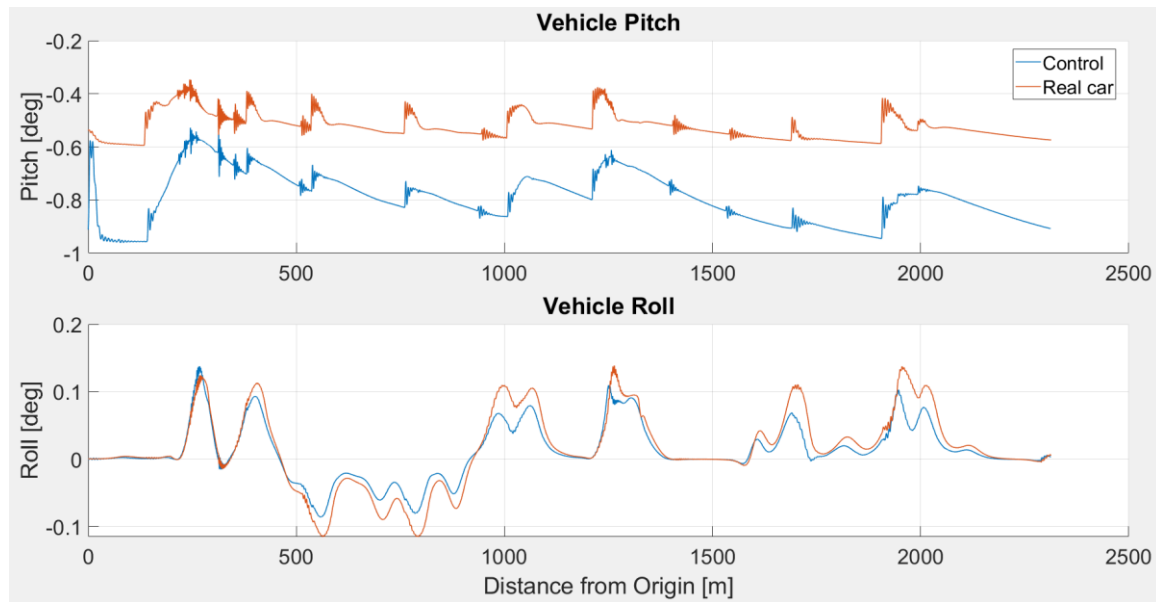


Figure 98 Vehicle Pitch and Roll Comparison Control Vehicle model vs Real Car (control with dampers)

On the other hand, a slight increase in roll motion can be observed in the vehicle with the new dampers. Although the difference is small, it suggests a trade-off: by limiting pitch sensitivity, the system allows marginally greater compliance in roll. Nevertheless, the roll values remain low in magnitude, indicating that the overall lateral stability of the car is not compromised.

In summary, the damping changes provide a beneficial compromise, reducing excessive pitch under braking while only minimally affecting roll behaviour. This adjustment aligns with the intended outcome of improving longitudinal stability without significantly degrading lateral balance.

For the upcoming analysis done on the car, regarding the change in dampers the control model will be using the new damper model which has lower damper characteristics (rate of energy absorption) as the driver is capable to taking the car to the limits without a higher damper stiffness to reach the desirable lap time and for the speed trace to be similar as with the lateral and longitudinal forces exerted on the car.

5.4 Anti Roll Bar (ARB)

The anti-roll bar (ARB) sweep was performed by scaling the baseline bar wheel rates simultaneously at the front and rear by ± 5 , ± 10 and ± 20 per cent relative to 68.5 N/mm (front) and 30.3 N/mm (rear). This isolates the influence of roll-stiffness distribution on vehicle attitude and tyre loading while keeping springs, dampers and aero unchanged. Figures 44 [Tyre Fz] and 45, 46 [Roll] compare chassis roll angle and the front-left tyre normal force across a representative lap. The colours go as; Red – Control (0% Change), Yellow – (-20% Stiffness), Orange – (-10% Stiffness), Light Blue – (-5% Stiffness), Dark Blue – (+5% Stiffness), Green – (+10% Stiffness), White – (+20% Stiffness).

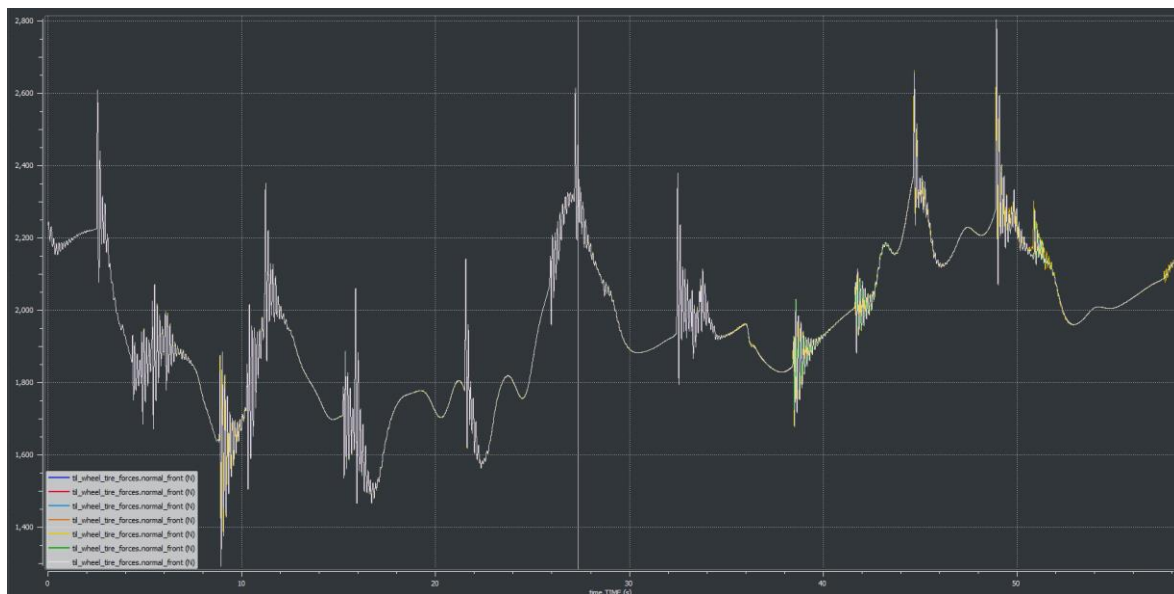


Figure 99 Front Left Tyre Load comparison between the different models

Increasing ARB stiffness produced a expected reduction in chassis roll (Bayrakceken, Tasgetiren and Aslantas, 2006). The roll traces converge as stiffness is increased, with the +10 per cent and +20 per cent variants showing the smallest peak roll and the tightest envelope through direction changes. Because ARBs act primarily in roll and not in heave, this reduction in attitude occurs without altering the platform height, which explains the near-identical pitch behaviour across all variants.

Table 17 ARB - Rate [N/mm] - Anti Roll Force at Wheel

Anti Roll bar - Rate[Newton/mm] - Anti Roll Force At Wheel							
	MCR ARB	-20%	-10%	-5%	5%	10%	20%
Front	68.5	54.8	61.65	65.075	71.925	75.35	82.2
Rear	30.3	24.24	27.27	28.785	31.815	33.33	36.36

The reduction in roll angle is accompanied by a rise in lateral load transfer through the ARB change. This is visible as a small but consistent increase in the outside-front normal force with stiffer bars and a corresponding reduction on the inside tyre. The effect is most apparent in long, high-lateral-g corners and over kerb sequences where single-wheel inputs put the ARB to work. The increase in F_z is small and remains within the operating envelope verified with the TireLimits model, confirming that the tested range does not exceed structural or friction limits for the chosen compound.

A localised handling slip appears at the last corner. The +10 % and +20 % settings exhibit a mild oversteer tendency that is not present in the softer variants. Corner traces show higher rear-axle μ utilisation linked to the increased ARB-induced load transfer, pushing the rear tyres slightly beyond their optimal combined-slip region under sustained lateral and drive demand. The event is recoverable and does not provoke instability, but it marks the point at which added roll stiffness begins to show the rear-axle margin in terms of load added into it.

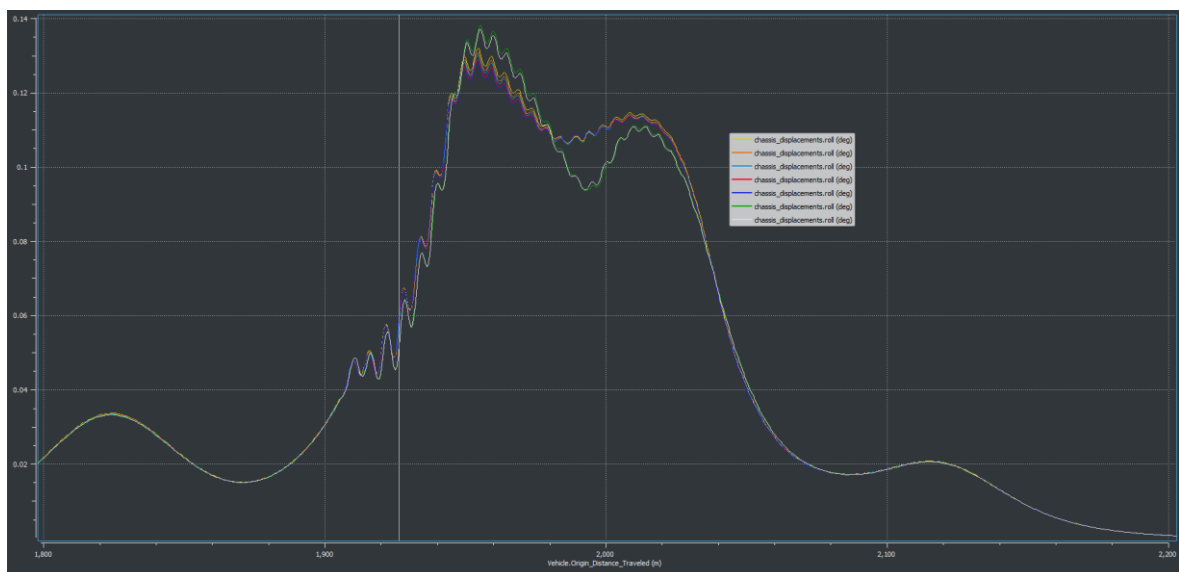


Figure 100 Roll (deg) vs Distance from Origin Graph Roll Analysis sweep comparing different ARB stiffness in the vehicle model

Despite the changes in the Anti Roll Bar stiffness, lap time does not improve with increase or decreases on ARB stiffness. This outcome is consistent with tyre load sensitivity. By shifting more of the lateral load to the outside tyre, the axle's combined lateral capacity remains flat or can decrease slightly, so the net grip does not rise even though the car feels more supported. In addition, a lower roll angle can move the outside tyre away from its camber sweet spot that was

implicitly tuned for the baseline roll gradient. Together these effects explain the absence of a raw-performance gain in lap time.

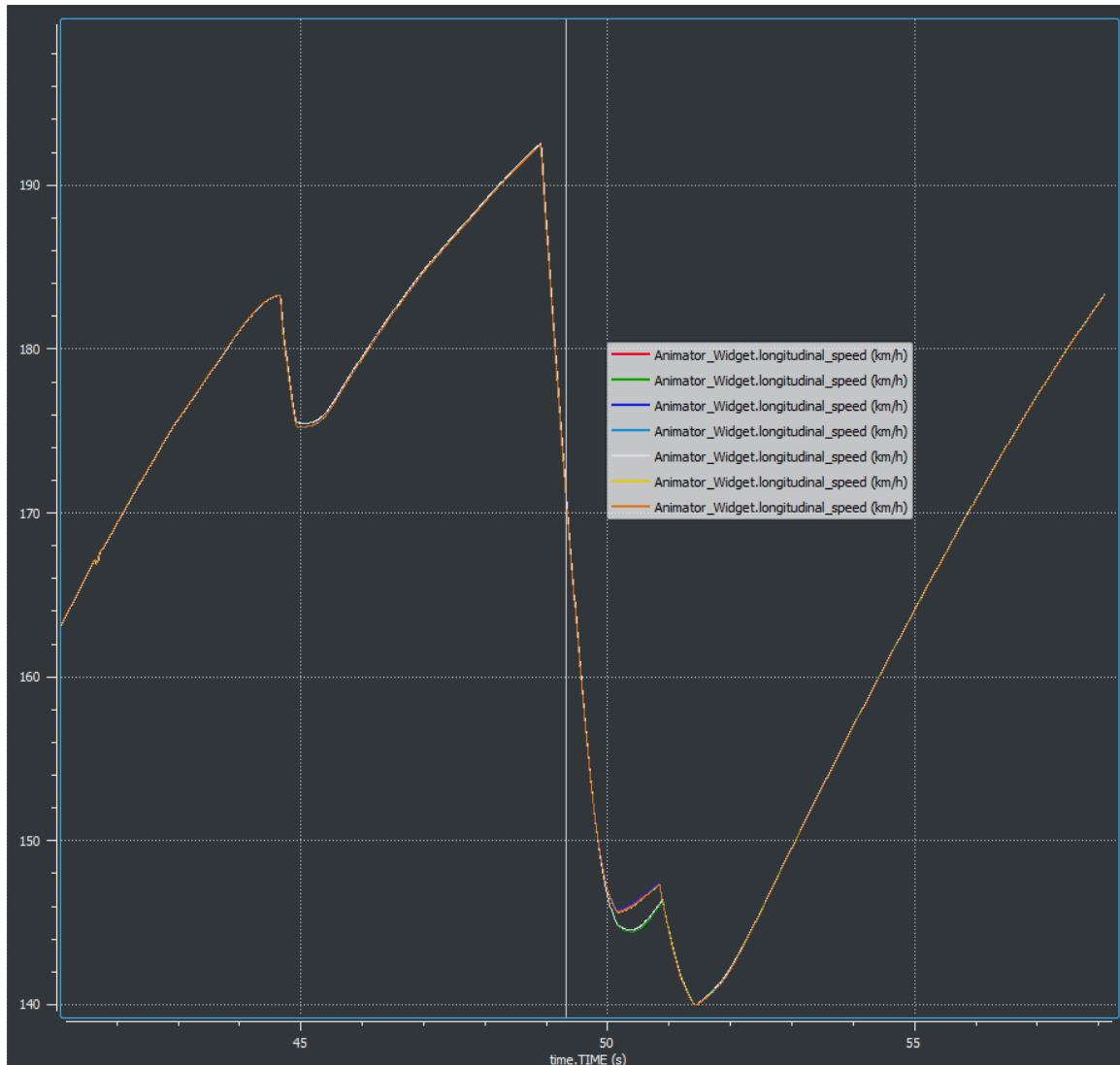


Figure 101 Speed Graph - Change in Roll -20% to 20% with steps of 5% using the MCR ARB

6. Optimisation Method – Box Behnken Method

6.1 Identifying the Optimal Setup

Before an optimization on chassis performance, it is essential to understand the fundamentals of load transfer and its impact on tyres and suspension systems. The main challenge does not lie in analysing load transfer itself, but rather in dealing with the vast number of interconnected factors that influence vehicle behaviour. In practice, chassis performance depends on achieving balance across all four tyres under a wide variety of dynamic conditions, a task made complex by the thousands of variables and millions of possible interactions that must be considered (Rowley, 2003).

Even with simplifications, a typical lap time simulation requires thousands of inputs to produce meaningful results. Mathematical models, such as those within VI-CRT, are therefore invaluable when calibrated accurately, as they highlight which parameters have the greatest influence on lap time and handling. Nevertheless, no simulation can fully replace physical testing, particularly in the early stages of learning how load transfer influences performance.

Chassis Performance can be broken down into three core elements:

1. Static Setup
2. Dynamic Load Transfer
3. Tyre Acceleration vs Normal Loads

The initial or shop setup provides the foundation for on-track performance. At this stage, the engineer defines springs, anti-roll bars, dampers, preloads, and ride heights. Steering behaviour is influenced by these choices, together with tyre properties, chassis balance, and circuit layout.

A closed-loop process then links expected acceleration forces (lateral, longitudinal, and combined) with the suspension and tyre models. Accurate acceleration data is crucial, as it defines the loads at the tyre contact patches, spring deflection, and suspension motion. If earlier setups were sub-optimal, the recorded track data will not reflect the car's maximum performance potential, limiting the ability to refine shop settings for optimal load transfer.

The final setup must combine appropriate rates (spring, damper, and anti-roll bar, together with motion ratios), kinematics (camber gain, jacking, and anti-geometry), and ride height targets. A chassis is designed to operate at a prescribed dynamic ride height, which influences pitch, roll, and suspension motion, and consequently camber gain/loss, bump steer, roll steer, and aerodynamics.

Ultimately, spring, anti-roll bar, and damper rates—together with their respective motion ratios—govern the balance between static and dynamic contact patch loads. For example, replacing a spring with a stiffer unit will not affect static loads, but it will alter dynamic load transfer across all four corners. Similarly, changes in anti-jacking geometry may have limited influence on static balance but can significantly affect dynamic load transfer and cornering loads. In this sense, dynamic corner load can be expressed as the sum of static corner load and inertial forces, meaning that dynamic performance always begins with the static setup as its baseline.

6.1.1 Data Correlation with final VI-CRT Vehicle Model vs Real Data

For starters the running model and the one that uses all the data from the car had this result against the real data this shows the resemblance in corner entry and exit as well as top speeds and minimum. The vehicle model does deviate at some points too like at the last corner of the track where it needs to apply the brakes for a brief moment to regain traction, but the upcoming corner which would be the first corner it takes it as if it were the real car.

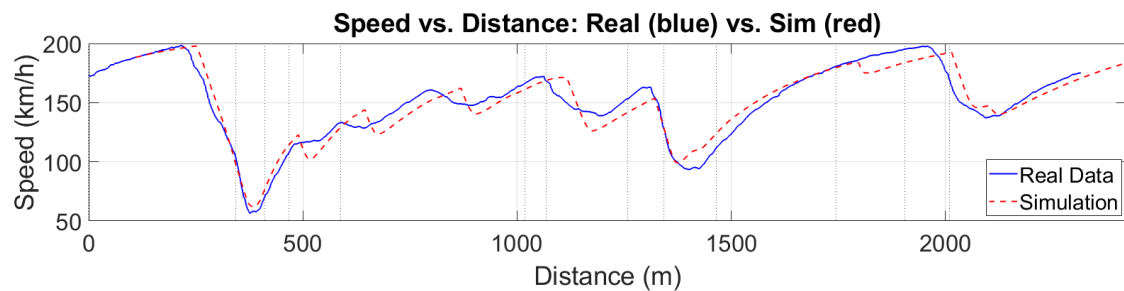


Figure 102 Speed vs Distance Graph VI-CRT Simulation vs Real Data

When talking about forces acting on the car it looks like the vehicle model from VI-CRT does behave accordingly to what the real data shows too.

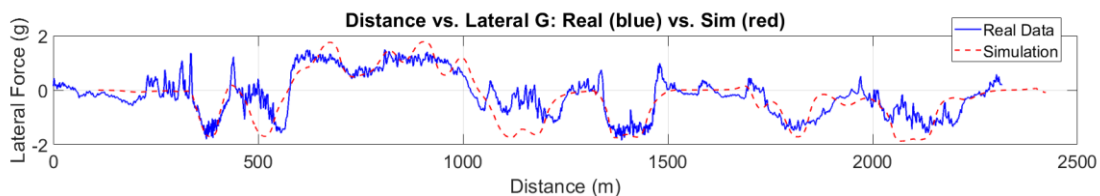


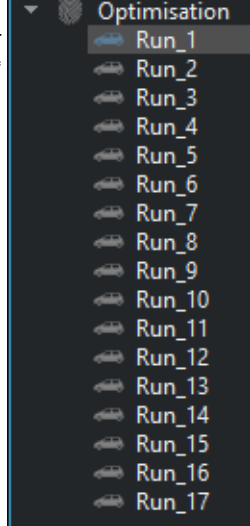
Figure 103 Lateral G Force vs Distance Graph VI-CRT Simulation vs Real Data

6.1.2 VI-CRT Set Up and run for the Optimization data

As now is known the vehicle model selected to do the optimization the study will investigate 3 variables which will be, Front Damper Scale Factor, Rear Damper Scale Factor, and Front and Rear Damper Angle.

Table 18 DOE Box-Behnken Runs | 3 Variables - FDS - RDS - Camber

Run	x1 FDS	x2 RDS	x3 Camber	FDS scale	RDS scale	Front and Rear Camber deg
1	0	0	0	1	1	-1
2	1	0	1	1.2	1	1
3	-1	-1	0	0.8	0.8	-1
4	0	0	0	1	1	-1
5	0	-1	-1	1	0.8	-3
6	0	0	0	1	1	-1
7	0	0	0	1	1	-1
8	1	-1	0	1.2	0.8	-1
9	0	-1	1	1	0.8	1
10	1	0	-1	1.2	1	-3
11	-1	0	1	0.8	1	1
12	0	1	-1	1	1.2	-3
13	1	1	0	1.2	1.2	-1
14	-1	1	0	0.8	1.2	-1
15	0	1	1	1	1.2	1
16	0	0	0	1	1	-1
17	-1	0	-1	0.8	1	-3



To make a simpler and streamline process, all the vehicle models were put together in one single fingerprint as the initial ideas was to run all of them separately however that idea was out of the question as it would have been hard to identify quickly from the software were the changes in performance occur throughout the lap.

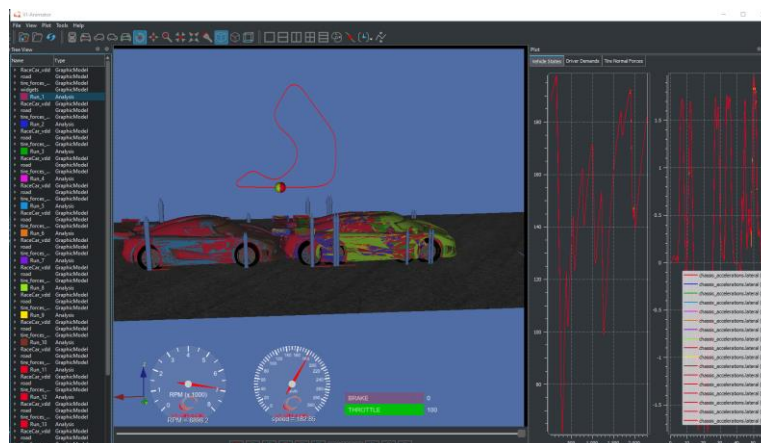


Figure 104 Visual Representation of VI-Animation showing the 17 runs used for the BBD DOE

As it is possible to observe there are actual changes in performance due to the changes in the variables set previously which is a positive sign meaning the vehicle model responds to variable change which was also proved in the sensitivity test -validation section #6.

6.2 MATLAB Box-Behnken DOE Visualization

The experimental design was implemented in MATLAB using the inbuilt `bbdesign` function, which generates Box–Behnken designs. For n factors, `bbdesign(n)` returns a numeric matrix containing the factor levels for all runs in the design, scaled between -1 and 1 . The resulting matrix is of size $m \times n$, where m is the number of design points and n the number of factors. This ensures a systematic and balanced exploration of the design space while minimising the total number of runs compared to a full factorial design.

In this study, the three factors investigated were front damper scale, rear damper scale, and camber angle. These parameters were chosen due to their strong influence on tyre load distribution, grip utilisation, and ultimately lap time performance. Each Box–Behnken run corresponded to a unique configuration of these variables, with the performance metric being the simulated lap time obtained from VI-CarRealTime.

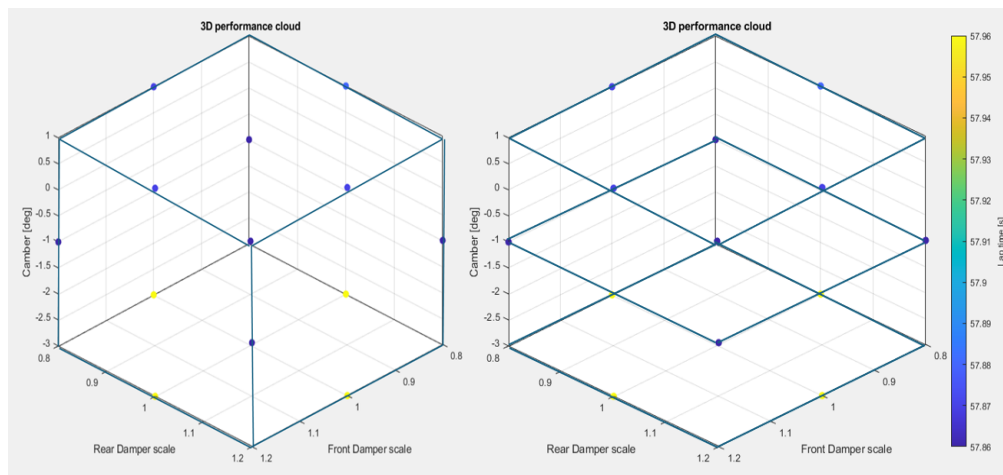


Figure 105 3D Performance Cloud - 3D Cloud Performance - 3 Variable Coordinate - Cube Representation

The colour scale in Figure 105 indicates lap time variation, with regions of lower lap times identifying potentially optimal setups.

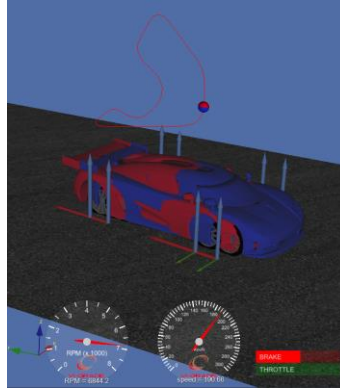


Figure 106 VI-Animator Red Car Final Vehicle Model vs Blue Car Theoretical Best Car (at the last corner of the lap)

In figure 106 the optimal case do proceed to be faster just before the last corner of the lap having a marginal 0.01 seconds of advantage.

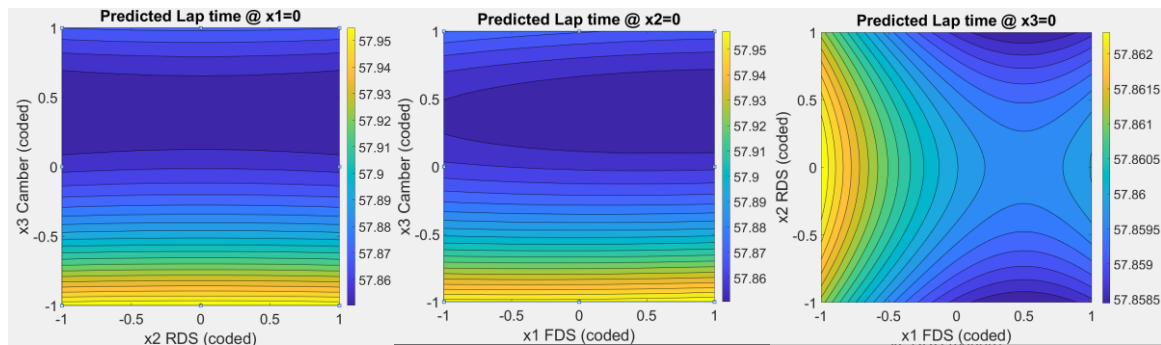


Figure 107 2D Sensitivity representation of the Variables on the Lap time

The 2D contour plots (Figure 107) and 3D surfaces (Figure 108) show how lap time changes with front damper scale, rear damper scale, and camber. Camber has the most consistent effect, with more negative values reducing lap time. The damper scales interact non-linearly, meaning their influence must be balanced rather than maximised individually. These results highlight that the optimal lap time comes from the combined tuning of camber and both damper scales, confirming the value of the Box–Behnken approach.

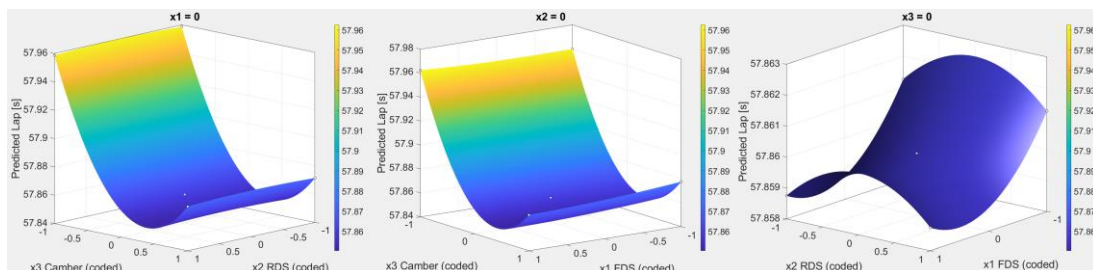


Figure 108 3D Sensitivity presentation of the Variables on the Lap Time

The regression analysis shows that camber (x_3) is the dominant factor influencing lap time, with a highly significant non-linear effect (x_3^2). The dampers on their own (x_1 , x_2) have limited impact, but their interactions with camber ($x_1:x_3$ and $x_2:x_3$) are significant, meaning that optimal lap time depends on tuning camber together with damper settings rather than in isolation. Overall, camber drives the largest performance gains, while damper effects are secondary but important in combination.

Linear regression model:
 $Y \sim 1 + x_1 \cdot x_2 + x_1 \cdot x_3 + x_2 \cdot x_3 + x_1^2 + x_2^2 + x_3^2$

Estimated Coefficients:

	Estimate	SE	tStat	pValue
(Intercept)	57.86	0.00084515	68461	3.747e-32
x_1	-0.00125	0.00066815	-1.8708	0.10355
x_2	-5.5199e-19	0.00066815	-8.2614e-16	1
x_3	-0.04375	0.00066815	-65.479	5.0916e-11
$x_1:x_2$	-3.5528e-15	0.00094491	-3.7599e-12	1
$x_1:x_3$	-0.0025	0.00094491	-2.6458	0.033146
$x_2:x_3$	-1.3045e-15	0.00094491	-1.3806e-12	1
x_1^2	0.00125	0.00092099	1.3572	0.21684
x_2^2	-0.00125	0.00092099	-1.3572	0.21684
x_3^2	0.05625	0.00092099	61.076	8.2824e-11

Figure 109 Linear Regression Model Variables and their coefficients - Box Behnken Method

As shown in the previous figures the effect of the dampers are not as critical to the lap time but to the handling of the car. In Figure 110 it is shown the estimated value in terms of lap time of the variables x_1 being the front damper, x_2 rear damper and x_3 camber angle. The results were impressive having that the camber angle does affect way more than the damper rates throughout this study.

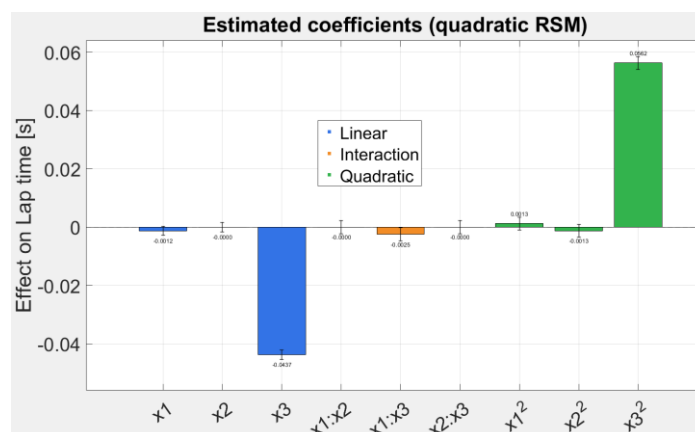


Figure 110 Estimated Coefficients and their Influence on the Lap Time

6.3 Model Feasibility

In terms of feasibility, the Box-Behnken Design predicted an optimal vehicle setup with a lap time improvement of approximately 0.01 s compared to the baseline. While this gain is marginal, it still represents a step toward refinement. Interestingly, when tested in simulation, the theoretical best setup did not outperform the Final Vehicle Model (FVM) along the same path—finishing slightly slower at 57.87 s compared to 57.86 s.

```
Predicted optimum (coded): [0.900 -1.000 0.400]
FDS=1.180 | RDS=0.800 | Camber=-0.200 deg | Pred Lap=57.8492 s
```

Figure 111 Predicted Optimum (Coded from Optimal Values)

This small discrepancy highlights an important point: the best theoretical model (BTM) derived from the design of experiments does not always translate directly into a performance gain in practice. However, this is an expected outcome in early optimisation stages. What is significant is the process itself, which demonstrates the capability of the methodology to identify trends and refine the model.

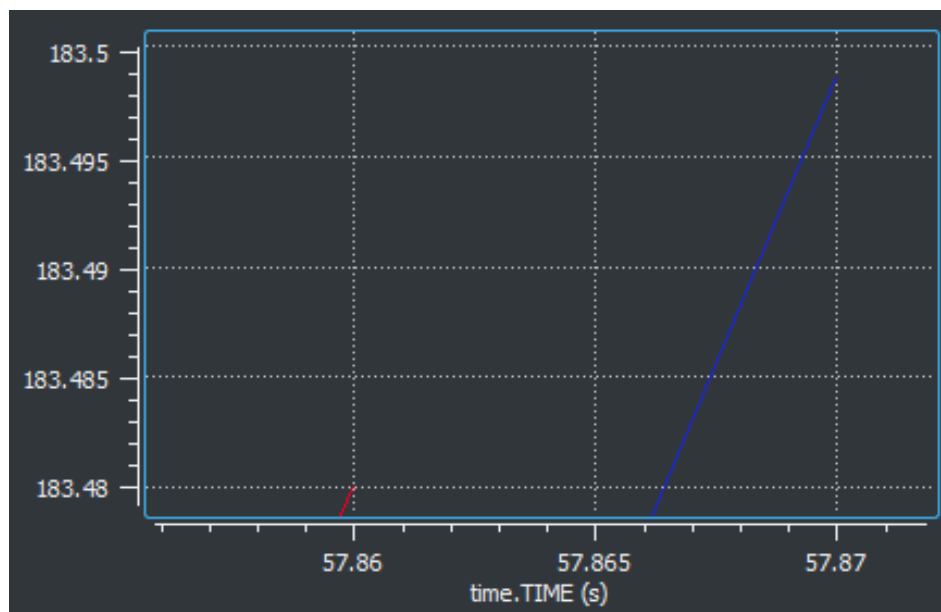


Figure 112 Final Vehicle model (Red) vs Theoretical Best using Box-Behnken Design of Experiment (Blue)

As shown in Figure 112, the difference is negligible, yet it validates that the Box-Behnken framework can be used to continuously improve model accuracy. Furthermore, when both the BTM and FVM follow an alternative racing line, the BTM begins to show its potential, suggesting that optimisation can unlock performance under different track conditions or driver strategies.

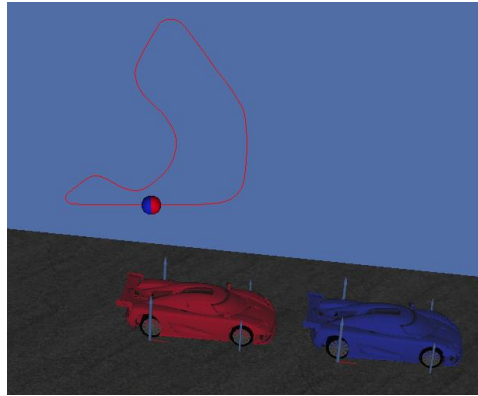


Figure 113 Using a slightly different racing line for both vehicle models

With this result, it is proven that the BTM can indeed perform better, but its effectiveness depends heavily on the circumstances and the chosen racing line. As is well known in motorsport, not every car delivers its best performance with the same setup; some drivers prefer a car with a tendency toward understeer, while others extract more from a car with slight oversteer characteristics. In this case, when the racing line was altered, the BTM finished more than a car length ahead, translating into a 0.11 s advantage. This difference is substantial in racing terms and demonstrates that the optimised setup is more suitable for a different line, where it delivers a clear performance gain.

7. Conclusion

This research set out not only to create a Driver-Out-of-the-Loop (DOL) simulation model of the MCR race car but to demonstrate the critical role of simulation-driven optimisation in motorsport engineering. The process of building and validating the model made clear that even the smallest detail whether in tyres, dampers, springs, or aerodynamics can alter the behaviour, stability, and performance of the vehicle.

The findings highlight that chassis and tyre performance are the result of countless interconnected variables. In practice, there are millions of potential combinations that could define the “ideal” setup. While external influences such as track surface, weather, and driver style were not considered in this study, the results show that even in controlled conditions, optimisation is far from simple.

By applying a structured Box-Behnken Design, this project isolated three key factors—front damper scaling, rear damper scaling, and camber angle—and proved that targeted optimisation can deliver measurable gains. Perhaps most importantly, the study revealed that camber, often underestimated compared to damping, had the most significant effect on lap time. This underlines that the path to performance is not always intuitive, and only systematic investigation can uncover the true priorities in vehicle setup.

Ultimately, this research proves the feasibility and value of simulation as a tool for guiding engineering decisions in motorsport. It demonstrates that with careful model creation, correlation, and the use of advanced design of experiments techniques, engineers can not only replicate but also improve upon real-world performance. The conclusions reached here provide a validated foundation for optimisation, showing that the integration of simulation and experimental methods is essential to extract the maximum potential from a race car.

8. Further Research

This study demonstrated that simulation-based optimisation can reliably highlight key factors affecting lap time and handling balance. However, the methodology can be extended in several directions to improve both the fidelity of the vehicle model and the breadth of the optimisation framework.

A first area of expansion is the inclusion of new kinematic factors. While this work focused on damper scaling and camber, future studies could evaluate toe settings (front and rear) in combination with camber to investigate how alignment changes influence yaw stability, understeer/oversteer gradients, and lap time. By applying the Box–Behnken methodology to these variables, it would be possible to quantify their interactions and build a more complete understanding of suspension kinematics.

Another promising area lies in exploring rate-package optimisation. Testing springs, dampers, and anti-roll bars together as three factors would provide valuable insights into how load transfer management and ride balance affect tyre utilisation. With the existing VI-CRT model and MATLAB code developed in this study, it is already possible to scale these parameters systematically and evaluate their combined effect. This approach would reveal whether specific stiffness combinations generate more predictable handling or improve lap time consistency.

Further research should also move beyond single-objective optimisation. While lap time was the main performance metric here, future work must often balance competing goals such as tyre life, driver workload, and aerodynamic efficiency. Incorporating multi-objective optimisation techniques, such as Derringer–Suich desirability functions, would allow multiple responses to be optimised simultaneously. For instance, a setup could be selected not only for lap time but also for minimising steering rack forces or controlling tyre temperatures within their optimal operating window.

Robustness testing should be introduced to assess how sensitive optimised setups are to variations in external factors such as grip level, ambient conditions, or tyre pressure. Monte Carlo or Taguchi methods could be applied within the DOE framework to identify configurations that are not only fast but also consistent across a range of scenarios. In addition, path sensitivity could be explored by treating different racing lines as factors in the optimisation as it was done in the model comparison between the vehicle model and the optimized version. This would reflect the reality that driver style and trajectory can influence whether a setup performs at its best.

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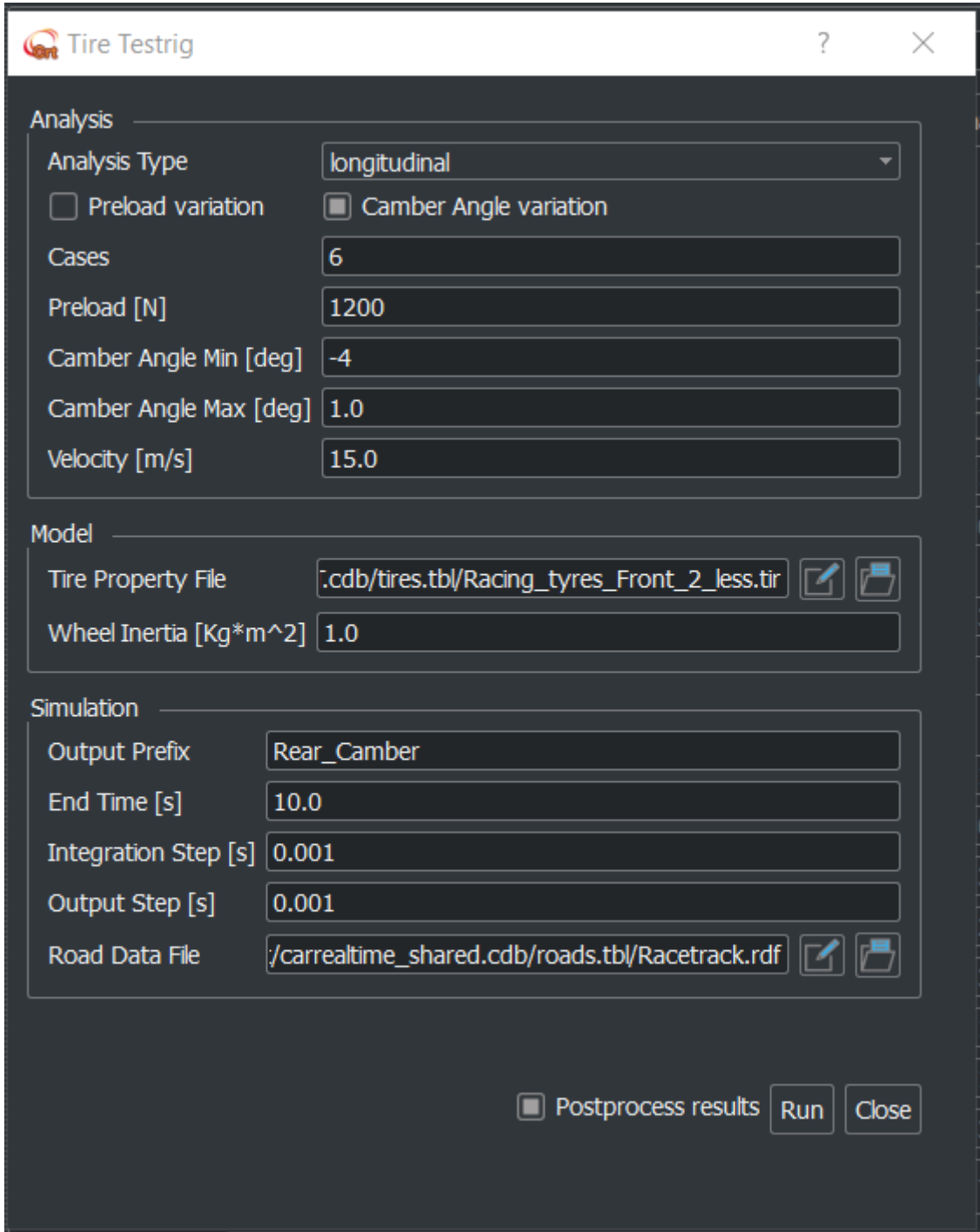
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10. Appendix

Appendix A

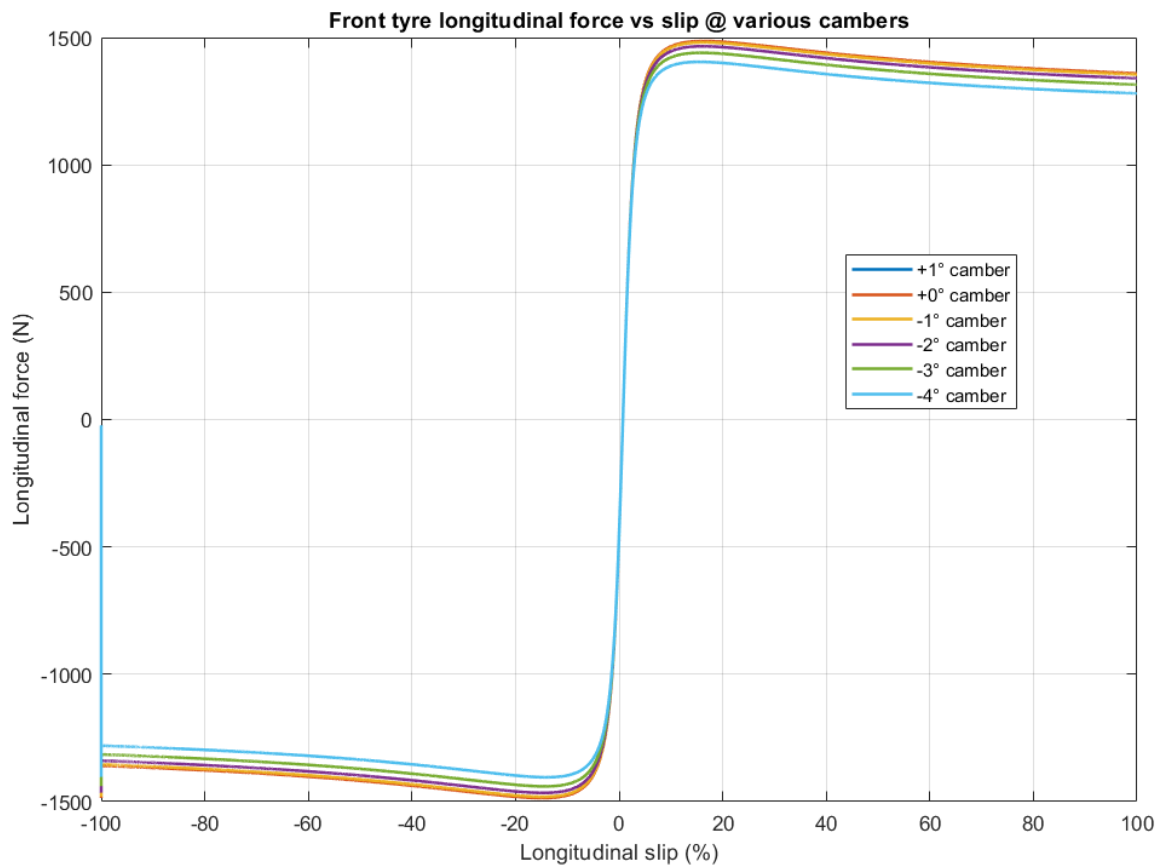
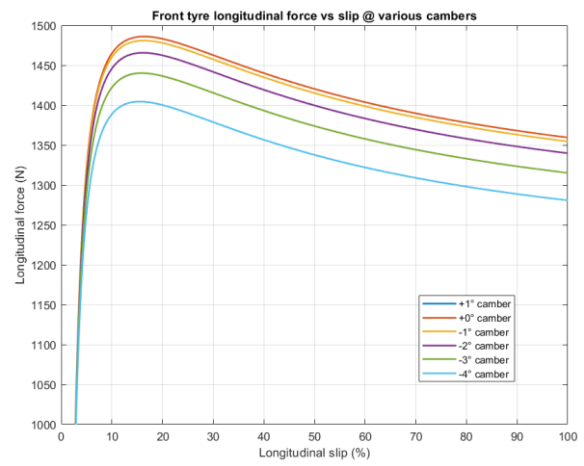
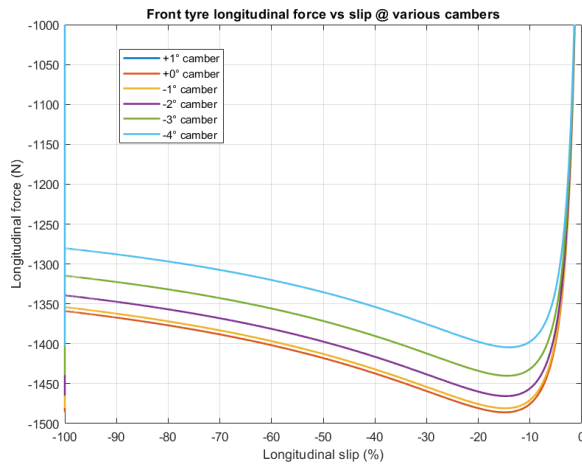
Longitudinal Response for camber change – Front Tyres



The screenshot shows the 'Tire Testrig' software interface with the following settings:

- Analysis**
 - Analysis Type: longitudinal
 - ☐ Preload variation
 - ☒ Camber Angle variation
 - Cases: 6
 - Preload [N]: 1200
 - Camber Angle Min [deg]: -4
 - Camber Angle Max [deg]: 1.0
 - Velocity [m/s]: 15.0
- Model**
 - Tire Property File: .cdb/tires.tbl/Racing_tyres_Front_2_less.tir
 - Wheel Inertia [Kg*m^2]: 1.0
- Simulation**
 - Output Prefix: Rear_Camber
 - End Time [s]: 10.0
 - Integration Step [s]: 0.001
 - Output Step [s]: 0.001
 - Road Data File: ./carrealtime_shared.cdb/roads.tbl/Racetrack.rdf

At the bottom, there is a checkbox for 'Postprocess results' and two buttons: 'Run' and 'Close'.



Tyre	Camber_deg	PeakLongForce_N	SlipAtPeak_deg
"Front tyre"	{ [1]}	1480.9	16.295
"Front tyre"	{ [0]}	1486	16.359
"Front tyre"	{ [-1]}	1480.9	16.295
"Front tyre"	{ [-2]}	1465.6	16.135
"Front tyre"	{ [-3]}	1440.1	15.879
"Front tyre"	{ [-4]}	1404.5	15.495

Longitudinal Response for camber change – Rear Tyres

 Tire Testrig

Analysis

Analysis Type: longitudinal
☐ Preload variation ☒ Camber Angle variation
 Cases: 6
 Preload [N]: 1750
 Camber Angle Min [deg]: -4
 Camber Angle Max [deg]: 1.0
 Velocity [m/s]: 15.0

Model

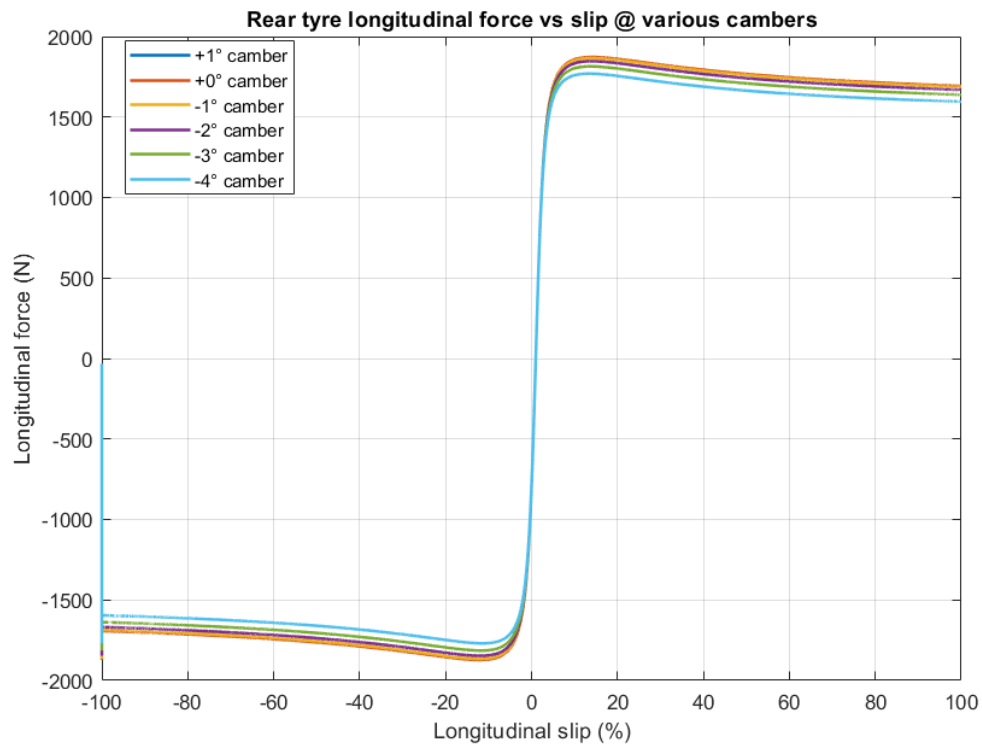
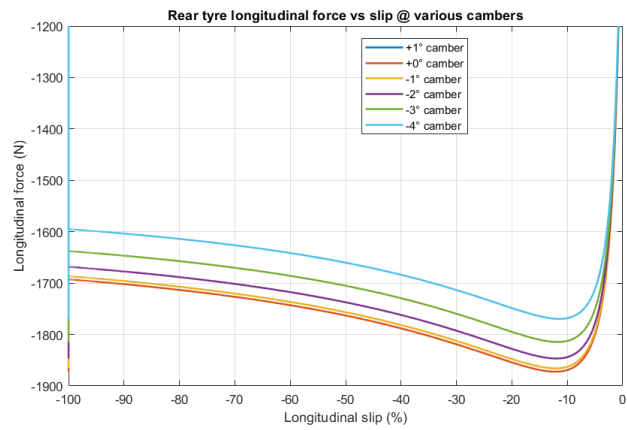
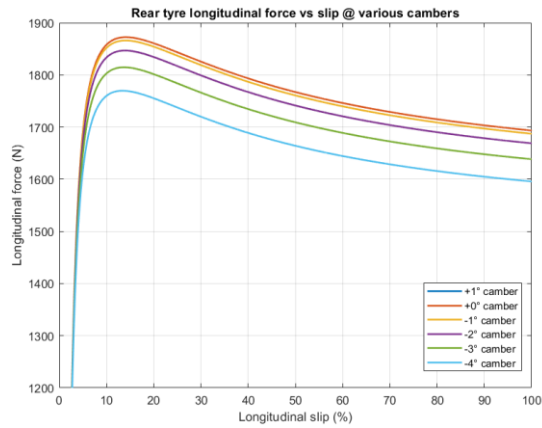
Tire Property File: I:\cdb\tires.tb\Racing_tyres_Rear_7_less.tir
 Wheel Inertia [Kg*m^2]: 1.0

Simulation

Output Prefix: Rear_Camber
 End Time [s]: 10.0
 Integration Step [s]: 0.001
 Output Step [s]: 0.001
 Road Data File: /carrealtime_shared.cdb/roads.tb/Racetrack.rdf

☐ Postprocess results

Run
 Close



Tyre	Camber_deg	PeakLongForce_N	SlipAtPeak_deg
"Rear tyre"	{ [1] }	1866	14.086
"Rear tyre"	{ [0] }	1872.4	14.118
"Rear tyre"	{ [-1] }	1866	14.086
"Rear tyre"	{ [-2] }	1846.7	13.926
"Rear tyre"	{ [-3] }	1814.6	13.702
"Rear tyre"	{ [-4] }	1769.7	13.413