University of Southern Queensland

Faculty of Engineering and Surveying

Suspension Development for a Short Circuit Racing Car

A dissertation submitted by

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Abstract

This report investigates the suspension parameters required to alter a modified mass produced car for use in the short circuit racing environment. The cornering force and handling characteristics of the car are to be altered to result in faster cornering speeds and resultantly reduce lap times.

The required research was undertaken and the necessary modifications were evaluated. The roll centre locations were found to have the largest effect on the cars weight transfer characteristics and balance, and hence the investigation into their location was undertaken in the interest of increasing their optimization. The fitment of front roll centre adjusters was analysed and the positive results justified their fitment to the front suspension. A Watts Linkage was designed and fitted to the rear suspension and the modification and adjustability of the rear roll centre was the result. The roll centres were effectively raised in the front and lowered in the rear to alter the weight transfer and jacking forces.

The race car was physically tested both before and after the modifications were completed to ensure that a fair ground for comparison was available. The differences in tyre temperatures, cornering g forces and the driver's evaluation of the modifications have resulted in positive outcomes for the cornering potential and handling parameters. The new roll centre locations have resulted in a car that is more neutral in its handling characteristics.

The modified roll centre locations have also introduced a new situation for the suspension development and further testing and suspension modification are recommended. Recommendations for supplementary investigations are included in the interest of further increasing the effectiveness of the performed modifications.

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ENG4111 Research Project Part 1 & ENG4112 Research Project Part 2

Limitations of Use

CERTIFICATION

I certify that the ideas, designs and experimental work, results, analyses and conclusions set out in this dissertation are entirely my own effort, except where otherwise indicated and acknowledged.

I further certify that the work is original and has not been previously submitted for assessment in any other course or institution, except where specifically stated.

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The completion of this report could not have been accomplished without the assistance of many personal. The application of the large number of required modifications could not have been performed without the help of the following people.

Beaurepaires Gatton has assisted in providing sponsorship for the development of the cars suspension. The use of their wheel aligning equipment throughout the suspension development has saved countless hours of measuring suspension parameters.

My father, Neville Nawratzki, is without a doubt the largest sponsor of the project and credit is definitely due for the work that he performed in the manufacturing and development of the suspension modifications. He also assisted with the resource requirements of the project and I hold the utmost gratitude for his involvement in this project.

Chris Snook supervised the project and his involvement throughout the project was much appreciated. The resource requirements which himself and the university have provided assisted greatly in the final dissertation.

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1. Introduction

In the current trend of modifying cars, many people have done so without the consideration on the handling characteristics of their modified vehicles. Such vehicles possess a safety hazard to all users, including persons both inside and outside the car. Larger more powerful engines add a considerable amount of weight to the front of the car and upset handling dramatically. Handling can be modified however to increase safety and enjoyment for all parties.

The aim of this project is to increase the cornering power and driver control of a short circuit racing car which contains an engine conversion. This is to be achieved through modifying and tuning the suspension setup to optimize cornering characteristics.

This report presents many issues which will need addressing throughout the project in order for it to be completed successfully. The background information for the project will be discussed, before the information sources and potential outcomes are reviewed. The relevant theory will be explored and related to the current situation and the methodology of the project will be presented. A risk assessment of the dangers likely to be encountered will be completed and resource requirements will be quantified. The modifications undertaken as part of the project will be explained and their results measured and compared to gauge the potential benefit. Conclusions on the effectiveness of the modifications will be discussed in both data form and drivers input. Suggestions will also be made as to areas which require further work and development.

1.1 Project Aims

This project seeks to decrease the current lap time of the race car. This is to be achieved through suspension modifications and tuning to increase cornering speed and driver control. The standard suspension setup is aimed towards comfort and is resultantly compromised in terms of the chassis's cornering power potential. The budget considerations present in a mass produced car have resulted in a car that has less than ideal suspension geometry and suspension tuning for a racing application. The available technology at the time of the cars creation is a limiting factor and further technological advancements have been made in terms of suspension design and tuning of the current setup. This project will seek to rectify these variables and increase the cornering power and driver control of the car.

1.2 Specific Objectives

The main objective of this project is to decrease the lap time. The lap time reduction is to be achieved through modifying the current suspension setup to increase the cornering power. Both front and rear suspension setups are not ideal for the racing application and as a result many modifications can be performed to increase the usage of the car in the racing field. These modifications will need to be categorised and prioritised. Due to the time and resource constraints within the project, all of the intended modifications will not be viable for completion. The modifications will therefore be performed depending on the expected benefit, financial situation and time input for the modification to be performed. The corner speed will be increased through modifying the suspension geometry to increase tyre contact with the road surface and alter the vertical weight applied to each tyre through lateral transfer. Suspension parameters such as springs, shock absorbers and anti-roll bars also affect the tuning of the suspension and will be briefly investigated to find their total system effects.

1.3 Background

1.3.1 Race Car

The Race Car to be used in the project is a 1974 Datsun 120Y. These cars were renowned for their slow acceleration and extremely compromised handling. They were an extremely low end budget car from the era which saw Japanese cars make a huge impact on the Australian, Europe and American markets. Although they were budget cars, they possessed good build quality and used extremely reliable components. The standard suspension on the 120Y is typical of the era with independent MacPherson struts in the front (fig. 1.3.1.1) and a solid leaf sprung rear end (fig. 1.3.1.2).

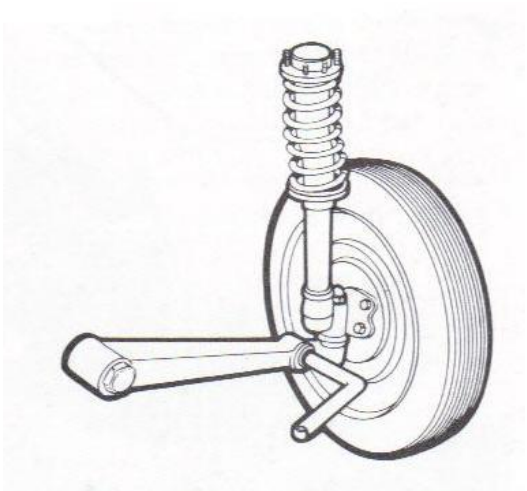

Fig 1.3.1.1 MacPherson Strut (F Puhn 1981, p30)

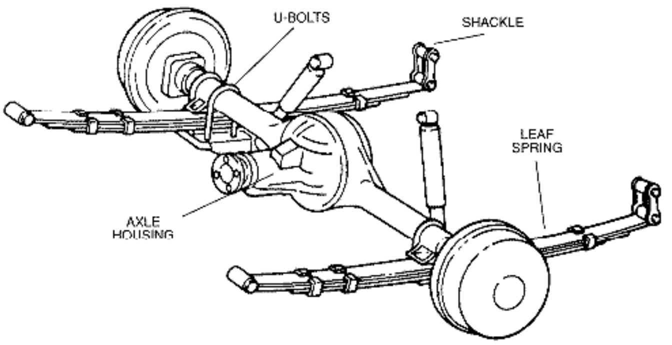

Fig 1.3.1.2 Solid Axle, Leaf Spring Suspension (Monroe Website)

The modification process has been consistently adapted over the previous 3 years. The car has been in the possession of the owner since the modification process began. In this manner it can be easily seen if modifications effectively make the car faster and easier to drive. All the modifications so far have decreased the lap time and resulted in a faster and more predictable handling car. The race car is a continually changing organism with many modifications made since it rolled of the factory floor over 35 years ago.

In its current state it is powered by a 3 litre, 6 cylinder, Nissan RB30e engine producing approximately 126 rear wheel horsepower. In a racing application it may sound like a small amount of horsepower, although the torque increase achieved has resulted in a car which accelerates quite rapidly. The standard 1.2 litre, 4 cylinder Nissan engine was also physically smaller than the current engine and a custom firewall, transmission tunnel and engine mounts were fabricated. The RB30e is also a lot heavier than the standard engine, and added a considerable amount of weight to the front of the car. The increased weight upsets the balance of the car, and suspension modifications to take into account the extra weight and torque produced are needed to be competitive in the racing class in which the car now competes. The gearbox is currently the standard RB30e close ratio 5 speed and the rear axle has also been replaced with a Hilux limited slip differential suitable to the increased torque and the racing application.

Fig 1.3.1.3 Photo showing the physical size of the engine replacement

The current suspension setup still utilises the standard arrangement of components, although having been modified to improve handling. The front suspension still consists of the standard MacPherson struts although the strut units themselves have been interchanged with R31 skyline units to allow for further bump travel with the decreased ride height. The skyline unit also offers a larger number of shock absorber selection options with valving more appropriate to the weight of the engine as the R31 was released with a RB30e engine.

The current setup has including the creation of more adjustable components as well as increasing the stiffness of the mountings. As can be seen by the following list of components present in the front suspension, little of the original geometry and suspension tuning remains.

- Adjustable height spring platforms set to the minimum height which allows clearance to the wheel and tyre
- Pedders sports rider shock absorbers, valving for r31, standard r31 length which is 50mm shorter than other commonly used datsun struts.
- 8kg/mm main spring, and 75lb/inch tender spring
- Adjustable camber top mount with spherical bearing
- R33 Skyline 4 pot callipers and 296mm slotted and vented rotors

- Lengthened lower control arms - 30mm and moved ball joint 5mm forward to increase positive castor. New ball joints fitted also.
- LJ torana radius rods
- Sway bar 26 mm in diameter

Fig. 1.3.1.4 Front Struts

Many of the components are adjustable and tuning is therefore made easier through simply changed components or settings.

The current rear suspension still utilises the standard leaf springs and solid axle arrangement. The standard springs are still present although their orientation has been altered to both lower the car and change the effective spring rate. In its current state there are 3 springs of decreasing length with the smallest spring reversed to pull against the others and lower the car. 2 small lowering blocks of 5 mm thick aluminium have also been used to further lower the rear of the car. The rear shock absorbers are standard 120Y replacement items of a gas construction and made in Australia to suit Australian roads. Caltracs have also been fitted to the rear suspension and axle to increase traction and reduce axle tramp.

The current setup has resulted in a fast car that can hold its own against many times more expensive cars on tight short circuits. The understeering nature however has always been an issue and is something that the project will aim to address. The cornering power of the car is to be increased by making the car more neutral in its handling and increasing the driver's control. The driver will remain the same throughout the testing process to ensure quality control. This also presents another tangible result basis for the assessment of the handling and control component.

1.3.2 Track Layout

The racing car needs to be setup to suit the type of tracks in which it will race on. Many different bitumen tracks are available in the South East Queensland area, all with their own different style and layout. The most common type of event in which the car is to compete is to be short circuit or sprint events. This includes hill climbs and limited lap track timed events. The racing is predominately a single car on the track at any time and the fastest time wins. That said, it is normally in the regulations that at least 75 percent of the runs be completed to be considered a contender for a position. There is also an event such as Stanthorpe, which contains 4 laps in which there are other cars on the track to indirectly race against. The total time for the 4 laps is the final result.

Many different track layouts are used, with the main one being closed off public roads. They are normally relatively smooth and hot mixed bitumen although the regular bumps and undulations are to be expected. They normally contain at least one chicane made up of large rubber witch's hats which contain a penalty if knocked over. The transient handling requirements for these chicanes may show when compared to other tracks that do not contain such characteristics.

1.3.3 Car Requirements

Racing cars have many different uses depending on the type of tracks raced on and the rules and regulations in which they must compete under. In the interest of this report the usage has been aimed at, but not limited to, the cams road racing rules. In terms of the suspension development phase, the cams rules are very open to the modification of all components of the car. The car competes in the Unregistered or Sports Sedan class (depending on the event regulations) and neither of these limits the car development in anyway. The only limiting factor is if the car was to be registered for road use at anytime. As this is not to be a consideration in the suspension development program, the options are left unrestricted. Due to the budget and time considerations the basic suspension type will remain the same, with modifications performed to increase the cornering power and drivability of the system.

1.4 Information Sources

Many different information sources are available to fulfil the requirements of the project. In order to perform the many different forms of suspension analysis, a thorough understanding of the principles which allow the vehicle dynamics to be altered and optimized must be known. The area of most information is in regards to race car dynamics. The basic information and principles allowing suspension tuning is easily found with many different researchers having covered the topic. The main suspension type focused on however is the independent double wishbone suspension as found in most high end racing cars. The Macpherson strut setup is common in cars, even to this day, and lots of information is present to its geometry and optimisation in the racing field. All aspects of its operations are covered in detail in almost all of the available literature.

The rear leaf spring suspension setup is however an older technology and its use in racing cars have been limited. In terms of circuit racing, the leaf spring rear end has always been removed in favour of a four link or similar suspension setup. Leaf springs are used mainly due to the cost advantage or load carrying capabilities associated with their simple arrangement. As a result the data obtained for rear suspension has limited leaf spring attributes when compared to other high end rear suspension setups. The leaf springs usage has however been somewhat useful in drag racing applications and hence applying these principles and some of the basic leaf spring principles may be sufficient enough to gain an acceptable understanding of the rear suspensions parameters and performance.

Many different methods of gaining information on the topic can be utilised. The main method is research books, such as those that attribute themselves to vehicle dynamics.

These are full of information and the real challenge is limiting what information is relevant to this exact project and situation. In order to limit the amount of factors that are addressed, there needs to be parameters which will not be altered, and only stated as being a consideration in the other variables. In order for the project to focus on only the important factors, the other parameters will remain constant. This will also keep the quality control as high as possible.

Other methods such as technical journals and handbooks may be used in the creation of the final product. The factors that they address are more specific to the application than the theory, and will be more helpful in terms of the actual modification process. Many other untraditional resources are also available including physical sources such as human knowledge. The art of chassis tuning is varied depending on the type of car and suspension setup. Many people may have been through similar development programs although without the depth of documentation. Talking to people in similar situations can unlock all sorts of tips and tricks to gain a better understanding of how your particular car works, or even more importantly, where it doesn't. Not all racers or developers are keen to share information, although many are eager to talk about their cars and a great deal of knowledge can be gained by talking to the right person.

1.5 Consequential Effects/Potential Outcomes

The benefits of the report are likely to increase the pool of knowledge that already exists in suspension development. In terms of racing development of the current setup there are no conclusive references. This report will increase the application based approach of people doing similar modifications. The actual data and analysis used is nothing new to the industry, although the application is specialized and therefore fits for a select application only. The modifications and results can be used by others to gain an insight into the viability of such modifications for their cars.

The current trend of putting large engines into small and often older cars has resulted in many poor handling cars that are good for only straight line acceleration. This project is aimed to attempt to prove that it is possible to get a reasonable handling car with this setup, even with the largely uneven weight distribution present. Other cars which have similar suspension systems can also benefit from the analysis. Cars from this era are cheap to buy and are becoming more highly supported by aftermarket parts suppliers due to the current surge of modified examples being driven on the streets and race tracks.

The results of this report can also be used in suspension development for new cars. Leaf springs are still used in modern high performance cars such as the turbo Falcon Ute series. The information gained from the project may be used to increase handling on such vehicles which use very similar rear suspension setups more than 3 decades later.

2. Literature Review

2.1 Slip Angle

Many different variables allow suspension tuning to be performed. The major physical characteristic which allows suspension systems to be tuned is tyre distortion, or slip angle. In essence, slip angle is the angle between the cars path and the direction the wheel is pointing (Puhn F, 1981). When cornering all contact between the road and the car must be performed through the tyres. The large side force during heavy cornering distorts the tyre sideways and results in an increasing slip angle. Cornering force increases with slip angle until a point, at which the slip angle has reached its maximum cornering force value. At this point the tyre will break away and lose traction with the surface, resulting in spinning and a decreasingly available cornering force. In terms of handling, the difference between front and rear slip angles will determine how a car will handle. The progressive nature of slip angle is what enables us to modify suspension to maximise cornering power.

In the figure 2.1.1 it is seen that the rear slip angle is larger than the front. This is due to the car in the figure containing a more rear weight biased setup. The larger the weight being placed on the tyre, the higher the slip angle that will be experience by that tyre. This is caused by the increased weight, causing more distortion in the tyre. The car in the diagram will result in oversteer due to this increased slip angle.

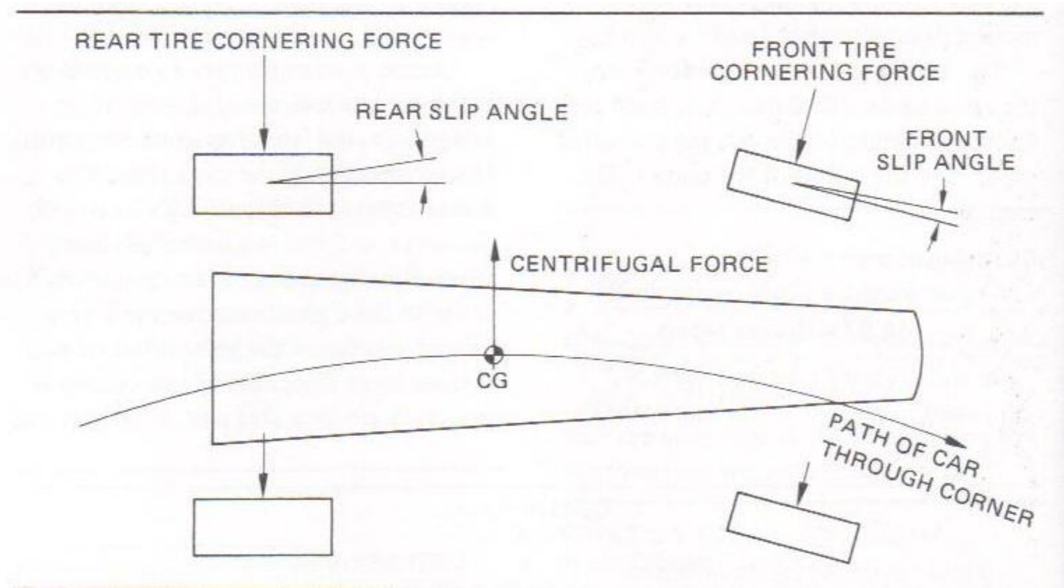

Fig. 2.1.1 Slip Angle (Puhn F 1981, p16)

Oversteer is a characteristic with which we define the handling of a certain car. In terms of the driver it can be defined as 'whether the front tyres reach the limit of cornering traction before or after the rear tyres do' (Smith C, 1978). If the front reaches its limit before the rear, this is called understeer as the car is steering less than intended. In the opposite case the rear reaches its limit before the front and results in oversteer as the car is turning more than intending. The ultimate for circuit racing is to have a car that is neutral, neither understeers nor oversteers, and results in all four wheels reaching their corning limits simultaneously.

In terms of the project a front heavy car is utilised. This presents the opposite situation to figure 2.1.1 and will result in as understeering condition through higher front slip angles. The natural tendency of understeering is the basis of the project, and the reason current handling is limited through understeering tendencies.

Another variable affecting tyre's grip is the coefficient of friction between the tyre and the road surface. The 'coefficient of friction varies with slip angle' (Smith, 1978). The coefficient increases with slip angle until the maximum value is achieved. This is the reason the cornering force increases with increasing slip angle until 'break away' occurs.

2.2 Lateral Weight Transfer

Weight transfer can be described as the movement of weight from one wheel to another due to forces acting on the car. Through a turn this phenomenon is known as lateral weight transfer. The centre of gravity of the car is important in understanding the lateral transfer of weight. The centre of gravity is the point at which the car would balance if picked up via a single point. This shows where the weight distribution is concentrated and how this weight will affect handling. Through a corner, there are centrifugal forces being generated through the tyres grip onto the road surface. This force takes a direction away from the centre of the corner and through the centre of gravity. 'Because the centre of gravity is some distance above the ground, weight is removed from the inside tyres and added to the outside tyres' (Puhn F, 1981). This is also affected by many other variables in the suspension design, although the basic weight transfer principle remains the same. The method and rate of transfer only changes, and this is what the designer modifies to change the weight transfer characteristics. As can be also be noticed, the lower the centre of gravity, the lower the weight transfer.

The increasing and decreasing weight on outside and inside tyres respectively results in different slip angles. Figure 2.2.1s how the slip angle and cornering force increases with increasing vertical load. The increase in vertical load present on the outside tyre will result in an increased slip angle. The inside tyre will see a reduction in weight and therefore will decrease slip angle and cornering force. The outside tyre is the one we are most concerned with as it does most of the corning due to uncontrolled weight transfer from the portion of centrifugal force. This force will always result in some weight transfer (regardless of suspension setup) and due to the nature of the force will never be eliminated. For this reason we are more concerned with the outside tyre condition. In this case the outside tyre has gained slip angle through increased vertical load. The nature of tyres results in a slight decrease in the available coefficient although the increase in vertical load overpowers these changes to result in more cornering force for the outside tyre. As shown by Smiths examples, lateral weight transfer while cornering will result in a lower total cornering potential while fore and aft transfer will result in more axle grip being available. As shown by Smith (1978), if a car has 500 pounds on each tyre and a coefficient of 1.35 then potential cornering forces generated are (1.35x500)x2 = 1350 lbs. If 100 lbs is added to each tyre the coefficient will decrease to 1.33 (calculated from graphs of tyre tests) the total traction potential has increased to (1.33x600)x2 = 1596 lbs, a gain in potential cornering power. If lateral weight transfer occurs a different situation is presented. In Smiths example, the front wheels have a vertical load of 400 lbs and a coefficient of 1.4. In the steady state condition, this offers (1.4x400)x2 = 1120 lbs of cornering force. If 80% of the weight is transferred to the outside wheel, not uncommon in racing cars, the potential is greatly reduced as is shown by the potential of both tyres. The outside gains grip with its increased vertical load equal to 400+(400x0.8) = 720 lbs while the inside only has 80 lbs vertical load. As can be seen from figure 2.2.1, the cornering forces generated are 936 lbs and 120 lbs, a total cornering force of 1056 lbs, a net reduction from the steady states potential cornering force.

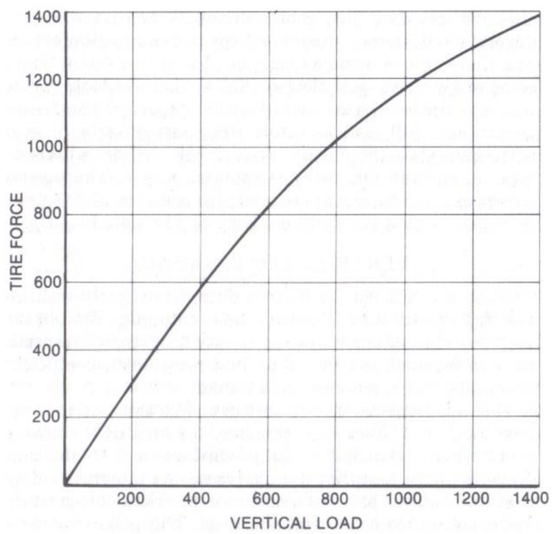

Fig. 2.2.1 Vertical Force vs Tyre Force (Cornering Potential) (Smith C 1978, p18)

2.3 Roll Centre and Axis

When a car rolls over in lateral weight transfer through turns, there is often a point at which the weight is effectively considered to roll around. This point is commonly an imaginary point in space, although some suspension systems have an actual suspension pivot which corresponds to this point. This point is called the roll centre. The height of this point above the ground is the contributing factor as it affects how much the centre of gravity affects the rolling amount or angle of roll. This point is therefore important when comparing suspension systems as it affects the weight transfer rate. The lower this point is, the more weight transfer that will exist due to a larger moment arm existing between the roll centre and the centre of gravity. The benefit of a lower roll centre is the reduction of jacking forces. Jacking forces exist when the roll centre is above ground level, and resultantly 'the line of action between the tyre contact patch and the roll centre will be inclined upwards towards the vehicle centre line' (Smith C 1978, p38). The tyre side force will resultantly develop a vertical component and result in jacking of the body. This jacking will result in an increase in the centre of gravity height and suspension which is operating in the droop region of its camber curve. The reduction in camber and vertical force decreases grip if high jacking forces are evident. Figure 2.3.1 shows the application of jacking forces and the origination of the force.

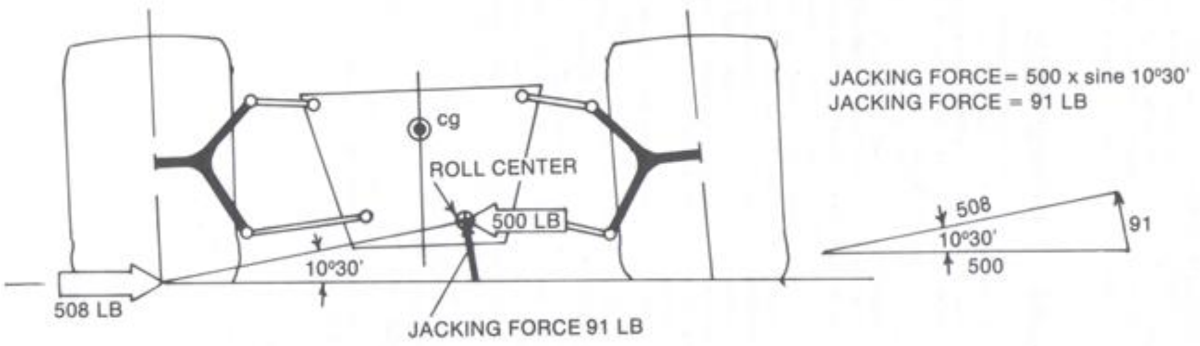

Fig. 2.3.1 Roll Centre Jacking Forces (Smith C 1978, p39)

The optimum roll centre is a compromise between weight transfer due to moment arm length and jacking forces. Milliken (1995) states that 'roll centre heights are trading off the relative effects of the rolling (centre of gravity) and nonrolling (jacking) moments.'

As can be seen in figure 3.2.2, the front roll centre is measured by taking into consideration the suspension setup. The front MacPherson Struts are independent in nature and hence have a low roll centre like other similar independent suspension systems. In terms of the project, this roll centre height is to be measured and altered accordingly. Independent suspension systems also result in lateral displacement of the roll centre under roll and similar suspension movements. The two dimensional motion of the roll centre results in an increasingly complicated calculation, and hence computational methods are often used for independent suspension systems analysis.

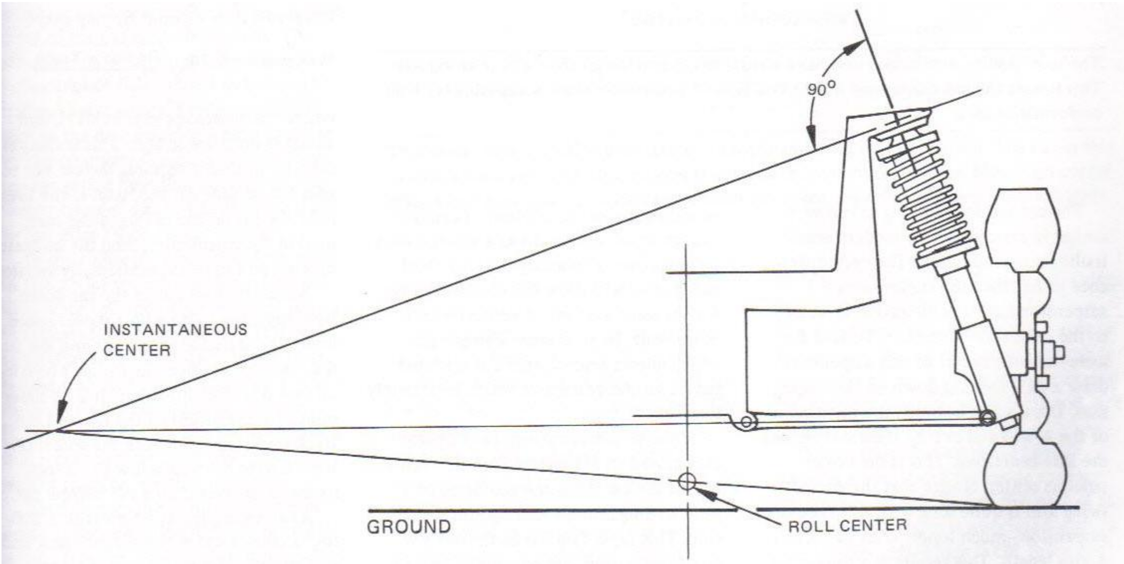

Fig. 2.3.2 Front Roll Centre (Puhn F 1981, p37)

Another factor affecting suspension operation is the instantaneous centre as shown in figure 2.3.2. The instantaneous centre of independent suspension systems addresses the progress of the wheel as it travels through its vertical suspension movement. The wheel however does not travel around the roll centre and hence has its own rotation point. ‘When a wheel has been tilted from vertical as a result of a bump or suspension movement, we can find some point about which the wheel could have rotated to assume the tilted position’. (Puhn F, 1981) This shows how the suspensions setting will change with height changes. No instantaneous centre exists for the rear suspension due to the solid rear axle, ensuring that both wheels are always vertical to the road surface.

Both the front and rear suspension systems have their own roll centres, which may be at different heights depending on the suspension setups employed. The imaginary axis which joins these two roll centres is known as the roll axis. This effectively shows how the body rolls through a turn.

The rear roll centre height is also important to this project and hence needs to be measured and adjusted accordingly. As seen below the roll centre for simple Hotchkiss drive rear ends, as in this project, are limited in their roll centre adjustment and height. Axle locating devices change the roll centre height and result in a lower roll centre location. The balance between the moment arm and the jacking forces can be altered to change the handling characteristics of the race car.

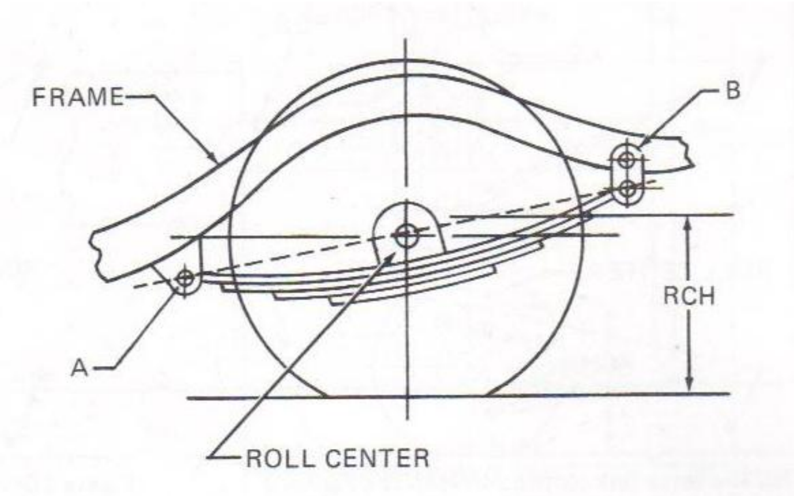

Fig. 2.3.3 Rear Roll Centre (Puhn F 1981, p33)

2.4 Camber

Camber is measured as the angle between the plane of the tyre and vertical. As can be seen in figure 2.4.1, camber can be either positive or negative. Positive camber is when the wheel leans away from the body of the car. Positive camber is undesirable in all racing applications and is always attempted to be eliminated. Negative camber as seen below is the situation where the wheel is leaning into the car. This is more ideal in racing applications and is used in varying amounts by different race cars.

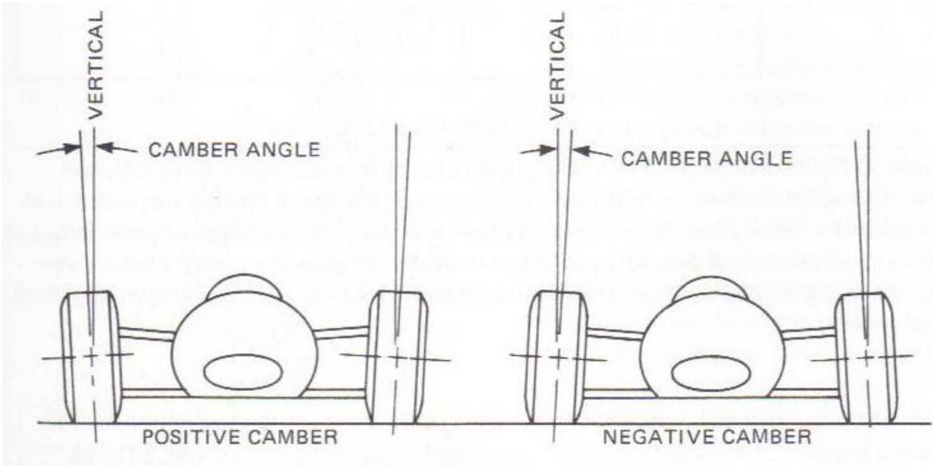

Fig. 2.4.1 Camber Angle (Puhn F 1981, p20)

Negative camber is normally set statically, while the car is not side loaded. Camber is however a dynamic setting and is set up to ensure the tyre remains in contact with the surface during weight transfer and subsequent roll through a turn. Negative values are chosen for this reason, and ensure that once rolled, the tyre will flatten out to near vertical. This maintains full tyre contact during hard cornering.

The amount of roll stiffness will determine how much static camber is required. If low roll stiffness is present, a higher static camber angle must be used to ensure chamber remains negative while cornering. Likewise, high roll stiffness needs lower static values. This is however dependant on the grip of the surface. If surfaces with low grip levels are used, the camber must be decreased as the actual roll of the car is decreased through less cornering force being available. Camber is set according to the roll stiffness, vertical camber change through suspension travel, and grip of the available surface to try and result in no positive camber situations and maximum traction while cornering. Rear camber is always at zero with a solid rear axle, and hence no adjustments are possible or required.

Camber thrust is another phenomenon which increases cornering potential and is the reason chamber angles are maintained at negative values. The cornering force varies with camber angles, and the maximum coefficient will be achieved at some small negative angle. 'This is due to the 'camber thrust' caused by the straightening out of the arc of the contact patch as the tread of a cambered tyre rolls over the ground. If a tyre is cambered in the negative sense, this force acts in the direction of the centre of curvature and increases cornering power.' (Smith C 1978, p.18) Wide tyres however have limited affects due to camber as the outside edge may result in a substantial tyre contact reduction. In the project cars case, the tyres are of a reasonably small width and these affects of camber are not likely to be an issue. Figure 2.4.2 shows the tyres coefficient of friction for each angle of camber of a given tyre. This data is specific for each tyre and hence the camber thrust curve will be different for each tyre.

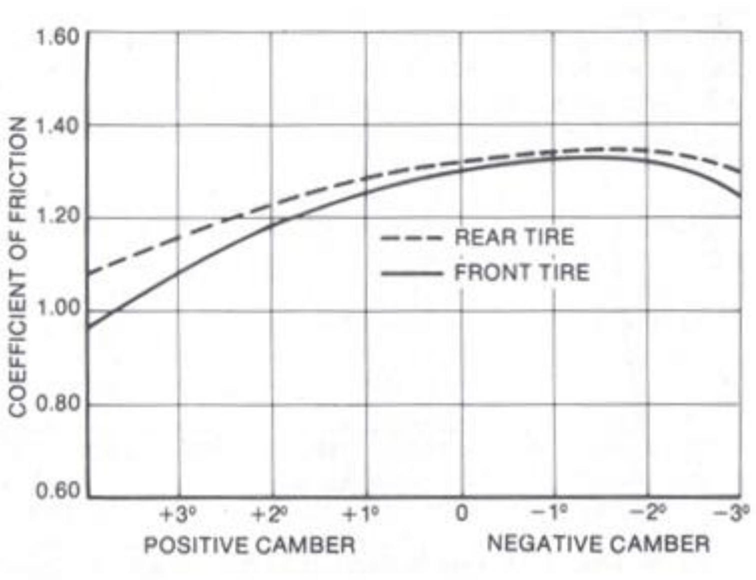

Fig. 2.4.2 Camber Thrust (Smith C 1978, p18)

2.5 Castor

Castor is one of the many different settings in which can be modified to alter the handling of a car. 'Castor is the angle made by a line between the upper and lower steering pivots and a vertical reference line'. (Puhn F, 1981) In race cars it is important that this line is always 'ahead of the tyre contact patch' as to 'promote straight line stability and provide steering feel to the driver'. (Smith C, 1975) This is known as positive castor and is generally used in light cars with small tyres, such as the car used for the project. Positive castor increases straight line stability and offers a self centring effect due to the side force generated during turning. 'This side force moves the tyre contact point to the left or right of the direction of travel and tends to return the wheel to the direction the car is travelling' (Puhn F, 1981). Castor also affects wheel camber during cornering, with large caster angles resulting in more camber with increased steering input. The front suspensions castor setting is found by the angle of the strut housing as it effectively contains the upper and lower ball joint as shown in figure 2.5.1.

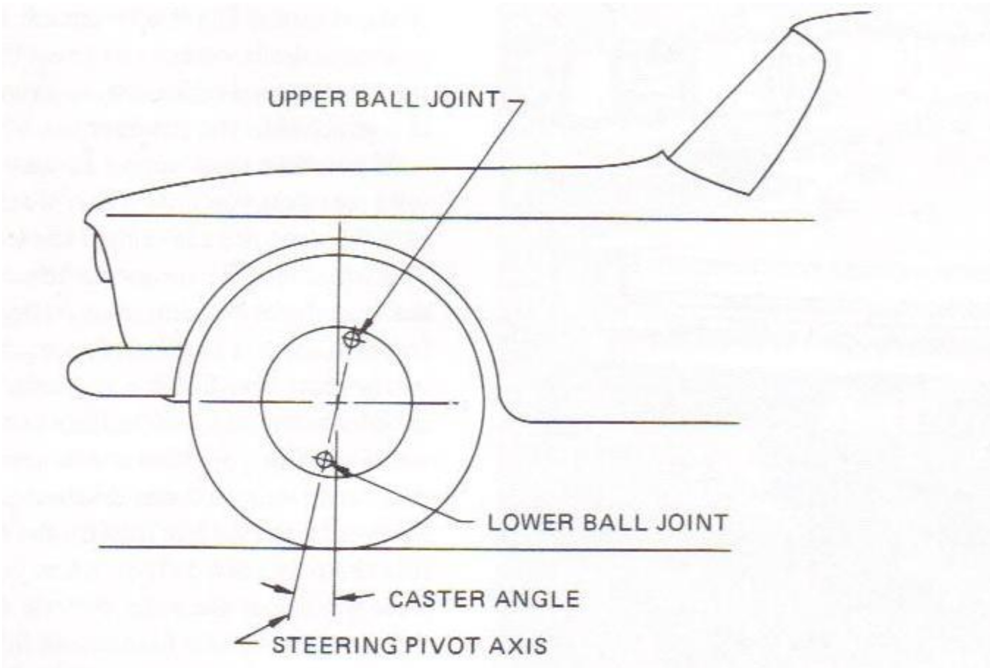

Fig. 2.5.1 Castor Angle (Puhn F 1981, p72)

2.6 Toe

Toe in or out is the angle or measurement of the wheels in comparison to the straight ahead condition. Toe in as seen in the figure 2.6.1 results in both wheels facing slightly towards each other. This is a stable condition, as each wheel will try to centre the car if turned or bumped by road undulations. Toe out as seen in figure 2.6.1 is when each wheel is facing apart slightly. This is an unstable condition that encourages the car to turn when a steering input is seen.

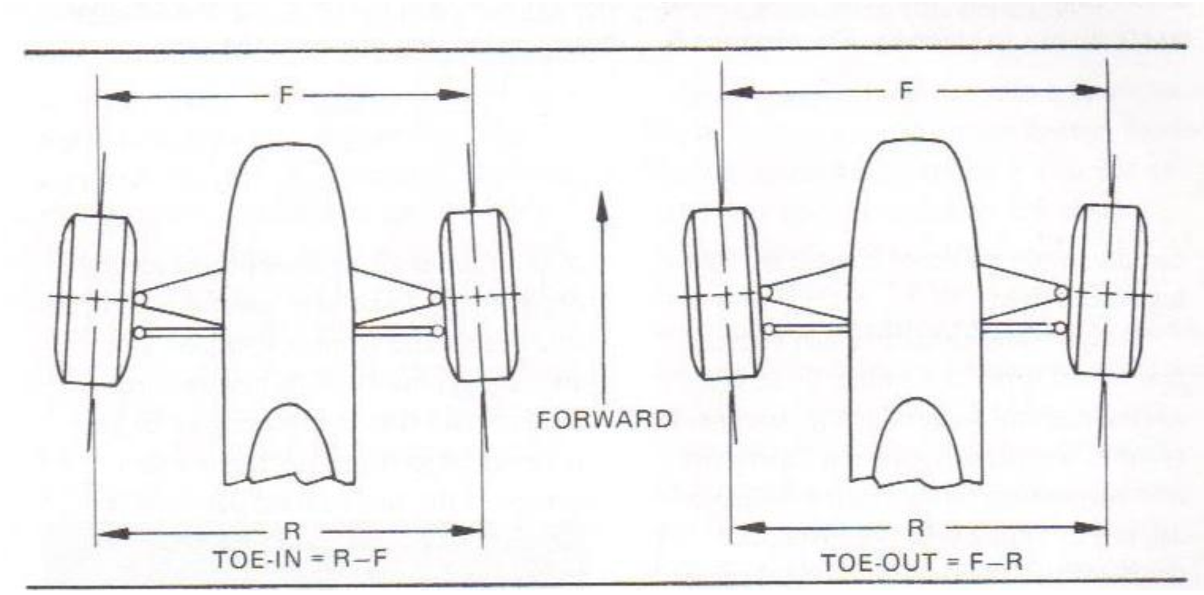

Fig. 2.6.1 Toe (Puhn F 1981, p23)

The project requirements have aimed for a car which understeers less, and as a result a higher amount of toe out has been used. This does have limits however due to scrubbing taking place during straight line driving. Too much toe in any direction will lower straight line speed, however very small adjustments are required before a handling altercation is noticed. This does introduce a small amount of tyre slip angle during operation and make the car less stable. This will result in initial oversteer for toe out settings as used in the project race car. This is used to try and alleviate the understeering tendency of the front weight biased chassis. Rear toe is not an issue in this project due to the solid rear axle.

2.7 Ackerman

Ackerman's principle is another factor affecting dynamic toe settings. In Smiths long career with racing cars he found that toe out helped in all situations to reduce understeer. The Ackerman's principle is simply designing the steering arms and required hardware to ensure that once steering was altered from straight, the toe out setting would increase with higher steering angle inputs. This is stated by Smith (1975) to increases the slip angle on the inside front tyre and increases cornering force. Ackerman's principle is hard to alter and outside the scope of this project, although its affects on the racing car need to be understood when looking at the steering arrangement.

2.8 Bump Steer

'Bump steer occurs when one or more wheels move up or down and the toe-in (or out) of that wheel changes' (Smith C, 1975). The effects of this toe change will drastically change the handling of the car. Through cornering, the suspension will compress on the outside and expand on the inside. If bump steer is evident, the toe setting will change while cornering, greatly affecting the handling and most importantly the predictability of the race car. If bumps are navigated, the toe change from bump steer will result in an unstable car due to this toe change. Any unwanted change in toe must be eliminated to improve driver control.

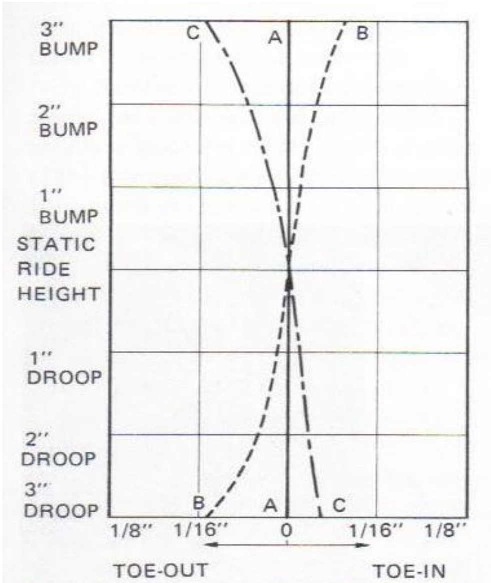

Fig. 2.8.1 Bump Steer (Puhn F 1981, p90)

Suspension setups each use different steering linkage arrangements depending on budget considerations and available technology. Older more budget oriented systems have some bump steer built in to them, although it is possible to eliminate this tendency. Modified and lowered suspension systems are also more prone to developing bump steer. Most cars that are lowered significantly required their bump steer to be adjusted or reset to as close to zero as can be achieved.

Most bump steer adjustments are made by changing the height of the tie rod (the steering arm which connects the steering rod to the strut or wheel bearing carrier). Many different bump steer characteristics can be present as can be seen in figure 2.8.1. Only A however is ideal, and the steering setup should be adjusted to result in as similar a bump steer graph as is possible. The method of changing the bump steer attributes is usually by using spherical rod ends on the steering connections (tie rods) and spacing these vertically according to the bump graph produced. This method has limited adjustment due to the nature of the moment being placed in the locating bolt through the bearing. The additional method of gaining further adjustment is by heating and bending the steering arm accordingly. This is however a critical component and only trained professionals should be used during the modification of steering components. The modification of steering arms will also affect the Ackerman of the steering system, and hence the effect on this value must also be included in the modification analysis.

2.9 Roll Steer

Roll steer is very similar to bump steer in that it results in a change of the toe setting with changing vertical wheel height. Roll steer does however result from the steering of the rear wheels due to rolling of the chassis from the centrifugal cornering force. As portrayed in figure 2.9.1, the condition becomes a rear steering situation if both rear wheels travel in opposite toe directions through a corner. With independent suspension if it 'is such that each rear wheel toes in in bump and out in rebound, then as the car rolls, both rear wheels will point into the centre of the corner' (Smith C, 1975). This is roll understeer as seen in figure 2.9.1 and presents the situation found in most production cars in the interest of car safety through predictability. Roll oversteer is when the outside wheel toes out and causes the car to oversteer in a roll condition. Road undulations also have an effect if rear steer is present. 'When one wheel hits a bump it acts the same as body roll, causing rear-wheel steering'. (Puhn F, 1981) For the most predictable handling it is advisable to have a neutral rear steer characteristic.

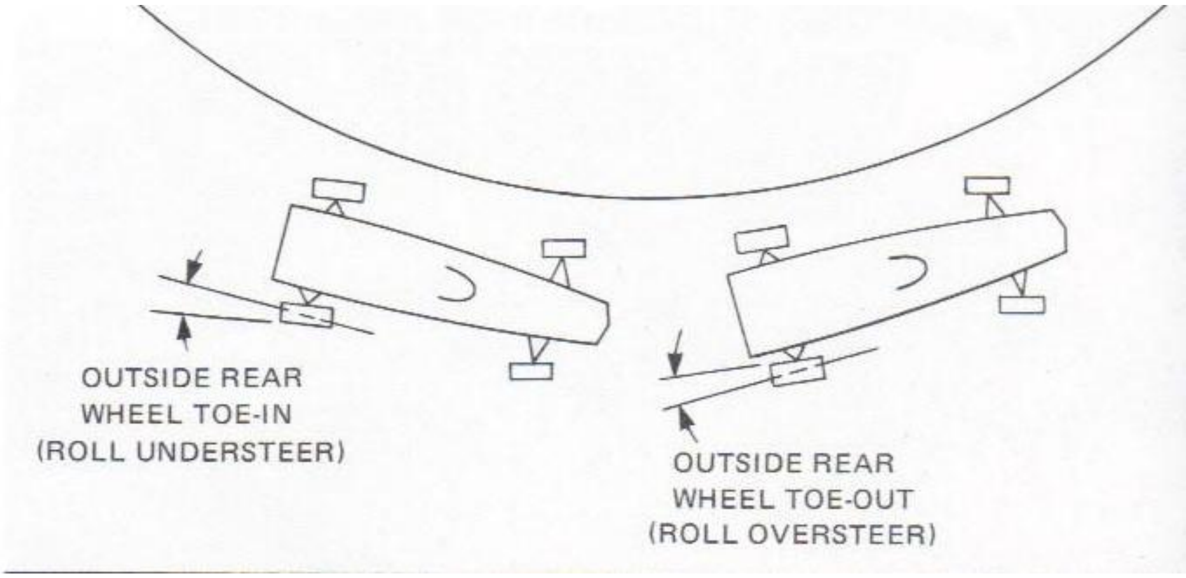

Fig. 2.9.1 Roll Steer (Puhn F 1981, p92)

Leaf sprung solid axle rear suspension is not immune from rear steer and is perhaps more susceptible to it than most may envision. In terms of leaf springs, the orientation of the springs and their mounting are the contributing factors. Due to the leaf spring, rear steer is achieved by one side of the axle moving forward while the other side moves rearward. In the understeering case, which all leaf spring rear suspension cars contain, the outside wheel will move forward while going into bump, while the inside wheel will move backward while going into rebound. This occurs as the rear mount of the leaf spring is higher than the front. As the wheel travels vertically it also travels forward and backward depending on the direction of vertical travel.

This can easily be remedied however, by ensuring the front and rear spring mounts are at the same height. This ensures the spring is mounted horizontal and therefore only vertical travel will be introduced by wheel movement. The spring should also be flat when the weight is applied to ensure this condition is met. To eliminate rear understeer in the project car it is necessary to lengthen the rear shackles to result in a more horizontal spring. This will however raise the rear of the car and further lowering will be required to return the ride height to its current value.

2.10 Anti-Roll Bars

Anti-roll bars greatly affect the handling characteristics of a car. The bar is in essence a connection between both sides of the suspension at the relevant end of the car that the bar is fitted to. As a result the behaviour of each wheel is affected by the bars effective stiffness rating. The anti-roll bar stiffness affects the amount of lateral weight which is transferred from the inside wheel to the outside wheel during cornering. The higher the level of resistance the bar has to twisting, the higher the roll resistance will be. This will result in the body rolling less throughout a turn, although the lateral weight transfer has increased through the anti-roll bars resistance to twisting. Stiffer anti-roll bars will result in less roll through corners, although more weight is being transferred laterally. In the event of a one wheeled bump the anti-roll bar will also increase the effective spring rate.

The stiffness of the bar is predominately a result of two variables. The diameter of the bar is the most commonly altered variable as it contains the highest dependence in the twisting stiffness. Figure 2.10.1 contains a very simple anti-roll bar. The equation for calculating the stiffness of the solid round steel anti-roll bar presents the diameter (D) of the bar as a variable which is raised to the power of four. The stiffness of the bar is therefore highly affected by the diameter.

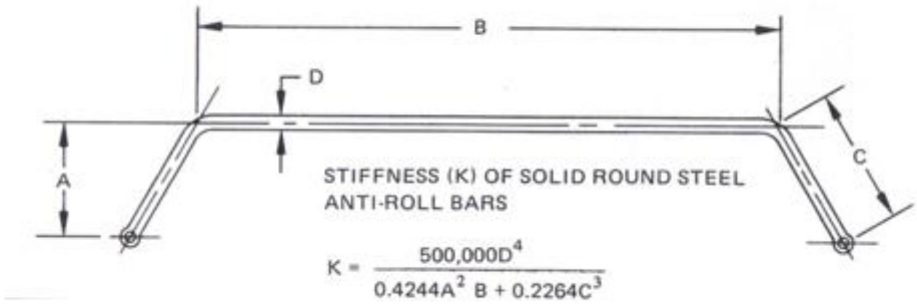

Fig. 2.10.1 Anti-Roll Bar (Puhn F 1981, p150)

The distance represented by C in figure 2.10.1 is the additional distance that is commonly altered in order to change the stiffness of the bar. This is predominately altered to fine tune the bars stiffness once the diameter has been set. The effect of changing the C length is minimized due to the 0.2264 factor to which it is applied.

Other factors also affect the effective stiffness of the bar, although most are not altered in the interest of changing the bars initial setting. The mechanical advantage of the bars outer mounting points will result in differing stiffness ratios. The further out on the suspension arms (closer to the wheel) the anti-roll bar is mounted, the more effective the bar will be at controlling roll angle. The bar should be mounted as close as is practical to the wheel to achieve the maximum effect from the bar size. The mounting of the bar also affects its overall stiffness with more compliant mounting resulting in a softer bar. Puhn tested different mountings on an anti-roll bar and found that a bar of 0.8 inch diameter mounted solidly was equal in stiffness to a 1.0 inch bar that was rubber mounted. The properties of the material will also affect the stiffness of the anti-roll bar, although generally steel used for the manufacture of anti-roll bar components will be fairly consistent to result in a meaningful size difference between bars.

The main benefit of anti-roll bars is in controlling camber curves on independently sprung suspension systems. The roll angle greatly affects the camber angle and high variations can be the result. The anti-roll bar limits the amount of roll, and attempts to keep the camber angle at an acceptable level to maintain acceptable tyre contact. The fitment of anti-roll bars will also increase responsiveness of the suspension by providing a positive reduction of initial turn roll.

The roll centre location will also affect the lateral weight transfer and the roll angle. Lower roll centres will result in an increased roll angle and this effect must be included in the sizing of the anti-roll bar. The front suspension in the 120y had an extremely low roll centre, a factor which resulted in high weight transfer and roll, with low jacking forces being present. The relative stiffness of the front anti-roll bar was required to control the roll angle in the interest of maintaining sufficient camber angles throughout the cornering sequence.

The rear suspension does not contain an anti-roll bar. The relatively high roll centre has resulted in limited amounts of roll being generated and resultantly the need for a rear bar has been limited. The modification of the rear roll centre height may present a different roll situation and a rear anti-roll bar may be required.

2.11 Springs

Springs are the only mechanisms which are containing the weight of the car. The spring settings are controlled by the stiffness of the springs and the mechanical advantage built into the suspension system. The spring stiffness affects many parameters, however lateral weight transfer is unaffected. The springs do affect roll resistance in all suspension systems, although this is due to the physical mounting position. The springs are always mounted away from the centre line of the car and hence resist roll due to the wide spring base. The resistance offered by this is not adequate because 'if the suspension springs are stiff enough to limit roll to our desired maximum, the wheel rate in ride inevitably would be too high for tyre compliance'. (Smith C 1978, p66) Tyre compliance is vital to ensure that the tyres remain in contact with the road at all times. The springs therefore need to be soft enough to maintain tyre contact with the surface and heavy enough to prevent the body from scrapping. The mechanical advantage of a suspension system will alter the rate which is actually seen by the wheel. The wheel rate is more important than the spring rate and the relationship between these variables must be calculated for the particular suspension system.

Springs in automotive applications generally have two different forms. The front of the 120y contains coil springs. The spring stiffness is calculated using the following formula as taken from Puhn 1981.

$$K = \frac{W^4 G}{8ND^3}$$

where K = stiffness of spring in lbs/in

W = diameter of the spring wire, in inches

G = 12,000,000 for steel springs (depends on material properties)

N = number of active coils

D = diameter of the coil measured to centre of wire, in inches

The spring force is similar in effect to the anti-roll bar with the diameter of the pipe making an incrementally large difference to the overall stiffness. The larger the spring diameter the heavier the spring will be, at the powered factor which is present in spring calculations. The number of active coils can also be changed within the realms of the standard suspension points unlike the diameter which requires new mounting positions. The diameter and number of active coils are most likely to be altered to change the spring rate which in turn alters the wheel rate.

The front springs in the 120y are 8kgs/mm and are a suitable stiffness for the weight of the vehicle. The stiffness of the suspension is dependent on the weight of the car and the inclusion of this factor into the variables presents the spring's natural frequency. The calculation of the natural frequency is taken from Puhn 1981 p139. Note: the wheel rate has been assumed to be the spring rate and the weight placed on the front wheel is assumed to be 300 kg.

$$natural\ frequency = 3.13\sqrt{\frac{suspension\ vertical\ stiffness\ per\ wheel}{sprung\ weight\ per\ wheel}}$$

$$natural\ frequency = 3.13\sqrt{\frac{446.9\ lbs/in}{661.4\ lbs}}$$

$$natural\ frequency = 2.57$$

Puhn has found that acceptable values are between 1 and 2, with the upper values being deemed for race cars only. The natural frequency of 2.57 is high and shows that further optimization of the spring rates could be achieved once other parameters have been set. The calculations for this analysis should also take into account the actual situation more closely once roll centres and their effects have been modified. The rear spring rates are considerably softer than the front although the required calculations for determining their effective spring rates are complicated and unjustified given the roll centre adjustments being made.

The 120y rear suspension contains leaf springs. The analysis of their spring rate is outside the scope of this report and further calculations into such variables should be considered after the roll centre location has been altered. The rear traction is extremely good and warrants the limited investigation into its parameters at the current stage.

2.12 Dampers

Dampers perform the simple function of transferring kinetic energy into heat energy. The operation and effect that they have on the dynamic race car however is far from simple. There are many different forms of dampers and each has their own attributes. The damper in basic principles resists the motion by pushing oil through an orifice. The oil is pushed through the orifice by the piston which is directly connected to the suspension's movement. Many dampers are also filled with pressurised gas to prevent the foaming of the oil during the stroke. Dampening levels depend greatly on the spring rates that are used, hence why most race teams use adjustable dampers.

The piston velocity is varied by the size or configuration of the orifices and this greatly affects the damping force. Fluid dynamics laws state that 'a fluid's resistance to flow through any given orifice will increase directly as the square function of flow velocity' (Smith C 1978, p74). Many different tricks are used to alleviate this phenomenon with spring loaded valves and progressive orifices used to result with any particular characteristics that are desired.

The three rates that can be tuned into the dampers are linear, progressive and degressive. Linear dampers result in dampening that increases at the same rate as piston velocity. Progressive dampers result in dampening which increases at a greater rate than piston speed and degressive dampers have the opposite effect. Dampers are dependent on velocity and as a result the load which applies the acceleration force is also a contributing factor to the dampening force. In racing, the tendency is towards little dampening at low speed in order to maintain suspension sensitivity.

The bump and rebound of the damper refers to the shortening and lengthening of the damper respectively. The rate of this is normally adjustable independently on racing dampers as each case presents different variables due to the different setups used.

The damper also has an effect on the load transfer. The total load transfer or roll angle is unchanged by the damper. The rate at which this takes place, or the transient handling, is however affected by the dampers stiffness. 'Stiff shocks (dampers) give rapid response and good transient characteristics - they help the race car 'take its set' quickly.' (Smith C 1978, p76) Most race cars are set up overdamped for this reason, although going too far with damper stiffness will result in decreased tyre compliance due to a 'choppy ride and wheel chatter' (Smith C 1978, p76).

Dampers are a highly unsorted area and most information sources suggest that settings be found through testing. The mechanics involved in the damper are extremely complex and beyond the scope of this report that is focusing on suspension geometry. The dampers currently used in the front of the 120y are sports oriented, although the rates of piston velocity to dampening forces are unavailable. Most damper producers do not release damper rates with their products and as a result the art of damper tuning is often met with confusion. The rear dampers are standard 120y items, a decision which was made in the interest of maximum rear tyre compliance. The damper rates have been left unchanged due to the large capital investiture required with their modification or replacement. The damper settings have also been unmodified in an effort to control the experimental variables.

2.13 Rear Axle Lateral Location

The 120y contains a simple Hotchkiss axle, as seen in figure 2.3.3, which sees the differential being sprung and laterally located by longitudinal leaf springs. The springs potential at locating the differential laterally are extremely compromised as the spring is designed to flex. The front of the spring will deform under hard cornering forces and result in the lateral motion of the wheels and differential with respect to the body. This is highly undesired and the result is a slow to respond and generally sloppy feeling location of the rear end. Many different methods exist of locating the differential laterally although each contains benefits and disadvantages.

The easiest lateral location device is the Panhard Rod as shown in figure 2.13.1. The single lateral location arm is installed in the rear suspension to prevent the sideways motion of the differential. The Panhard Rod does however result in horizontal movement of the axle as it is raised or lowered through the suspension movements. The horizontal motion of the differential can be reduced by constructing the Panhard Rod as long and horizontal as is applicable. The horizontal motion can cause binding of the rear suspension if values are high enough or the height change is great enough. The instillation of a Panhard Rod will also alter the roll centre to the point at which the bar crosses the centre of the car. In order for the rear roll centre to be adjusted this point must be altered.

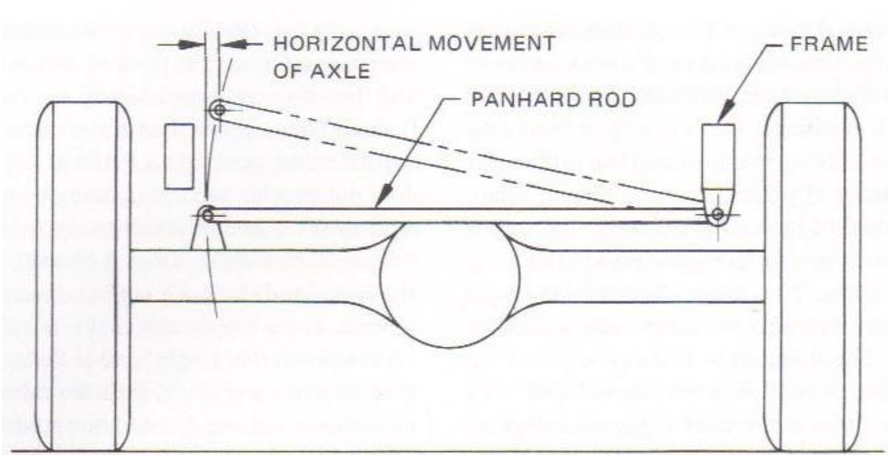

Fig. 2.13.1 Panhard Rod (Puhn F 1981, p152)

The Watts Link is an alternative means of locating the differential. The Watts Link setup is more complex and time consuming to design and construct although the benefits over the Panhard Rod are fairly substantial. The Watts Link will result in no horizontal movement throughout its vertical travel and hence no binding will occur through the leaf springs lateral movement. The Watts Link also changes the roll centre location, a variable that can then be easily changed if the pivot is positioned on the body as shown in figure 2.13.2. The pivot bolt is the rear roll centre and any height change associated with this will result in the associated roll centre adjustment. The Mumford Link is an adaptation from the Watts Link and its use is mainly highlighted in its potential to lower the rear roll centre below the ground level. This is not required and hence the Watts Link is sufficient.

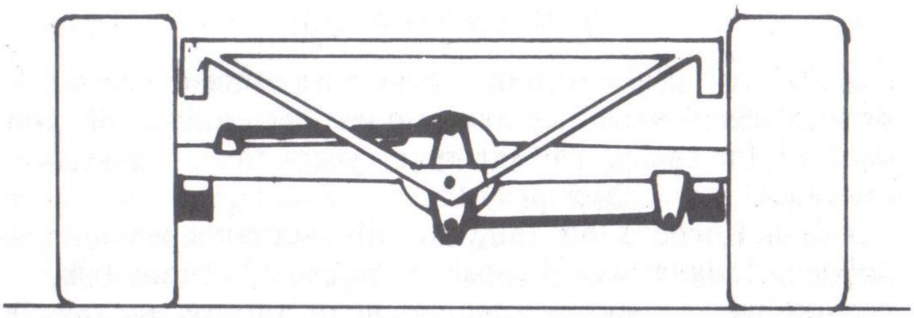

Fig. 2.13.2 Watts Link (Smith C 1978, p156)

Other methods of locating the differential are possible through the inclusion of additional diagonal arms. The roll centre is altered with the inclusion of such arms and generally they are included to try and attempt to eliminate spring wrap up. The fixed roll centre location does not provide a tuning tool to balance the car and hence their application was limited in the projects aims of providing a more balanced car.

Leaf sprung rear ends are also limited in their application of high torque values. The spring wrapping up under hard acceleration and creating an S shaped spring profile limits the amount of torque that can be transmitted to the ground by the tyres. The resulting loss of traction due to the spring being fluctuated around this shape means that less drive is achieved on corner exit. The Caltracs setup that is currently used allows the suspension to operate while eliminating the spring's tendency to wrap up under hard acceleration. The Caltracs were fitted before the report data acquisition began and was resultantly left unchanged. The setup had resulted in good traction with no noticed negative effects and hence was left unmodified.

2.14 Summary and Modifications

The review of the available literature that is applicable to the current situation has resulted in many different modifications which could be performed. All the applicable modifications were analysed in terms of their capital expenditure, time considerations and potential benefit to the aims of the project.

The first section considered the roll centre location and its effect to the lateral weight transfer. The roll centre is vitally important and every effort should be made to control its location and optimize its affect. This area had also had no consideration placed on it during the suspension development so reasonable gains could be expected. The camber, castor and toe settings had all been modified and slight optimization had occurred. The gain to be had through the further development of these parameters was expected to be minimal. The bump steer characteristics and Ackerman effects required special considerations, however the restrictions placed in this area from size constraints reduced the overall effect that could be achieved by such modifications. The roll steer characteristics could be easily and cheaply altered and presented a relatively good base for improvements. Anti-roll bar rates could be optimized, however this is part of a package which must be selected in fitting with the spring rates and roll angles. The roll centre will also affect the anti-roll bar rates and altering these values will result in modifications which are less than ideal in the new situation. The dampers are also tuned to the springs and are dependent on the system parameters such as spring rates. The increased lateral location of the differential will greatly increase the driver's feedback and result in an altered and adjustable rear roll centre.

The decision was made to look into methods of modifying the front roll centre and find its effects on other parameters. The rear roll steer will be reduced through modifying the spring's orientation and a Watts link will be fitted to the rear suspension. The Watts link will contain adjustment to allow the modification of the rear roll centre with the benefit of adjusting the cars balance. Smith (1978) states that vehicle balance and driveability are the most important from a lap time point of view, and therefore in the projects objectives the efforts required in having an adjustable rear roll centre is worth the required investiture.

3. Methodology

The required stages in completing the project were systematic and aimed to cover all relevance sections of the suspension analysis and development. In order to conduct these in a logical and progressive manner it was advisable that a clear direction and focus be placed on each section.

The first stage of familiarisation was the required literature. In order to gain a good understanding of the current and required parameters of the short circuit racing car, research in the area was required. Once the conditions and needs for suspension systems in short circuit racing cars were quantified, the project then moved to other sections and continued its development. A background understanding of the requirements is critical in order to modify and tune the suspension accordingly.

The data acquisition system needed quantifying from the beginning of the project to ensure that the data will be legible and useful in comparing previous and current systems. The use of the GPS data logger proved to be reasonably reliable in operation, and a good indication as to what cornering speeds were achieved. Tyre temperatures were also utilized to setup some parameters, while driver feedback also played a vital role.

Analysis of the current suspension system was conducted in fitting with the literature review to determine methods of increasing cornering power. This could have taken many different forms, and modifications were expected to be extensive. All of the required modifications were then enlisted to a further selection process to ensure that only the best modifications were made, based on varied inputs such as cost, time for modification and predicted outcome.

In order for the baseline parameters to be met for the cars operation, they must be clearly stated. In order for the car to fulfil its requirements, these parameters needed to be applied to all sections and therefore provisions needed to be made to ensure this project direction was followed.

The modification process began with evaluating the current suspension system performance. This was conducted through the use of the data achieved before the project modifications began. This was gained from events which were conducted during the research phase. The data was then analysed and modifications were prioritised accordingly.

Once modifications or tuning were completed the difference to the cornering power and driver control were measured. The benefit was then measured through more track time, with data being compared to previous models. The modification and tuning process was then repeated as time and resources permitted.

3.1 Risk Assessment

3.1.1 Working on the Car

Safety must always be paramount when working on automobiles. Many different dangers exist in this environment and must be understood fully in order to eliminate or decrease the risk of car damage and most importantly personal damage. Almost all aspects of working with a physical material present some sort of danger to the mechanic. Even trivial injuries such as cuts from hose clamps and cable ties can be avoided or decreased by paying more attention to the job at hand and noticing things around the working area. Many of these can be avoided by using the right sized clamps or positioning the potential cutting surface away from the area most likely used by the mechanic. Preventative safety precautions such at trimming cable ties back flush and smooth may remove all chances of injury. Even simple tasks such as loosening and tightening bolts present dangers if the wrong size or type of tool is used. Leverage should be used wherever practical to decrease the mechanics strain and increase the control over the tool. The predicted tool travel path should also be analysed before attempting to undo any tight nuts or bolts to decrease the chance of skinning knuckles and fingers. Welding and cutting of components for the modification process are also potential dangers if handled by untrained or inexperienced personnel. This danger will be decreased by using only experienced welders and crafts people in the modification of components.

Many of the suspension modifications and measurements must be made from underneath the car. In terms of safety, this is perhaps the biggest danger that exists in the project. The chance of injury can however be decreased remarkably by using common sense and using tools applicable to the job at hand. Firstly, jacking of the car to the required height should be done both carefully and slowly enough to ensure that it is done safely. The jack point used is just as important to eliminate the chance of the jack sliding or slipping on the component. For this reason it is advisable to always inspect the jacking surface and jack head to ensure they are free from contaminates such as oil and dirt. Once the car is jacked up, stands should always be placed under the car to ensure the safety of the operator. Even if the jack is still holding the cars weight, it is always advisable to use stands to limit the distance the car may fall in the event of a slippage or jack leak. Trolley jacks also move forward and back while moving vertically and this movement needs to be accounted for when jacking and positioning stands to ensure stands do not twist and fall over.

Fig. 3.1.1.1 Floor Jack

Fig. 3.1.1.2 Axle Stands

Large components also present a danger to the modifier. Not only does the weight mean that care must be taken in shifting the components, it also increases the danger of fitting such components as differentials. Ensure correct lifting techniques are used under all circumstances and that all care is taken to foresee the travel path and team lifts are used wherever necessary. Many components such as springs and gas shock absorbers present their own mechanical danger in the release of the stored mechanical energy. Extreme care must be taken when removing these components, as a spring can cause serious injury if released incorrectly. In this case it is recommended that the correct clamping procedures be used to remove the spring, and then slowly unwound and released safely.

Although the project is focused on the suspension development, other aspects of the cars operations will affect the safe operation of the intended aims. Electrical considerations need to be made to ensure no damage is made to either the car or the operator. The car is fitted with an isolation switch, which cuts power to all operations. This switch is always turned off when working on the car to eliminate the chance of electrical damage. Fuel in the cars fuel tank and lines is a major fire hazard. During the modification process, welding on the body may be required, and the large heat creation can spell disaster if not contained. For this reason it is advisable to remove fuel and vapour from the fuel tank and lines well prior to the welding or cutting stage. This is only necessary if the heat (including sparks and weld splatter) is at all likely to come in contact with the fuel or vapour.

3.1.2 Racing and Testing

Racing and testing of the car will present a major component of the safety considerations. The safety of the actual physical testing of the race car will present a large portion of the risk involved in the project. Many different processes and precautions are however put in place to limit the danger to the driver and spectators and prevent damage to the car. Many driver aids are used to increase safety, and are mandatory in most racing applications. Roll cages are not mandatory although highly advised in these classes. Roll cages protect the driver by preventing roof cave-ins and are highly critical in the event a car should roll. The race car used for the project is fitted with a roll cage, in both a safety regard and to increase chassis stiffness. The drivers must also wear approved helmets and fire retardant clothing in order to compete. In the likelihood of an accident, seat belts are also mandatory. In this particular situation, a four point harness has been fitted to comply with the regulations as shown in figure 3.1.2.2. This also increases driver restraint through hard corners and allows a more enjoyable and controlled atmosphere.

Fig. 3.1.2.1 Race Helmet

Fig. 3.1.2.2 Race Harness fitted

Other factors such as concrete and water barriers and tyre walls are also erected to prevent damage in the case of an accident. Gravel traps are also evident at some race tracks and stop the car with little damage in the case of a driver error or component failure. Street circuits are fairly unforgiving in terms of track run off, and care needs to be taken to ensure that these areas are not relied on. Flag marshals are present at all events and warn other drivers of a spun car in the event they need to slow down. Being predominately single car events there is plenty of time to slow down in the event of a blockage in the racing line or track.

Working on the car at the track is possibly the most likely place in which an injury may occur. In the rush to get the car fixed or modified between races is where the greatest risk is observed. In these times of increased stress, hazards may not be seen as easily and injury could result. It is important that safe operations are still used, and that all parties remain calm. Safety should always be considered first, regardless of the importance of the race.

3.1.3 Researching and Modelling

Compared to the other dangers involved in the project, the researching and modelling safety issues are minor in both the exposure and consequences. Sitting at a desk during the background research stage involves less physical strain then actually working on or under the car. Although this is the case, correct posture should be maintained under all conditions and correct lighting should be available. These simple factors may not sound like much, although during long study and modelling periods the results can make a difference to productivity as well as user health.

3.2 Resource Requirements

The resource requirements for this project are reasonably high in comparison to other projects that may be undertaken in a strictly lab or program based environment. Many different expenses are present in such a varied project and limitations may need to be applied to the end results due to these limitations.

The first call of resources is in the research field. Many different books are available on the subject matter, most of which are available directly through libraries such as those found at universities. Many different styles of explaining concepts help with the understanding of how each individual setting and variable interacts with each other. Internet sites are also available on different sections of the research and are important for ironing out any other questions that may exist.

The next requirement is in the computer programming of the suspension geometry to reduce the actual measuring and trailing on the car. This will reduce the time taken in establishing just how the suspension is formatted during all stages of the project. The program used to evaluate the suspension geometry is WinGeo 3 Version 4.00 and is provided by the University of Southern Queensland for use within this project.

Once the required components are calculated the resource focus will then change to obtaining the required components. As the car is a personal possession of an individual, it has been approved through communications that the individual is to financially support the ventures undertaken in the project. This does however have limits in terms of time to perform modifications and financial funding. Parts that are already owned or can be modified to fit will largely be used to keep in fitting with the budget orientation of the project. Sponsors are also already established to assist with the financial aspects of modifications through offering discounts and free services. A workshop is also available with full tool facilities and measuring equipment. Financial provisions have also been made in the unlikely event of a crash or similar failure to ensure the project will continue with a minimum of interference.

The testing equipment to be used at the race track will also represent a large input of financial resources. The GPS data logger (Fig. 3.2.1) to be used in the project is provided by the car owner. The usefulness of such a device in comparing modifications is invaluable and will add a measureable difference to the modifications performed. Other test equipment such as tyre temperature thermometers (Fig. 3.2.2) are also owned by the car's owner and present another reduction in financial stress placed on the project.

Fig. 3.2.1 GPS Data Logger

Fig. 3.2.2 Tyre Temperature Thermometer

The track time required for this project represents a large financial resource requirement. Events such as street sprints and hill climbs are relatively expensive and without the support of sponsors, may well be overbearing. The car owner and driver have however envisioned that the full cost of entries and consumables will be covered. The data that is gained from these events and the following modifications that will be performed have been deemed great enough to warrant such ventures.

4. Front Suspension and Steering Analysis

The McPherson front suspension, as found in the front of the Datsun 120y, is relatively basic in its operations. It contains limited moving parts and is easily modified to alter the characteristics of the suspension system. As previously found there are many parameters which affect the overall effectiveness of a suspension system, and few are more noticeable than the roll centre location. The roll centre location can be modified by the fitting of roll centre adjusters.

Fig 4.0.1 Roll Centre Adjusters

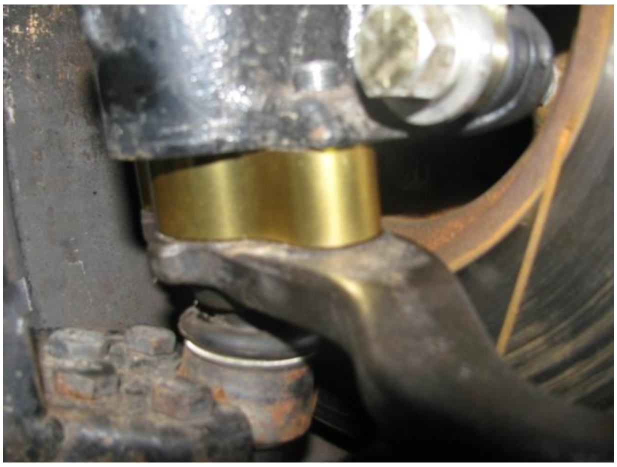

Fig 4.0.2 Roll Centre Adjuster location

Front roll centre adjusters are spacers which fit between the strut housing and the lower ball joint, as shown above in figure 4.0.2. They effectively lower the ball joint end of the lower control arm and steering tie rod. They are commonly available from many suspension performance shops for a small initial outlay. The effect of the roll centre adjusters is to raise the roll centre on lowered cars, and return the suspension geometry to more standard specifications. Roll centre adjusters of 25 mm spacing height are the standard production item so the analysis is based around the application of this commercially available size.

4.1 Win Geo 3 Analysis

The settings and migration of suspension parameters greatly affect the handling of a car. A suspension geometry program was required to evaluate the performance of the independent front suspension. WinGeo 3 is a suspension geometry program which outputs many different variables for analysis, and allows iterations to be performed through a range of suspension and steering inputs. The front suspension parameters were measured and placed into WinGeo 3. The measurements are taken from a centre point in the front suspension, which was found with a string line as shown below in figure 4.1.2.

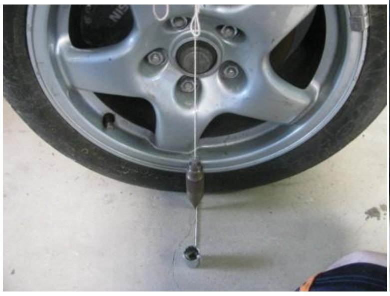

Fig 4.1.1 Measuring Front Suspension

Fig 4.1.2 Centre Point of Front Suspension

The altercations for the spacer fitment were then calculated and the analysis was performed as a means of comparison. The exact measurements of the car used for the analysis are included in Appendix L.

A comparison was required to evaluate the performance of the front roll centre adjusters. Ride height, roll angle and steering angle iterations were made through an acceptable range of values before a cornering sequence was evaluated to find their total effects on the dynamic race car. All aspects of suspension and steering changes as a result of the modification were analysed and the results compared.

The program presented a major limitation in that a McPherson strut suspension system could not be analysed with the current drag link steering arrangement. The model was constructed in such a manner that assumed the car contained a rack and pinion steering system. The actual results will give a good understanding of the changes that the roll centre adjusters will make, although actual values may vary in their genuine application.

4.1.1 Ride Iteration: -50mm to +50mm

Camber Curve

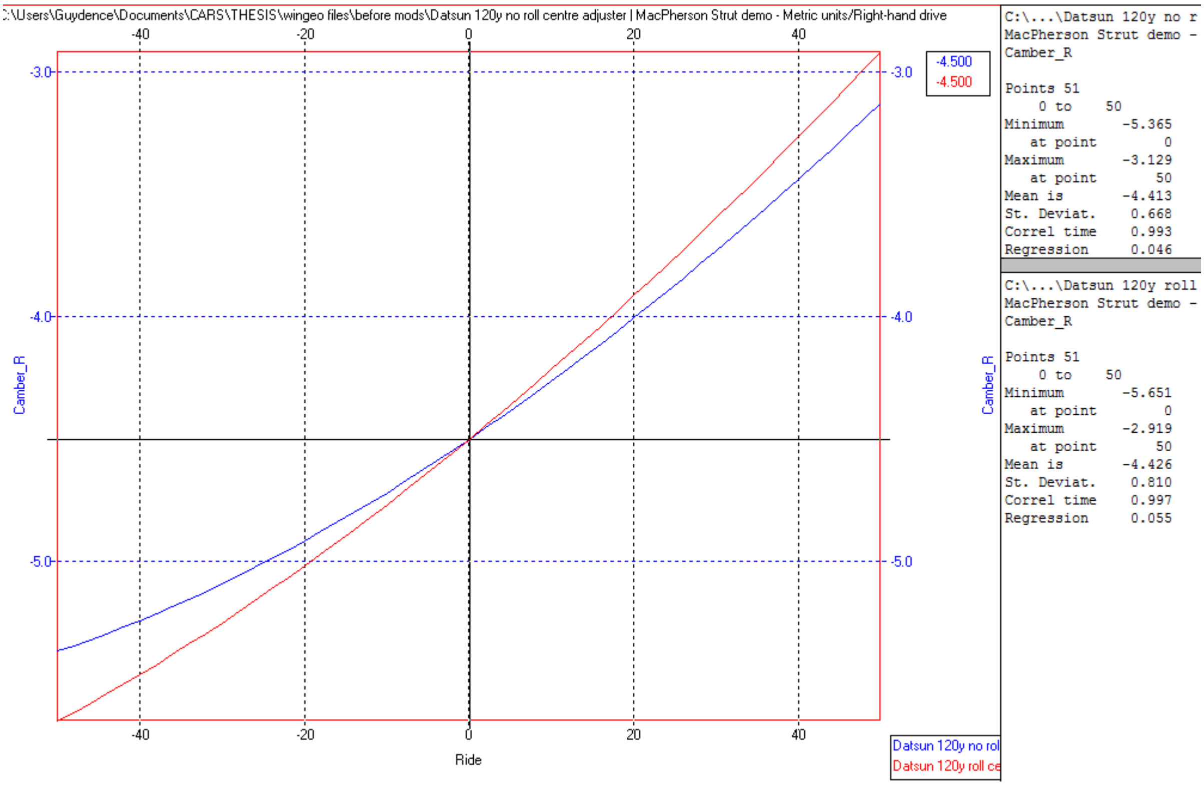

Fig. 4.1.1.1 Ride Iteration – Camber Curve

The camber curve has changed slightly due to the different angle of the lower control arm. The result is more negative camber as the ride height is reduced and less negative camber as the height is raised when compared to running no spacers. The braking affect will be slightly reduced with increased camber although the likelihood of running too much corner exit camber will be reduced. Both camber curves nature also promotes good turn in with high camber values and reducing turn out with decreasing camber values.

Castor

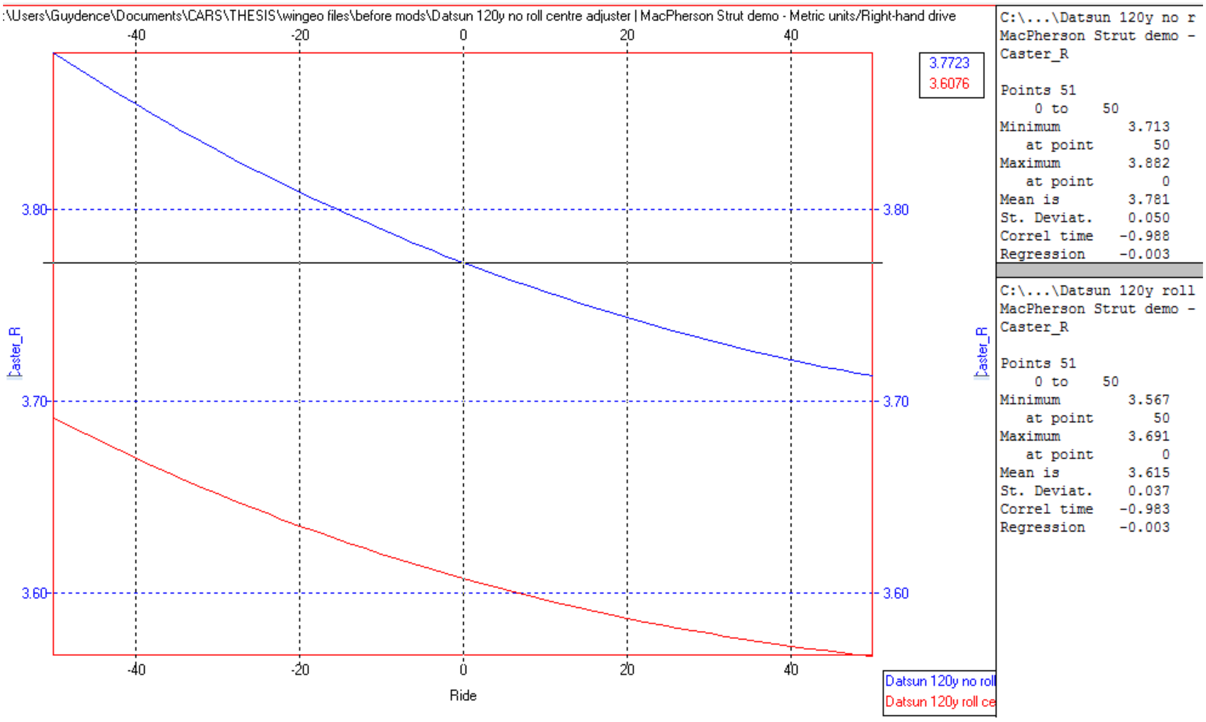

Fig 4.1.1.2 Ride Iteration – Castor

The castor values have only been changed by very limited amounts through the fitment of the spacers. This has resulted in a reduced amount of castor being achieved throughout all of the ride variations. The slight benefit is that the standard deviation of the castor change has been slightly reduced from 0.050 to 0.037, although this change is likely to be unfelt. The mean castor change from 3.781 degrees to 3.615 degrees is only a small change and the effect of this change is also unlikely to affect handling in a largely noticeable manner.

Net Steer

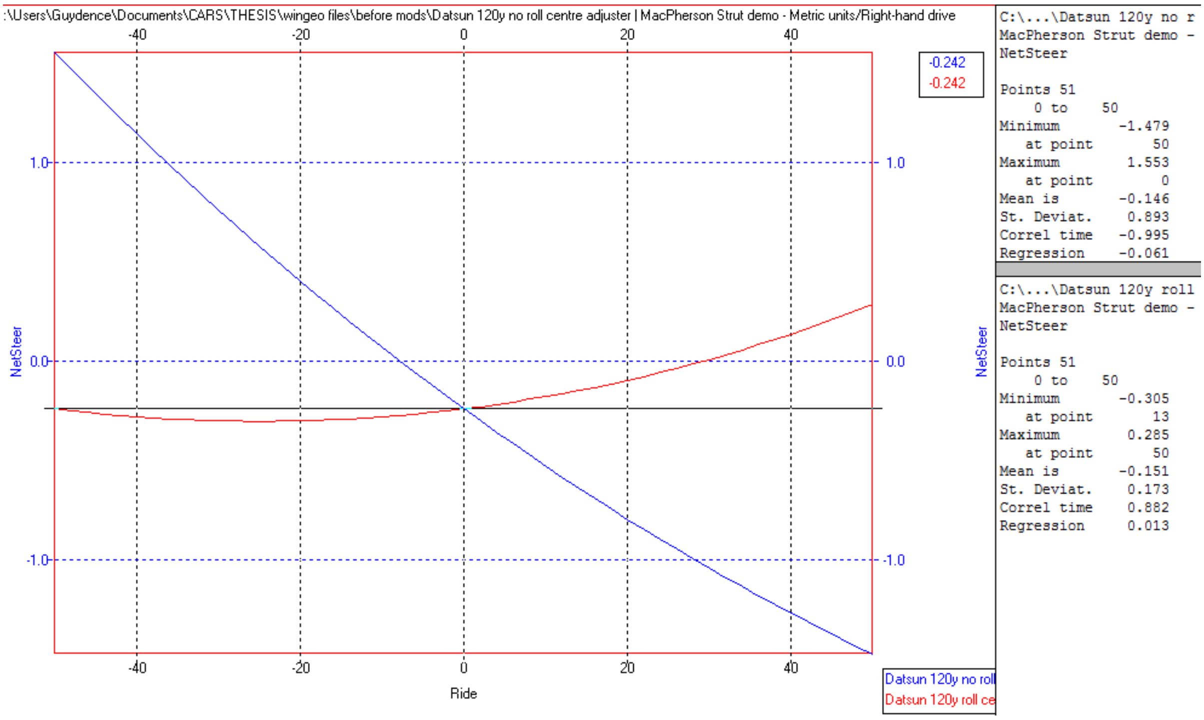

Fig 4.1.1.3 Ride Iteration – Net Steer

The net steer effect of the car presents the bump steer analysis. As can be seen the bump steer has been greatly reduced with the addition of the spacers. Without the spacers it is clearly seen that the steering bumps in throughout ride reduction and out during ride increases. The nature of this has been altered with slight steering out during ride reduction and steering in during ride increases. The actual variation of the values is of the largest concern and should be kept as low as possible to increase the predictability of the steering system. The standard deviation of the bump steer has been greatly reduced in the range specified, dropping from 0.893 to 0.173.

Ackerman

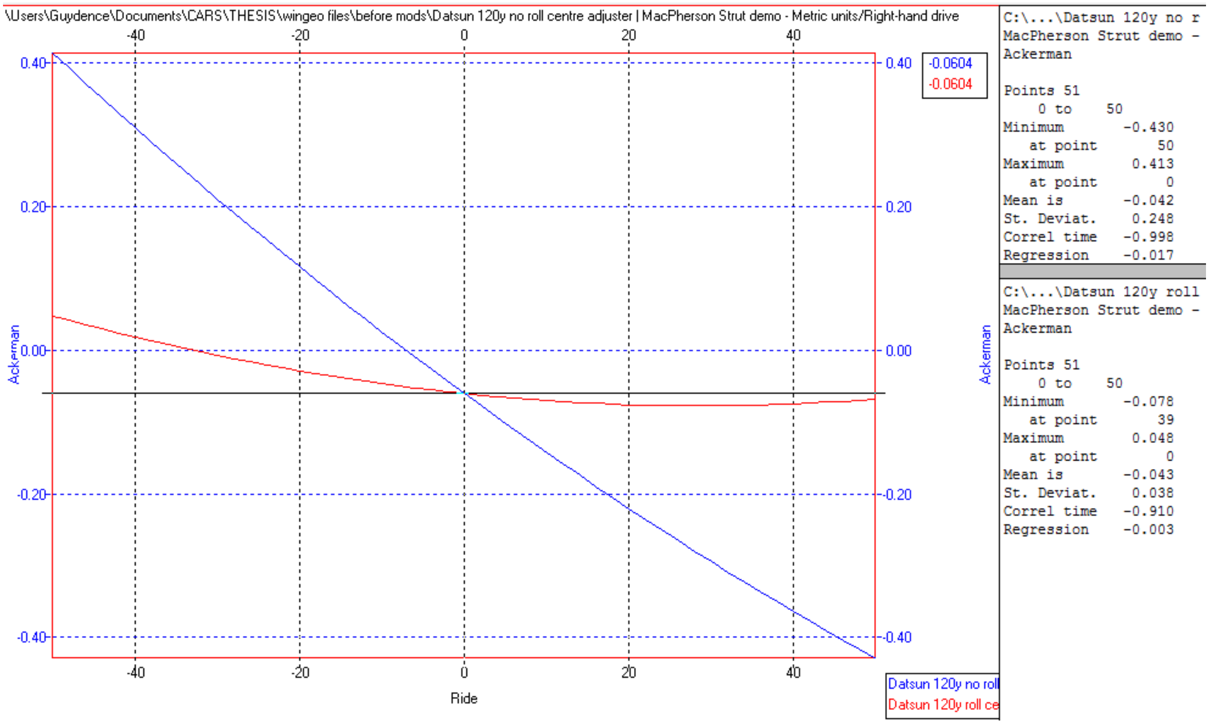

Fig 4.1.1.4 – Ride Iteration – Ackerman

The Ackerman amount built into the steering system has also been greatly changed. The spacers have a similar effect on the Ackerman as they do to the bump steer. The reduction of variance in these values has again resulted in a car with more consistent handling characteristics. The standard deviation of these values also shows a reduction in the variance from 0.248 to 0.038.

Net Scrub (Track Width Change)

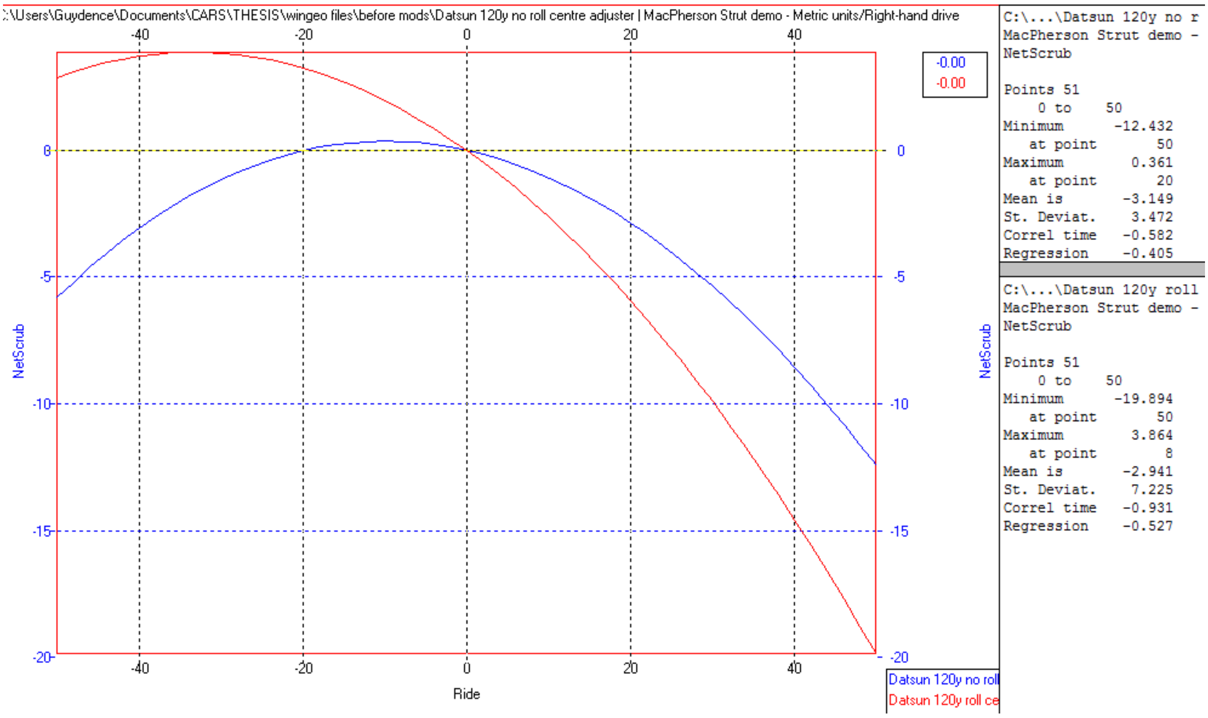

Fig 4.1.1.5 Ride Iteration – Net Scrub

The track width affects the cornering potential that can be achieved. A wider track is better for cornering and, if all other parameters are equal, a car with a wider track can corner with more force. The altered lower control arm angle has shifted the track curve to result in more track change throughout the height reductions but reduced track in height increases. The increased track in ride reductions will promote good turn in but may result in decreased cornering force on the exit.

Roll Centre Height

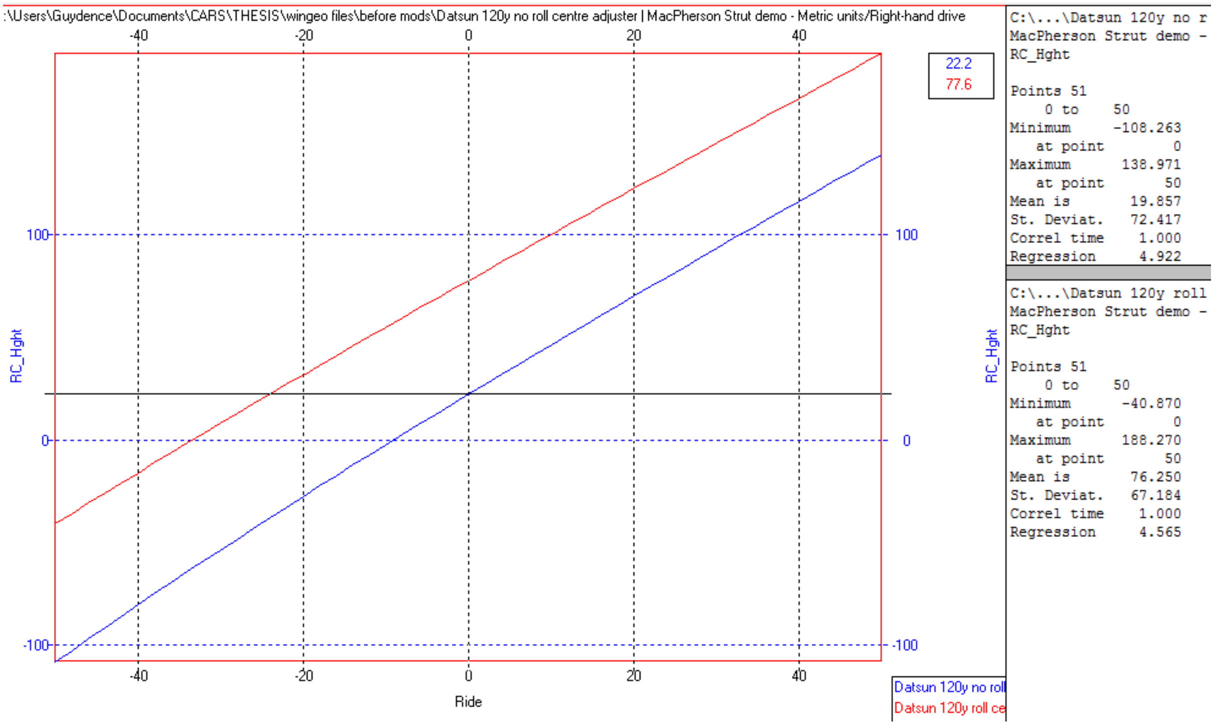

Fig 4.1.1.6 Ride Iteration – Roll Centre Height

The roll centre adjusters have changed the roll centre height. As can be seen in figure 4.1.1.6, the mean roll centre height was 19.857 mm before the spacers were fitted. The addition of the spacers has resulted in a mean roll centre location of 76.250 mm. The raised roll centre has benefits due to the large amount of variation that occurs with altered ride height. With the spacer fitted, the roll centre has less likelihood of moving under the ground and hence transferring a large amount of weight. The downside is that higher jacking forces will exist.

Roll Centre Width – unchanged with height

Roll Centre Moment Arm

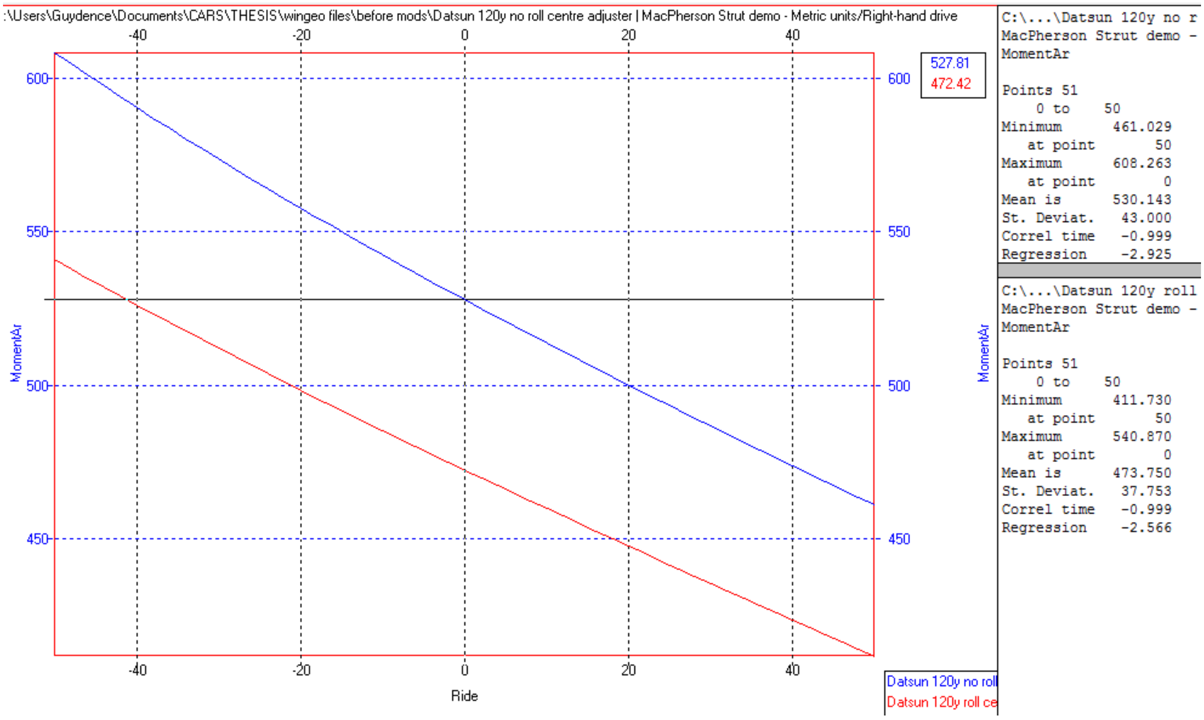

Fig. 4.1.1.7 Ride Iteration – Roll Centre Moment Arm

The roll centre moment arm is affected in a very similar way to the roll centre. The overall difference of a reduced moment arm will result in less weight being transferred laterally between the front tyres. This is in fitting with the reduction of understeer required and does not affect the overall progressive nature of the moment arm through suspension travel.

Jacking Centre of Gravity – Right

The Jacking Centre of Gravity refers to the height of the force application point. This is the point directly beneath the centre of gravity and is on the line connecting the tyre contact point and the front view instant centre. The consistency of this value is more important than its actual value, as this will change with roll centre height.

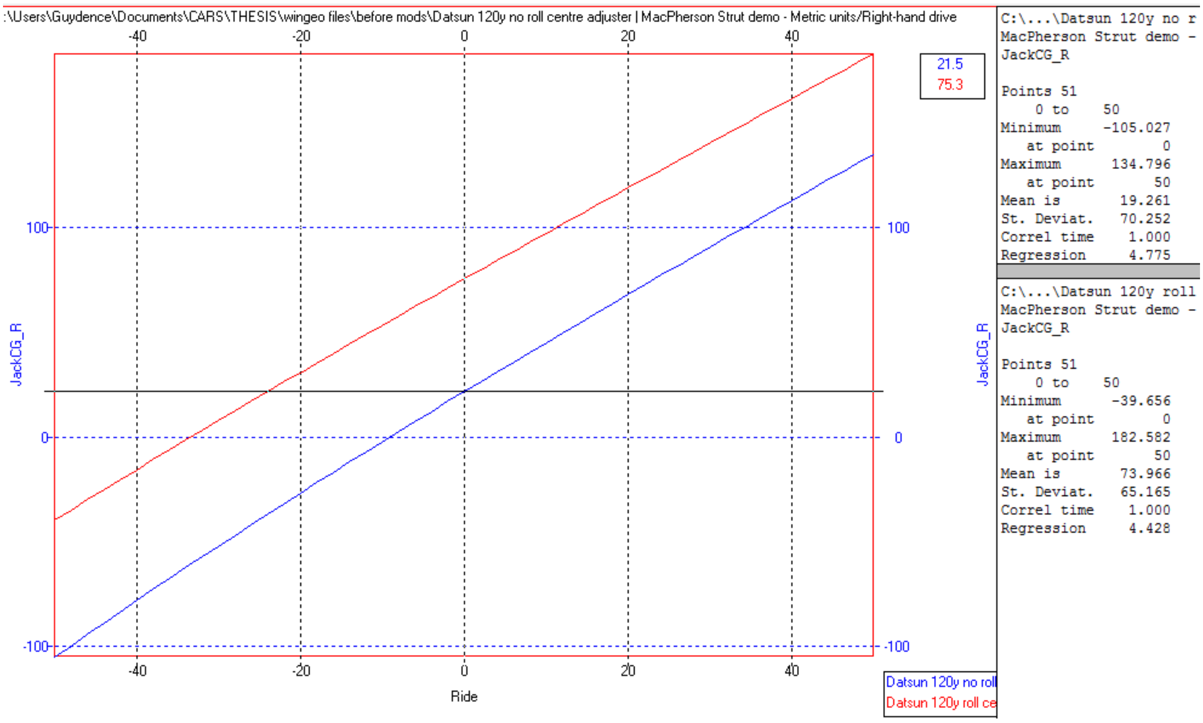

Fig 4.1.1.8 Ride Iteration – Jacking Centre of Gravity, Right

Jacking Centre of Gravity – Left

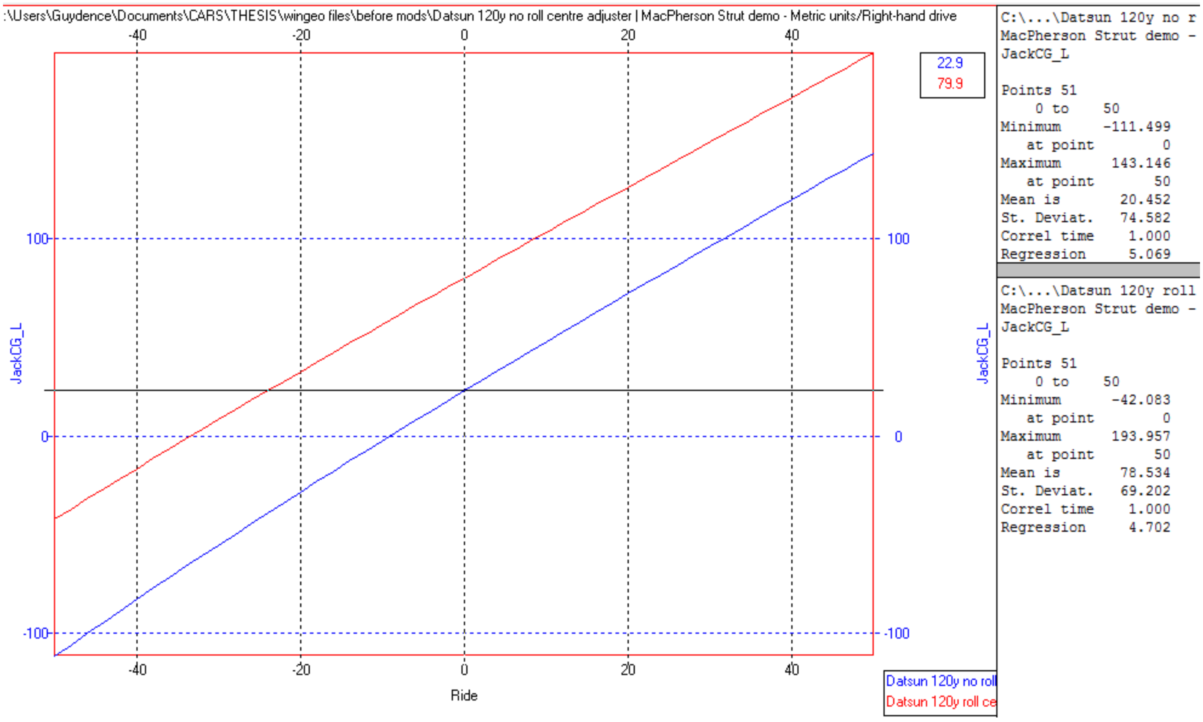

Fig 4.1.1.9 Ride Iteration – Jacking Centre of Gravity, Left

The jacking centre of gravity is also exhibiting the same tendency as the roll centre and moment arm. This is expected as the jacking point is dependent on the other values. The main benefit of the roll centre adjusters is that the deviation of these points is slightly reduced. The standard deviations of the right and left side points have been reduced from 70.252 and 74.582 to 65.165 and 69.202 respectively. The overall raising of these points has resulted in more jacking forces although the increase in the force application points are a trade off to reduce weight transfer from a large moment arm as a result of a reasonably high centre of gravity, at least when compared to most purpose built racing cars.

Clearance Point

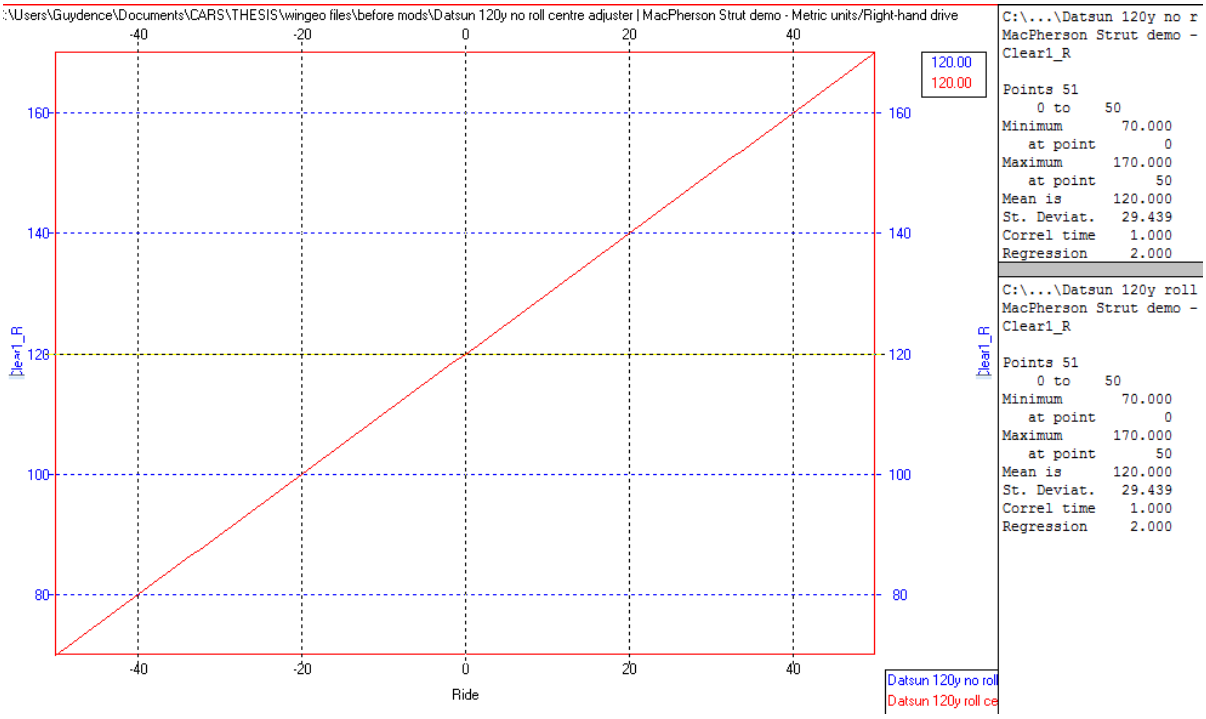

Fig. 4.1.1.10 Ride Iteration – Clearance Point

The clearance point presents no problems during the change of ride height and as a result no further action was required.

The ride iteration has shown that the roll centre adjusters have altered the roll centre height and resulted in a reduction of the moment arm. The net product of this is reduced lateral weight transfer, although higher jacking forces will be present, as shown by the higher jacking force application points. The bump steer and subsequence variables have also be greatly reduced.

4.1.2 Roll Iteration: 0 to 4 degrees

The data range will show suspension movements to 4 degrees of body roll due to the likelihood of reducing the front anti-roll bar stiffness.

Camber - Right (Outside)

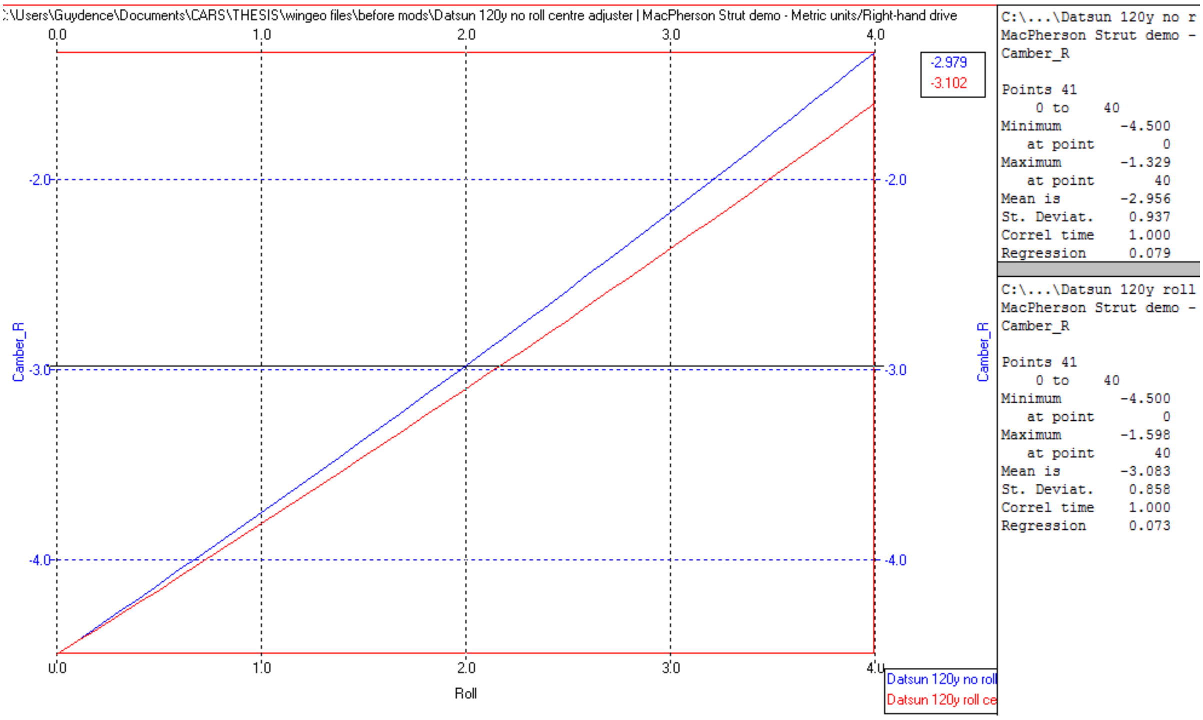

Fig. 4.1.2.1 Roll Iteration – Camber - Right (Outside)

The outside camber angle is the most important in maintaining the correct camber thrust throughout the roll characteristics. As shown by the outside wheels camber curve, the camber reduction through body roll has been reduced. The benefit of this is that less static camber can be used to maintain the same camber angle during cornering. The standard deviation has resultantly been reduced from 0.937 to 0.858, resulting in more consistent camber angles.

Camber - Left (Inside)

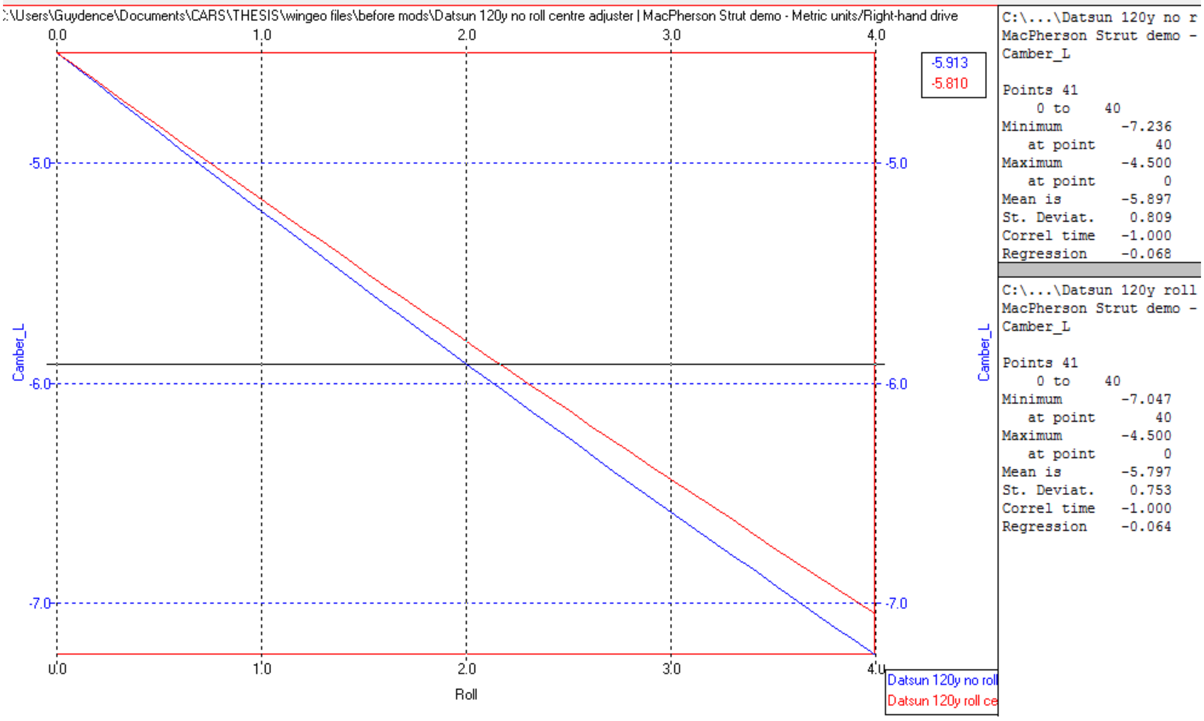

Fig. 4.1.2.2 Roll Iteration – Camber – Left (Inside)

The camber of the inside wheel is highly irrelevant in hard cornering applications due to the large amount of weight that is being transferred to the outside wheel. After the fitting of the spacers, the inside wheel has a reduced amount of camber. This is beneficial in that camber angles are excessive for the inside wheel and any reductions in this area will have a positive effect of total front grip.

Castor – Right (Outside)

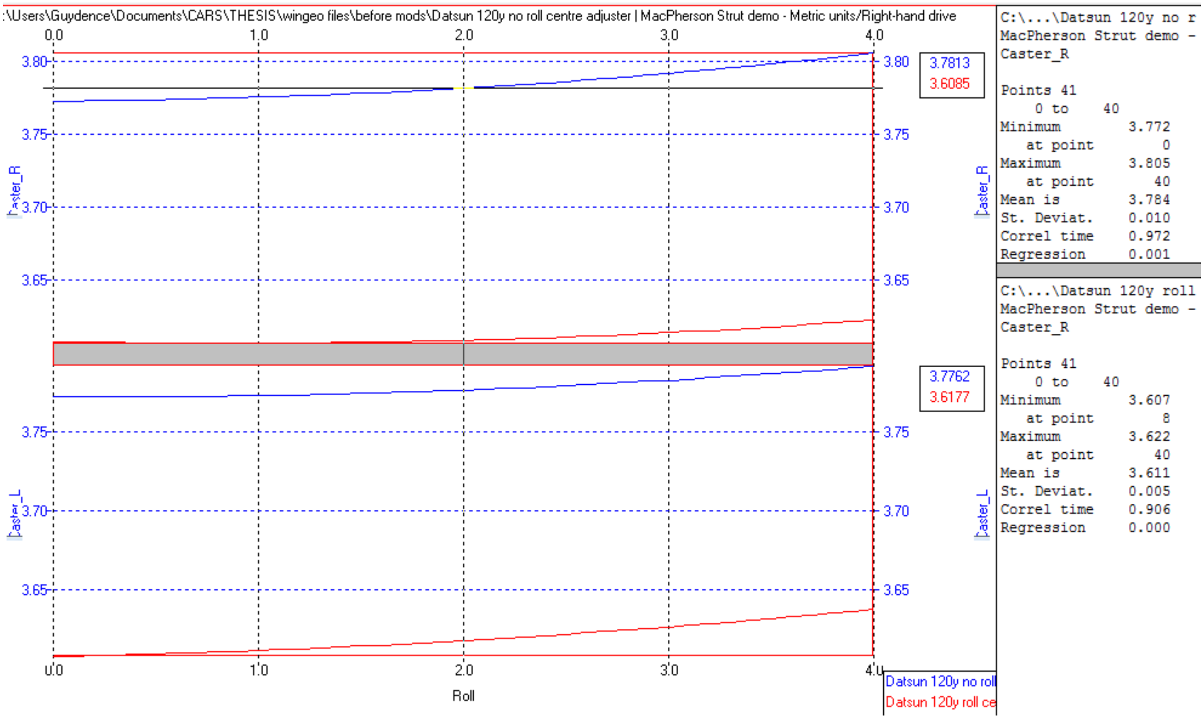

Fig. 4.1.2.3 Roll Iteration – Castor

The castor has again shown very little change and the same effects were found as per the ride iteration.

Net Steer

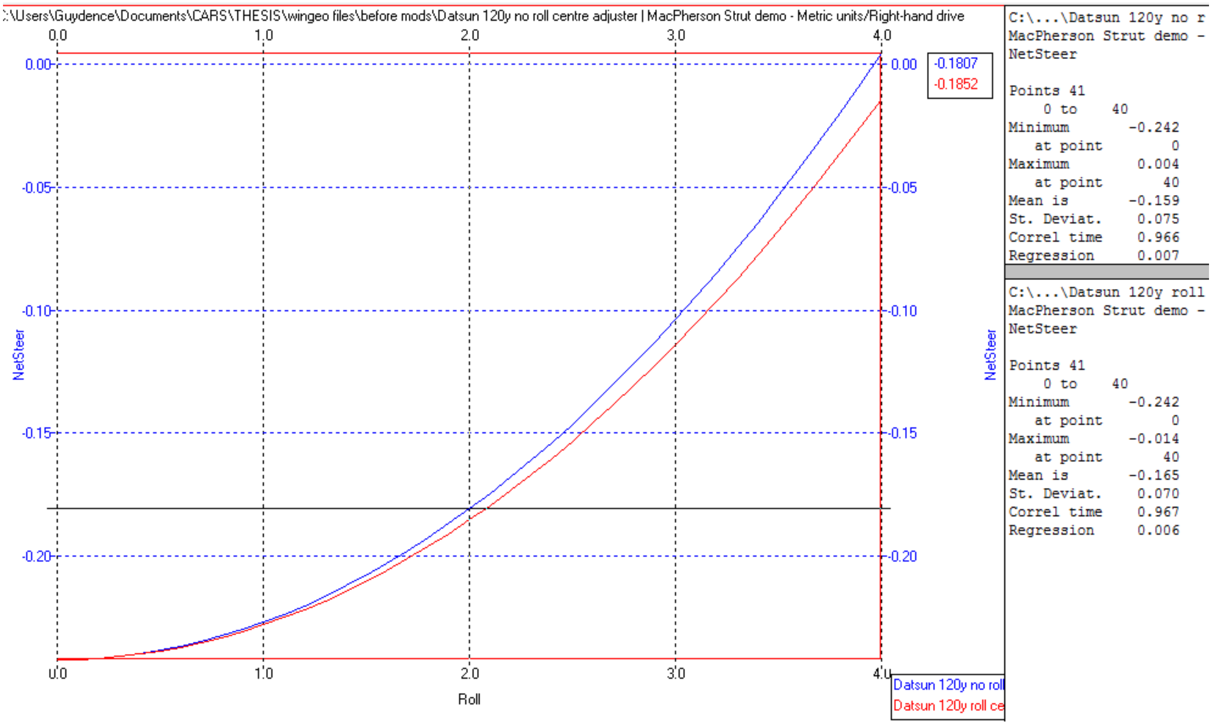

Fig. 4.1.2.4 Roll Iteration – Net Steer

The net steer of the car during roll is always towards toeing in. The above graph shows that the further the car rolls during a turn the further the steering will toe in. This will result in a reduction of the static toe out and a car that will understeer more as the roll angle is increased towards the apex of a corner. The spacers reduce the rate of toe out reduction and will result in a car that will understeer less at the apex of the corner.

Ackerman

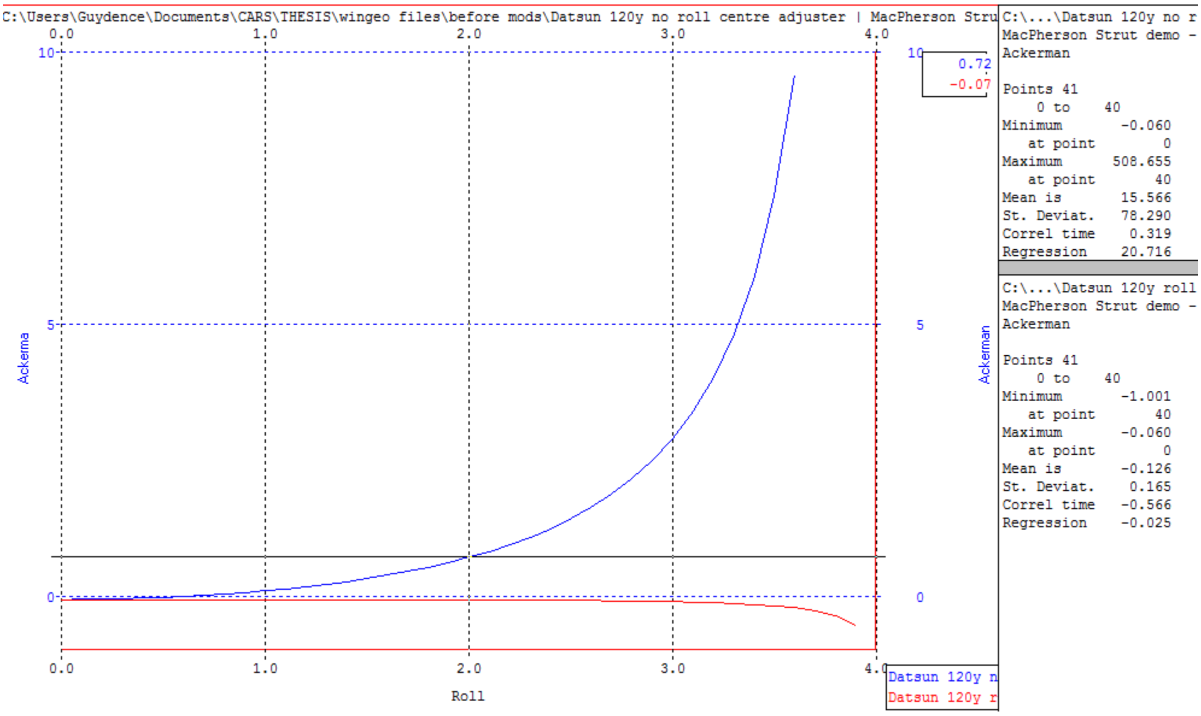

Fig. 4.1.2.5 Roll Iteration – Ackerman

Body roll greatly affects the Ackerman being seen by the steering system. As noticed the Ackerman has a great deal of variation with high values being achieved through high body roll. The spacers have limited the variation of the Ackerman greatly and the result is a much more consistent steering effect. The standard deviation change of 78.290 to 0.165 shows the magnitude of the benefits that the spacers offer in terms of Ackerman management.

Scrub – Right (Outside)

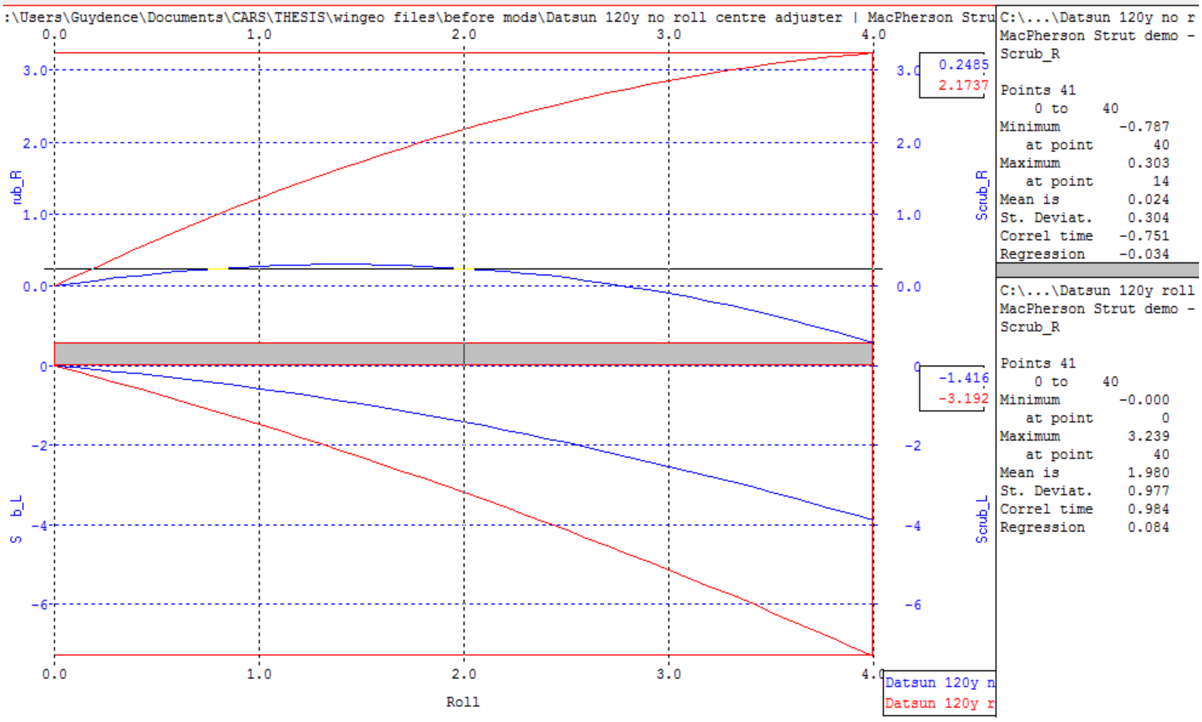

Fig. 4.1.2.6 Roll Iteration – Scrub

The overall effect of the scrub is more important than the individual wheel scrub, although the above graphs have been included to show the effect that each wheel has to the net scrub.

Net Scrub (Track Width Change)

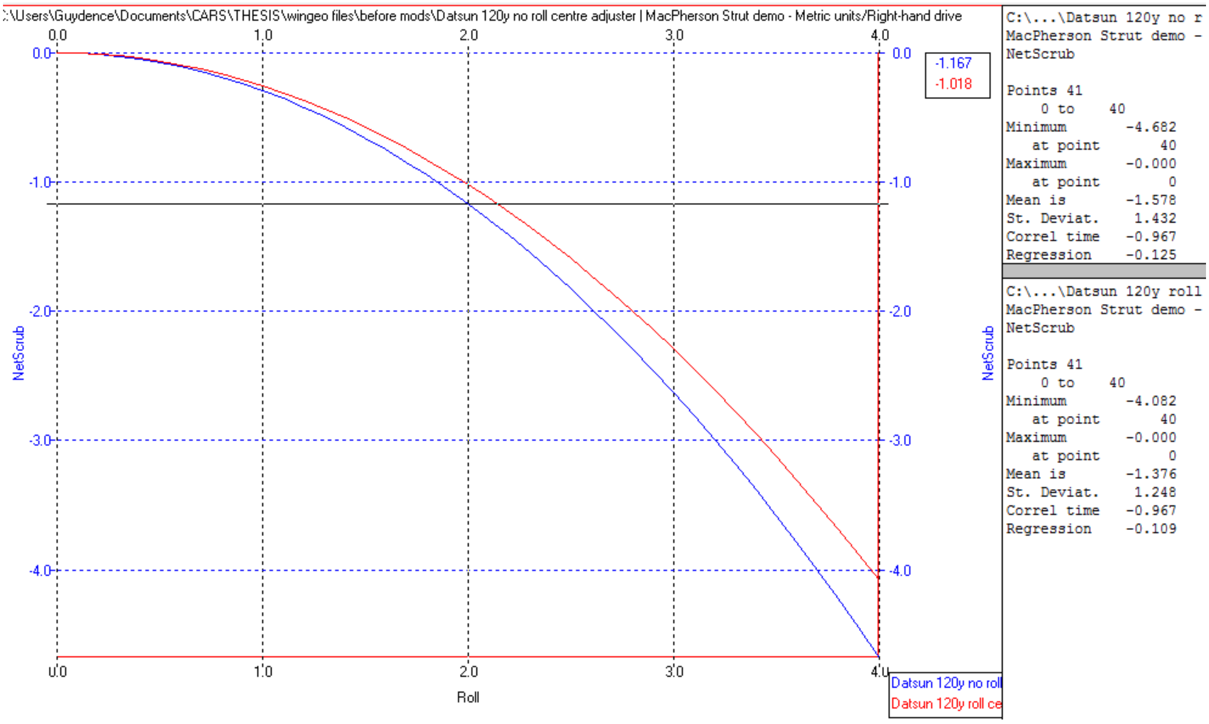

Fig. 4.1.2.7 Roll Iteration – Net Scrub

The overall track change will affect the manner in which the car handles corners. The ultimate for pure cornering ability is to increase the track width. In all instances, roll results in a decreased track, a bad condition for the generation of cornering force. The fitting of the spacers has however reduced this track reduction and resulted in a larger track than was previously being achieved.

Roll Centre Height

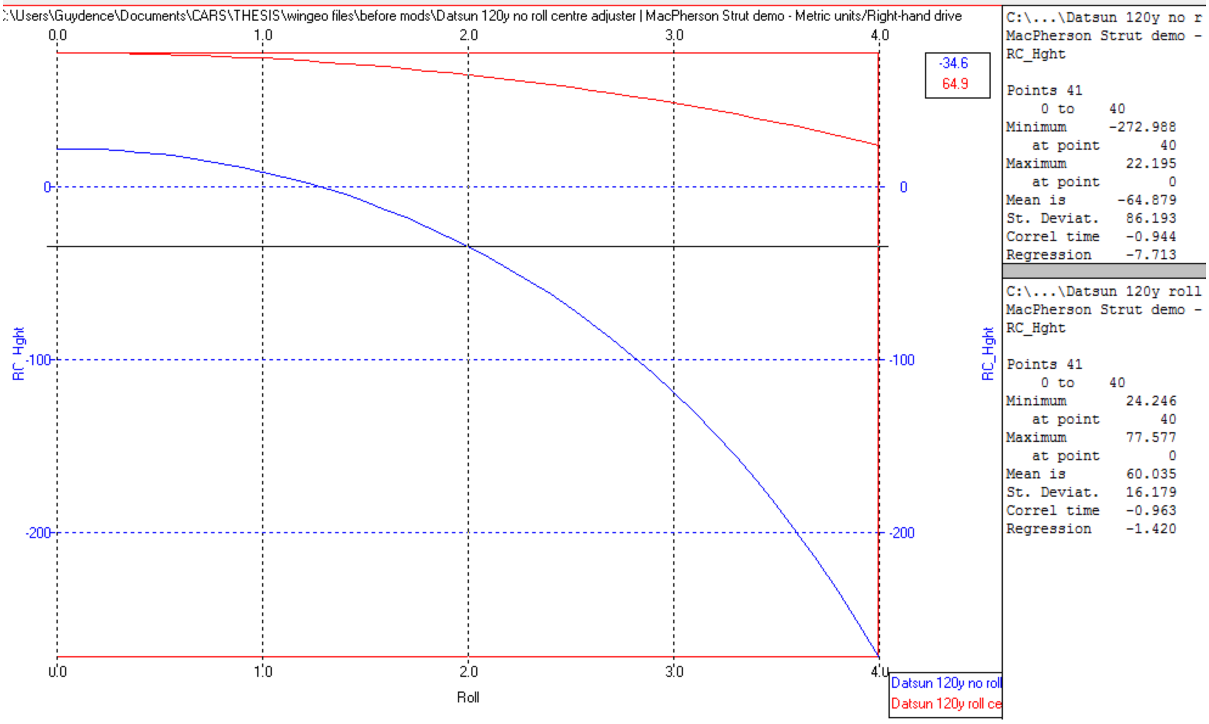

Fig. 4.1.2.8 Roll Iteration – Roll Centre Height

The roll centre height and its migration greatly affect the handling and weight transfer characteristics of the car. Without the spacers it is clear that the roll centre height reduces dramatically with roll to result in a roll centre that is well below the ground level. With the spacers fitted the roll centre is statically higher, although the reduction in height occurs at a much lower rate and remains above the ground level under all circumstances. The standard deviation also supports the increased consistency of the spacers with a reduction from 86.193 to 16.179.

Roll Centre Width

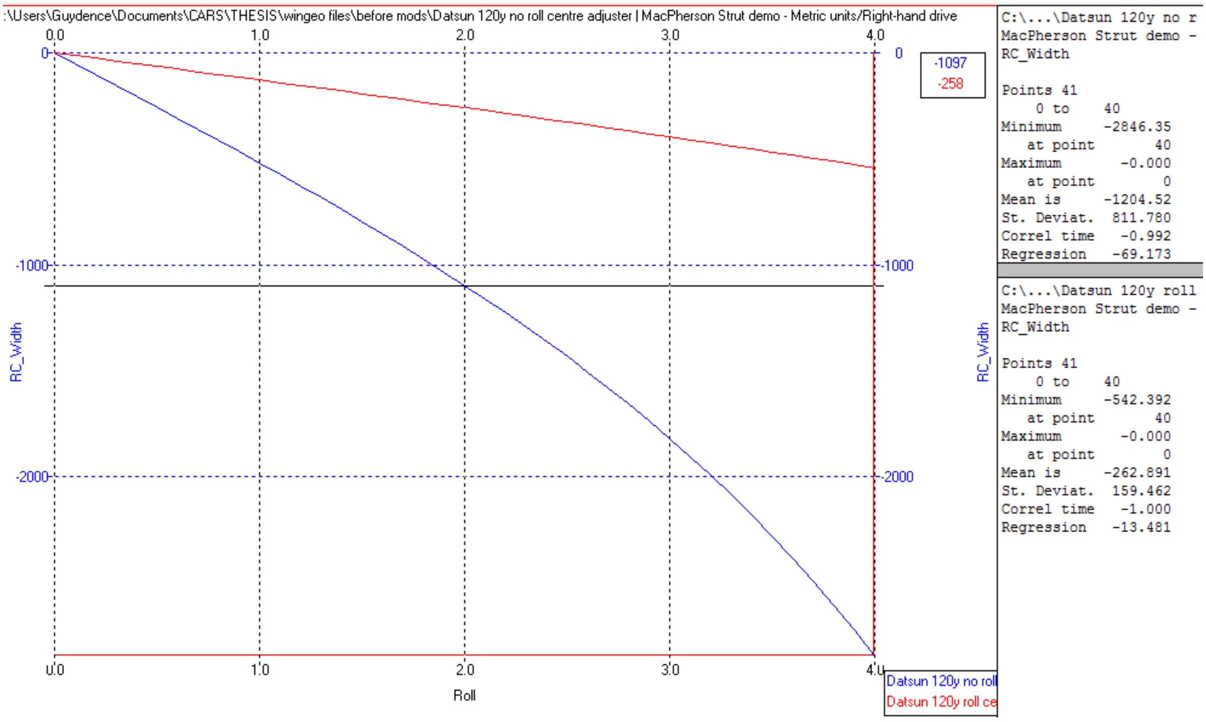

Fig. 4.1.2.9 Roll Iteration – Roll Centre Width

Roll centre width is similar in its application to the roll centre height. The lateral variation of the roll centre has been greatly reduced with the spacers, as can clearly be seen in figure 4.1.2.9. The standard deviation again shows that the spacers allow the suspension to better control the roll centre location with a reduction from 811.780 to 159.462 being achieved.

Roll Centre Moment Arm

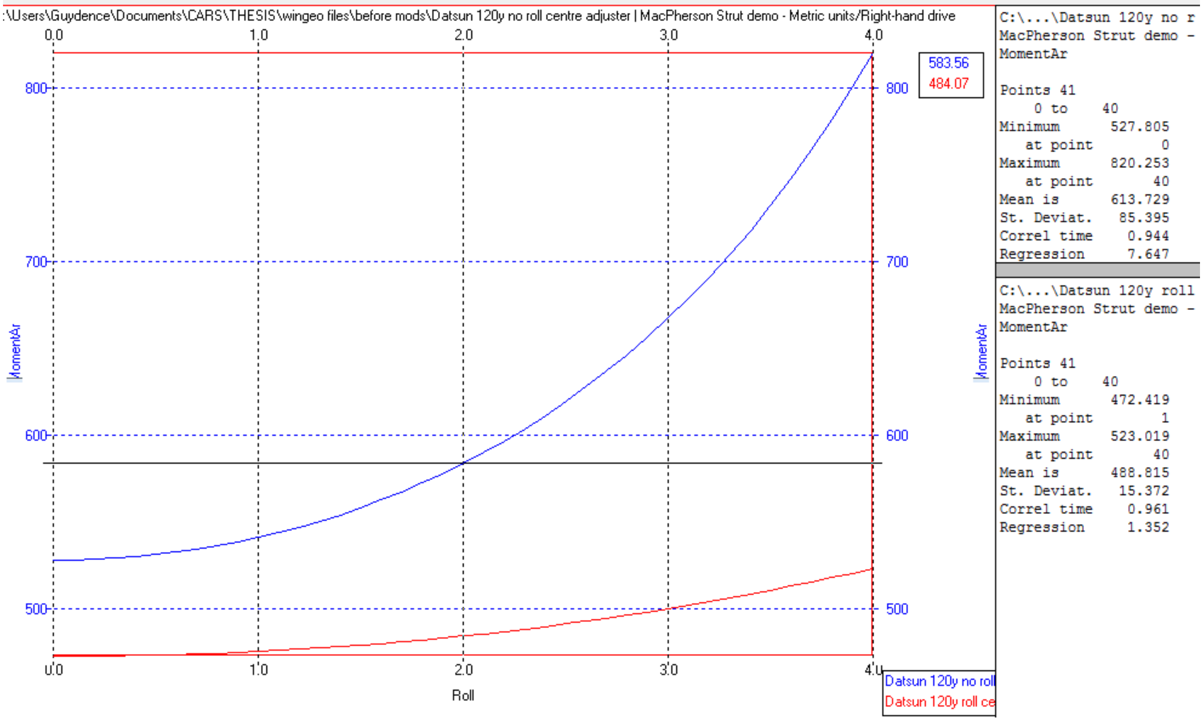

Fig. 4.1.2.10 Roll Iteration – Roll Centre Moment Arm

The roll centre moment arm is a combination of the effects of both the roll centre height and width. The moment arm has been reduced in all angles of roll with the fitment of the spacers. The lower value will result in less lateral weight transfer although the roll centre has been slightly raised. The progression of the moment arm has also been greatly reduced, with the tendency of an increasing moment being drastically lessened. The standard deviation again supports the increased consistency with an improvement from 85.395 to 15.372 showing that the weight transfer should be more consistent as well as reduced.

Jacking Centre of Gravity - Right

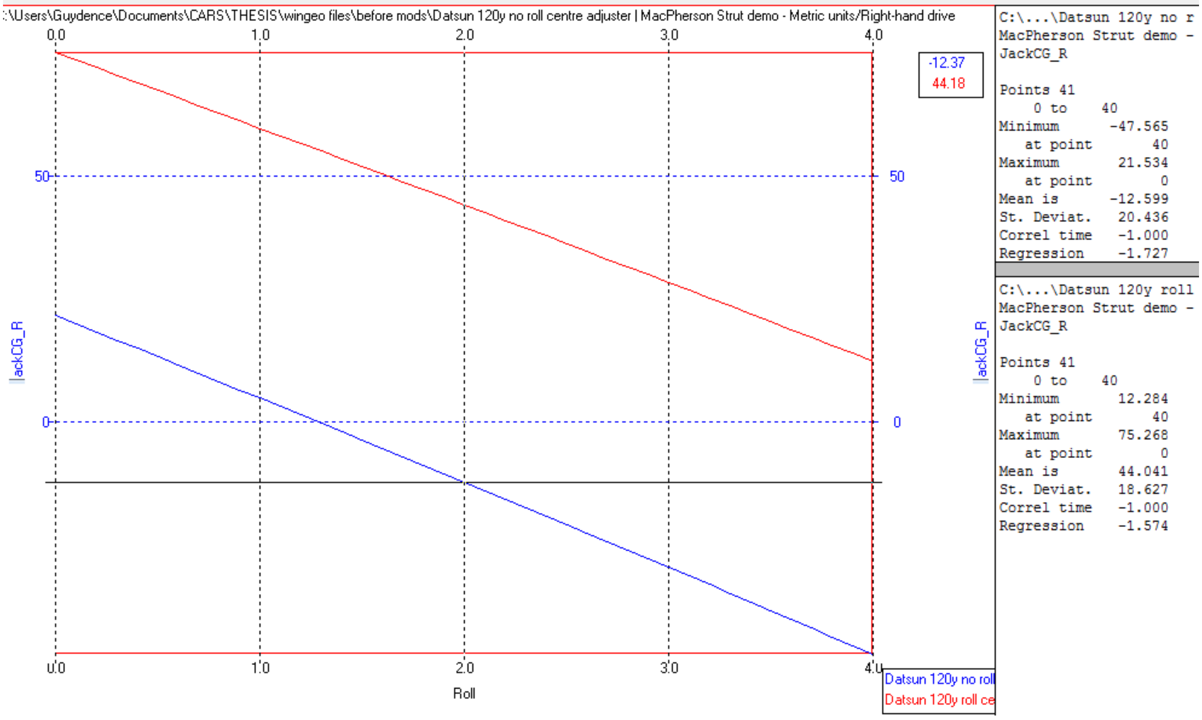

Fig. 4.1.2.11 Roll Iteration – Jacking Centre of Gravity – Right

Jacking Centre of Gravity - Left

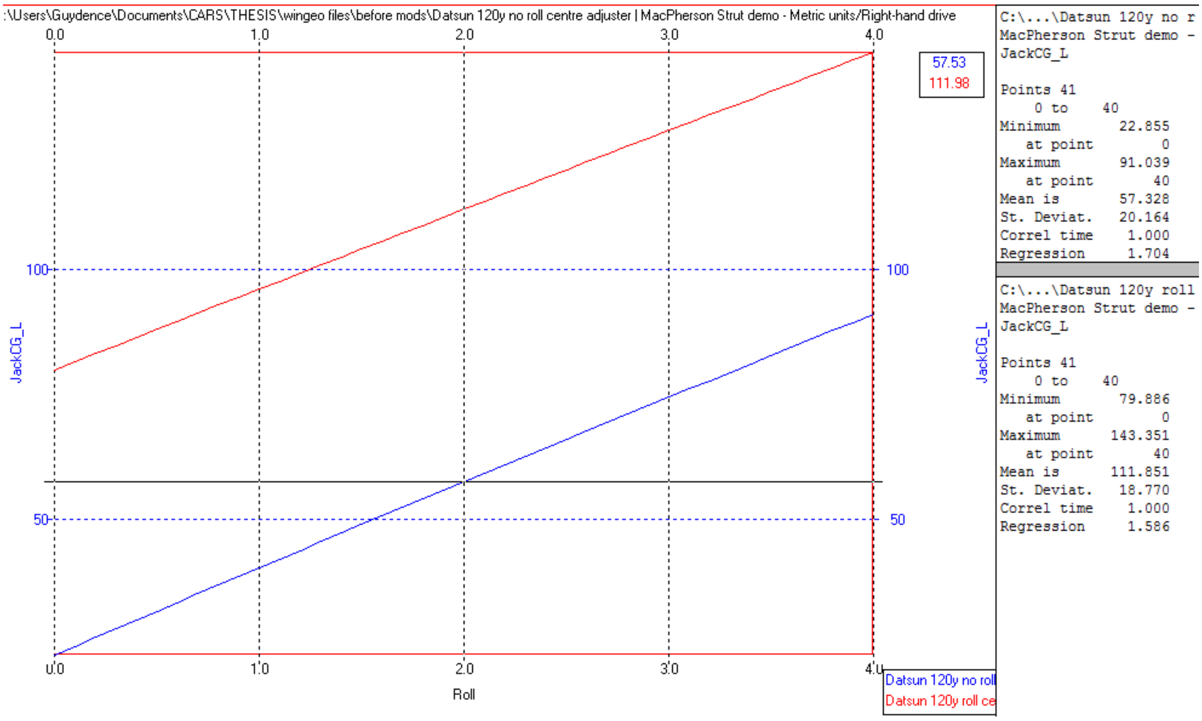

Fig. 4.1.2.12 Roll Iteration – Jacking Centre of Gravity – Left

The jacking centre of gravity has been increased with the roll centre and the progression of these points has been largely unchanged. The progression of these points have however been slightly reduced with both right and left sides standard deviation showing a reduction from 20.436 and 20.164 to 18.627 and 18.770 respectively.

The roll iteration has shown that the roll centre adjusters have greatly increased the roll centre control as well as bump steer characteristics. The results of the increased roll centre consistency are likely to result in less weight transfer and a more consistent handling car. The positive attributes of the roll centre adjusters are highlighted in the role iteration, with most variables producing more ideal values.

4.1.3 Steer Iteration: 0 to 40 degrees

Camber - Right (Outside)

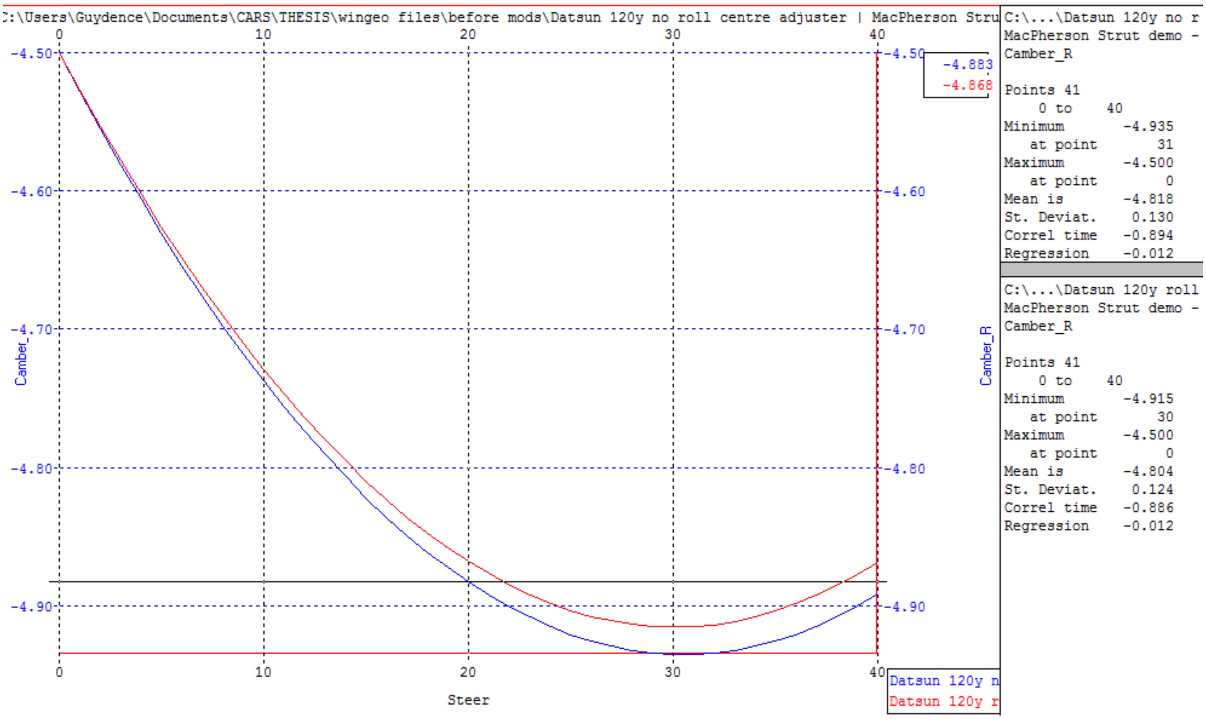

Fig. 4.1.3.1 Steer Iteration – Camber – Right (Outside)

The outside wheel carries the most load and hence its camber is of the most importance. The camber difference presented by the spacers fitment is minute and only shows up as a difference of 0.006 in the standard deviation comparison. The spacers result in a slight decrease of outside wheel camber, although the values are unlikely to make a noticeable difference by themselves.

Camber - Left (Inside)

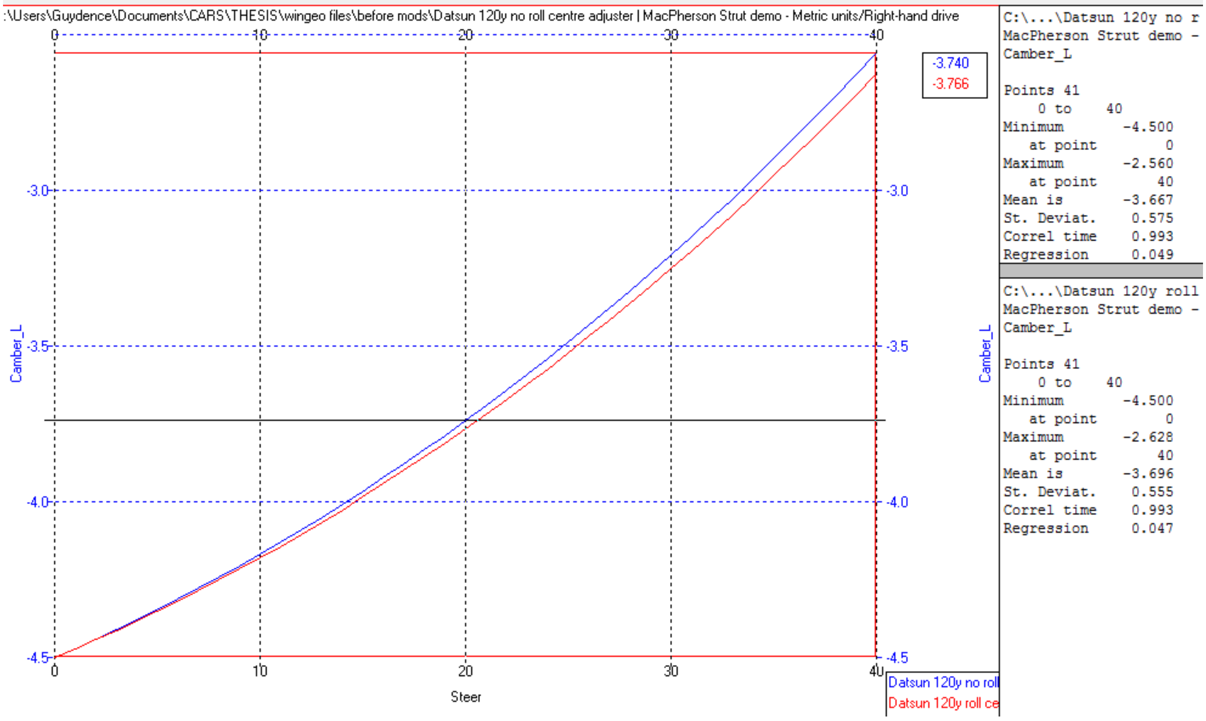

Fig. 4.1.3.2 Steer Iteration – Camber – Left (Inside)

The inside wheels camber angle has been slightly increased, an initiative that is less than ideal. The difference achieved in the camber is however very insignificant over the range of steering motion. The reduced weight on this wheel during cornering also reduces the significance of this value.

Castor – Right (Outside)

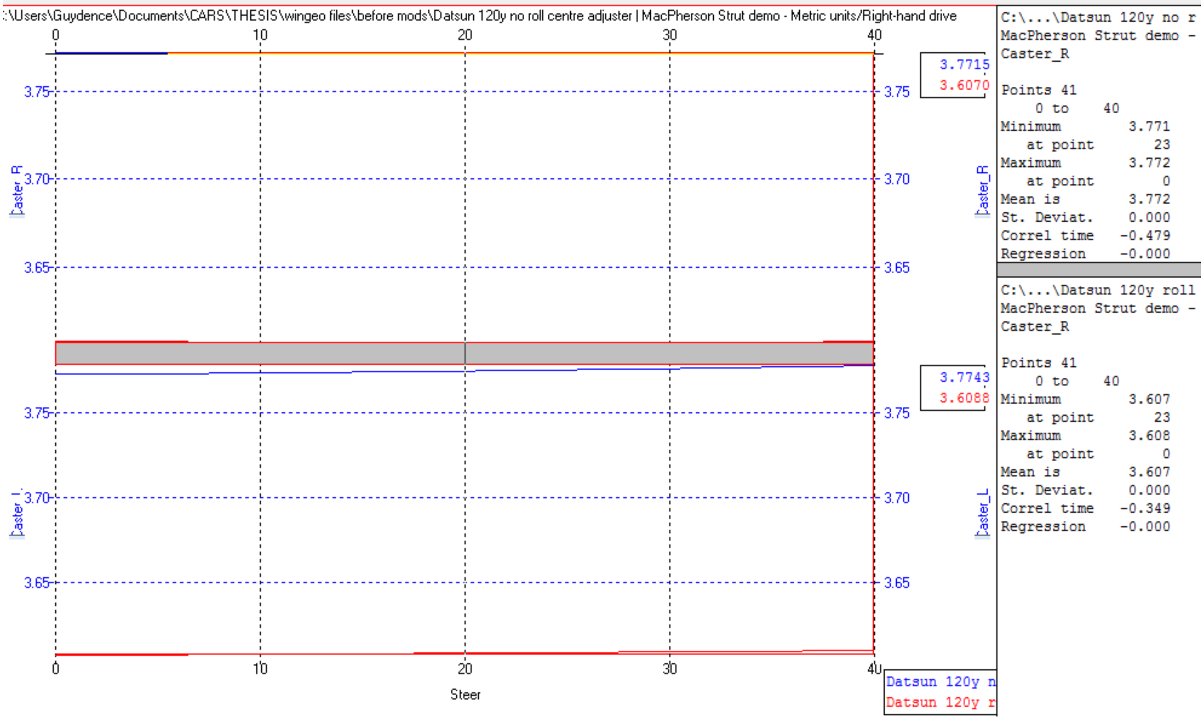

Fig. 4.1.3.3 Steer Iteration – Castor

The castor is highly unaffected by the steering input. The general reduction of castor remains with the spacers fitted and little to no variation is found in the values over the full range of steering angles.

Net Steer

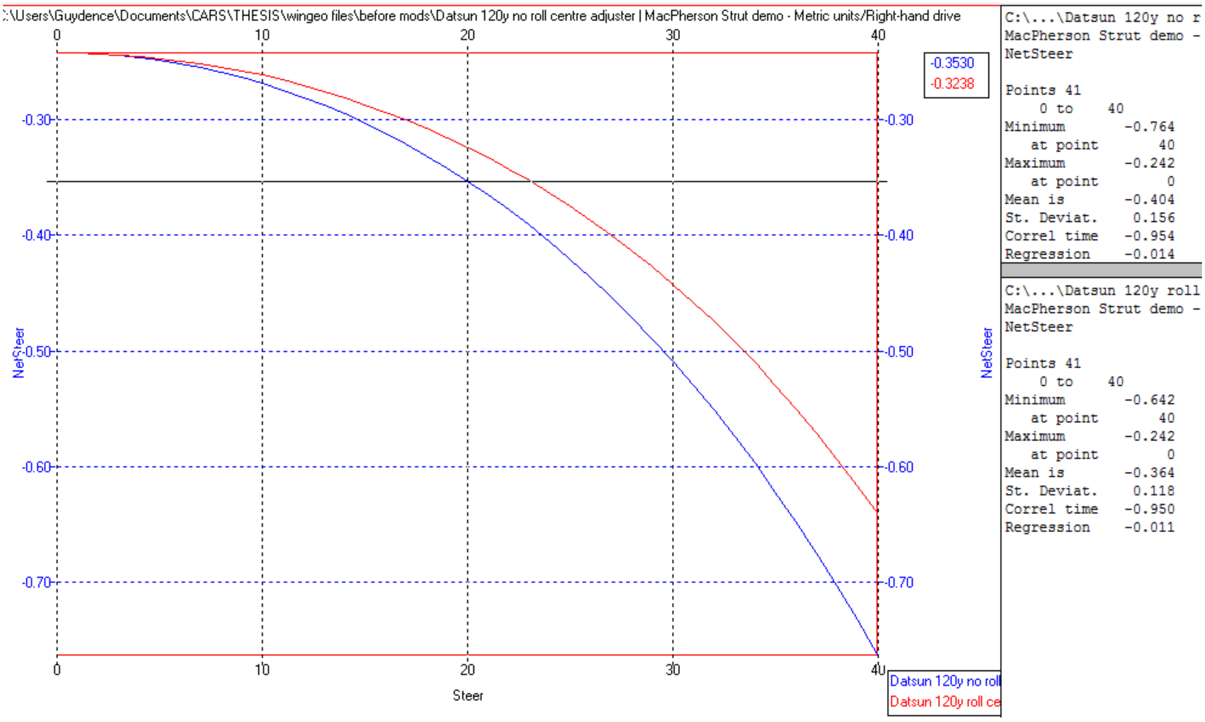

Fig 4.1.3.4 Steer Iteration – Net Steer

The net steer through the application of steering input is paramount to ensure the cars toe settings are operating to promote consistency and good turning ability. The steering input results in an increasing amount of toe out. This is beneficial to helping the car turn throughout the corner although too much toe out will result in an unstable car and excessive tyre scrubbing. The spacers reduce the maximum toe out setting and promote a more consistent steering process.

Ackerman

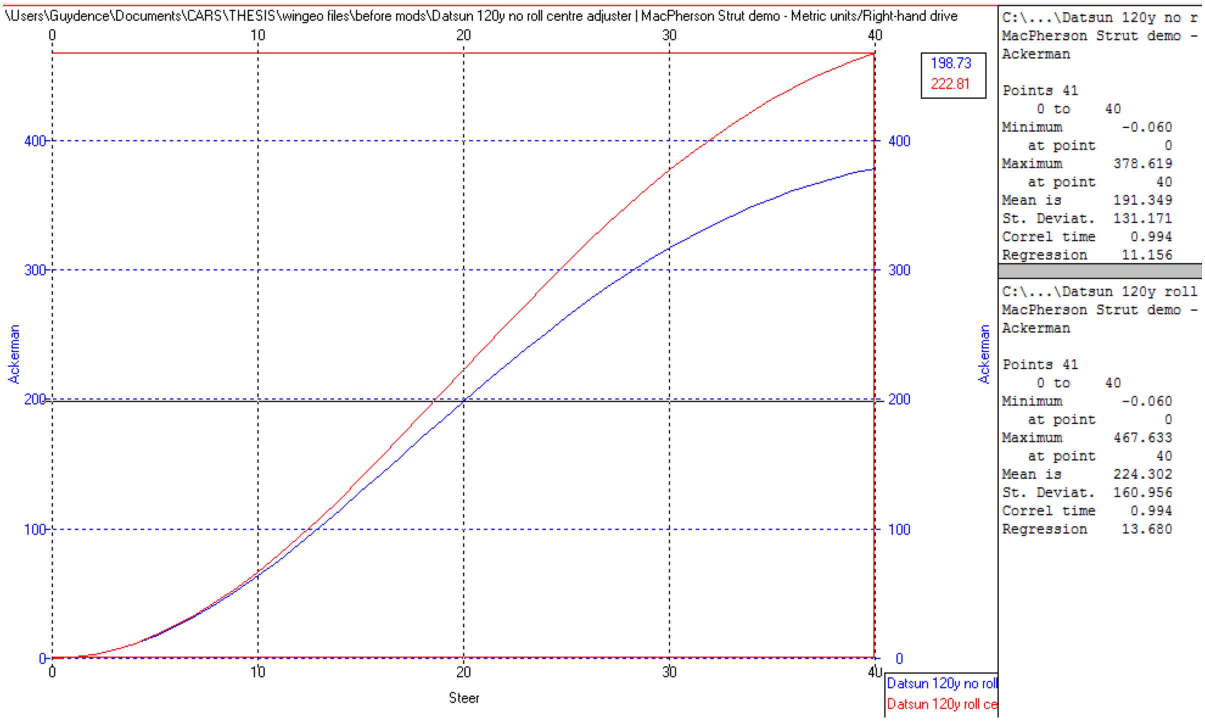

Fig. 4.1.3.5 Steer Iteration – Ackerman

The application of the Ackerman principle is directly related to the toe out nature of the steering system. The increase in Ackerman shown in the graph by the application of the spacers should help with the removal of some of the understeering tendencies that remain in the car.

Scrub – Right (Outside)

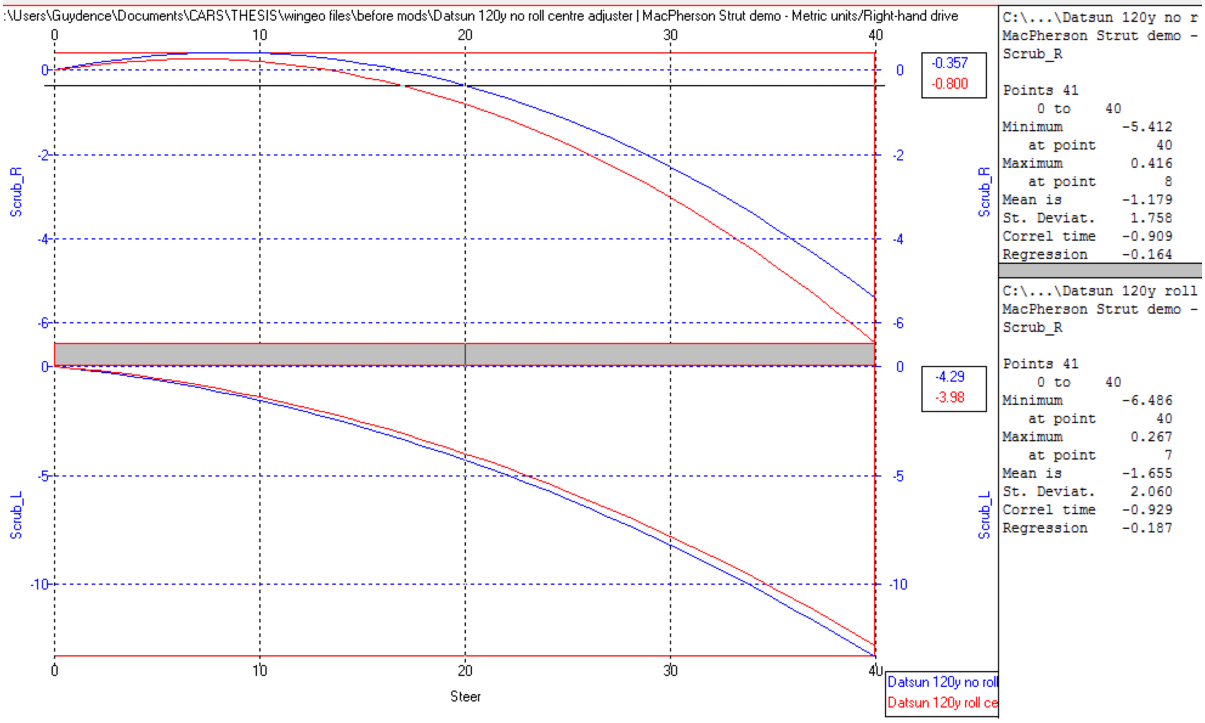

Fig. 4.1.3.6 Steer Iteration – Scrub

The overall effect of the scrub is more important than the individual wheel scrub, although the above graphs have been included to show the effect that each wheel has to the net scrub.

Net Scrub

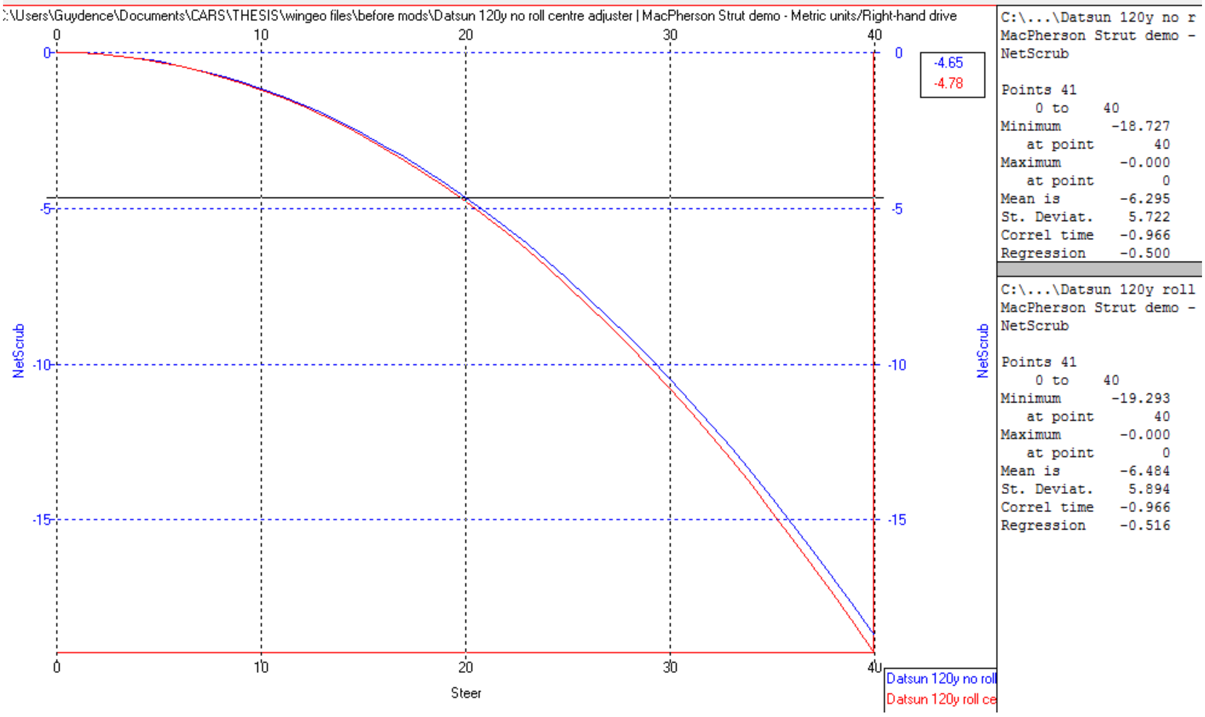

Fig. 4.1.3.7 Steer Iteration – Net Scrub

The net scrub or track width change is marginal. The difference that the spacers make to the overall track presents little difference to the potential of the car with only 0.566 mm of track change being found at 40 degrees of steering angle. The tendency towards a reduced track with higher steering inputs is built into the standard steering arms and suspension geometry. This difference is however low, so the effect is expected to be negligible.

Roll Centre Height

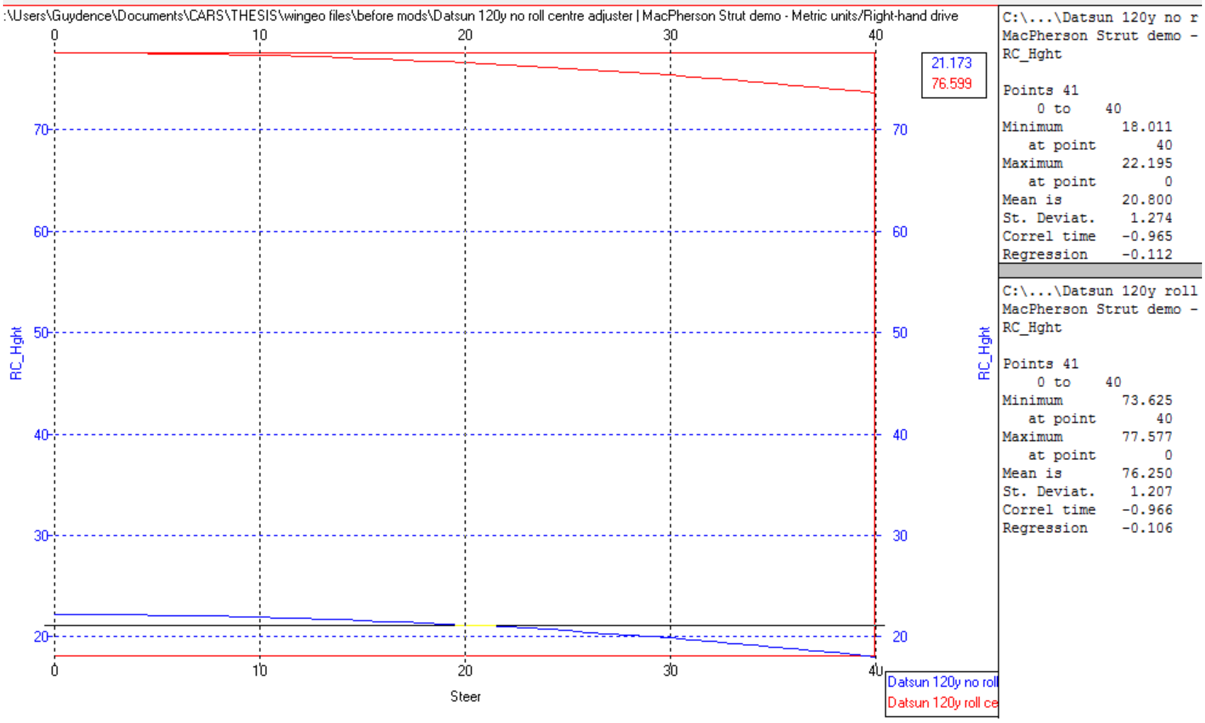

Fig. 4.1.3.8 Steer Iteration – Roll Centre Height

The role centre height and migration is only slightly affected by the steering input. The roll centre height starts at the static values and continues to reduce as steering input is increased. This trend is consistent in both situations, although the spacers offer a slight roll centre height control benefit through a decreased standard deviation of 1.207 compared to 1.274 for the standard fixture.

Roll Centre Width

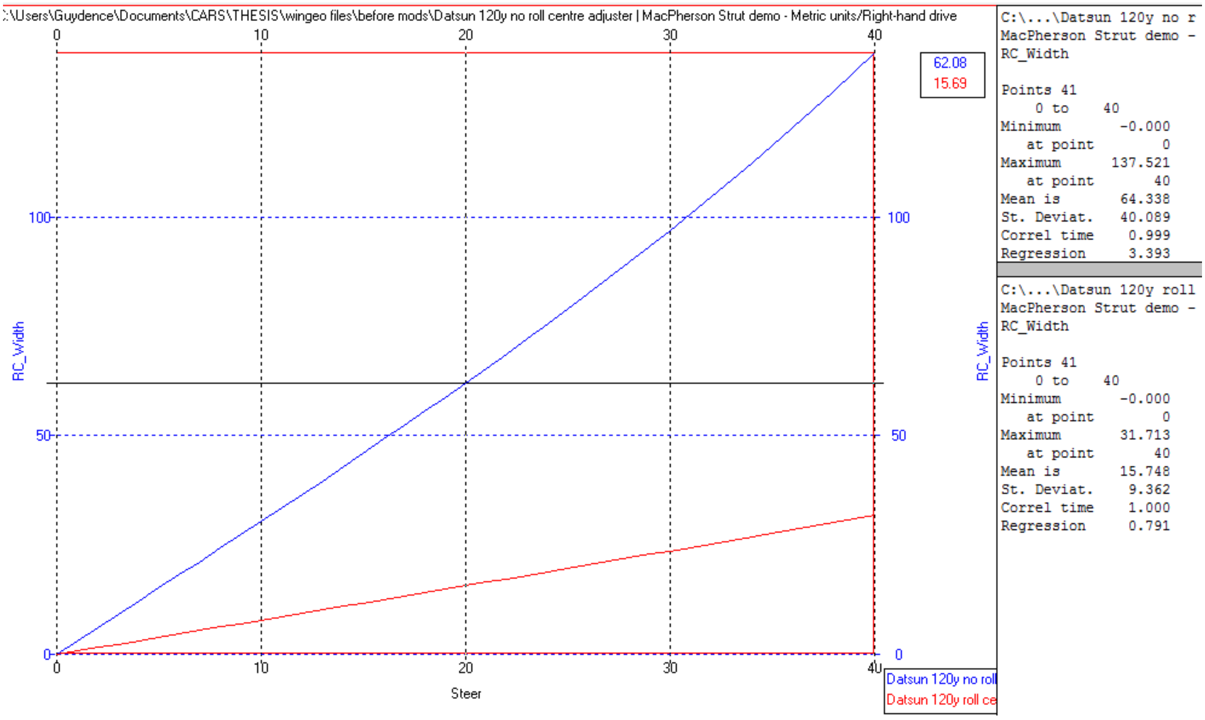

Fig. 4.1.3.9 Steer Iteration – Roll Centre Width

The roll centres lateral location is greatly affected by the application of roll centre adjusters. As noticed in the above graph the roll centre migration has been greatly reduced. The total roll centre width has been reduced from 62.08 mm to 15.69 mm at 20 degrees of steering input. This reduction shows an increased level of roll centre control from the suspension and steering system and results in a more consistent performance. The standard deviation of the system also shows a more consistent system with a reduction being achieved from 40.089 to 9.362 over the full 40 degrees of steering travel.

Roll Centre Moment Arm

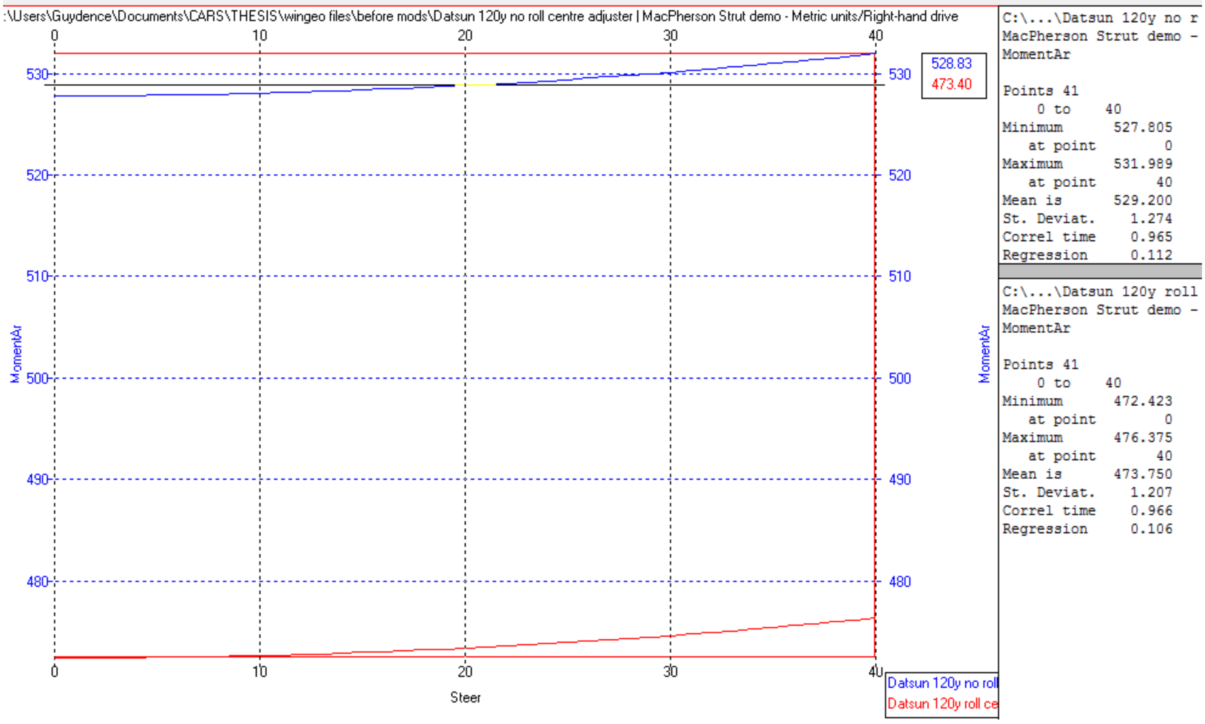

Fig. 4.1.3.10 Steer Iteration – Roll Centre Moment Arm

The combination of both the roll centre height and width has resulted in a moment arm that has changed very little in overall progression. The overall length of the moment arm has however remained in the reduced state as achieved by the fitment of the spacers. A standard deviation difference of 0.067 shows that a very slight increase in moment arm length management has been achieved.

Jacking Centre of Gravity – Right

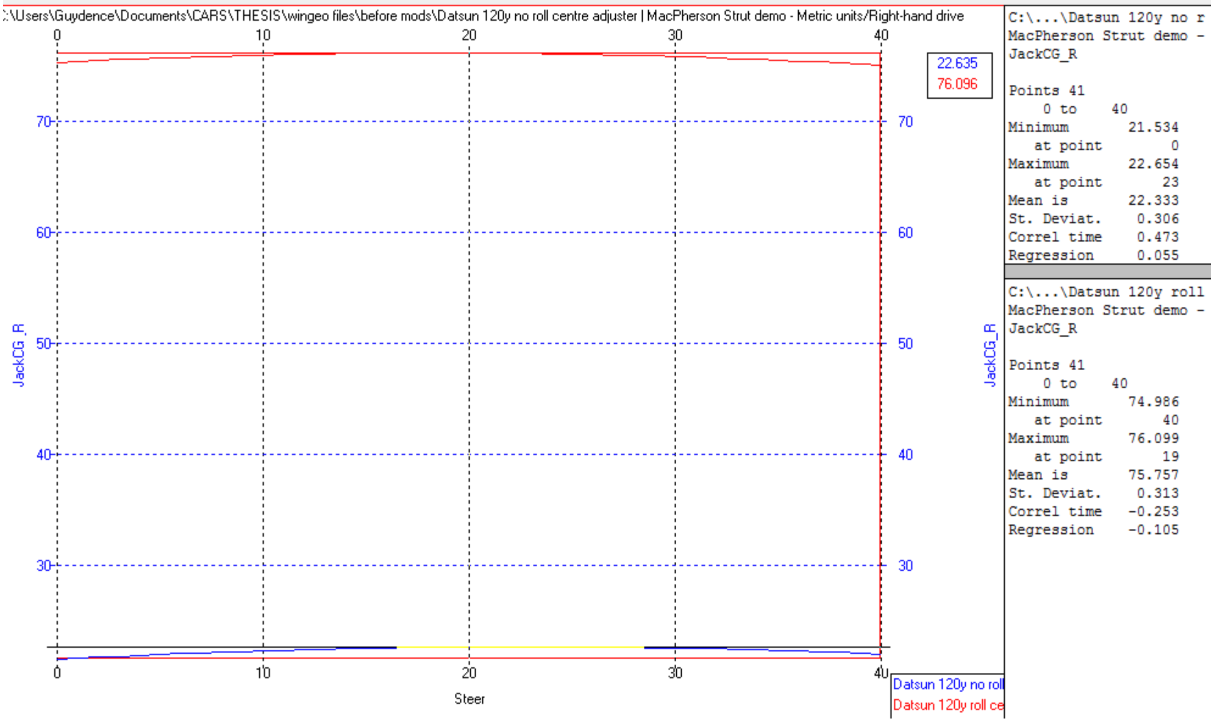

Fig 4.1.3.11 Steer Iteration – Jacking Centre of Gravity – Right

Jacking Centre of Gravity – Left

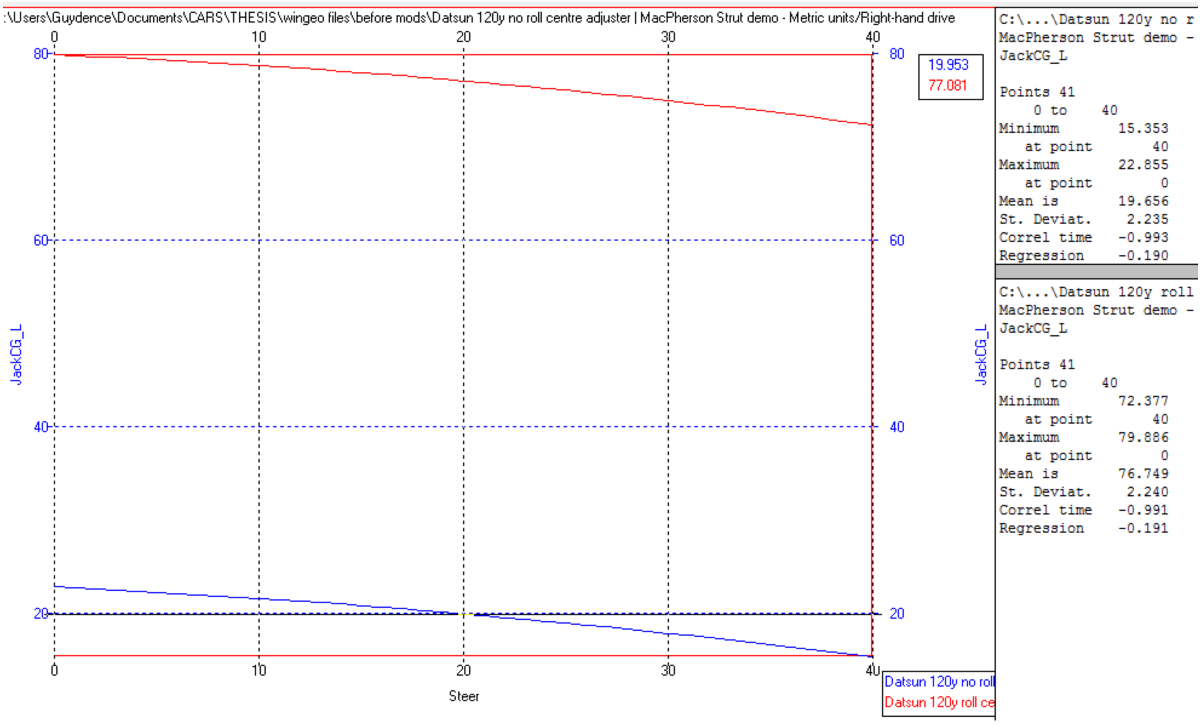

Fig 4.1.3.12 Steer Iteration – Jacking Centre of Gravity – Left

Steering inputs do affect the jacking centre of gravity, although the overall affects of the spacers do not alter the nature of the migration. The height altercations found by the spacers remain through steering inputs and generally show that a higher jacking force will be envisioned.

The steer iteration has generally been unaffected by the roll centre adjusters. The overall affect of the spacers have still been achieved and the nature of the variables is highly unchanged. The main benefit of Ackerman and net toe steer show that the spacers present a slightly more ideal situation for the steering system.

4.1.4 Cornering Sequence

The actual variables reached by the car during the cornering process must be evaluated to ensure the data previously compared is not misleading. The product of all the variables will also tell a more informative story as to the actual values the car is likely to achieve at different segments of the corner.

As different amounts of travel and roll are experienced in different parts of the corner, the following values will be used. The front axle weight is assumed to be 600kg and the g force data is taken as an average of the events completed before the modifications. The braking g force is a combination of the front and rear tyres although most braking is performed by the front. It was then deemed that the g force data for the braking performance would b used. This is overestimating the ride height change in all directions although this may be useful if softer spring rates are to be used in the future. Limitations of this model are that jacking forces and there affect on the ride height is not included.

$$ride\ height\ change = \frac{g\ force\ \times weight\ on\ axle}{spring\ rate\ \times 2}$$

$$ride\ height\ change = \frac{1.13\ \times 600}{8\ \times 2}$$

= 42.38mm lower than static (full braking)

$$ride\ height\ change = \frac{0.7\ \times 600}{8\ \times 2}$$

= 26.25mm higher than static (full acceleration)

The roll angle was then measured by using a front only image as shown in figure 4.1.4.1.

Fig. 4.1.4.1 Roll Angle Calculation

Left = 33

Right = 41

Distance between = 178

Height difference = 8

Angle of roll = $\tan^{-1}$ (8/178)

= 2.57 degrees

As can be seen, the photo is not perfectly straight on to the car. This will result in the calculated value being larger than the actual roll angle being achieved. For the purpose of the analysis 2.5 degrees will be sufficient.

Full roll angle = 2.5 degrees

Full steer through the corner was deemed to be 20 degrees.

Figure 4.1.4.2 below shows the cornering process used for the analysis. It was deemed that the onset of the ride, roll and steer would be linear up til the apex and following the apex. This is simulating an ideal driver on a flat corner achieving an extremely smooth driving style.

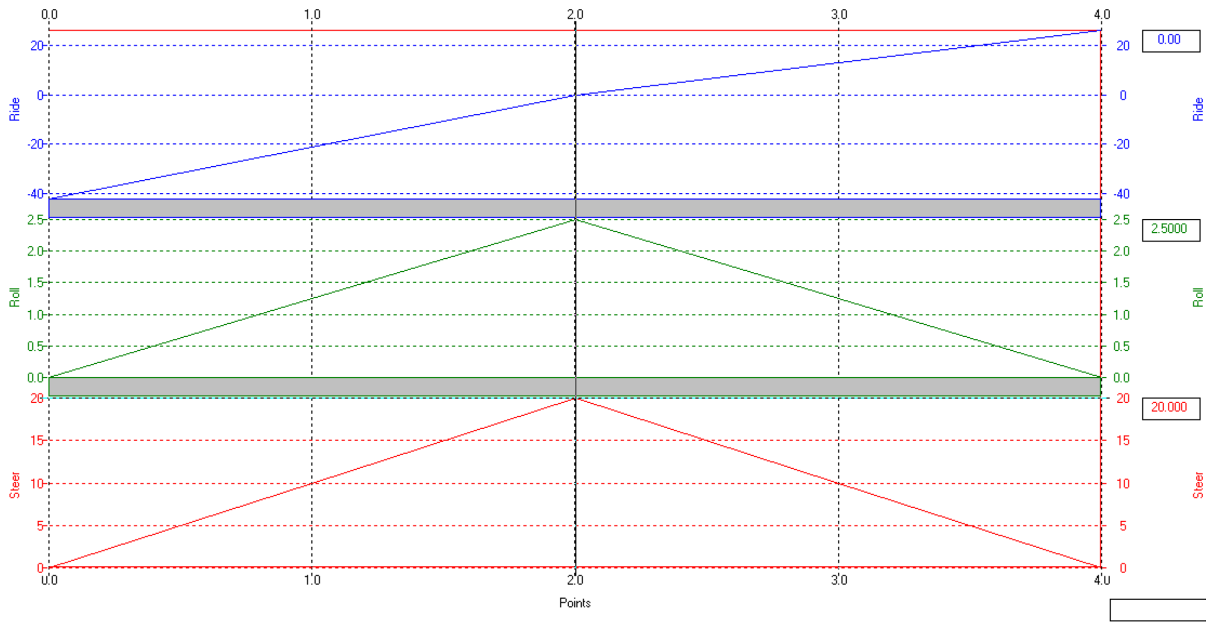

Fig. 4.1.4.2 Cornering Sequence used

Camber - Right (Outside)

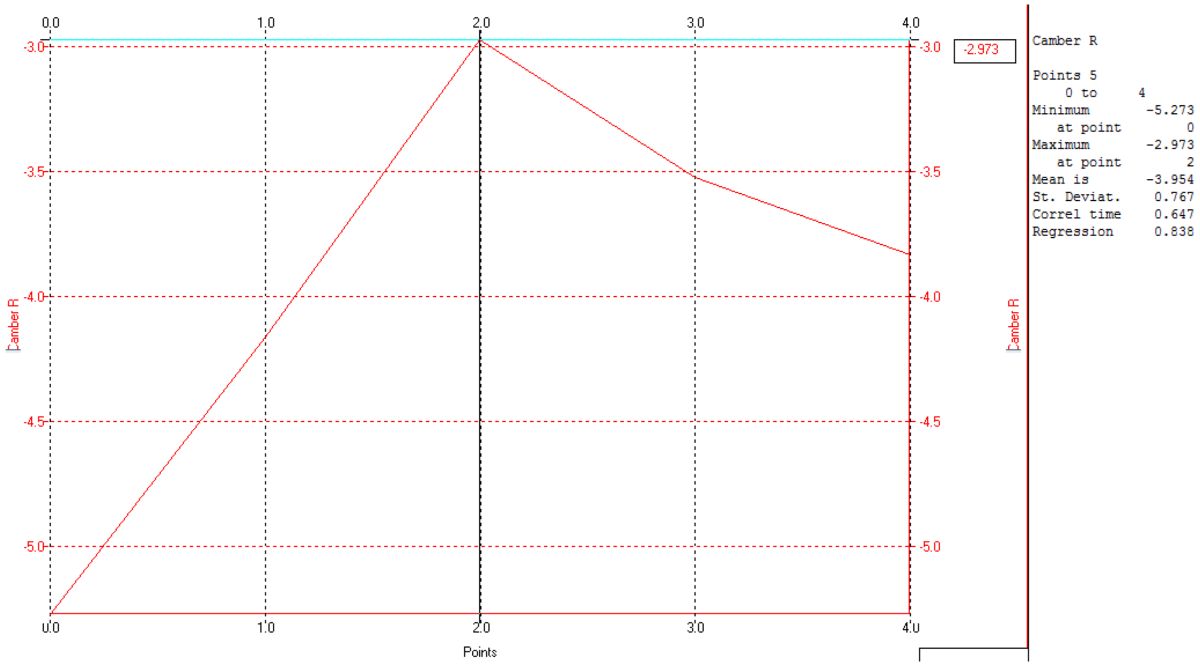

Fig. 4.1.4.3 Cornering Sequence – Camber – Right – No RCA's

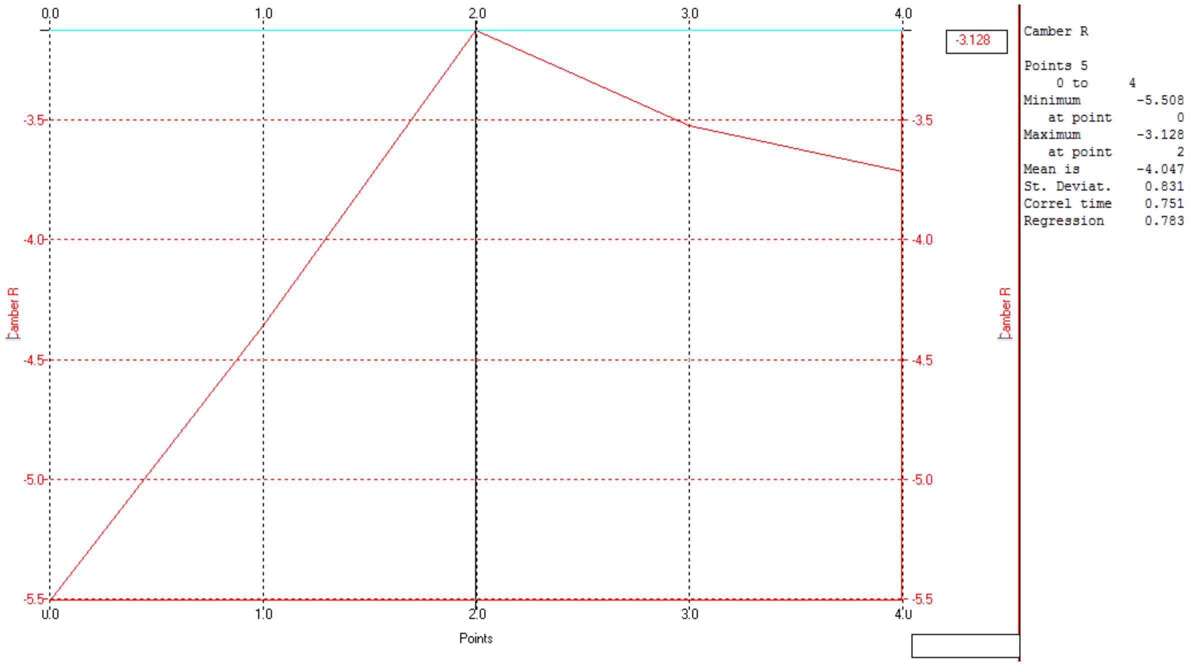

Fig. 4.1.4.4 Cornering Sequence- Camber – Left – RCA's fitted

The camber change throughout the cornering sequence has remained in the same succession, with camber decreasing to the apex and then increasing once acceleration has begun. The overall value of the camber has increased with the addition of the roll centre adjusters, a result which shows that the fitment of the spacers can be accompanied by a reduction in the static negative camber levels. This may help alleviate some of the additional camber that is generated during braking applications. The slight decrease in negative camber on corner exit will promote better turn out of the corner due to more closely aligning the camber with the maximum camber thrust available.

Net Steer

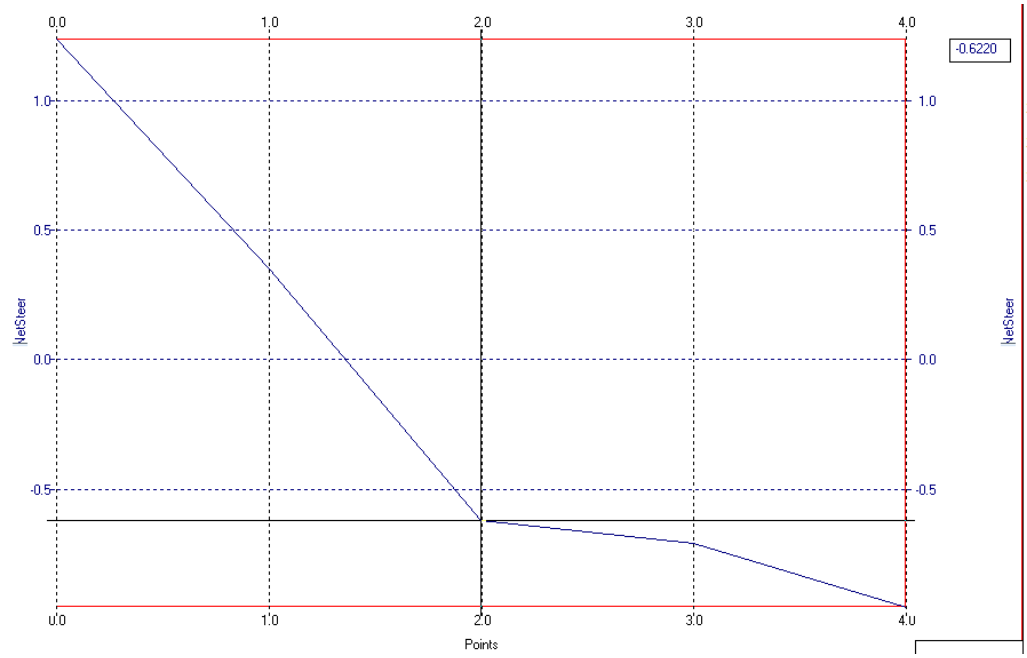

Fig. 4.1.4.5 Cornering Sequence – Net Steer – No RCA's

The net steer has been greatly reduced. As seen in the accompanying two graphs, the roll centre adjusters have made a substantial difference. Before the spacers were fitted the steering progressed from a large amount of toe in to a large amount of toe out just before the corner apex. In theory this provided corner entry understeer which led to oversteer just before the apex. This was not envisioned in the current settings of the car and may lead to the fact that other parameters such as roll centres where far from optimal.

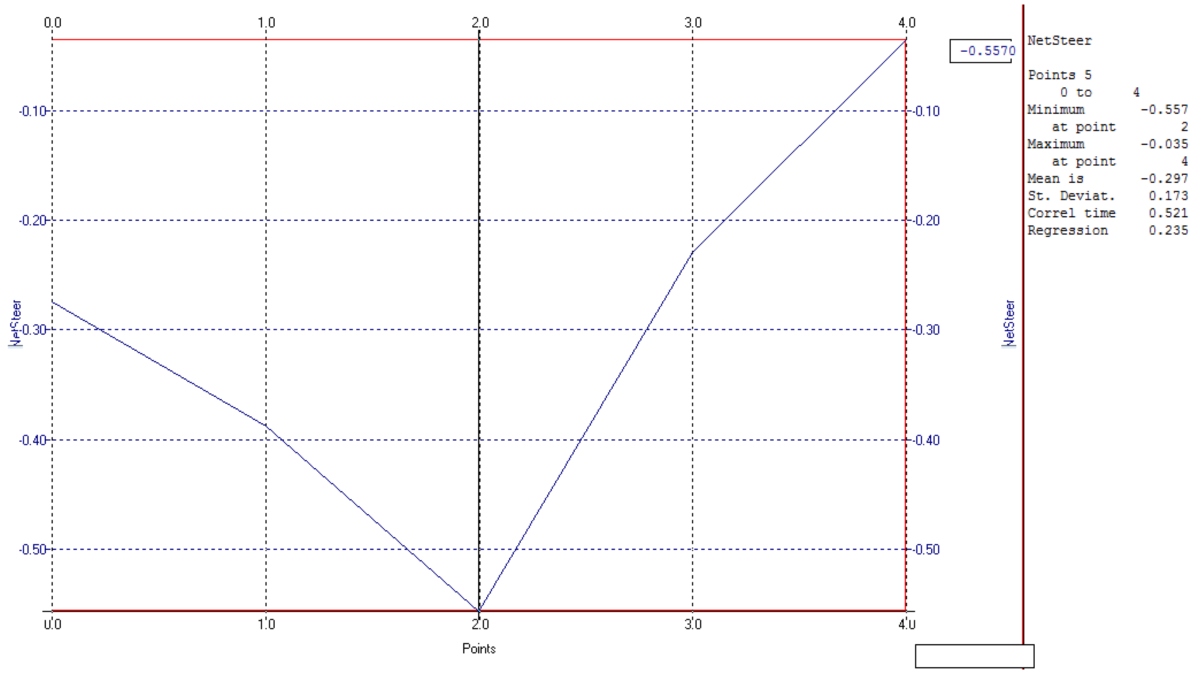

Fig. 4.1.4.6 Cornering Sequence – Net Steer – RCA's fitted

Once the spacers were fitted the net steer was greatly altered to result in toe out under braking. This value continues to rise until the apex at which point the toe out is slowly reducing until no toe out is achieved at full exit. The result of this is more turn in, although maybe slightly unsettled under braking. The apex presents the same values and then the decreasing values of toe out should result in a fairly stable car on corner exit. The overall feeling should have less understeer and be more progressive and consistent with a reduction in the standard deviation value from 0.802 to 0.173.

Ackerman

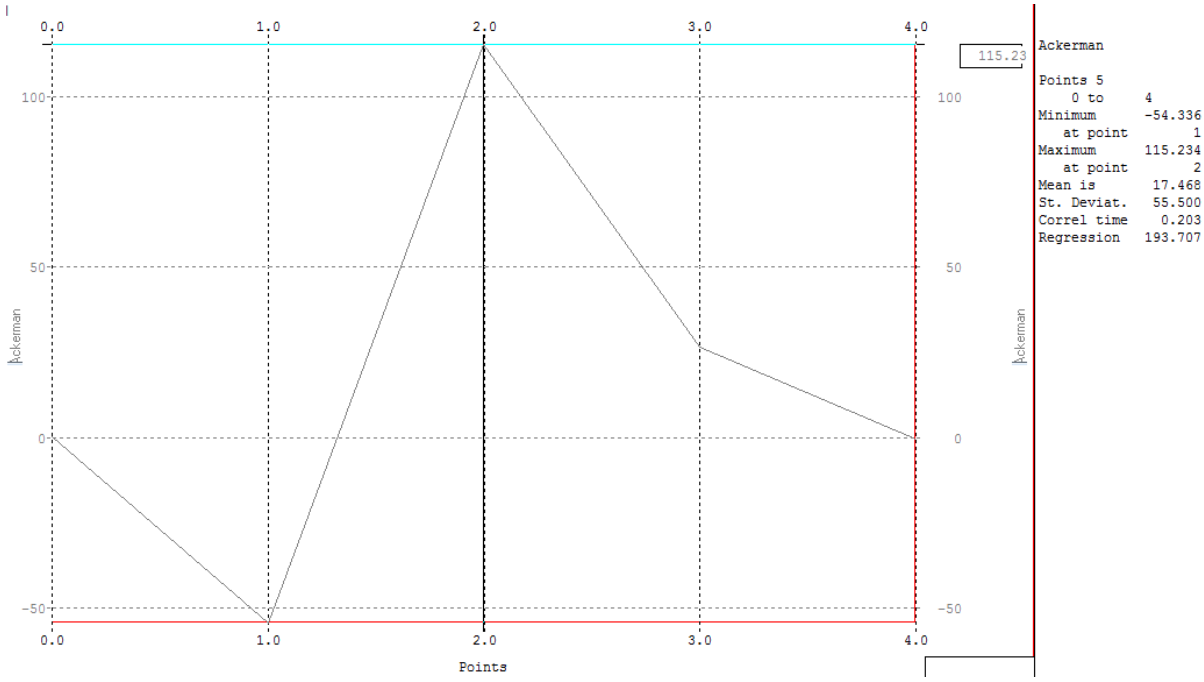

Fig. 4.1.4.7 Corning Sequence – Ackerman – No RCA's

The Ackerman throughout the corner process has been successfully altered to increase consistency of the values. The beginning value of 0 is then turned into a negative value on the approach to the corner apex at which stage it reaches its maximum positive value. During the corner exit the Ackerman returns to 0 in a relatively smooth fashion.

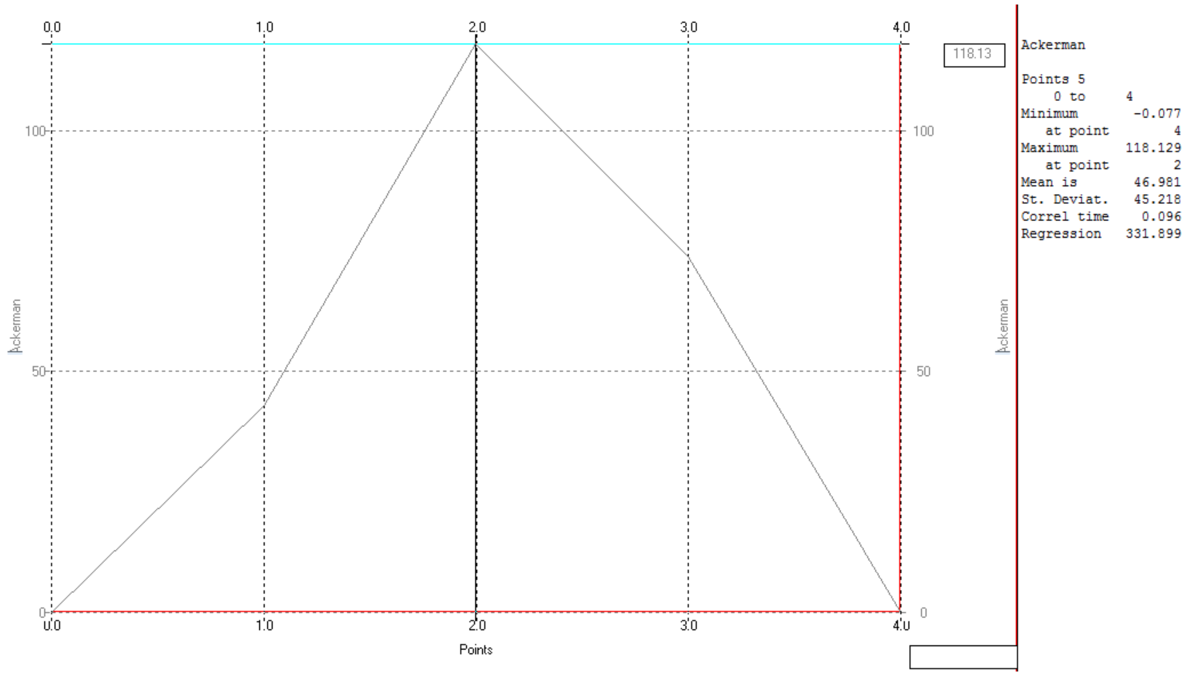

Fig 4.1.4.8 Cornering Sequence – Ackerman – RCA's fitted

The spacers do little to alter the overall effects of the Ackerman with it starting, finishing and even reaching a similar value at the apex (maximum). The progression however is greatly improved and both the corner entry and exit have a smooth transition to and from the maximum value. The result is a more consistent steering system that should promote an increased amount of turn in. The standard deviation of the system also supports the more consistent result with a decrease from 55.500 to 45.218.

Net Scrub – Track Width Change

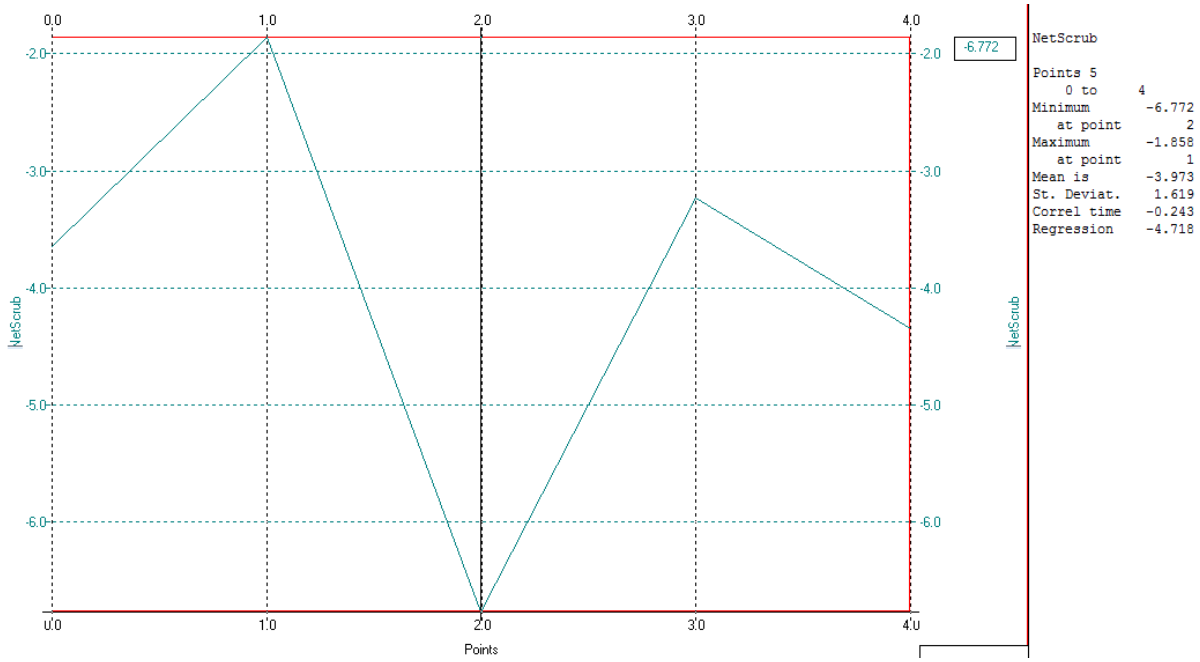

Fig 4.1.4.9 Cornering Sequence – Net Scrub – No RCA's

The track width change presents many different variables to the overall handling considerations. Without the spacers the suspension begins with a reduced track that increases slightly before it decreases towards the apex of the corner. This reduction in track towards the apex will result in reduced corning potential. The track then widens during corner exit to promote an increased cornering potential.

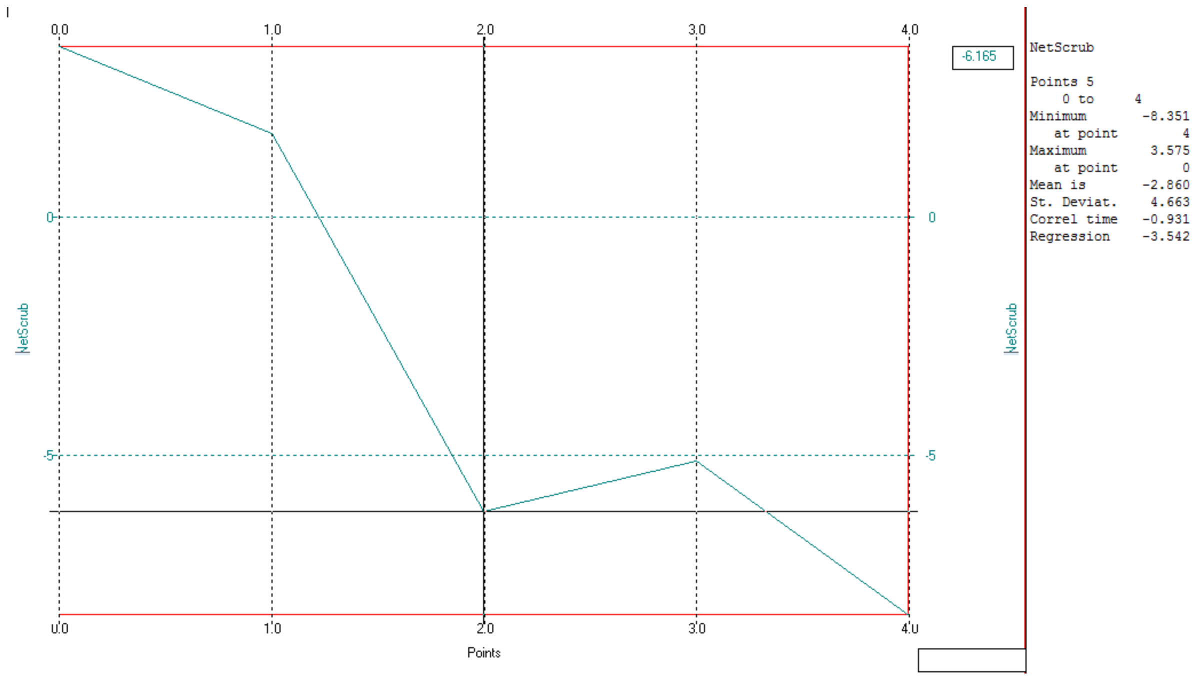

Figure 4.1.4.10 Cornering Sequence – Net Scrub – RCA's fitted

The spacers alter most aspects of the track migration. The track is actually increased to begin with and will promote good turn in under brakes. The actual track at the apex will be reduced to a similar value of that achieved without the roll centre adjusters before the track reduces further in the corner exit. This will promote less front cornering potential on the corner exit and possibly corner exit understeer. The spacers do however result in a smoother and more progressive scrub graph. The overall effect is an average value of -2.860 instead of -3.973, showing that although the nature of the progression may have changed, the overall effect is a net increase in cornering potential.

Roll Centre Height

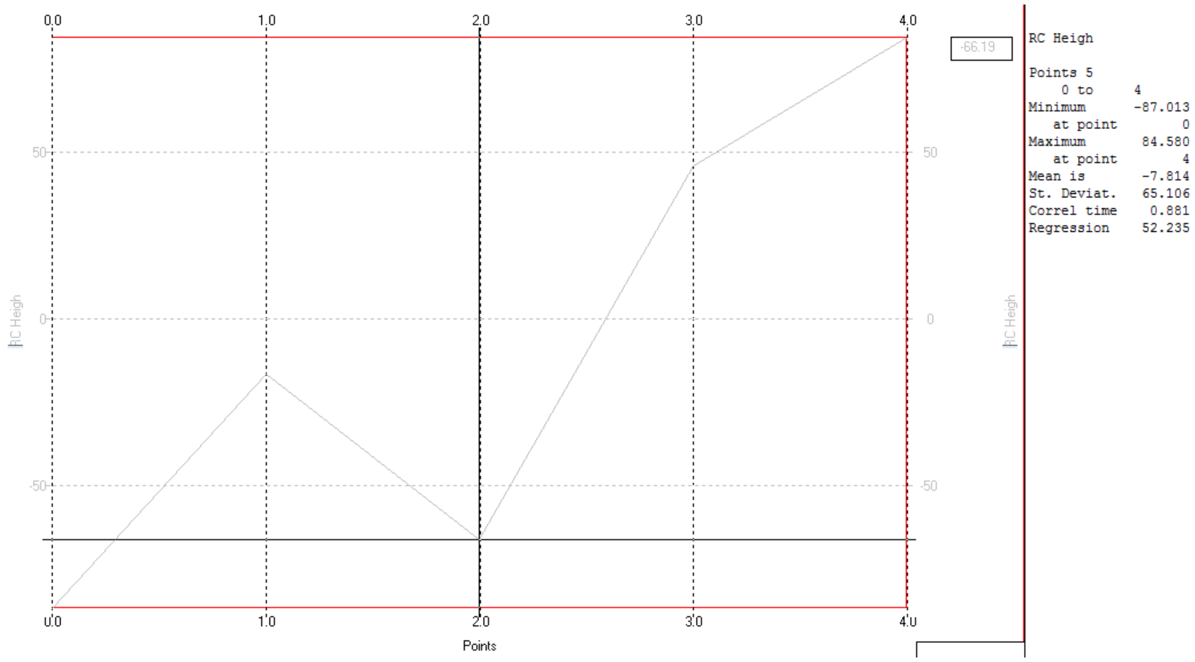

Fig 4.1.4.11 Cornering Sequence – Roll Centre Height – No RCA's

The roll centre height has changed completely due to the fitting of the roll centre adjusters. Before fitting, in corner entry the roll centre is 87 mm below the ground, a very low point that results in large weight transfer characteristics. The roll centre remains below the ground, albeit at a higher position, until just after the corner apex. At this stage the roll centre is raised above the ground and continues to climb until a maximum of 85 mm above the ground at full power exit.

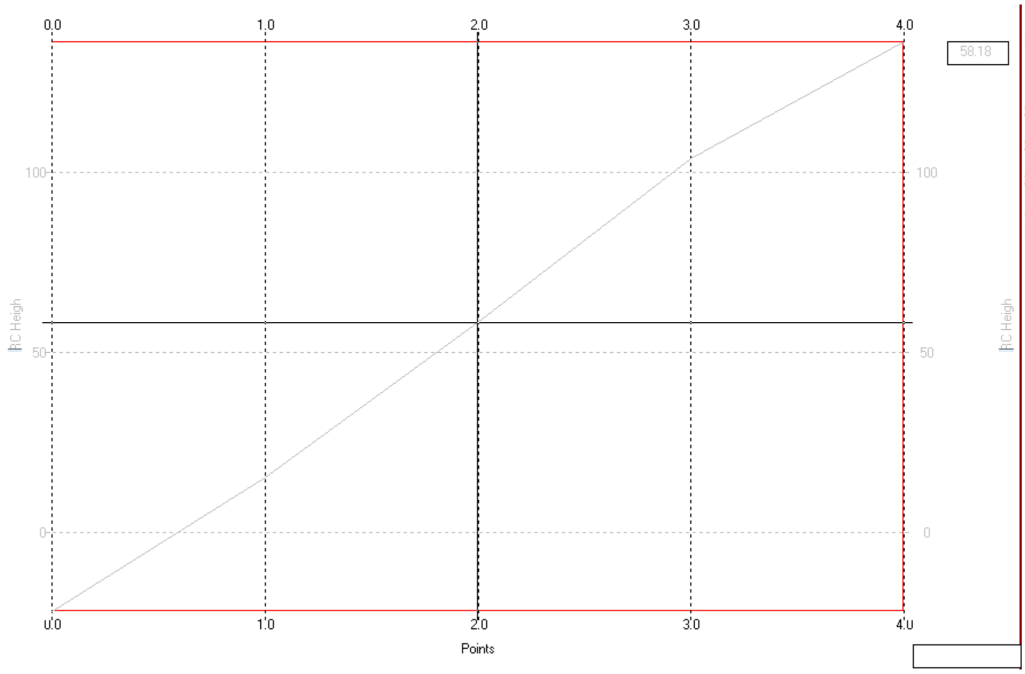

Fig. 4.1.4.12 Cornering Sequence – Roll Centre Height – RCA's fitted

The spacers alter the roll centre height and result in a much more linear roll centre height migration. The turn initiation is made with a roll centre that is now only 22 mm below the ground. The higher roll centre location will reduce weight transfer in turn in. The progression through the corner now results in a much more linear raise in roll centre height, a condition which is important in ensuring the car remains predictable. The final roll centre is much higher than previously found. Ideally the roll centre would remain at the same height, although limitations in the type of suspension will restrict this from occurring. The overall result is a mean roll centre height of 58.287 mm instead of -7.814, and a reduction in weight transfer albeit with an increase in jacking forces. The roll centre control has also been increased with the standard deviation being reduced from 65.106 mm to 57.303 mm.

Roll Centre Width

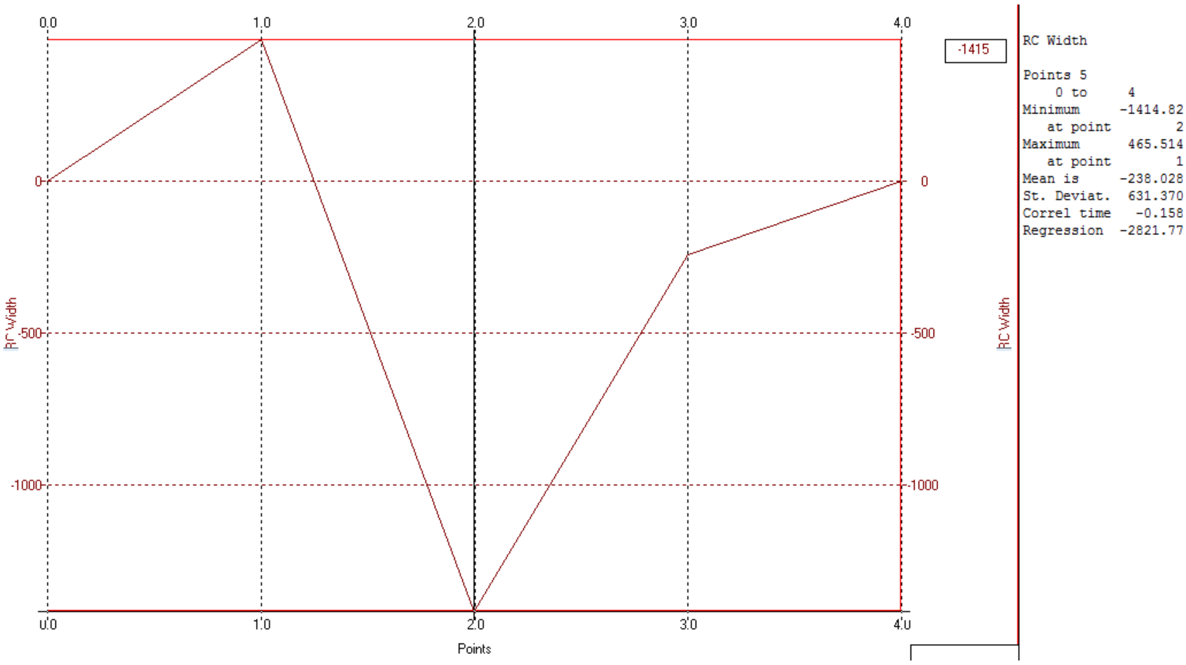

Fig 4.1.4.13 Cornering Sequence – Roll Centre Width – No RCA's

The roll centre width is perhaps the variable that the roll centre adjusters make the most difference in. Before the spacers were fitted the roll centre width was extremely unconstrained and large values were the result. In the corner entry phase the result varied from 466 mm to -1415 mm at the apex. The large variation shows that the roll centre is limited in its lateral control through the cornering process. After the apex the point returns to the centre relatively quickly, although the extremely large value at the apex has already resulted in large contributions to the moment arm.

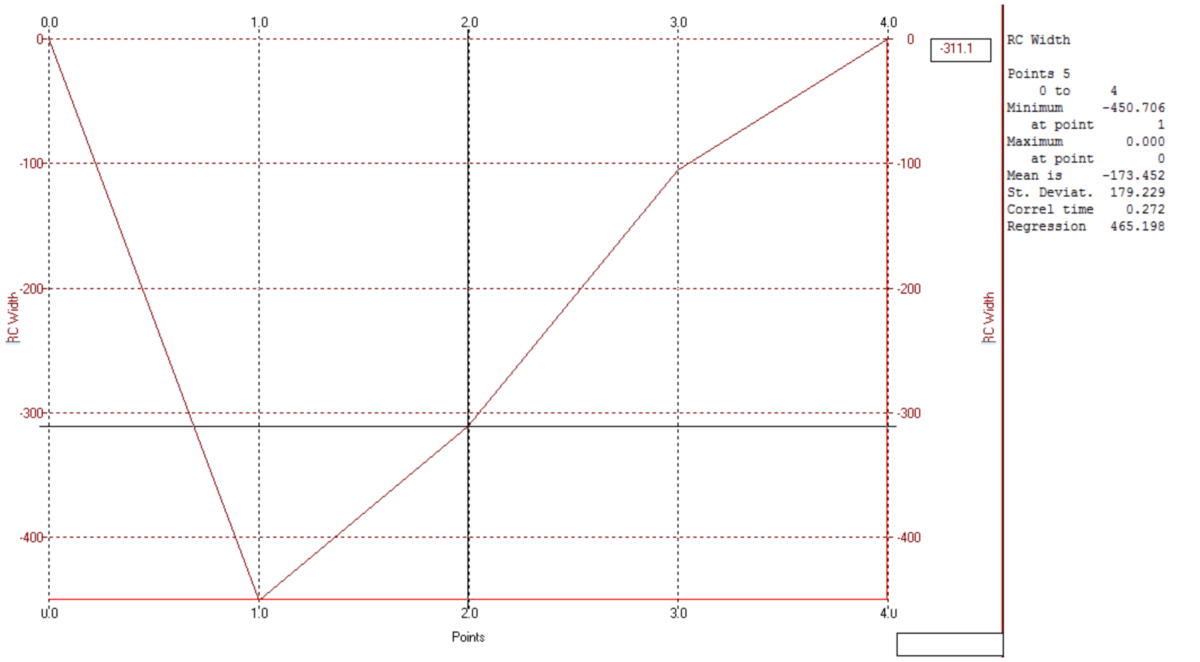

Fig. 4.1.4.14 Cornering Sequence – Roll Centre Width – RCA's fitted

The roll centre adjusters have greatly increased the control on the lateral location of the roll centre. The roll centre is always in the same direction and sees a smooth migration to the new maximum of -451 mm, a reduction in roll centre migration of 964 mm. After the maximum value during the corner entry phase, the lateral location of the roll centre is progressively brought back to the centre. The mean roll centre width shows the decreased contribution to the moment arm, with the decreased average lowered from -238.028 mm to -173.452 mm. The standard deviation shows the higher consistency of the location with a massive decrease from 631.370 to 179.229.

Roll Centre Moment Arm

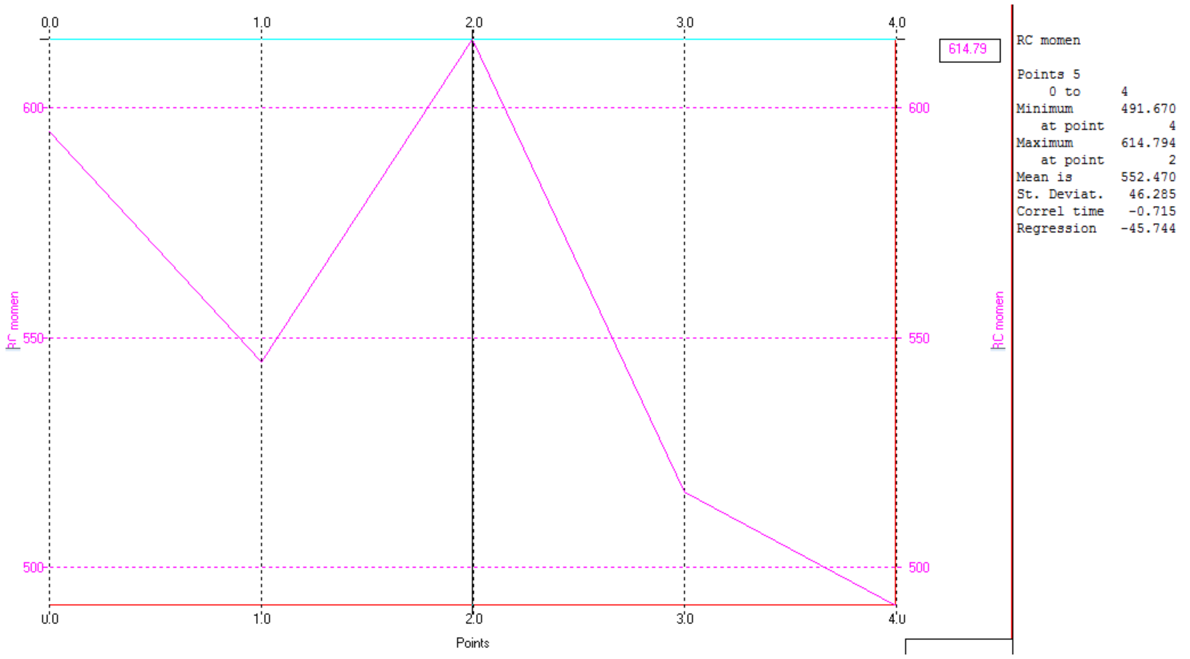

Fig. 4.1.4.15 Cornering Sequence – Roll Centre Moment Arm – No RCA's

The roll centre moment arm is a representation of the total roll centre migration. Before the spacers where fitted the moment arm started around 590 mm and decreased on initial corner entry. The moment arm was then increased to a maximum of 615 before decreasing steadily to 492 mm on corner exit. The relatively high values of the roll centre moment arm show that a large amount of weight transfer is being performed.

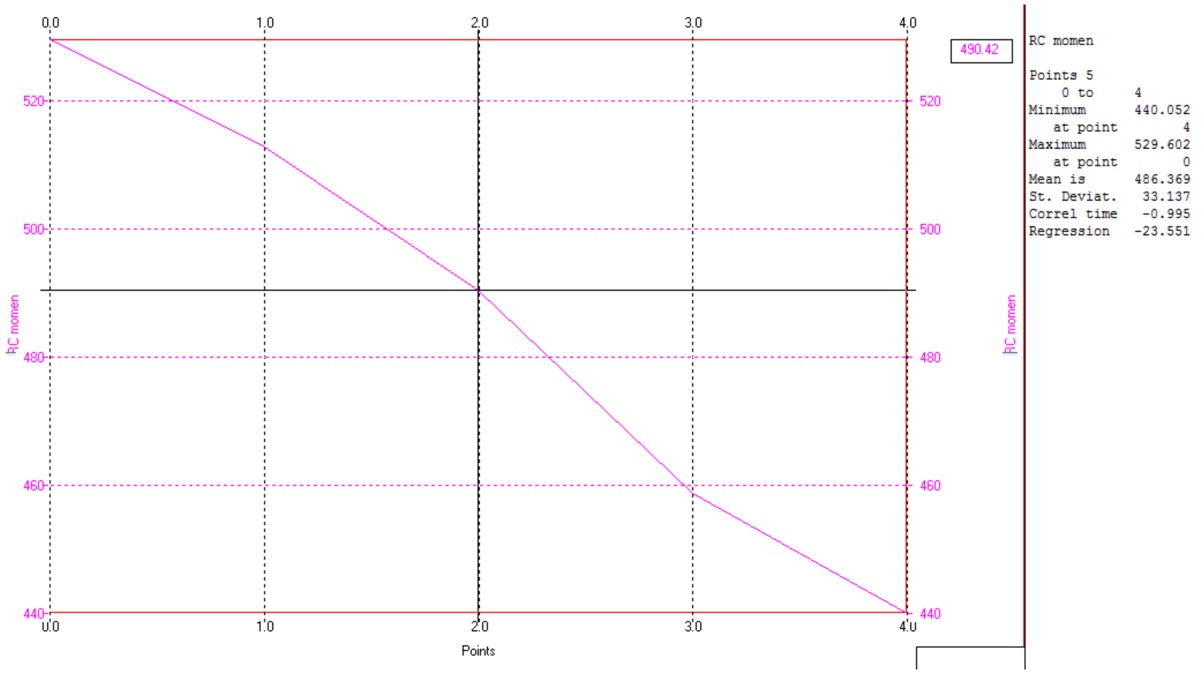

Fig. 4.1.4.16 Cornering Sequence – Roll Centre Moment Arm – RCA's fitted

The roll centre adjusters have decreased the moment arm. The overall effect is a reduction in the lateral weight transfer through limiting the effect of the moment arm. The progression of the moment arm is also particularly linear and a good indication of a consistent and more predictable handling car. The lower values are supported by the average moment arm length which has now been decreased from 552.470mm to 486.369mm. The more consistent and linear progression has also resulted in a lower standard deviation of 33.137 compared to 46.285 before the spacers were fitted.

The roll centre adjusters have increased the roll centre control and decreased the moment arm through all aspects of the cornering sequence. The front suspension has generally performed better in almost all categories with the fitment of the spacers. The analysis found that it was in the best interest of both the cornering potential and consistency considerations that the roll centre adjusters be fitted.

4.2 Roll Centre Adjuster Fitting

Physical fitting of the roll centre adjusters is a straightforward process of unbolting the strut housing from the ball joint/steering arm and placing the spacers between the two components. This will lower the outer end of the lower control arm and steering tie rod end by 25 mm. This presented its own problems due to tight constraints that are evident in the front suspension and steering. The outer tie rod end to wheel clearance is as issue which manifested from the application of highly positive offset wheels. The positive offset is required to ensure that the tyre does no contact the outer guard, a condition that has also limited the use of less negative camber. The steering tie rod end required modification to ensure that it would not foul on the wheel.

4.2.1 Steering Modifications

In order for the roll centre adjusters to fit in the designated position and at the designed height, the steering tie rod end height had to be reduced. There was the option to reduce the height of the spacers, although this will limit the effect they have on the suspension and steering parameters. For these reasons it was deemed viable to alter the tie rod end to gain the full potential from the spacers.

The tie rod end was a limiting factor in wheel choice. We could escape the problems of the tie rod end clearance by increasing the size of the wheels, although this presented a larger initial outlay due to the cost of new wheels and tyres. This would also alter one of the settings which were being kept constant to try and control the variables.

The tie rod end had very limited clearance to the wheel. With the fitting of the roll centre adjusters the tie rod end would foul on the inside of the wheel as shown in the below figures.

Fig. 4.2.1.1 Outer Tie Rod Clearance

Fig. 4.2.1.2 Outer Tie Rod Clearance

The tie rod end is enlarged in the section below the actual joint. As can be seen in figure 4.2.1.1 the removal of the excess tie rod end would result in the 25 mm clearance required for the fitting of the roll centre adjusters.

The decision was then made to purchase rod ends for the wheel end of the tie rod. The available rod ends were reduced in the overall length of the rod end, a fact which required new tie rods with an overall increased length to be purchased. The available product was purchased from a suspension and steering specialist outlet. On the passenger side, the inner tie rod end was replaced with one from the driver’s side. This enabled the use of normal right hand threads for the outer tie rods. This also made parts cheaper and more available in the event of a failure.

Fig. 4.2.1.3 Tie Rod End - Length difference

Fig. 4.2.1.4 New Steering Components

The tapered nature of the steering arm connection presented its own problems as no components were available to facilitate the use of the standard taper. It was then decided that the high grade (10.9) bolt that was used in the new tie rod end would be laterally located by threading the steering arm hole to the required size. The thread was larger than the taper so a high-quality deep thread could be cut. The bolt was heavily tightened to ensure that the preload induced by the bolt, resulted in the stress being transferred through the surface contact between the tie rod and the steering arm. The bolt was then locked with a nylon nut to ensure the thread would not unwind under any circumstances.

Fig. 4.2.1.5 Treaded Steering Arm

Fig. 4.2.1.6 Tie Rod End fitted

During the fitting and manufacturing process the binding of the rod end was monitored and altered accordingly. It was found that at full rebound the steering required a small washer between the steering arm and the rod end. The application of the small washer solved all binding issues under all suspension travel parameters. This will alter the bump steer characteristics of the steering system, although the difference from such an altercation will only make a collectively small contribution to the handling. The figures below show the tie rod is still free to move and hence no binding is occurring.

Fig 4.2.1.7 Tie Rod free from binding

Fig. 4.2.1.8 Tie Rod free from binding

Fig. 4.2.1.9 Tie Rod free from binding

Fig 4.2.1.10 Tie Rod free from binding

Once the steering modifications were made the available clearance enabled the front roll centre adjusters to be fitted. The overall clearance to the bolt head is limited. This is not expected to be an issue due to the positive lock between the steering arm and the wheel. The only deflection expected is from the steering arm (unlikely due to physical steel properties) and the wheel itself (well constructed alloy wheel). The clearance was deemed to be at a safe working limit and the new steering system was used. The system has been used at a few events and no problems have been encountered with its operation or clearance.

Fig. 4.2.1.11 Tie Rod Bolt Clearance


Fig. 4.2.1.12 Completed Steering System

5. Watts Link

The fitment of the Watts Link in fitting with the requirements of the project aims presented many complications which required overcoming. The end result is a lateral location device which allows full rear roll centre height adjustability. Each component of the Watts Link required custom fabrication to ensure correct fitment in the tight constraints. The Watts Link was constructed from commercially available steel in the interest of lowering the outlay for the initial construction as well as ease of manufacturing. The completed Watts Link has the required strength to service the requirements of the race car as shown in the subsequent sections. The rear roll centre can now be adjusted to alter the balance of the car and fine tune the handling as per the characteristics of the track and or the car conditions.

Fig. 5.0.1 Watts Link fitted

5.1 Centre Pivot

The centre pivot was the first component to be constructed. The requirements for the pivot were highly restrictive. The overall width of the pivot was highly restricted due to the clearance that was evident between the differential and the boot floor as shown in figure 5.1.1. The boot floor was able to be modified to result in increased clearance, although the pivot width was still highly restricted.

Fig. 5.1.1 Standard Clearance for Watts Link

The pivot bolt is fitted in the machined centre section through the application of a small bearing to allow easy rotation of the pivot under suspension changes. The overall size of the pivot was also restricted to keep clearance high and result in further adjustability of the roll centre location. Figure 5.1.2 and 5.1.3 shows that the travel range of the pivot will easily meet the required range of vertical travel.

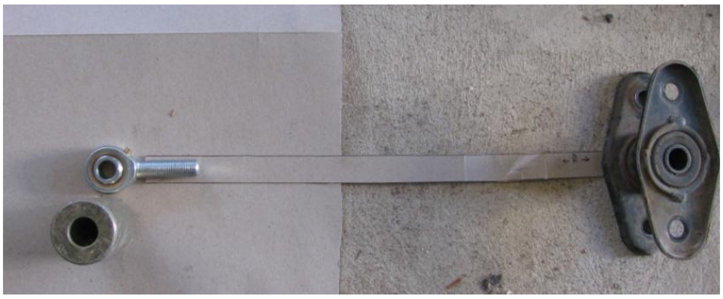

Fig. 5.1.2 Watts Link Travel – Standard Height

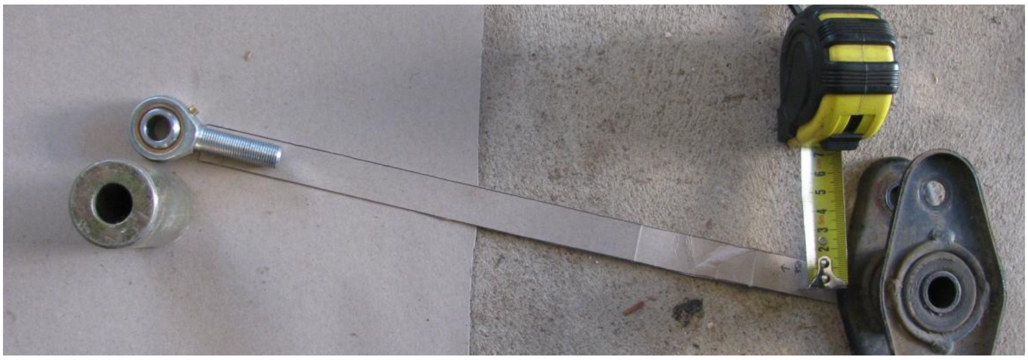

Fig. 5.1.3 Watts Link Travel – Differential 70 mm Higher

The pivot was then constructed as per the measurements found in Appendix J. The overall width of the pivot was restricted to the width of the bearings. The pivot was constructed from steel plates which were then welded to the centre bearing housing. Bolts and spacers were placed in the pivot, as shown in figure 5.1.4, during welding to prevent the distortion of the metal through the heat affected zone.

Fig. 5.1.4 Pivot before Welding

5.2 A Frame

The A frame construction is the main limitation of the Watts Link. The limited clearance has resulted in the pivot only being applied in single shear. This is not ideal, as a bolts potential to carry load is greatly reduced if single shear is employed. Bolts are limited in their strength in thread shear applications, and hence the limited strength of the system will reside with the centre pivot bolt that acts on the A Frame.

5.2.1 Pivot Bolt

The pivot bolts potential to withstand the force applied to it is integral to the systems safe operation. The bolt strength must therefore be calculated to ensure that it will not fail under the forces expected to be seen during the race conditions. Many assumptions have been made in the calculations however the general affect is towards an increased factor of safety.

If a bolt is to resist shear through its thread, without the effects of surface friction, the variables must be evaluated and calculated for the single shear situation. The following equation calculates the yield strength through single shear as stated by Juvinall & Marshek (2006).

$$Yield\ strength\ \ of\ bolt = Ssy\ \times At$$

The bolts available for use in the centre pivot were ½ inch - 13UNC grade 5 or grade 8 bolts.

The respective strength limits of the bolts were found as per Table 10.4 p407 (Juvinall & Marshek, 2006)

5 grade – Yield Strength = 92 ksi

- Tensile Strength = 120 ksi

8 grade – Yield Strength = 130 ksi

- Tensile Strength = 150 ksi

The effective area of the ½ inch bolt was then found in Table 10.1 p387 (Juvinall & Marshek, 2006)

A_t= 0.1419 in^2

'The distortion energy theory gives a good estimate of shear yield strength for ductile materials' (Juvinall & Marshek, 2006). The bolts in question are deemed to be ductile enough to warrant the use of this analysis due to being a relatively low grade bolt in comparison to other more highly heat treated fasteners.

$$Ssy = 0.58 \times Sy$$

$$Ssy\ 5\ grade = 0.58 \times 92{,}000$$

$$Ssy\ 5\ grade = 53{,}360\ ksi$$

$$Ssy\ 8\ grade = 0.58 \times 130{,}000$$

$$Ssy\ 8\ grade = 75{,}400\ ksi$$

The yield strength of the bolts was then calculated by inserting the variables into the applicable equation.

$$Yield\ strength\ \ of\ bolt = Ssy \times At$$

$$Yield\ strength\ \ of\ 5\ grade\ bolt = 53{,}360 \times 0.1419$$

$$Yield\ strength\ \ of\ 5\ grade\ bolt = 7{,}572\ lb$$

$$Yield\ strength\ \ of\ 8\ grade\ bolt = 75{,}400\ \times 0.1419$$

$$Yield\ strength\ \ of\ 8\ grade\ bolt = 10{,}699\ lb$$

The weight attributed to this force in metric units is found by converting the force to kg.

$$Yield\ weight\ (kg) =\ lb\ \times \frac{4.448}{9.81}$$

$$Yield\ weight\ 5\ grade(kg) =\ 7{,}572\ \times \frac{4.448}{9.81}$$

$$Yield\ weight\ 5\ grade(kg) =\ 3{,}433\ kg$$

$$Yield\ weight\ 8\ grade(kg) =\ 10{,}699\ \times \frac{4.448}{9.81}$$

$$Yield\ weight\ 8\ grade(kg) =\ 4{,}851\ kg$$

The ultimate failure strength of the bolts in the currently loaded situation will now be analysed.

Fisher and Struik studied the effects of bolts in reference to their shear strengths and concluded the following approximation as stated in eq. 10.16 (Juvinall & Marshek, 2006).

$$Sus\ \approx 0.62\ Su$$

$$Sus\ 5\ grade \approx 0.62\ \times 120{,}000$$

$$Sus\ 5\ grade\ \approx 74{,}000\ ksi$$

$$Sus\ 8\ grade \approx 0.62\ \times\ 150{,}000$$

$$Sus\ 8\ grade \approx 93{,}000\ ksi$$

Shear failure strength is found in a similar manner to yield strength through the following equation from Juvinall & Marshek (2008).

$$Shear\ failure\ force \approx Sus\ At$$

$$Shear\ failure\ force\ 5\ grade \approx 74{,}000\ \times 0.1419$$

$$Shear\ failure\ force\ 5\ grade \approx 10{,}500\ lb$$

$$Shear\ failure\ force\ 8\ grade \approx 93{,}000\ \times 0.1419$$

$$Shear\ failure\ force\ 8\ grade \approx 13{,}197\ lb$$

The converted forces are then calculated.

$$Shear\ failure\ weight\ 5\ grade(kg) =\ 10{,}500\ \times \frac{4.448}{9.81}$$

$$Shear\ failure\ weight\ 5\ grade(kg) =\ 4{,}761\ kg$$

$$Shear\ failure\ weight\ 8\ grade(kg) =\ 13{,}197\ \times \frac{4.448}{9.81}$$

$$Yield\ weight\ 8\ grade(kg) =\ 5{,}984\ kg$$

The force likely to be seen by the bolt must be analysed to ensure failure will not occur. The bolt force is seen by the application of the cornering force being transferred through the single point. The point therefore sees the following force.

$$Bolt\ force = cornering\ g\ force\ \times weight\ at\ rear\ axle$$

The highest cornering force achieved throughout the testing was 1.28 g and therefore this value will be used to calculate the force seen by the bolt. The weight on the rear axle is assumed to be 400 kg. This is assuming the car weighs 1000 kg and has a weight distribution of 60/40 from front to rear. This is overestimating the weight of the rear as the unsprung mass will not apply direct force to the bolt. This has incorporated another factor of safety in the calculation.

$$Bolt\ force = 1.28\ \times 400$$

$$Bolt\ force = 512\ kg$$

The forces that can be contained by the bolt are extremely high, even in the single shear application. The force that the bolt is likely to see is much lower than the yield force that the bolt can withstand and therefore the bolt should withstand its load. The effect of surface contact is also missing from the calculation and therefore the bolt is expected to remain in tension under all circumstances. This is a valid assumption that has reduced the impact loading seen on the bolt. The higher grade bolt was used although the lower grade would have been sufficient. The correct tensioning of the bolt within the bearing will be used to ensure that bolt fatigue failure will not occur as the surface contact will transfer the load efficiently.

5.2.2 A Frame Construction

The A frame itself must be strong enough to resist deflecting excessively under the application of cornering force. The highest bending forces will be present when the roll centre is set in the lowest position. The lowest position was deemed to be at the base of the differential carrier and the A frame was designed around being adjustable between the standard roll centre location (differential centre line) and this point. The A frame was constructed as per the measurements contained in Appendix J and shown in figure 5.2.2.1. The fitting of the frame to the car was performed as shown in figure 5.2.2.2.

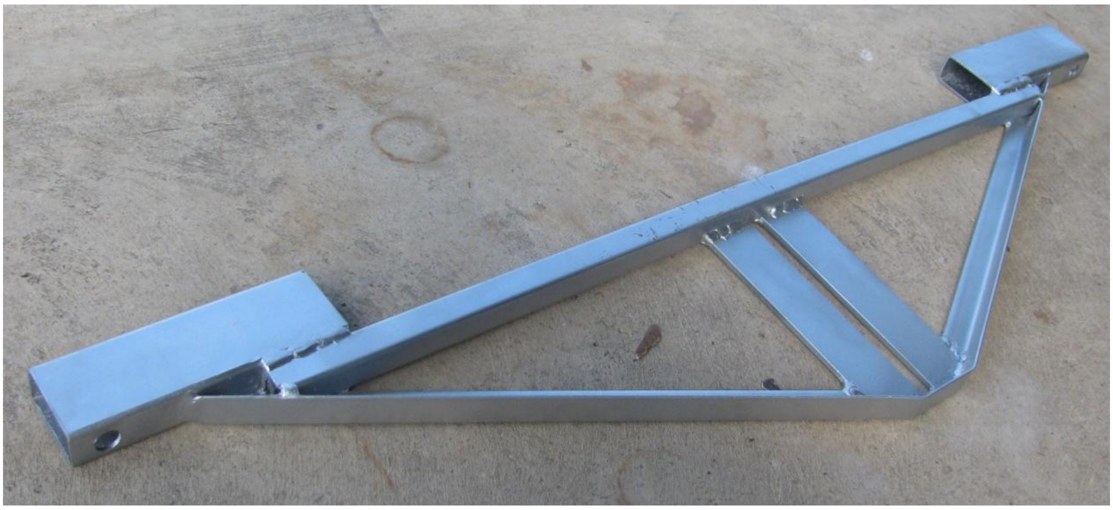

Fig. 5.2.2.1 Watts Link A Frame

Fig. 5.2.2.2 Watts Link A Frame Fitted

The actual fitting of the frame to the car requires that a large level of rigidity is utilised. The frame itself mounted directly through the chassis rails and as close as practical to the rail to reduce crushing as shown in figure 5.2.2.3. The frame is also supported with thick plates on the top of the chassis to prevent bending during operation. Figure 5.2.2.4 shows the bolts within the boot that also add longitudinal support to the frame.

Fig. 5.2.2.3 A Frame Mounting Location

Fig. 5.2.2.4 A Frame Mounting Support

The frames strength at the lowest roll centre location will be the limiting factor in the A frames construction and resultantly a basic FEA stress analysis was performed of a simplified version of the frame. SolidWorks 2010 was used to model the simplified frame and the total maximum cornering force was applied.

$$Frame\ force = 512\ \times 9.81$$
$$Frame\ force = 5022.72\ N$$

The simulation was run in SolidWorks and the maximum stresses and displacements were found across the frames construction. The simulation used AISI 1010 steel – hot rolled, in the interesting of ensuring that steel properties will not affect the strength of the designed frame. Cheap commercially available steel was used in the manufacture and the low grade steel is the most applicable to incorporate an increased margin of safety.

The simulation was performed and the following results were achieved.

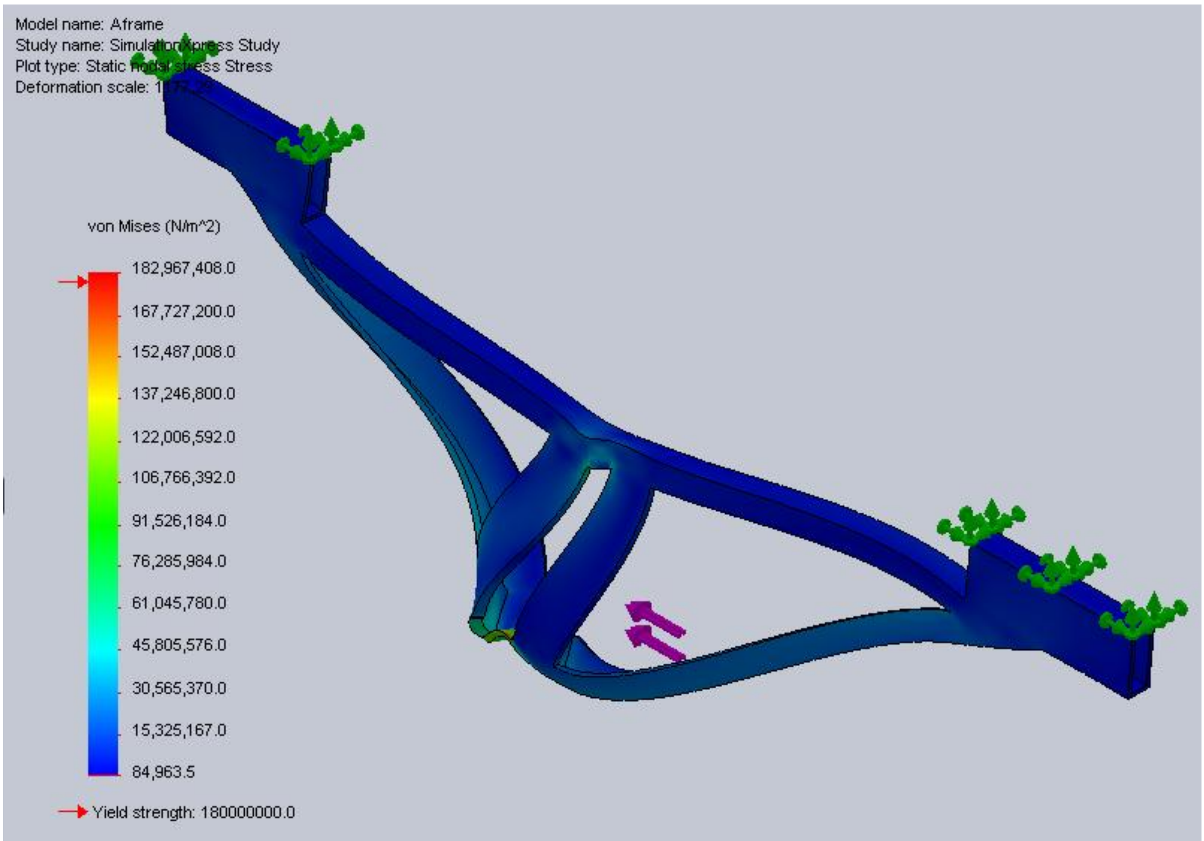

Fig. 5.2.2.5 A Frame Lowest Load Point – Von Mises Stress

Max Von Mises Stress = 1.82967×10^8 N/m^2

AISI 1010 steel – hot rolled yield stress = 1.8×10^8 N/m^2

The factor of safety is marginally less than 1 although this simulation does not include the metal that is contained between the two adjustment plates. As seen in figure 5.2.2.5 the highest stress is found in this location. The frame is also braced to the boot, a factor which will further lower the stresses seen in the frame. The frame is therefore deemed to be strong enough to resist yielding even if the lowest grade steel is used.

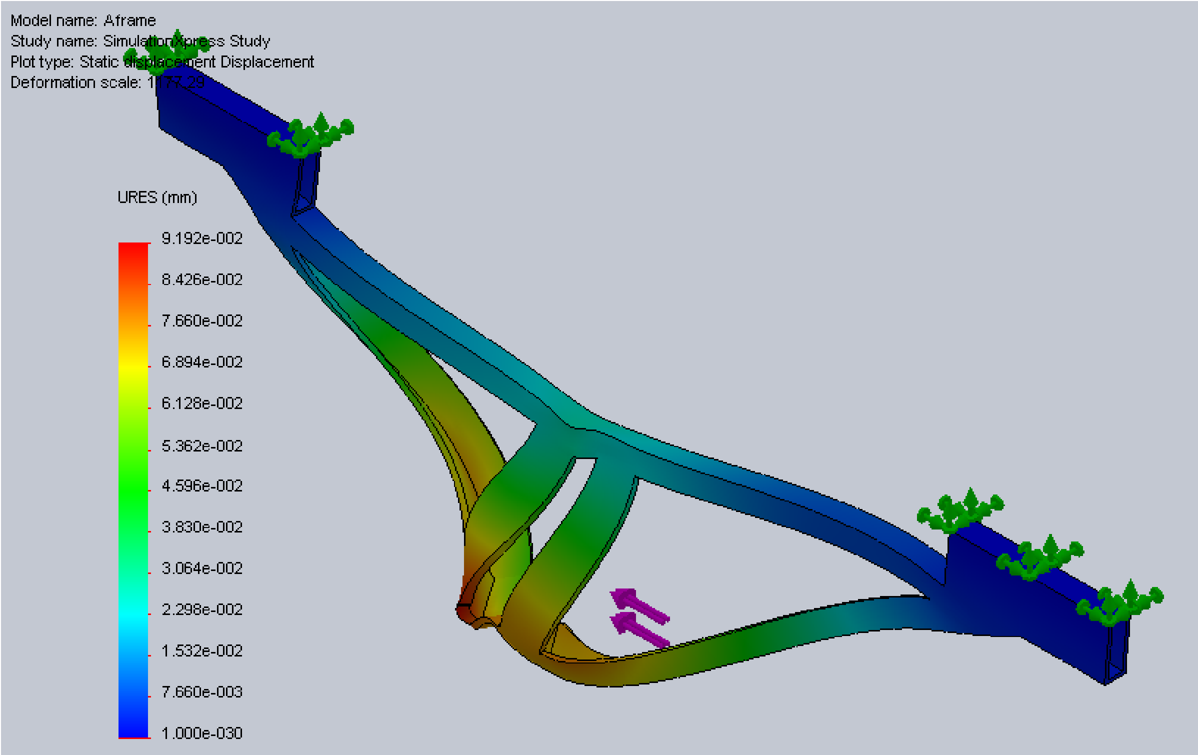

Fig. 5.2.2.6 A Frame Lowest Load Point - Displacement

The displacement of the frame under the applied load is vital to ensure the correct operation of the Watts Linkage. The lateral location of the differential depends on the frames potential to limit the movement of the differential under cornering loads. The maximum displacement experienced under the applied load is 0.0919156 mm. The small amount of deflection seen in the frame will result in a solid base for the lateral location of the differential.

The middle load point may also result in high stresses or deflections due to the limited size of the adjustment plates. The same stress analysis was again performed with the load being applied at the centre roll centre location.

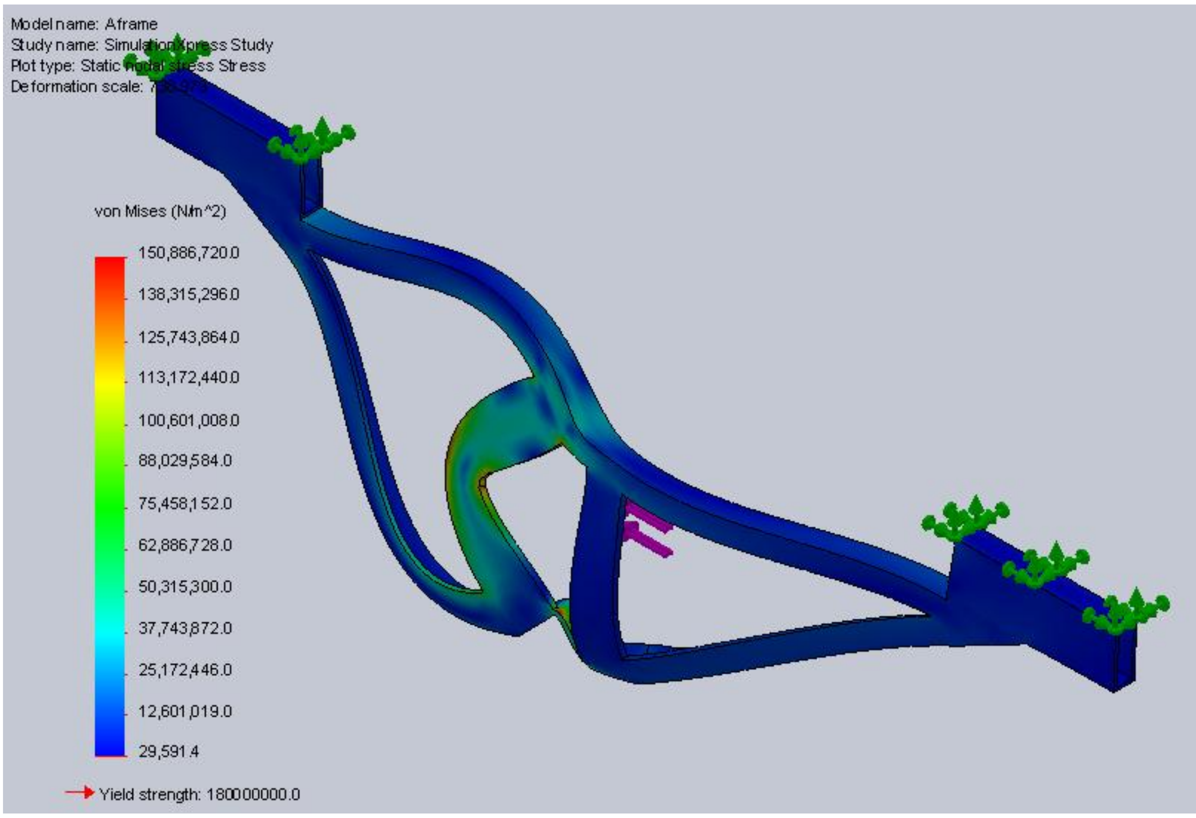

Fig. 5.2.2.7 A Frame Centre Load Point – Von Mises Stress

Max Von Mises Stress = 1.50887×10^8 N/m^2

Factor of Safety = 1.193

The factor of safety has increased to greater than one and hence a lower stress situation has been applied. The frame can withstand all positions of roll centre location and the result is a frame which meets the required criteria even with the lowest grade steel used in production.

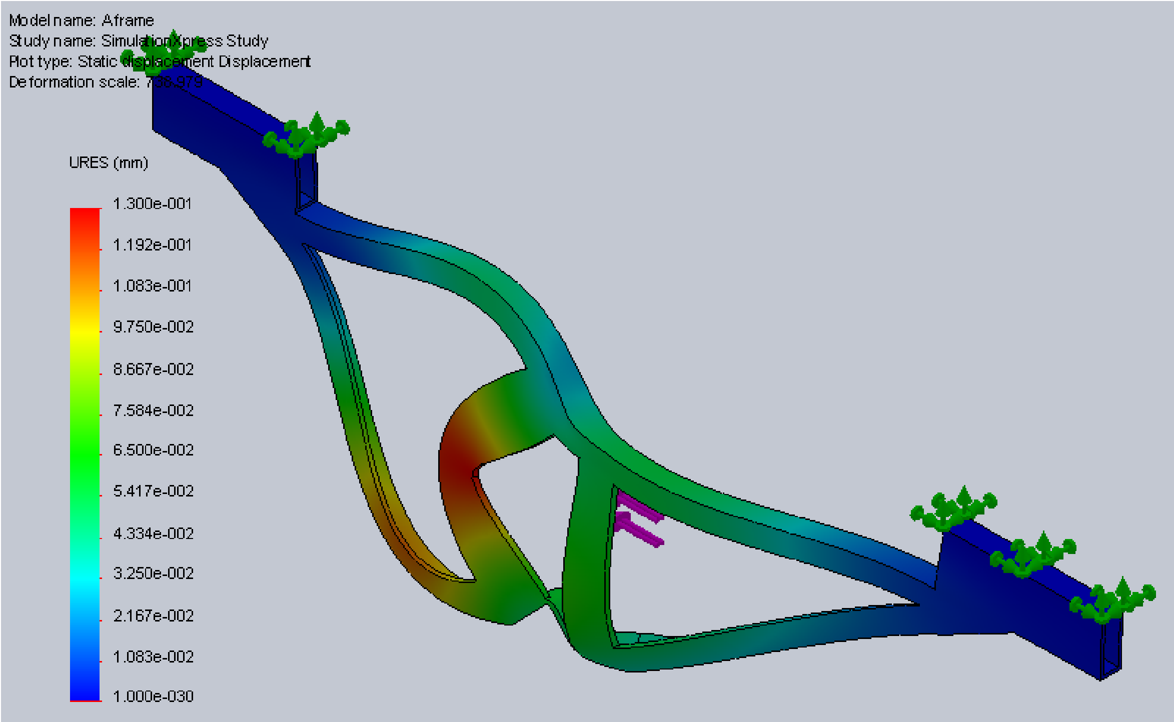

Fig. 5.2.2.8 A Frame Centre Load Point - Displacement

The displacement of the frame under the centre load point application is within the acceptable bounds of the lateral location requirements. The maximum displacement of 0.130006 mm is an increased value compared to the lowest load point, although the value is still low enough to ensure the frame will operate without any noticeable deflection.

The A frame is considerably strong enough to perform its function of laterally locating the differential. The frame also possesses the required rigidity to resist deflection and increase the driver's feel of the rear suspension.

5.3 Connecting Rods

The connecting arms and joints were manufactured to connect the pivot to the outer differential mounts. The arms must be the same length and setting of their length was performed before fitment was finalised. The arms were manufactured to the length specified in Appendix J, which was the longest length that could be achieved for both sides. This limits the effect that the roll centre height has on the pivot angle and resultantly the total vertical travel of the Watts Link. The arms were manufactured by threading thick section pipe to the required thread and locking the joints with lock nuts to remove all movement.

5.4 Passenger Side Differential Mount

The passenger side differential mount was manufactured to mount directly off the spring location plate welded to the differential carrier. Figures 5.4.1 and 5.4.2 show the mount as it is designed in Appendix J. The mount is extremely rigid and thick section steel was used to ensure the solid location of the outer mounts. The mounts are small and hence thicker material could be used to ensure their deflection was reduced. The connecting arm bolts are also being applied in double shear, a factor which given the pivot bolt analysis will result in extremely reliable service. The adjustment holes in the mount are used in reference to the driver's side mount and the application of the spacer plates to lower the car. The mount has also been positioned so that the spacers can be removed and the mount will not interfere with the spring's operation.

Fig. 5.4.1 Passenger Side Differential Mount

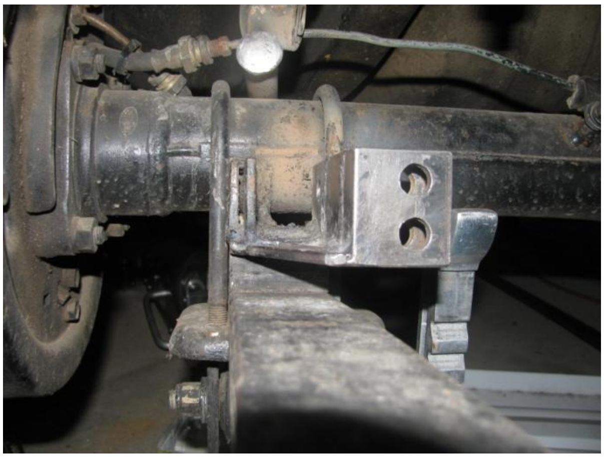

Fig. 5.4.2 Passenger Side Differential Mount

5.5 Driver's Side Differential Mount

The driver's side differential mount presented a similar situation to that present on the passenger side, although the point of mounting is 76 mm lower, as per the spacing between the arm connections in the pivot. This resulted in the construction of the mount as seen in figure 5.5.1 and 5.5.2 and Appendix J. The mount is again extremely rigid and thick section material and bracing was incorporated into the design. Double shear again results in extremely low bolt stresses. The adjustment holes in the design are used in the event that the spacer plates between the differential and the springs are removed. The correct height setting of the Watts Link is important to ensure that the proper geometry exists to ensure the smooth operation of the Watts Link is maintained. The mount is also welded to the spring plate slightly lower than the spring (fig. 5.5.1) to ensure that spring deflection can still occur without contact being made between the two surfaces. The differential mounts and bolts are extremely rigid in comparison to the A frame and hence they are also expected to fulfil the requirements of the Watts Links force transfer.

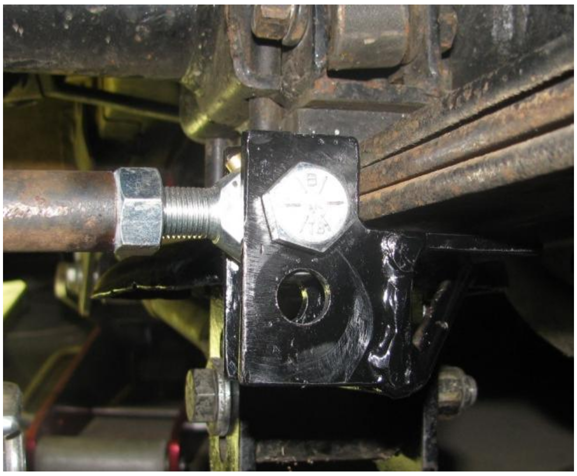

Fig. 5.5.1 Driver's Side Differential Mount

Fig. 5.5.2 Driver's Side Differential Mount

6. Additional Modifications

During the process of implementing the modifications many other issues where found with the suspension system. In fitting with the required use of the vehicle the necessary modifications were made to ensure the continual development of the car. The following issues were also attending to in fitting with the increased knowledge attained from the research undertaken.

6.1 Rear Leaf Spring Shackles

The rear leaf springs have a certain amount of roll understeer built in. In a standard car, all the variables are tuned to ensure that understeer is achieved in all circumstances that are over the limits of the tyres and suspension. Understeer is safe as the most common thing for a driver to do is slow down in an event of the cars uncontrolled behaviour. The slower speed increases front end grip and the car then turns. Racing requires that the understeer characteristics be greatly reduced. As discussed in the previous section the roll understeer manifested by the inclination angle of the rear springs has to be removed to ensure that roll understeer is reduced.

A certain amount of understeer is beneficial in ensuring the car remains stable and predictable once the limit has been reached and overstepped. The decision was made to ensure that the understeer characteristics remained, albeit at a reduced rate. The height of the spring mounts determines the amount of roll steer that is evident in the suspension system. In order to change the roll steer characteristics, the height of either the front or rear spring mounts need altering.

Fig. 6.1.1 Front Leaf Spring Shackle

The front of the spring presents many problems associated with altering its mounting location. The actual height increase required for the reduction of roll understeer would require the front spring mount to be fitted through the chassis. The amount of modification required would be extensive and far outweigh the predicted benefits. The caltracs setup, as seen in figure 6.1.1, also uses the front spring mount as its pivot point. The caltracs would also require modification to ensure proper functionality remained. The raising of this point would also lower the car, a situation which would be beneficial if more bump clearance was available. The overall complexity and reduced viability ruled out the front spring mount as a contender for potential roll steer adjustment.

Fig. 6.1.2 Rear Leaf Spring Shackle

The rear spring mount is a perfect contender for the roll steer adjustment. The rear of the spring also contains little force for locating the differential laterally. The axle location is however irrelevant due to the Watts link construction and fitting mentioned in the previous section. The roll steer can be greatly affected by the change in this height due to the available space for modifications. The height change associated with the longer shackles will also increase the rear ride height. This is beneficial in ensure that adequate bump clearance is maintained. A lower centre of gravity would be beneficial, however extremely limited bump clearance has been an issue with the rear suspension. The decision was made to modify the rear shackle to reduce the roll understeer.

The angle of the rear shackle is also important in ensuring that correct spring rates can be applied. The angle of the rear spring mount affects the rate rise of the rear spring system. The rear spring mount is currently leaning towards the ground behind the car, or as shown in Case 1 in figure 6.1.3. This results in a rising spring rate and is beneficial in all applications. The spring is longer than the distance between the shackle mounts, so rising rate will be achieved in all circumstances of shackle length, albeit at a differing rate of increase.

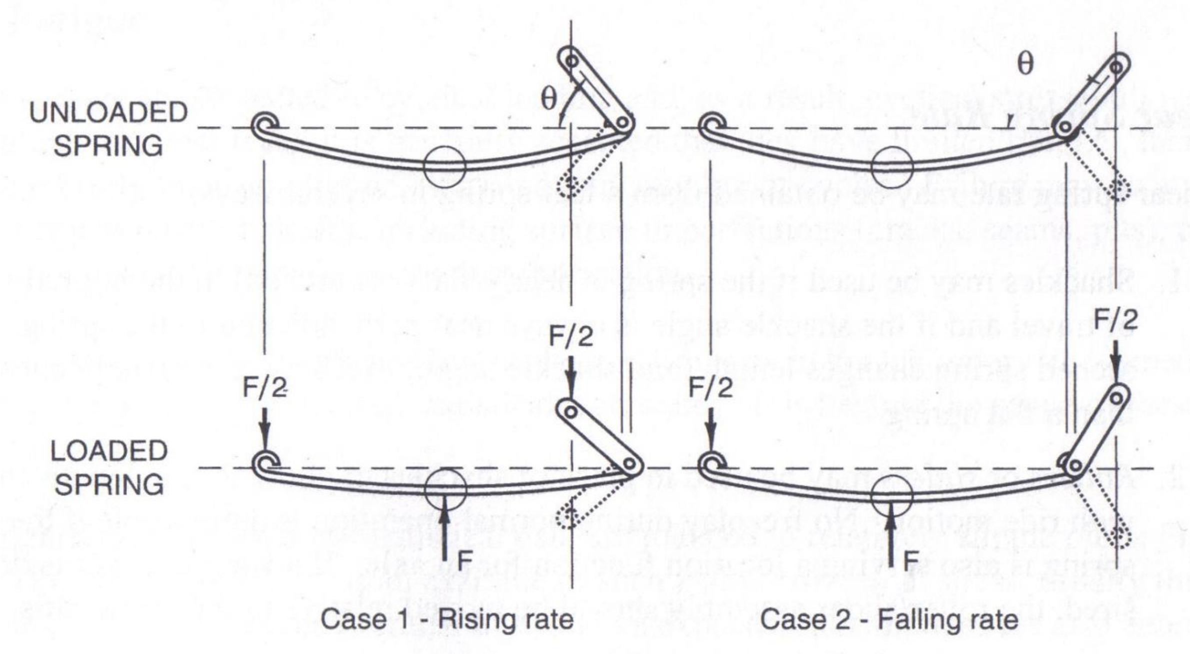

Fig. 6.1.3 Rear Shackle Effect on Spring Rate (WF & DL Milliken 1995, p.775)

The decision was made to alter the rear shackle length and a suitable process was obtained. The standard rear shackles were modified due to the fact that a curve is evident in the shackles construction. This limited the use of aluminium plates as a curve could not be made cheaply and safely. The required modifications were performed by lengthening the rear shackles to allow for adjustable roll steer and height.

The first step required that the extra adjustment plates be designed and fabricated. The required height change was measured and adjustment settings were placed 30mm apart. The steel used in the production of the shackles was of the same thickness as the standard shackles. The force through the rear shackles is only implemented in the spring application direction, and hence no bracing is required for the increased length. The standard spring mounting bolts were used to ensure that the strength of the system was not reduced. The additional adjustment plates were then welded to the standard rear spring shackles as shown in figure 6.1.5 to allow for the necessary adjustment.

Fig 6.1.4 Addition to Rear Shackle

Fig. 6.1.5 Extended Adjustable Rear Shackle

The modified rear spring shackles were fitted and the height changes were made. In fitting the shackles the required height was optimized to be in fitting with the other modifications listed in this section.

6.1.1 Rear Leaf Spring Shackle Height Setting

The modification of the pinion snubber/bump stop resulted in a slightly reduced rear height. With the rear shackles set in the standard location, the amount of tension placed on the pinion snubber/bump stop was noticeable. Figure 6.1.1.1 shows the height with the rubber while figure 6.1.1.2 shows the height reduction without the rubber.


Fig. 6.1.1.1 Height with Standard Bump Stop


Fig. 6.1.1.2 Height with modified Bump Stop

The decision was made to set the height as close as possible to the pre modification height. This required the lowering of the rear spring mount, which in turn, also reduced the amount of roll understeer. The rear spring mount was set in the second adjustment hole and figure 6.1.1.3 shows the final ride height achieved at that setting. The final product on the car is also shown in figure 6.1.1.4 with the applicable variables being set in their current location.


Fig. 6.1.1.3 Height with Modified Bump Stop and Second Lowest Height Setting


Fig. 6.1.1.4 Modified Rear Shackle and Current Setting

6.2 Pinion Snubber/Bump Stop

The bump stop is placed in a suspension system to limit the amount of suspension compression over large bumps and act as a positive stop to prevent the bottoming of dampers or other clearance issues such as tyres in guards or the body hitting the ground. The application of correct bump stops is recommended in all situations. The problem with bump stops on lowered cars is that they can be incorrectly sized for the reduced clearance.

When a bump stop is reached the spring rate becomes infinitely hard (as hard as the material used for the bump stop), and a loss of traction is noticed by the suspensions reduced ability to follow the roads surface. The correct application of bump stops is hence highly recommended on lowered cars.

The standard bump stop had remained throughout the suspension modifications, and many issues were found once a more detailed analysis was performed. The main bump stop on the 120y is also positioned to act as a pinion snubber once it has been reached. The application of a pinion snubber reduces the amount that the pinion angle raises under acceleration. The pinion snubber increases traction by ensuring that the pinion angle is always controlled. The caltracs mounted to the leaf springs act as an anti-squat device and hence will limit the amount that the pinion will raise during hard acceleration. A pinion snubber is therefore not required on the car and its operation as a bump stop is its sole purpose.

Fig 6.2.1 Standard Bump Stop Clearance

Figure 6.2.1 shows the bump stop in its standard form. It was clearly found that it was hitting the differential pinion carrier and resulting in an immediate bump stop. The bump stop itself was showing signs of wearing due to the constant load which it was required to carry.

The modification of the bump stop to allow the required travel was performed. The actual height of the rear bump stop was quite large with it extending approximately 50 mm from the floor level. The required height change was to be performed by cutting the standard rubber to result in a lower and alternatively shaped bump stop. The overall height was reduced to 25 mm and the rubber shape altered. The increased point on the bump stop is necessary to result in a reduced initial rate of bump reduction. The altered angle will result in a rubber which will begin at a reduced rate (to remain more consistent when bump stop operation begins) and increase in rate to a similar value of the standard profile (to adequately provide bump reduction). The altered profile and height is shown in figure 6.2.3.

Fig. 6.2.2 Standard Bump Stop

Fig. 6.2.3 Modified Bump Stop

The modified bump stops were fitted and the required clearance was achieved as shown in figure 6.2.3. There now exists a more predictable operation and their application in the event of a large bump should result in a more consistent handling vehicle. The modified bump stop will allow the suspension to operate as it should and reduce the tendency to lose traction over bumpy surfaces.

Fig. 6.2.3 Modified Bump Stop Clearance

The front suspension contains a large bump clearance and suitable bump stops are fitted. No further action was required. The suspension has a greater range of motion and can now more closely follow the road surface.

6.3 Guard Rolling

The clearance between the tyre and the guard is often a restriction in terms of both tyre size and suspension variables. If the tyre is large enough and the offset allows it, the guard may make contact due to being in a direct path of the tyres motion. The suspension settings and inbuilt variables also affect the total clearance achieved.

If the suspension is soft, the tyre to guard clearance will decrease, resulting in a higher likelihood of contact occurring. The static camber and camber curve will also affect the clearance at both static and dynamic load conditions. The steering inputs will also affect the amount of clearance, with increased steering input generally resulting in decreased clearance. It is advised that no contact between the inner or outer guards is encountered under all conditions.

The front inner guards are designed well within the range of steering and suspension travel. The inner clearance is however reached in a limitation that sees the wheel or tyre hitting the strut body or lower spring perch. The current wheels satisfy these factors and therefore the inner clearance is satisfactory.

Fig. 6.3.1 Standard Front Guards

Fig. 6.3.2 Modified Front Guards

The front outer guards have been rolled and flared in preparation for the fitment of the larger wheels and tyres. The track has also been increased slightly in the front, another factor which will attribute to outer clearance issues. The current flare built into the guards is sufficient to operate at the current -4.5 degrees camber and maximum achieved steering angle. The steering angle results in the lowest clearance value, and hence the static camber remains relatively large to accompany the wider tyres clearance requirements. The current guard profile as shown in figure 6.3.2 is sufficient, although larger flares would be required if static camber was to be decreased.

The rear tyres inner clearance is mainly limited by the spring's location. The tyre is very unlikely to hit the spring as the wheel is lacking the amount of positive offset required to cause clearance issues. The inner guard itself, at the top of the tyre, could be a restriction; however an extremely large roll angle would have to be achieved before clearance would be an issue.

The rear outer guard clearance was an area that needed addressing. The guards have remained standard and clearance issues were found before modifications began. The live rear axles tendency to twist (in reference to the body) during roll, causes clearance problems for the inside tyre. As the body rolls, it has the effect of increasing negative camber on the outside tyre and increasing positive camber on the inside tyre. The higher the value of positive camber, the lower the amount of clearance that is available. This also depends on the suspension pivots which are being altered, however in terms of a solid rear axle, clearance issues are likely to be highlighted on the inside outer guard. The 120y also contained these characteristics and additional clearance was required. The increased motion from the rear suspension will mean that the tyres position is likely to vary by a greater amount. The decision was hence made to increase the outer guard's clearance by use of a hand-held guard rolling machine.

Fig. 6.3.3 Standard Rear Guards

Fig. 6.3.4 Modified Rear Guards

The guard roller achieved the required clearance with the lowest initial outlay requirement. The increased range of suspension movement can be fully utilised without fear of the tyre touching any components of the cars guards or suspension. Other clearance factors, such as the caltracs, may be limiting the clearance properties. These are however controlled by the spring, anti-roll bar rates and bump stops and are irrespective of the tyres clearance to other components.

7. Testing and Data Analysis

The physical testing of the modifications was required to validate the theory that has been applied. This required that many different events be attended to gain information for comparisons. Events were attended both before and after the modifications were performed to create a benchmark from which improvements could be made.

7.1 Gatton 20th/21st March 2010 – Before Modifications

Tyre Temperatures

No tyre temperature data was recorded due to only being a single lap event.

Pressures were maintained at 30 psi front and rear.

GPS Data Logger

This was the first event the data logger was used at. It performed faultlessly and good data was gained from it. The cornering forces and speeds were both measured around the complete track. The fastest lap was used to gain the greatest understanding of the car on or towards the limit. Light averaging was used in all results gained from the data logger.

X – left (negative) and right (positive), cornering force

Maximum negative g force of -1.26 at left turn 3 – dropping corner over camber change

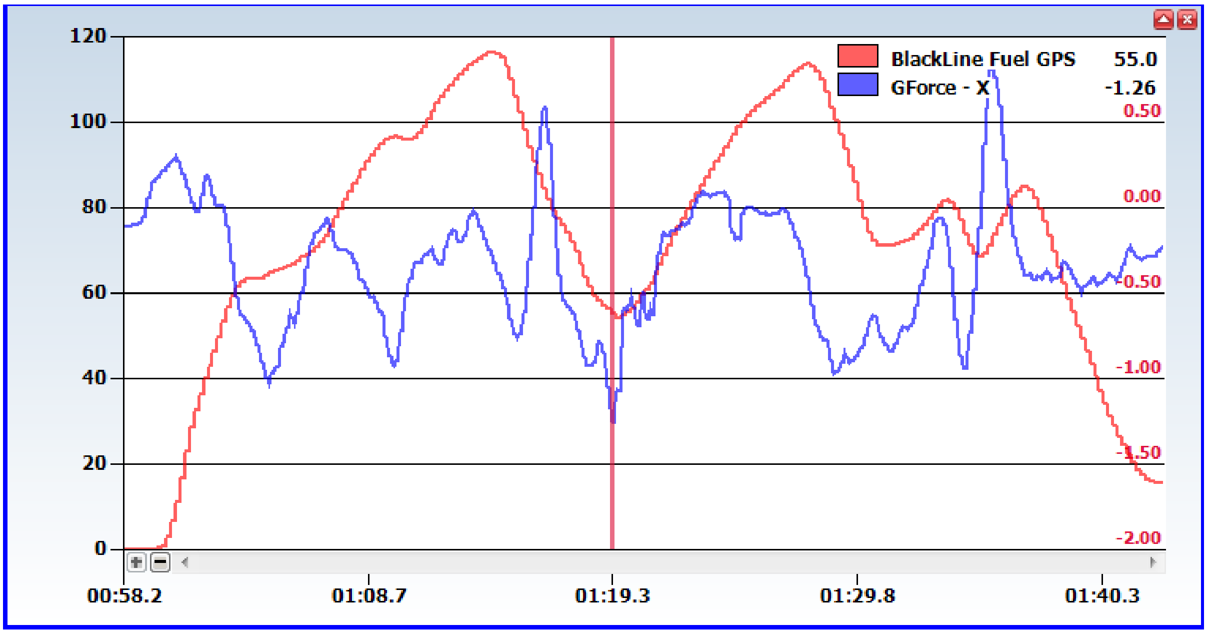

Fig. 7.1.1 Gatton – Max Negative X Direction G Force

Maximum positive g force of 0.81 at right turn in last chichane - flat corner

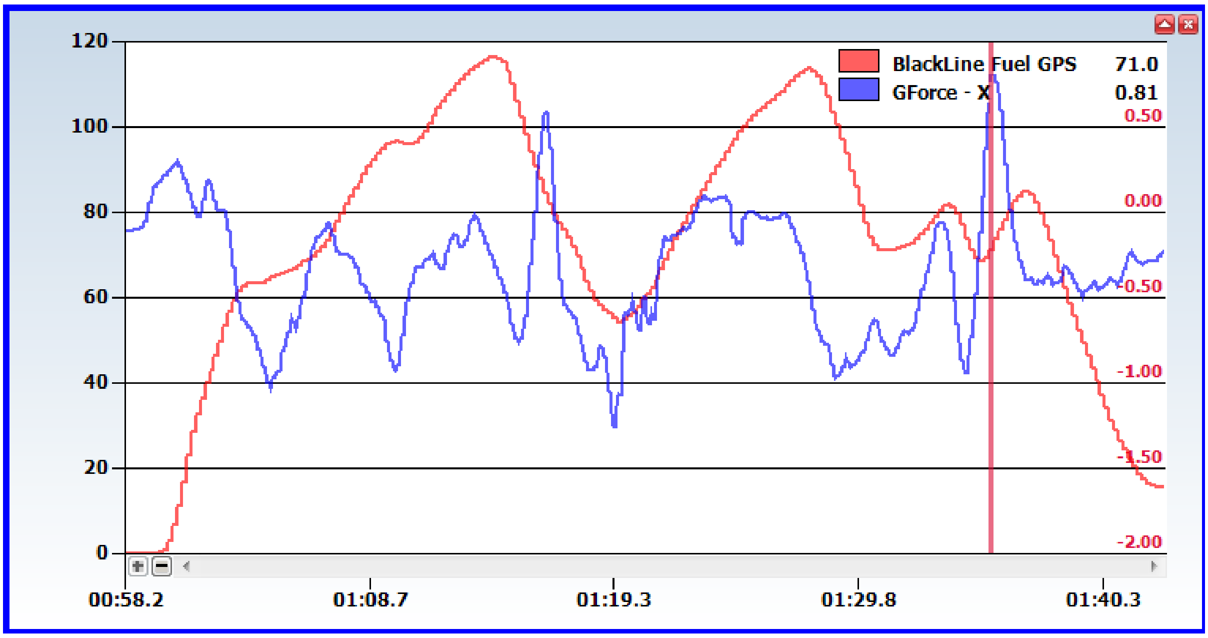

Fig. 7.1.2 Gatton – Max Positive X Direction G Force

Y – up and down (negative), gravity

Maximum negative g force of -1.52 at turn 3 – dropping corner over camber change

Turn 1 also contains a drop into the apex as shown by the high negative value.

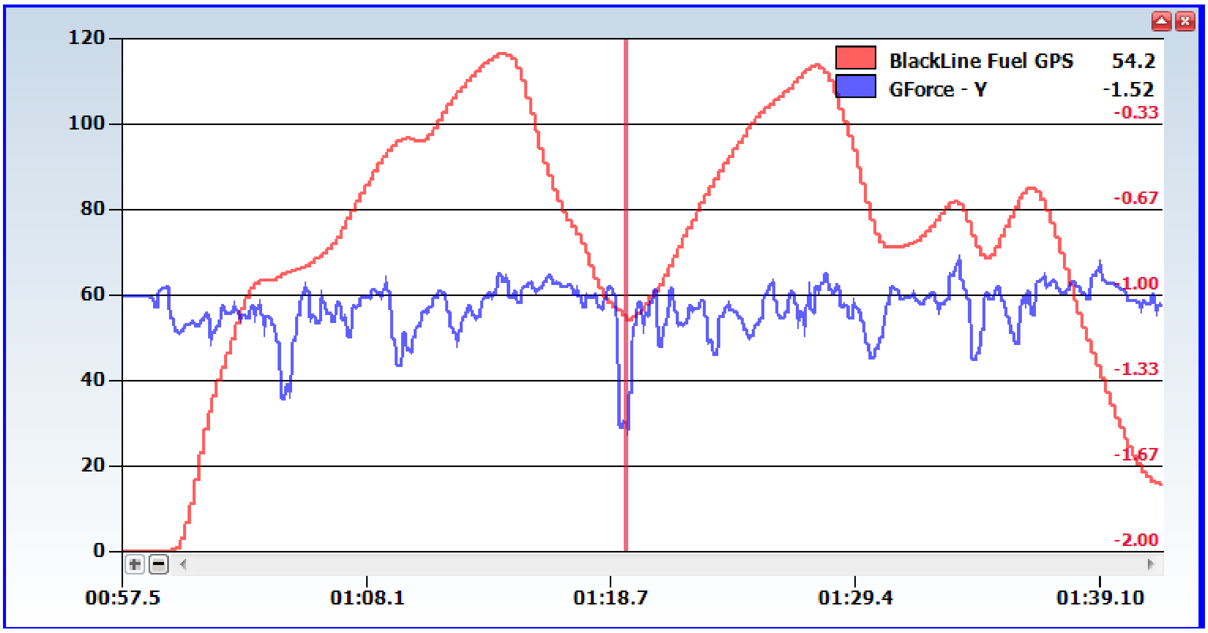

Fig. 7.1.3 Gatton – Max Negative Y direction G Force

Z – acceleration (negative) and braking (positive)

Maximum negative g force of -0.70 at turn 3 - dropping corner over camber change

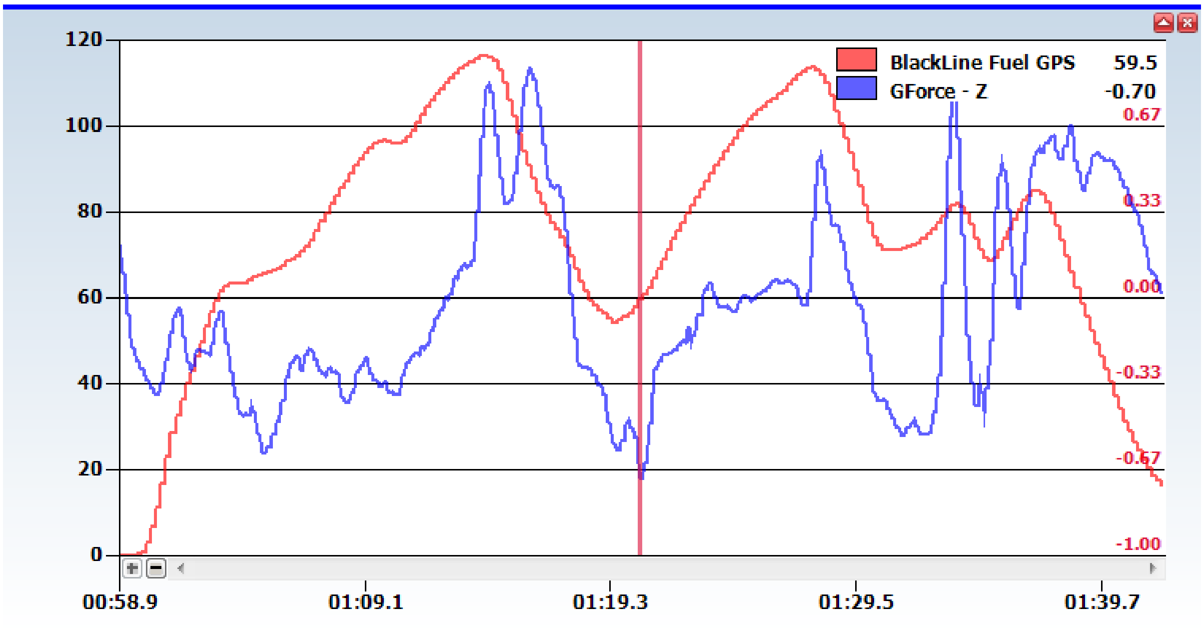

Fig. 7.1.4 Gatton – Max Negative Z Direction G Force

Maximum positive g force of 0.89 on the approach to the first chichane with heavy trail braking - Flat surface

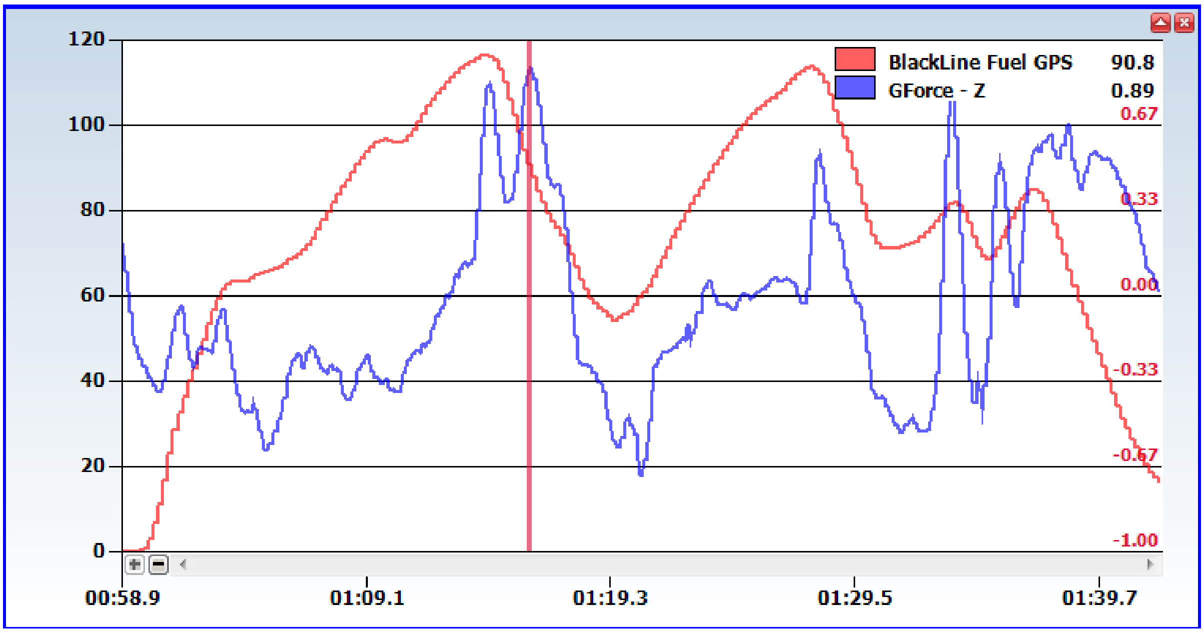

Fig 7.1.5 Gatton – Max Positive Z Direction G Force

Driver Evaluation

Handling was greatly improved compared to previous events. This however may have been predominately due to the semi-slick tyres front and rear. Turn 1 turn in was excellent with slight understeer through mid and late turnout. Turn two was similar although the rear lost traction slightly on corner exit. The first chicane was slippery on the first day resulting in many cars spinning. The 120y was no different and did struggle with front end grip. Once the correct line was developed and the driving style for the first chicane was sorted, few problems were encountered. The third corner had slight turn in understeer just before the apex, although turned out quite well with good grip even on the slightly bumpy surface on the outside of the corner. The rear remained in contact with the road at all times and good drive was achieved out of the turn. The fourth turn resulted in initial understeer under breaking but once slowed and power applied, the car handled very well and was fairly neutral in its handling. The last chicane before the finish line really suited the car and it handled extremely well in this section. Through the middle chicane leading on to the last hat, the front wheel could be felt lifting off the ground. The car responded very well to the sudden change of direction and felt very controlled, even with one wheel off the ground through the latter section of the chicane.

In general the car did quite well considering the competition that it was capable of beating. The main contenders were 4wd turbo evolution lancers, wrx's, and well established sports sedans with larger slicks and higher horsepower outputs. There were a few more standard setup cars that were able to set a faster lap time, and it is more important that the car is capable of beating such cars. In almost all other classes than the cars current class, it would have obtained a podium finish. Generally the weekend gave a very good indicator of how well the cars handling is balanced. Considering that it was capable of beating many a more powerful and in some instances, superior factory setup (IRS) vehicles, it has shown great potential for further development.

7.2 Stanthorpe 3rd/4th July 2010 – Before Modifications

Tyre Temperatures

Tyre temperatures were recorded as deemed necessary. The first run of both days was not recorded due to the track temperature being extremely low. As can be seen the tyre temperatures increased as the weekend progressed. This is due to the car being pushed increasingly harder, and often past its limits. The data is presented in the tables as they would appear on the car when looking from above. ie top left = front left, bottom right = back right.

Table 7.2.1 Stanthorpe Tyre Temperatures – 2nd Run Saturday

32.7	32.8	32.0		29.1	24.6	31.8
29psi	Pressure good	Camber good		28psi	Pressure too low	Camber more negative
38.5	39.3	32.8		35.4	36.9	33.9
29psi	Pressure high	Camber irrelevant		28psi	Pressure high	Camber irrelevant

Table 7.2.2 Stanthorpe Tyre Temperatures – 3rd Run Saturday

49.3	39.0	43.7		36.1	34.6	35.1
29psi	Pressure low	Camber more negative		28psi	Pressure low	Camber less negative
49.3	47.4	43.1		40.7	40.8	37.4
29psi	Pressure good			28psi	Pressure high	

The first few runs where to ensure consistent results were being achieved. The data presented from the runs all gave the same results and hence actions were taken accordingly. Camber seemed reasonably good and was left at -4.5 degrees. Tyre pressures where adjusted accordingly with the front raised to 32psi and the rear lowered to 28psi. This is aiming to increase the cornering power of the car, not tune the cars handling characteristics.

Table 7.2.3 Stanthorpe Tyre Temperatures – 4th Run Saturday

49.7	44.7	45.8		47.5	35.8	37.4
32psi	Pressure low	Camber more negative		32psi	Pressure low	Camber less negative
51.8	47.9	47.4		46.7	45	41.8
28psi	Pressure good			28psi	Pressure good	

Table 7.2.3 Stanthorpe Tyre Temperatures – 2nd run Sunday

56.0	49.6	53.6		48.9	39.5	48.2
32psi	Pressure low	Camber more negative		33psi	Pressure low	Camber good
48.1	44.3	43.7		36.9	41.5	50.1
28psi	Pressure good			28psi	Pressure good	

Table 7.2.4 Stanthorpe Tyre Temperatures – 3rd Run Sunday

58.0	50.7	53.1		57.7	41.6	44.2
32psi	Pressure low	Camber more negative		32psi	Pressure low	Camber less negative
47.7	44.4	40.7		38.5	41.1	51.5
28psi	Pressure good			28psi	Pressure good	

Table 7.2.5 Stanthorpe Tyre Temperatures - 4th run Sunday

61.2	52.1	51.8		53.3	39.6	42.3
32psi	Pressure good	Camber more negative		32psi	Pressure low	Camber less negative
47.0	46.3	45.4		36.5	40.9	41.9
28psi	Pressure good			28psi	Pressure good	

As can be seen the rear pressures are achieving their best performance at 28psi. The rear camber is irrelevant as the solid rear axle prevents camber from being adjustable. The variation across the tyre is presented by the cornering force transferring weight to the outside of the tyre.

The front however presents a differing story with pressures generally being considered too low. This is however in terms of the temperature of the tyres. When looking at the front tyres wear pattern, it was seen that the middle of the tyre was slightly wearing first. This contradicts the temperature data, even though the wear was only very slight and unmeasurable. For these reasons the pressures of the front tyres where left at 32 psi. The front camber is considered adequate. The left front is the outside tyre for most of the corners and hence its readings are more relevant. As it has been noticed, the front requires more negative camber. 4.5 degrees is already dialled in and is the limit of negative camber that can be currently achieved. Additional camber will result in less tyre being available for braking and straight line stability. The roll angle maybe responsible for this, although roll has generally been reduced substantially. Rear roll does transfer a lot of the weight to the rear and outside resulting in front inside wheel lifting. This may well be the reason it appears that the front outside tyre requires more negative camber.

The front tyres are also operating at a higher temperature than the rears. As the weekend progressed this phenomenon was evident in an increasing manner. The higher front temperatures show that the car is working the front tyres harder than the rear. This is a direct result of the understeer that is felt by the driver. A more balanced handling car with increased cornering potential would show tyre temperatures that are more balanced from front to rear.

GPS Data Logger

The GPS data logger was used for all runs except the first run on Saturday morning. A few technical problems have resulted in data that is less than ideal. Most runs have errors due to electrical interference. This was due to the GPS locator being placed in a different location than used at Gatton. More electrical interference was evident and that combined with the lower number of satellite connections resulted in some data being useless. A few runs did however work out fine and the data from these runs will still be used. For the record of the analysis, the 4th run on Saturday will be used due to setting the fastest lap in this heat. Light averaging was used.

X – left (negative) and right (positive), cornering force

Maximum negative g force of -1.07 at the initial turn of turn 2 – flat to uphill after apex

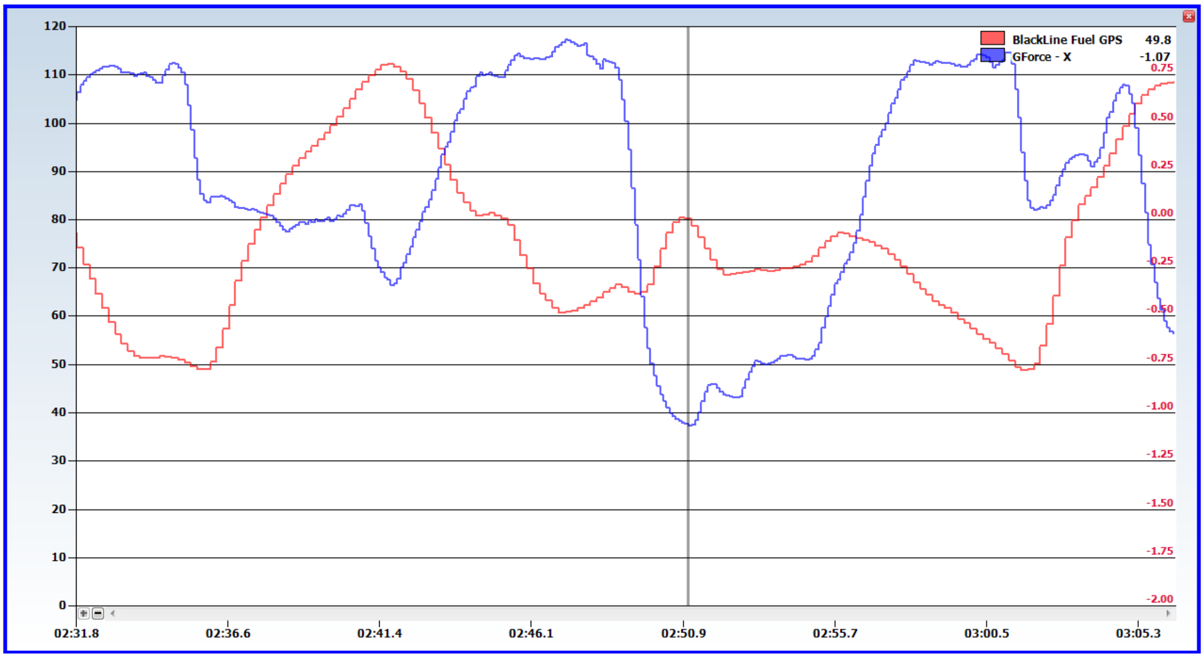

Fig. 7.2.1 Stanthorpe – Max Negative X Direction G Force

Maximum positive g force of 0.96 at turn 3 – flat corner on top of hill

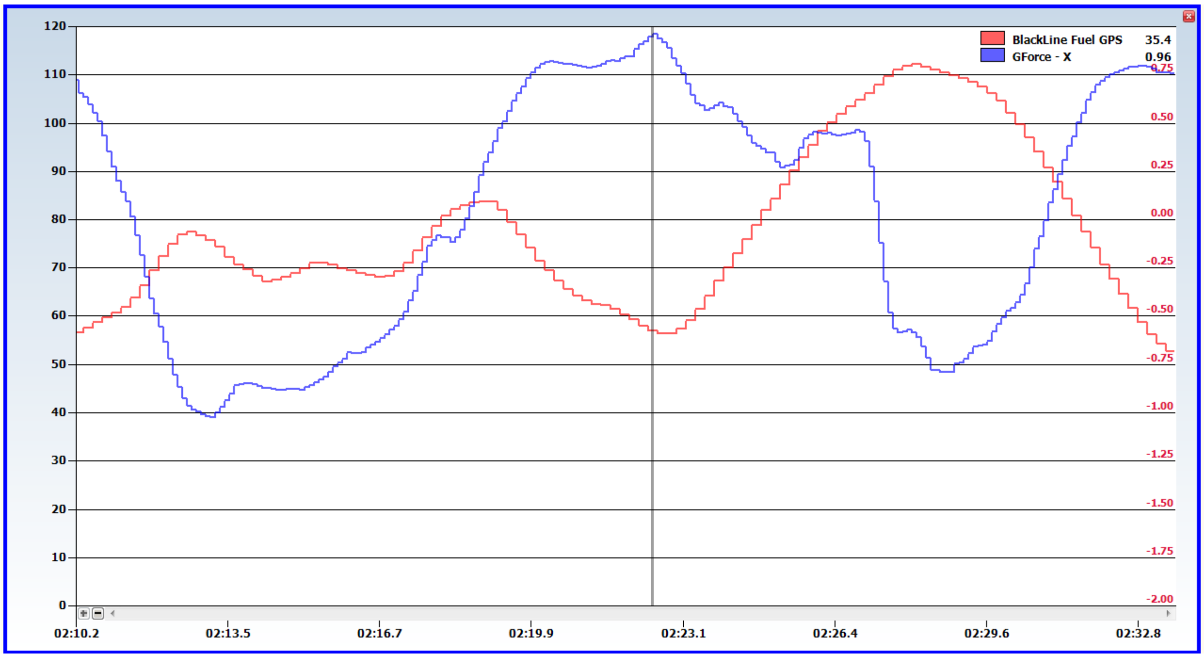

Fig. 7.2.2 Stanthorpe – Max Positive X Direction G Force

Y – up and down (negative), gravity

Maximum negative g force of -1.36 at turn 2 apex – as begins to climb uphill

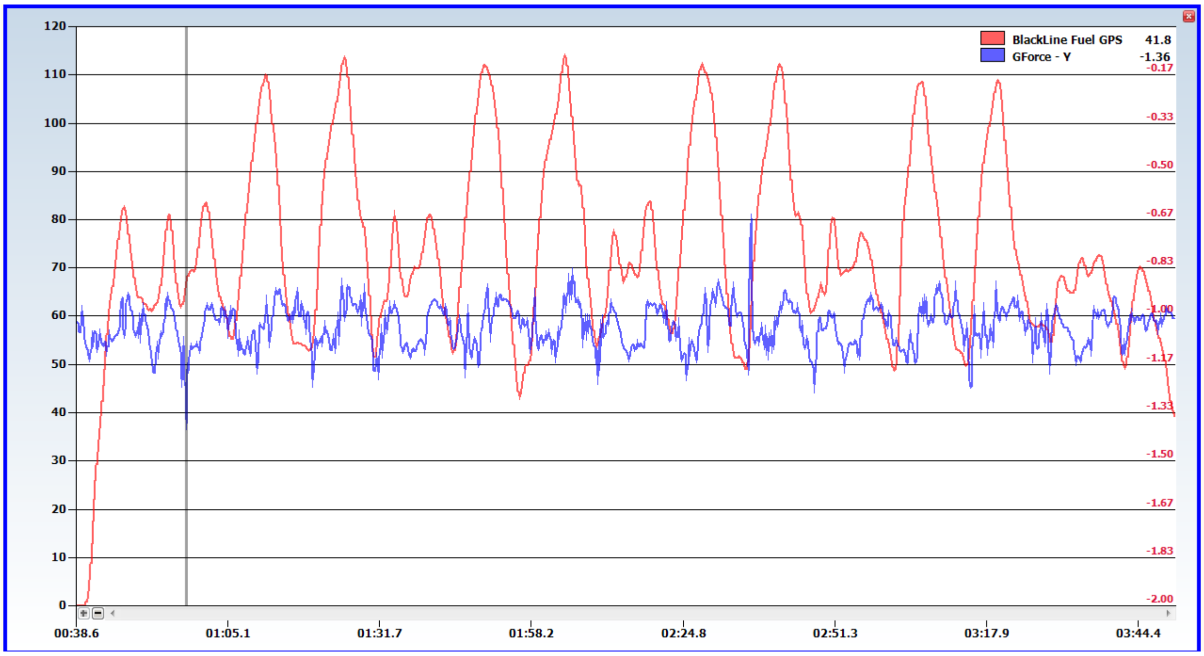

Fig. 7.2.3 Stanthorpe - Max Negative Y Direction G Force

Z – acceleration (negative) and braking (positive)

Maximum negative g force of -0.63 at turn 2 exit – uphill exit

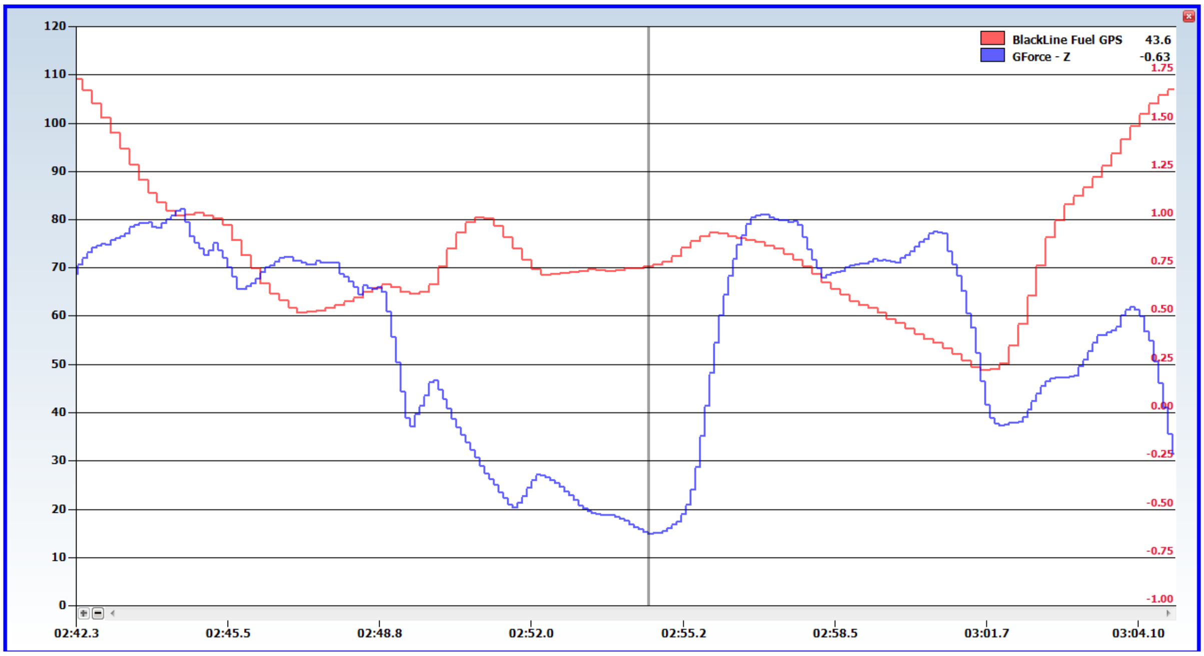

Fig. 7.2.4 Stanthorpe – Max Negative Z Direction G Force

Maximum positive g force of 1.14 at turn 1 entry – flat surface

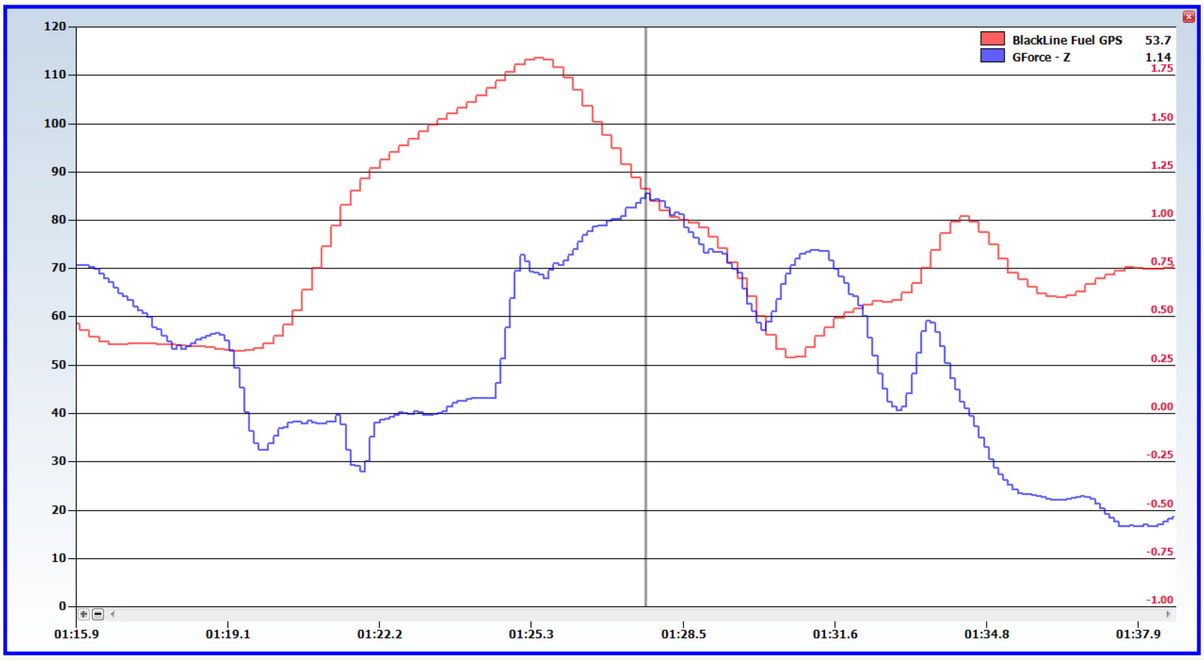

Fig. 7.2.5 Stanthorpe- Max Positive Z Direction G Force

Driver Evaluation

Handling was by far the best it had been at this track. This track results in understeer problems with most cars finding that front end grip is the limiting variable. The 120y was no different, with understeer being the main culprit in the handling department. In general, it was handling remarkably well and was beating/remaining with some competition that was previously faster at this track. Understeer was not majorly evident until pushed past the limit, and as a result the car cornering was safe and predictable. The engine developed problems over the weekend and was down on power slightly, a factor which may show in the data. The torque was highly unaffected by the slight reduction in horsepower, and all relevant data should be considered legitimate. All runs were achieved however and the results were extremely promising in reference to the corning power of the car and predictability.

Turn one turn in was good if not pushed too hard. If too much speed was carried into the turn the front would loose grip and push wide. Mid and corner exit handling was great, with excellent grip achieved by both ends of the car.

Turn two was similar to turn one, although with a much higher chance of corner exit understeer. On the exit the car begins to climb the hill to turn three and as a result, transfers weight to the rear resulting in front inside wheel lifting and slight understeer being noticed due to weight transfer diagonally.

Turn three results in good traction in most sectors and little problems where found in this corner. If pushed extremely hard the tendency was still towards corner exit understeer.

Turn four was a good vantage point as the car comes into and gets out of the corner extremely well. Most cars have problems getting the power down on corner exit, although no rear traction problems were found with the race car. Turn in and exit were good, with a slight tendency to understeer if pushed hard.

Stanthorpe also contains ripple strips, unlike most of the other street tracks. The 120y rode the strips well, with few problems being experienced by using them. The car remained predictable and supported confident driving up to and over the curbs.

7.3 Noosa 17th/18th July 2010 – Before Modifications

Tyre Temperatures

No tyre temperature data was recorded due to only being a single lap event.

Pressures maintained at 28psi front, 24psi rear – this is the cold setting pressures to achieve 32psi and 28psi hot operating pressures.

GPS Data Logger

The GPS data logger was run on the Sunday and reasonable data was achieved. The nature of the event did result in slight errors in the results. This is mainly due to the limited number of signals that the logger could communicate through. The G-force data will however be accurate and a good measure of the performance of the car. The last lap was recorded successfully and this will be used in the analysis as it was the fastest run for the weekend.

X – left (negative) and right (positive), cornering force

Maximum negative g force of -1.14 at turn 4 – cambered corner

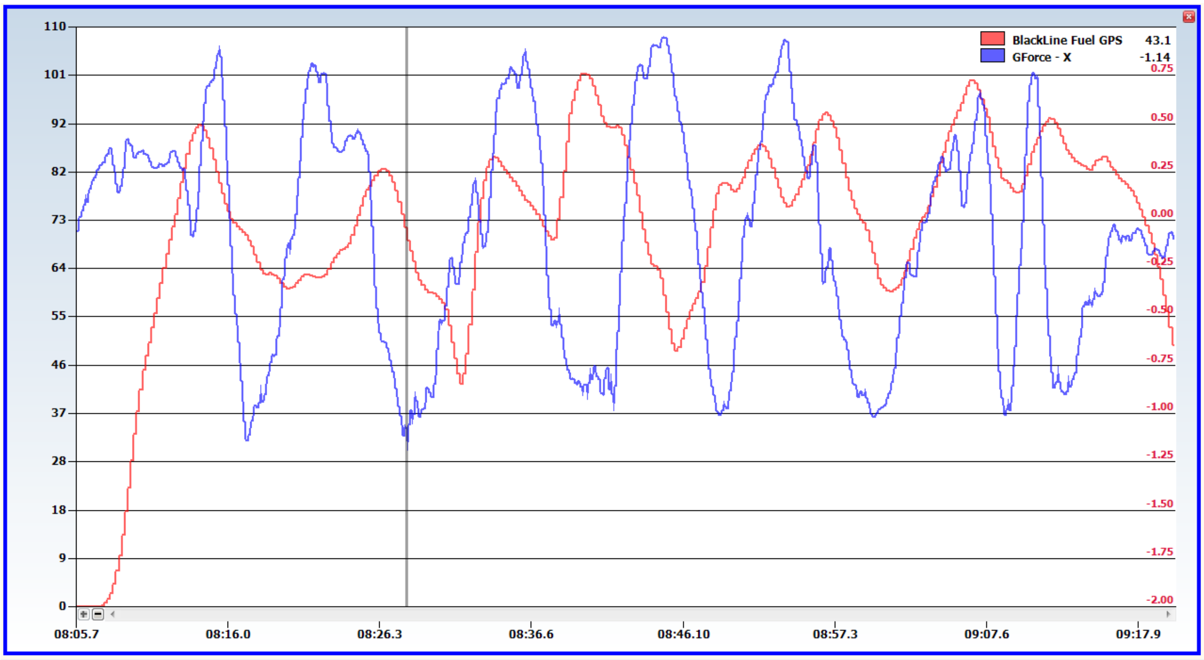

Fig. 7.3.1 Noosa- Max Negative X Direction G Force

Maximum positive g force of 0.95 at turn 7 – bottom of hill

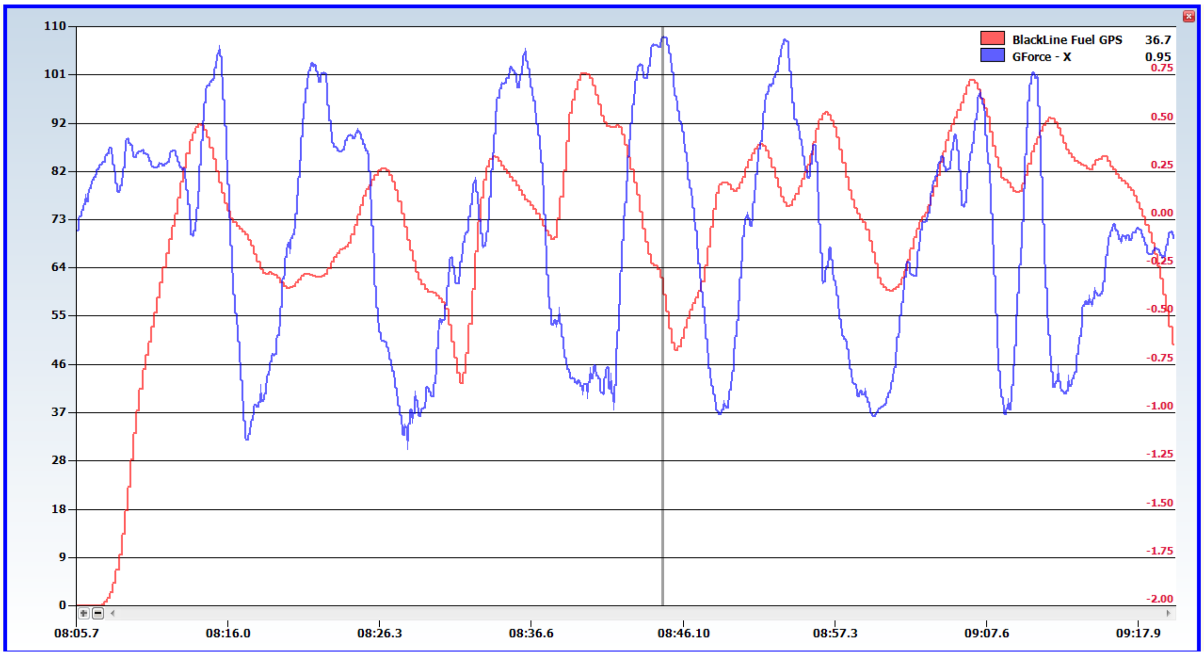

Fig. 7.3.2 Noosa- Max Positive X Direction G Force

Y – up and down (negative), gravity

Maximum negative g force of -1.48 at turn 13 – across flip flop

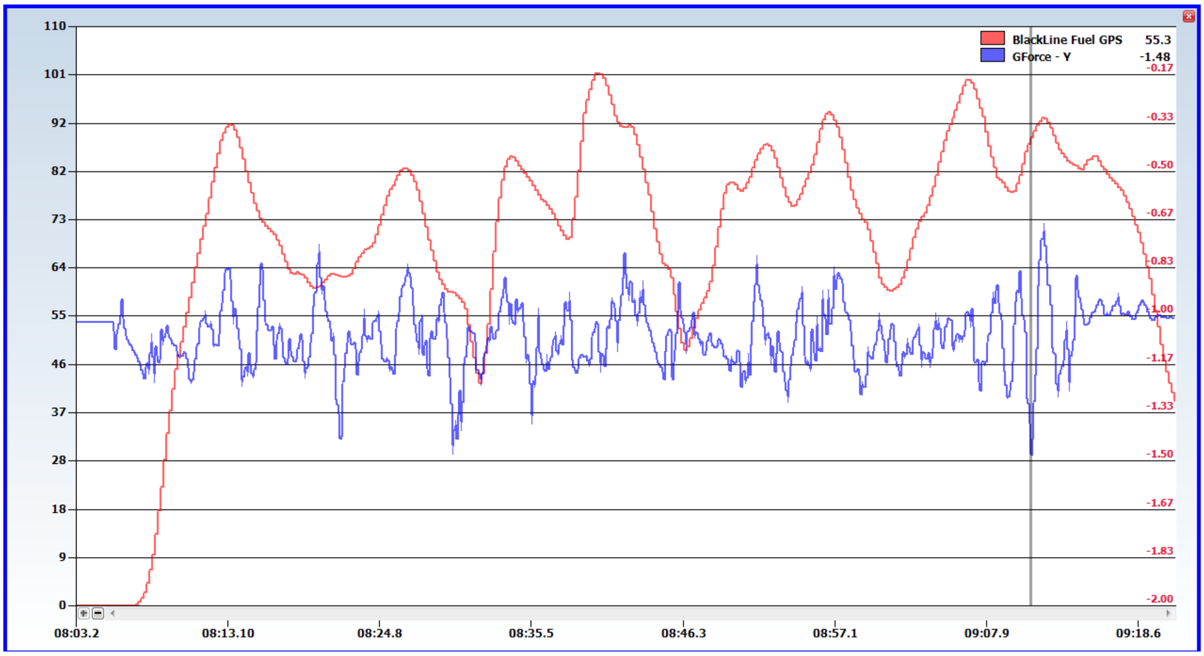

Fig. 7.3.3 Noosa- Max Negative Y Direction G Force

Z – acceleration (negative) and braking (positive)

Maximum negative g force of -0.76 at turn 14 – cambered corner, rolls over for exit

Note: very similar values achieved for most corner exits

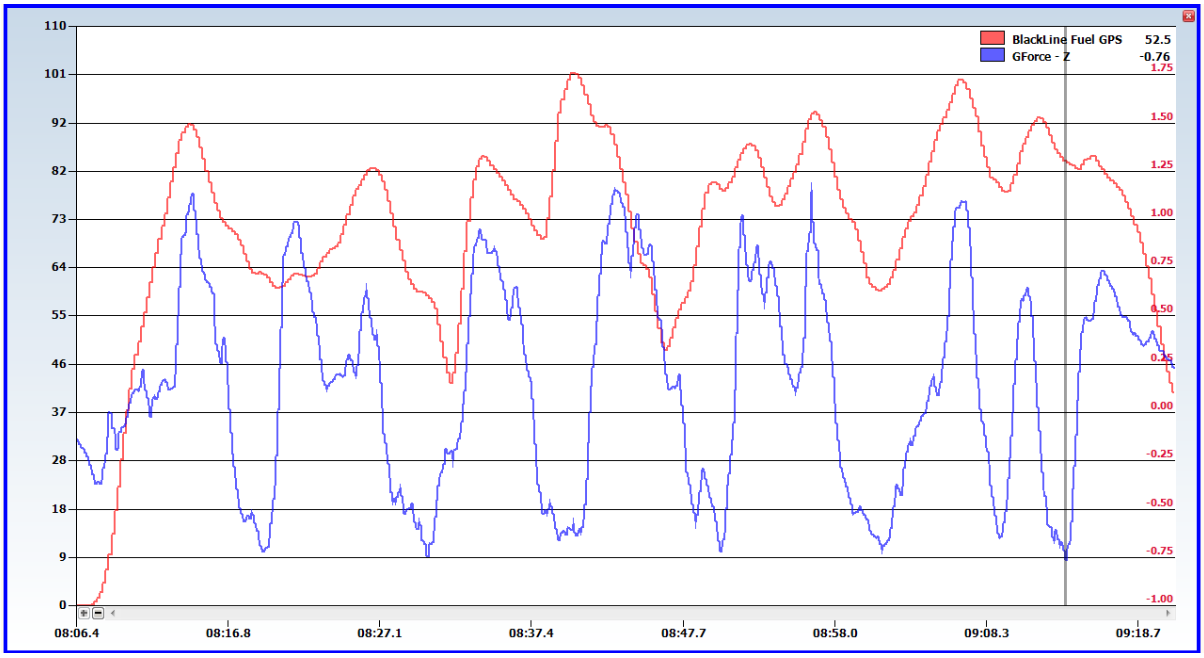

Fig. 7.3.4 Noosa- Max Negative Z Direction G Force

Maximum positive g force of -1.16 at braking approach to turn 7- flat surface

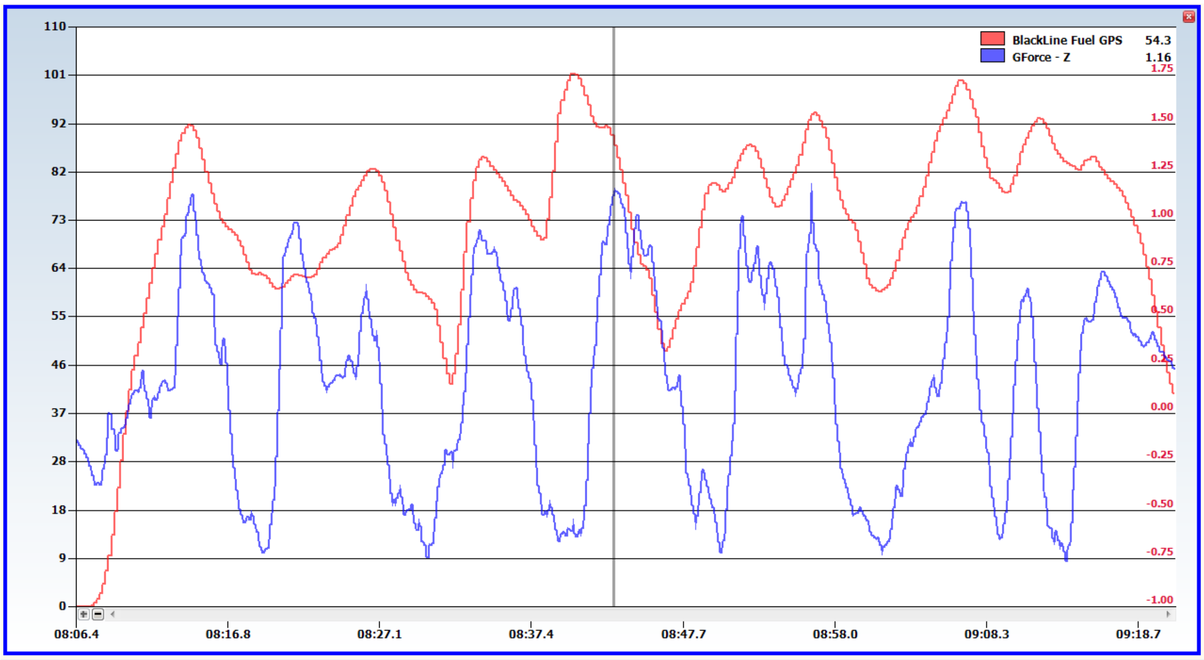

Fig. 7.3.5 Noosa- Max Positive Z Direction G Force

Drivers Evaluation

The high amount of corners presented at Noosa Hill Climb presents an extremely demanding situation. As seen in Appendix C the extremely twisty nature of the track presented the car with a very challenging course to perform on. The nature of the corners did however tend to result in the same situation and the handling over the similar corners was quite predictable. The corners were predominately filled with camber changes and a similar handling package was found in most circumstances.

The corner entry of most corners was cambered into the corner, and good turn in was the main result of these surface changes. The corner apex had good turn if the car remained in the cambered section. If this level was overshot the car resulted in extreme understeer, a process which required a much reduced corner speed. The corner exit always crossed from beneficial to detrimental road camber and this unsettled the car. If extremely harsh acceleration was used the rear would become loose over the roads camber change. The most likely phenomenon is understeer as the front weight is transferred to the rear, resulting in a decreased amount of vertical force with which to steer the car.

The handling was predictable however, an attribute which is extremely important given the highly dangerous nature of the event. The grip level was also generally good, with the car providing a high level of confidence to the driver.

7.4 Warwick 31st July/1st August 2010 – Circuit B, 1200m – Before Modifications

Tyre Temperatures

Tyre Temperatures were not taken as the pressures had been set at Stanthorpe earlier in the year. As no modifications to roll or weight transfer characteristics were being made, there was no need to remeasure the tyre temperatures.

Pressures were maintained at 32psi front and 28psi rear.

GPS Data Logger

The GPS data logger was run on the last 3 runs on Sunday. This decision was made due to rain on Saturday and a wet track Sunday morning. The data logger worked extremely well at this track with the data being repeatable and more accurate than other tracks. The software enabled a track map to be overlayed on top of the data, although this map was slightly off relative to the car position. This is due to slight errors in the accuracy of the GPS system. This does however give a much greater understanding to the results. The start/finish line was not set in the GPS due to time considerations and the nature of the event, although this will be set when returning to this track for a test day. The accuracy of the GPS results is a major consideration in selecting a test track.

The results of the data logger can be used to compare modifications made and tuning performed. The data from these runs is important in benchmarking the car before modifications begin. The test day will be performed on the longer track, however the similar corners will offer a good comparison from the driver’s perspective. The last run will be used as it was the fastest heat of the weekend and also contained the fastest lap.

X – left (negative) and right (positive), cornering force

Maximum negative g force of -0.95 at turn 3 – bottom of hill

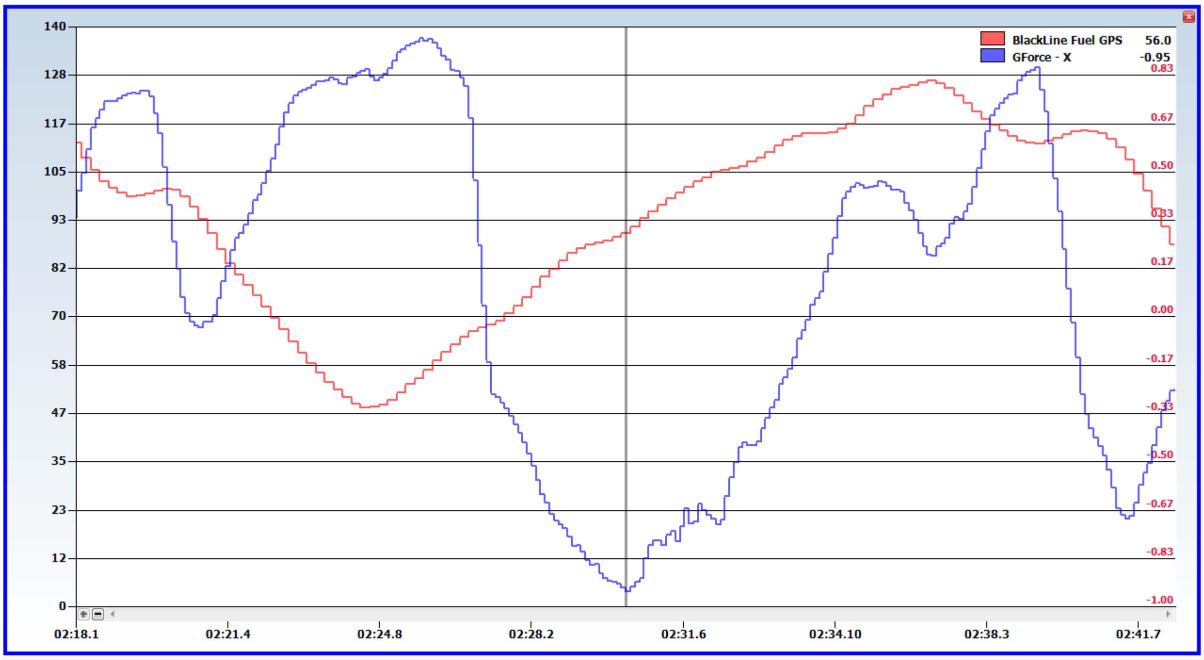

Fig. 7.4.1 Warwick – Max Negative X Direction G Force

Maximum positive g force of 0.99 at 7, flat surface

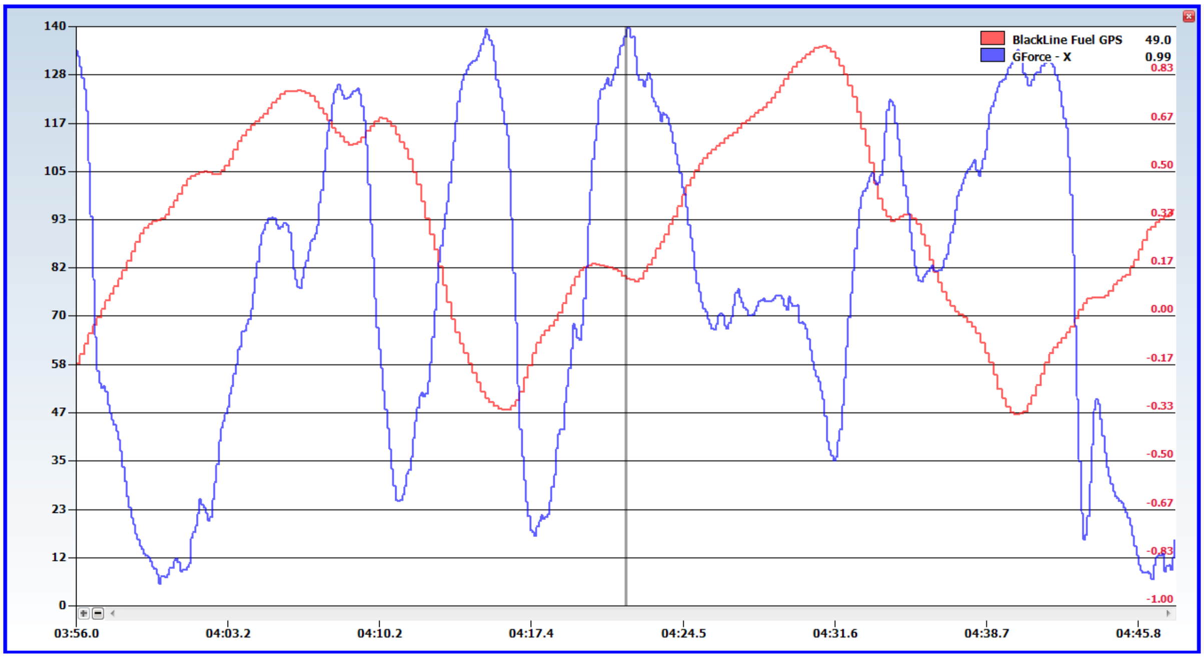

Fig. 7.4.2 Warwick- Max Positive X Direction G Force

Y – up and down (negative), gravity

Maximum negative g force of -1.32 at bump on straight

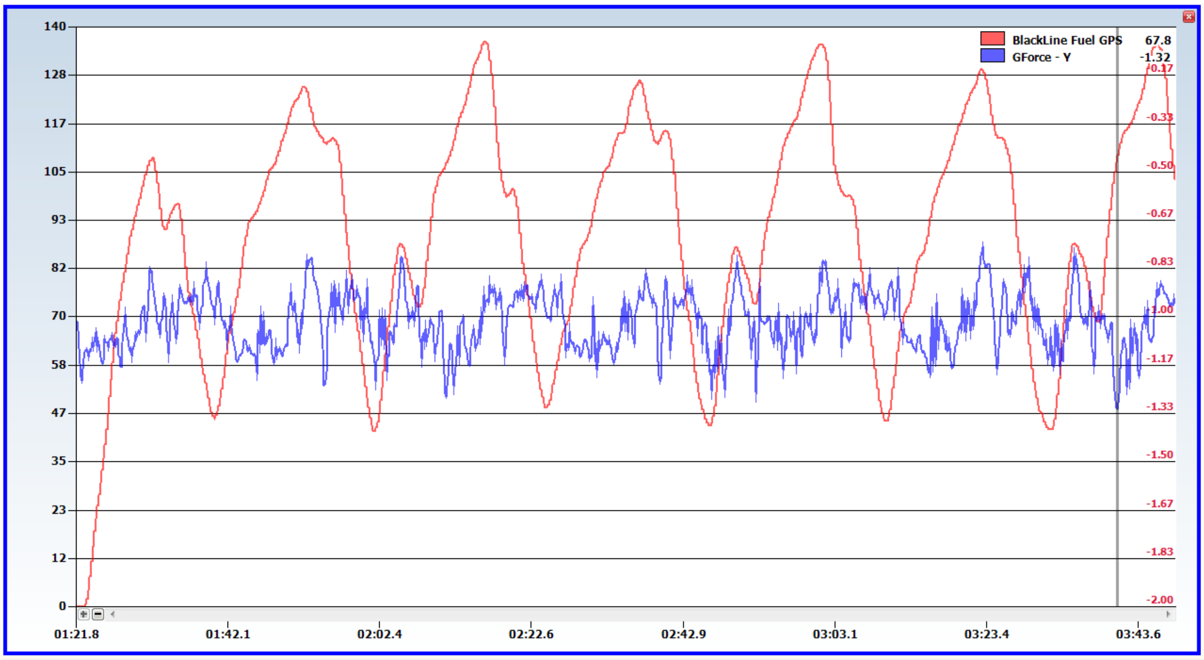

Fig. 7.4.3 Warwick- Max Negative Y Direction G Force

Z – acceleration (negative) and braking (positive)

Maximum negative g force of -0.64 at turn 5 exit, flat surface

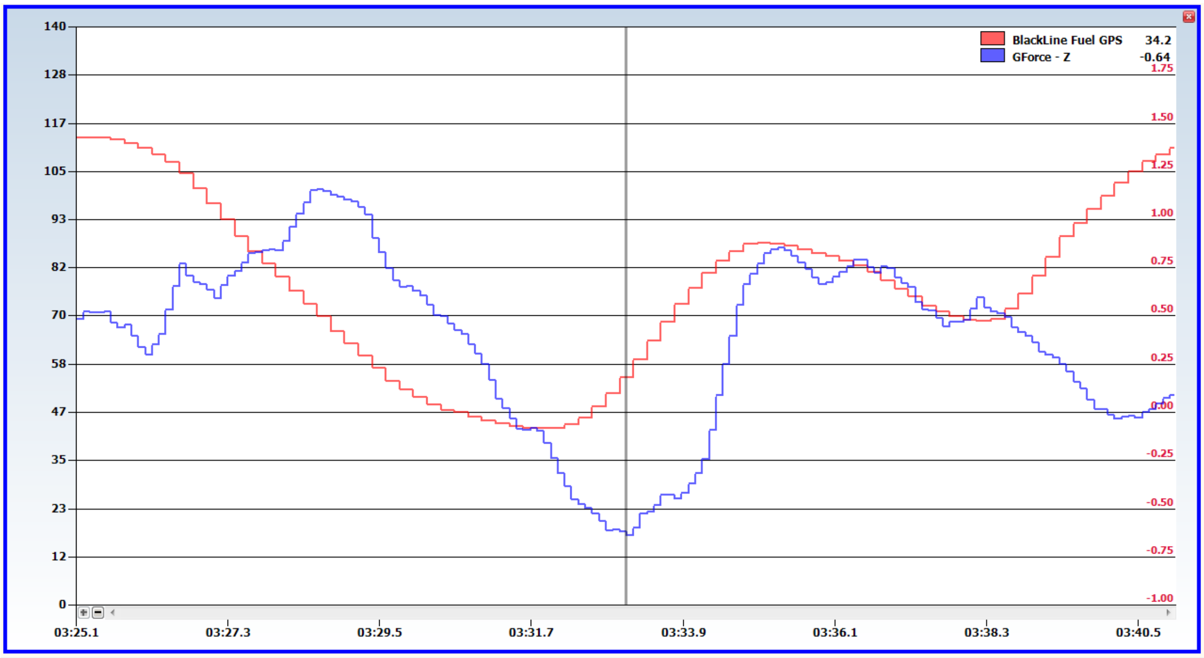

Fig 7.4.4 Warwick- Max Negative Z Direction G Force

Maximum positive g force of 1.22 at turn 5 braking area, down hill

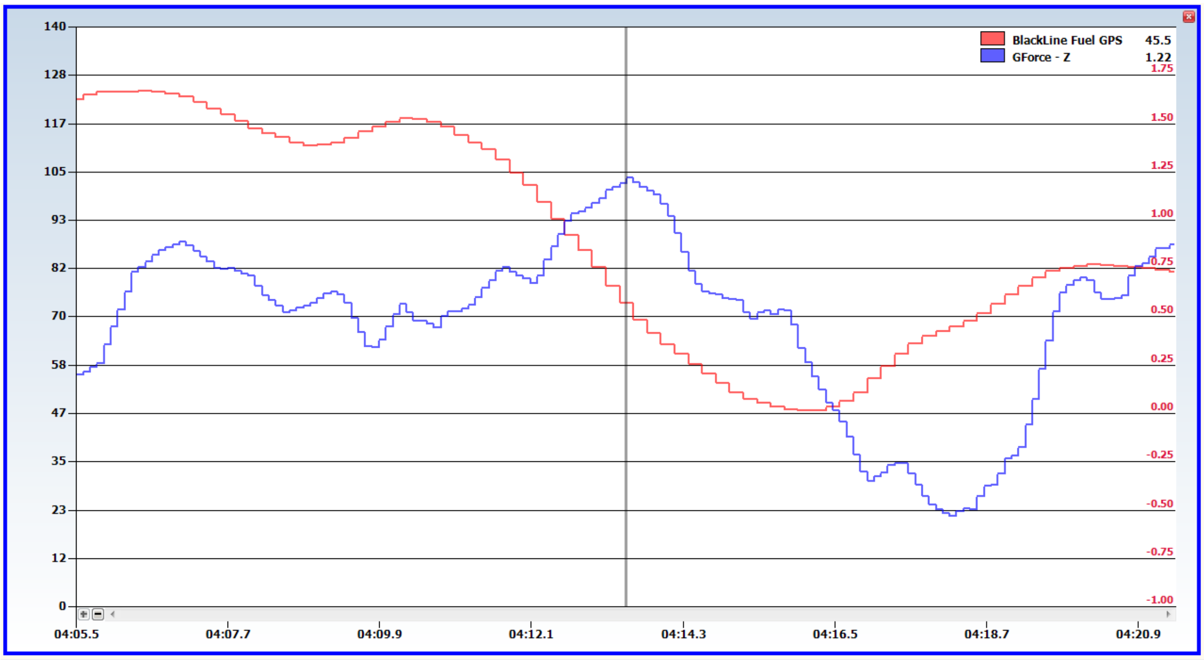

Fig. 7.4.5 Warwick- Max Positive Z Direction G Force

Drivers Evaluation

The car promoted relative confidence in its handling package and a good hard weekend of driving to the limit could be achieved. The early rain resulted in many different handling parameters. The limit of traction could be achieved much easier, a variable which made it easier to find the natural tendency of the vehicles dynamics.

In turn one the car was generally unstable and little confidence was found through the higher speed corner. The car would settle as the brakes where applied, although the corner apex and exit understeer would have the car continually running wide on the approach to turn two.

Turn two did result in good turn and grip levels generally felt high. The car felt as though it was being pulled through the apex of the corner and good drive was found leading into turn three. In the wet however, bad understeer was found if pushed hard.

Turn three was one section where cars with similar lap times would pull away from the 120y. The long corner exit would result in understeer; however the net result of this was limited due to the lower corner apex speeds. The exit also contains a fairly large bump just after the apex. The driver would not lose control of the car over the bump; however the drive was greatly reduced. The rear suspension would not absorb the bump, a factor which was later attributed to the lack of rear bump clearance.

Turn four resulted in understeer over the top of the hill, a factor which pushed the car wide and resulted in a slowing lap time. The understeer at this point of the track could be attributed to the higher speeds resulting in lifting.

Turn five is the slowest corner on the track. The hard braking into the corner was an area where good gains could be made, although extreme care had to be taken due to gradient issues on the corner approach. The turn in and mid corner grip was excellent, with this point being a real benefit when compared to many other competitors. The corner exit also achieved good drive and steer with the car being able to launch out of the corner extremely well.

Turn six is only a small kink in the road and requires a short shift into third gear to prepare for the next corner. The limit of turn six was not approached as it was only ever used as an approach for turn seven linking onto the straight.

Turn 7 results in corner exit understeer. The run onto the straight would be more ideal if less understeer was evident and power could be applied with more confidence earlier in the corner.

The general cornering limit was found through understeer. The front was generally the section which lost traction first. In the wet conditions the rear could lose traction under hard acceleration although the car felt twitchy and generally uncontrolled once the rear traction limit had been reached. The rear felt like it snapped back into line once power was reduced and the confident application of power in such conditions was limited.

7.5 Mt Cotton 7th/8th August 2010 – Before Modifications

Tyre Temperatures

No tyre temperature data was recorded due to only being a single lap event.

Pressures were set at 28psi front and 24psi rear. Little pressure gain was found and hence pressures were increased to 32psi front and 28psi rear. This resulted in a 2 second lap reduction. The front pressures were then increased to 34psi and a further 1 second lap time reduction was found. Note- this was the first event we had attended at this track.

GPS Data Logger

The GPS data logger was used on Sunday's runs. The times progressively got faster with each lap time being reduced. The data from this event can be used to compare with that from later events as more events are held on the same track later in the year. This is another method of benchmarking to verify the drivers perceptions of the cars performance. The last run was used for the analysis as it was the fastest for the weekend. The second fastest run was only 0.01 seconds slower, so the values obtained in this run have been used to find an average value for the comparison in section eight.

X – left (negative) and right (positive), cornering force

Maximum negative g force is -1.28 at turn 4, flat corner

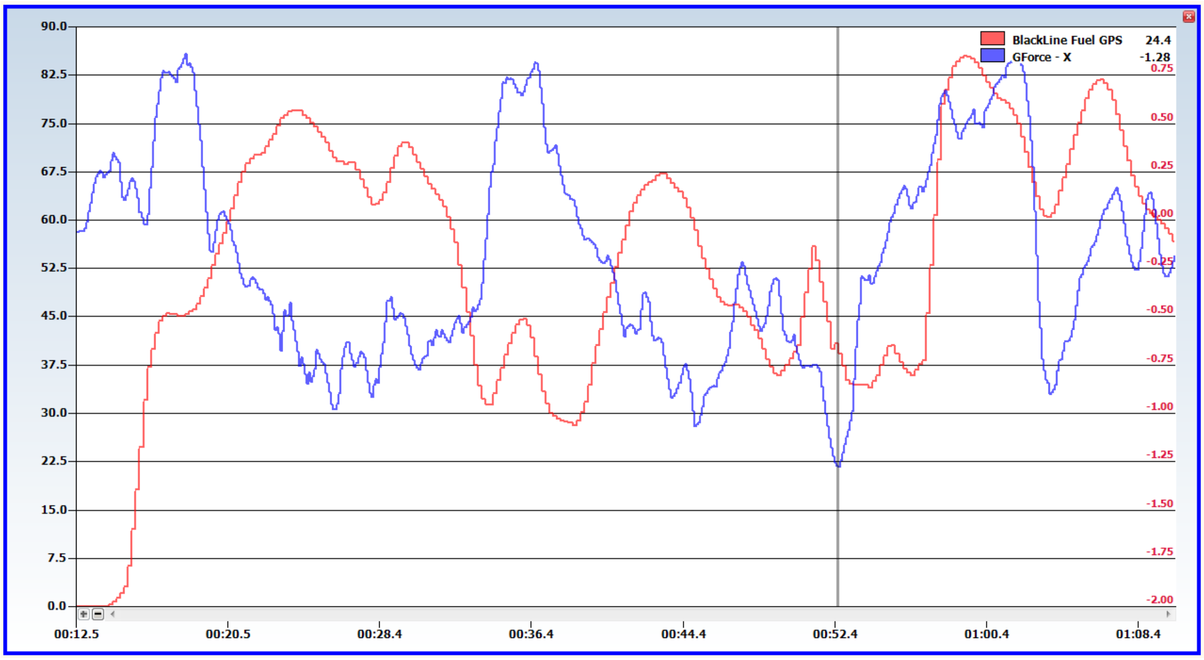

Fig. 7.5.1 Mt Cotton- Max Negative X Direction G Force

Maximum positive g force of 0.86 at turn 1, rolling to off camber

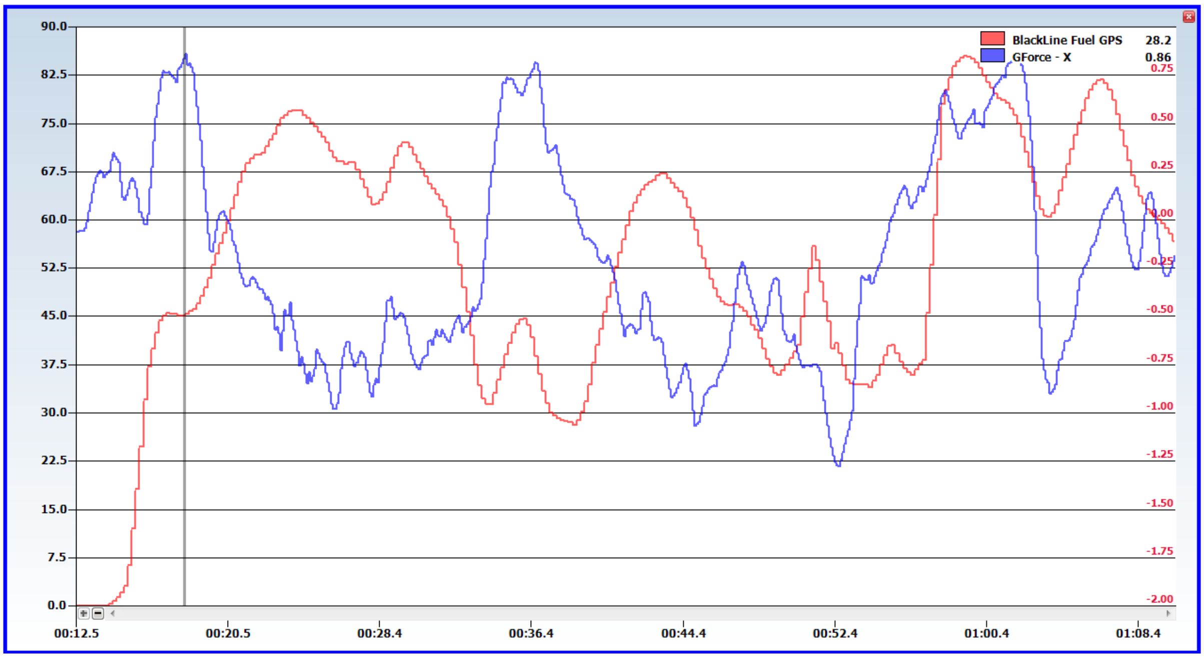

Fig. 7.5.2 Mt Cotton- Max Positive X Direction G Force

Y – up and down (negative), gravity

Maximum negative g force of -1.54 between turn 6 and 7, drop between corners

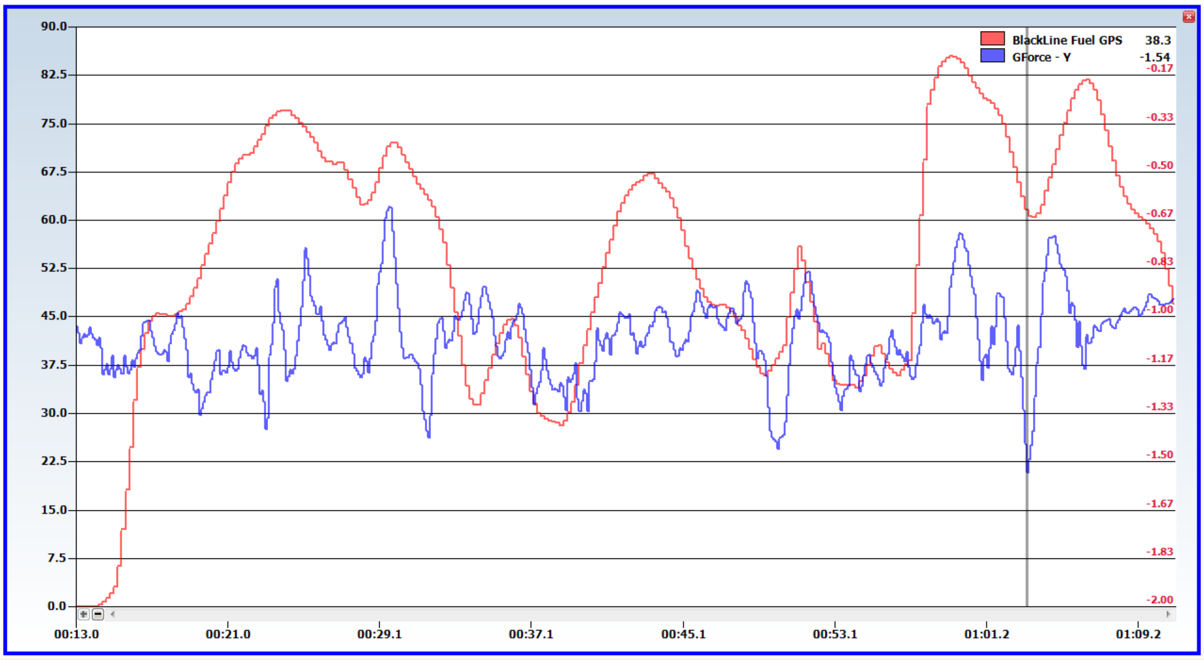

Fig. 7.5.3 Mt Cotton- Max Negative Y Direction G Force

Z – acceleration (negative) and braking (positive)

Maximum negative g force is -0.76 between turn 1 and 2

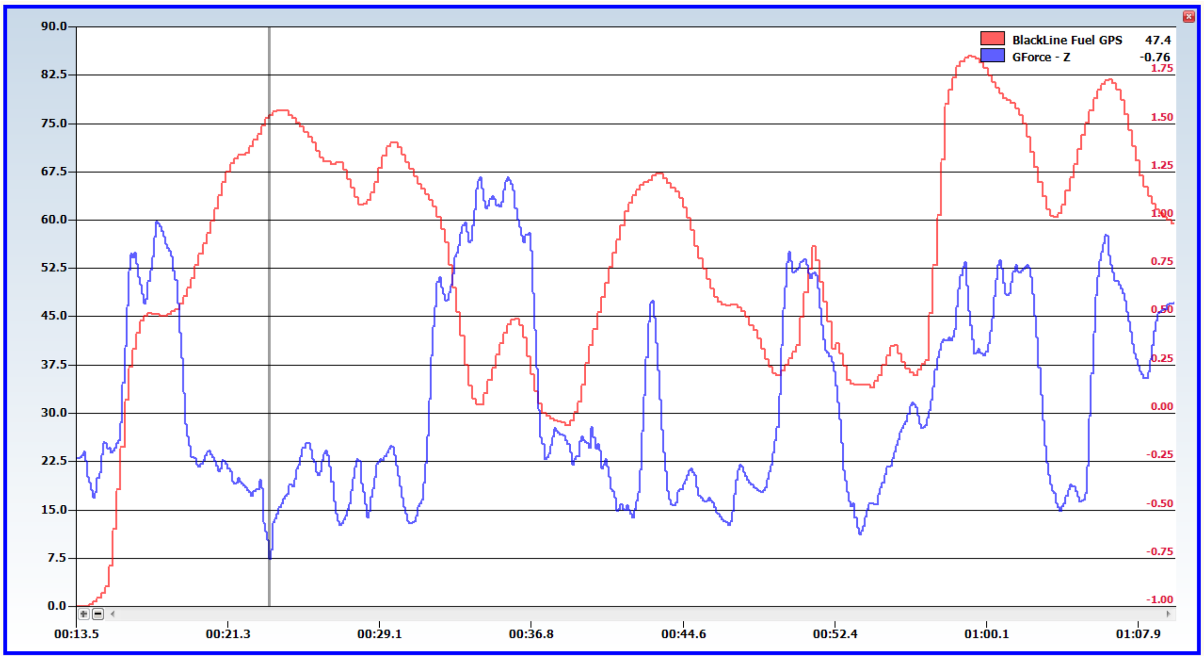

Fig. 7.5.4 Mt Cotton- Max Negative Z Direction G Force

Maximum positive g force of 1.22 at turn 3 entry

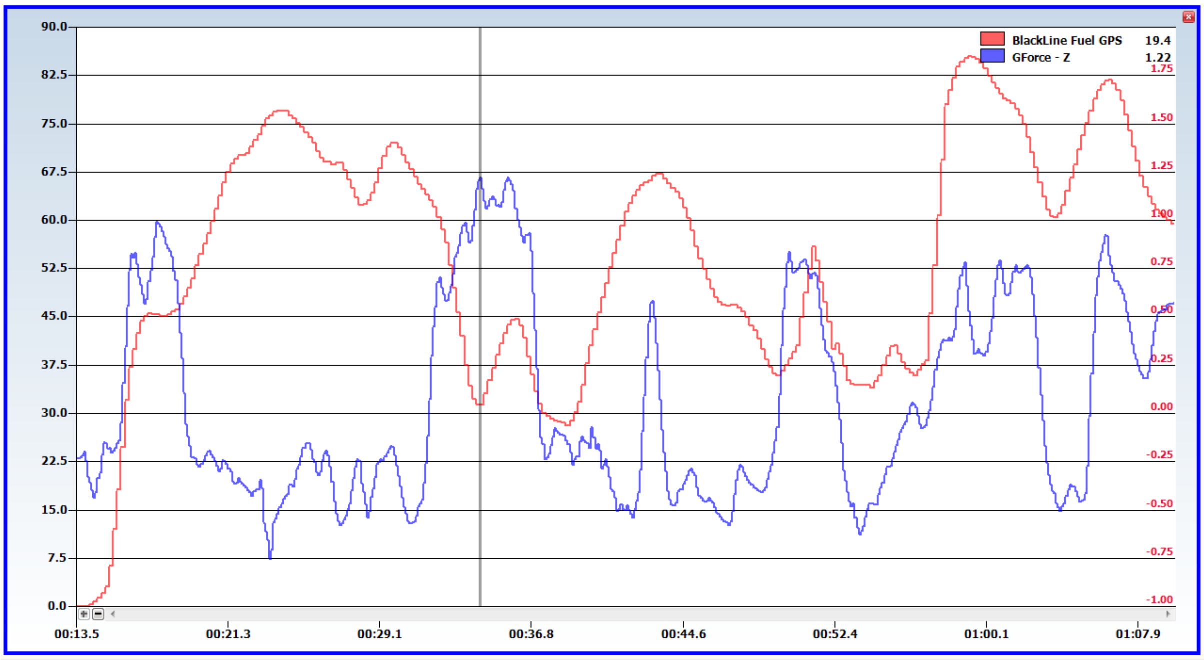

Fig. 7.5.5 Mt Cotton- Max Positive Z Direction G Force

Drivers Evaluation

The first event at this track resulted in an extremely high learning curve. The car parameters for this track were highly unknown and hence the handling characteristics at the limit were only found late in the weekend. The car again tended towards understeer and limited the lap times due to slowing corner speeds.

Turn one turn in was good resulting in good corner entry speeds. The apex to corner exit resulted in understeer and the car was generally running wide on the exit due to this handling condition.

Turn two is on top of the hill and the limiting device throughout the corner is apex understeer. The car then responds well once the grip has returned.

Turn three is highly cambered and few problems are encountered throughout this corner. Unless suspension parameters are extremely limiting, the general grip level is obtained by the tyres potential.

Turn four is again on the top of the hill and similar handling was found as was encountered in turn two.

Turn five resulted in slight understeer on the corner entry. Once the camber section of the corner apex was reached the car turned well and a similar situation to turn three was encountered.

Across the top of the hill through turn six and seven the cars tendency towards understeer was found the most noticeable. The transition between these corners resulted in a car that was limited by its front end grip.

The 120y generally handled the track well, although the understeering nature was evident across the top sections of the hill. The figures of cornering force (7.5.1 and 7.5.2) show that the top sections of the hill result in much lower g forces. This is also a contribution of the cambered nature of the bottom corners, although the nature of the cars handling also supports such differences.

7.6 Pittsworth 4th/5th September 2010 – Watts Link fitted

Tyre Temperatures

No tyre temperature data was recorded due to only being a single lap event.

Pressures were maintained at 32psi front and 28 psi rear.

GPS Data Logger

No data was recorded. The data logger is only run on the Sunday as the best runs are always later in the weekend once the correct lines and driving style have been found. The fastest runs where however run on the Saturday due to rain on Sunday and a patchy wet track being the result. Unfortunately the data logger was not run on the Saturday due to these unforseen issues. If it had been, the data would have showed information corresponding to the limited potential of the car. The car is only pushed to the limit on Sundays runs as Saturdays are seen as familiarisation runs for new tracks.

Drivers Evaluation

The drivers input was however positive to the changes, and the car handled well, with lap times that were closing in on other more sorted and prepared race cars. The Watts link greatly altered the cars attitude and successfully removed almost all understeer.

The first chichane was dealt with extremely well, a situation which has always been a strong point for the car. Turn one is extremely tight and good turn in was generally found although slight understeer did limit corner speed slightly. The corner exit was well controlled with slight power oversteer being the result. The second corner resulted in the same condition as the first and subsequently so did the third. The fourth corner is also a left hand flat gradient corner that had similar effects on the cars handling.

The fifth corner did however result in a touch of corner entry oversteer as the surface change unsettled the car. The car remained predictable throughout the corner and the oversteer could be trimmed to control the cornering amount.

The second chichane also presented few problems, like the first, and leading into the final corner the car remained settled. The last corner resulted in slight corner entry understeer which transferred to apex to corner exit oversteer. The oversteer was again controlled and the amount of turning could be easily controlled with throttle and steering inputs.

The Watts link has greatly reduced the corner exit understeer. The result of the rear roll centre in its current location (differential carrier mid height – unchanged from before Watts link fitment) has shown that oversteer is the result. The Watts link has increased the rear end feel and responsiveness and resulted in a car that has a much crisper and touter feeling rear suspension setup.

7.7 Mt Cotton 2nd/3rd October 2010 – After Modifications

Tyre Temperatures

No tyre temperature readings were taken due to the single lap nature of the event.

Pressures were maintained at 34psi front and 28psi rear.

GPS Data Logger

The data logger was operational on both days however rain prevented the readings from being ideal. The best run was performed on Saturday, when a partially dry run was possible. The data presented from this run will be included, although its relevance as a means of comparison is limited.

X – left (negative) and right (positive), cornering force

Maximum negative g force of -1.11 at turn 5, slightly cambered

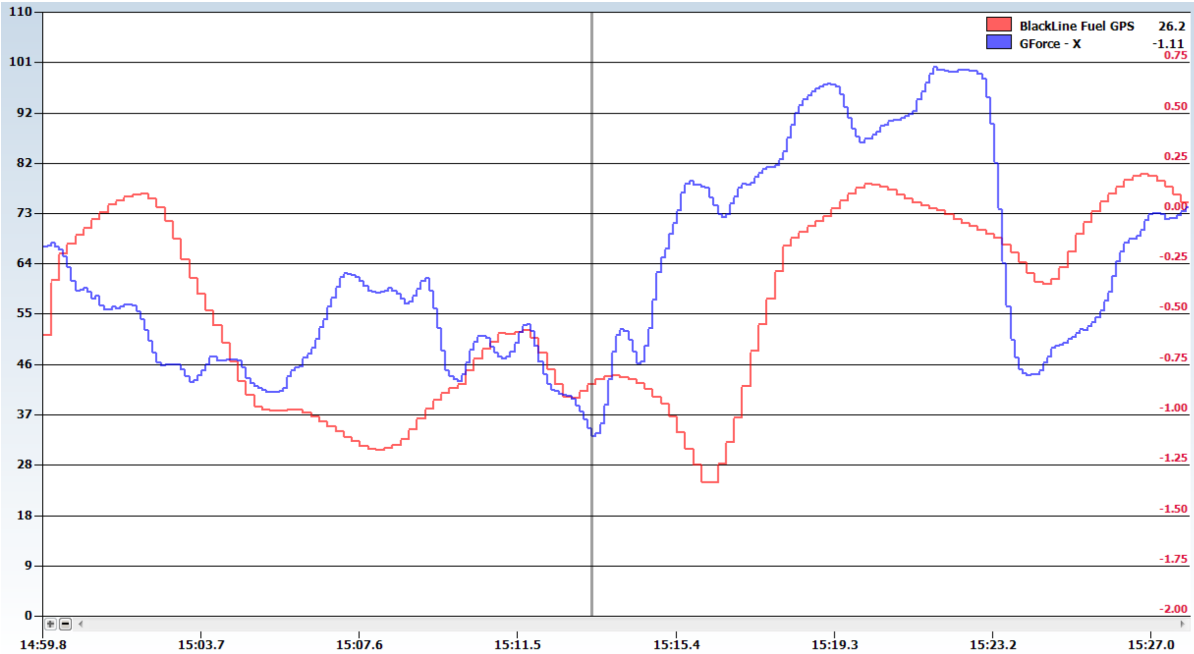

Fig. 7.7.1 Mt Cotton- Max Negative X Direction G Force

Maximum positive g force of 0.77 at turn 1, cambered - uphill exit

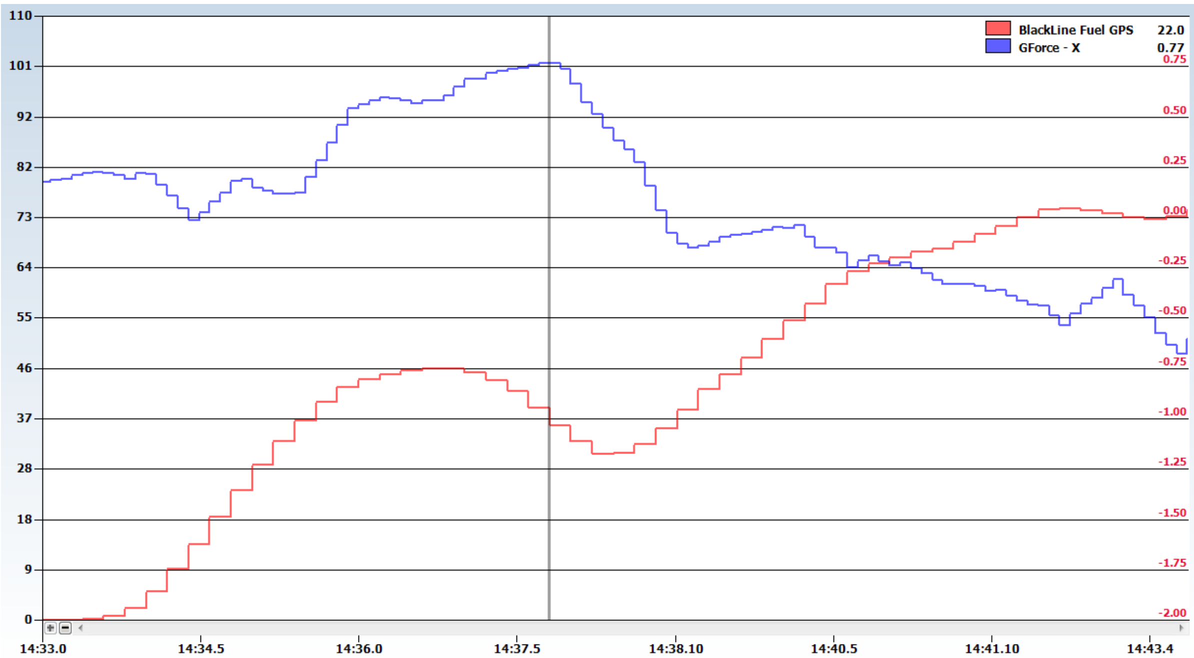

Fig 7.7.2 Mt Cotton- Max Positive X Direction G Force

Y – up and down (negative), gravity

Maximum negative g force of -1.39 at turn 5 entry, drops into cambered corner

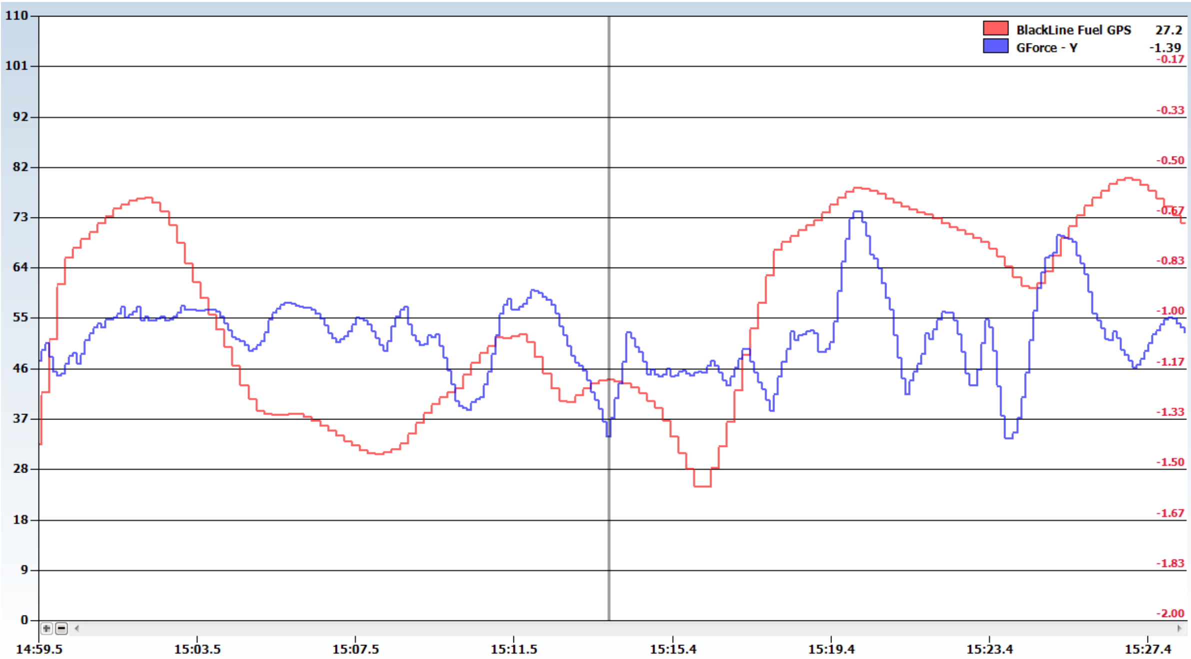

Fig. 7.7.3 Mt Cotton- Max Negative Y Direction G Force

Z – acceleration (negative) and braking (positive)

Maximum negative g force of -0.69 at turn 5 exit, cambered corner flattens out

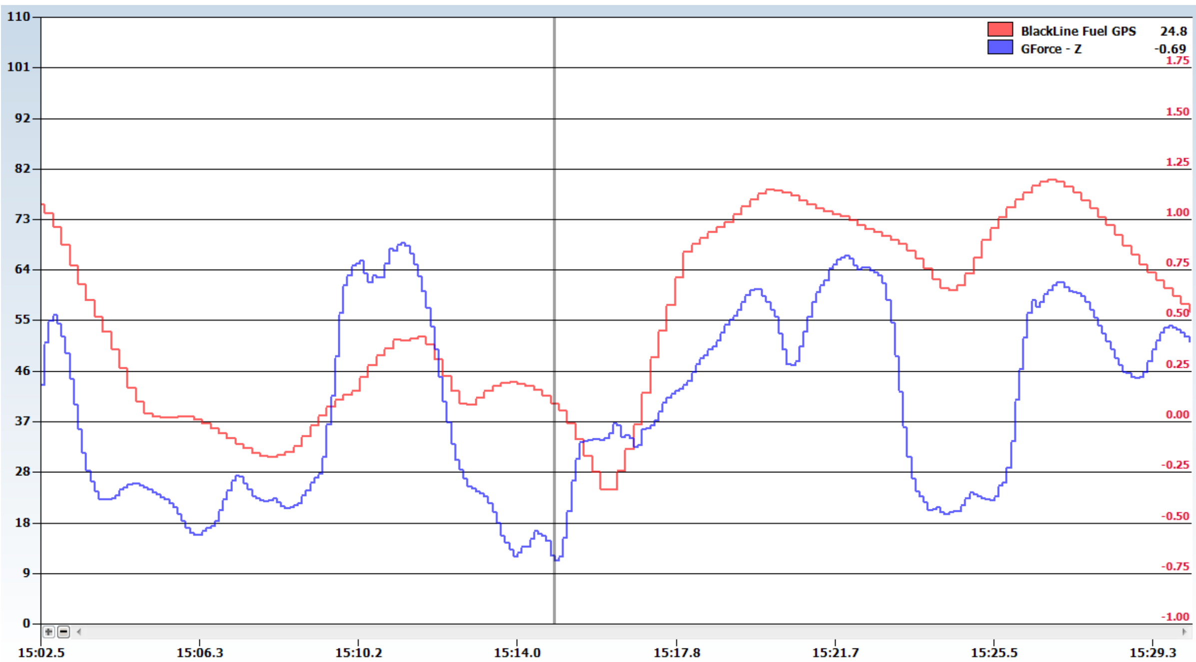

Fig. 7.7.4 Mt Cotton- Max Negative Z Direction G Force

Maximum positive g force of 1.19 at turn 3 braking area, flat

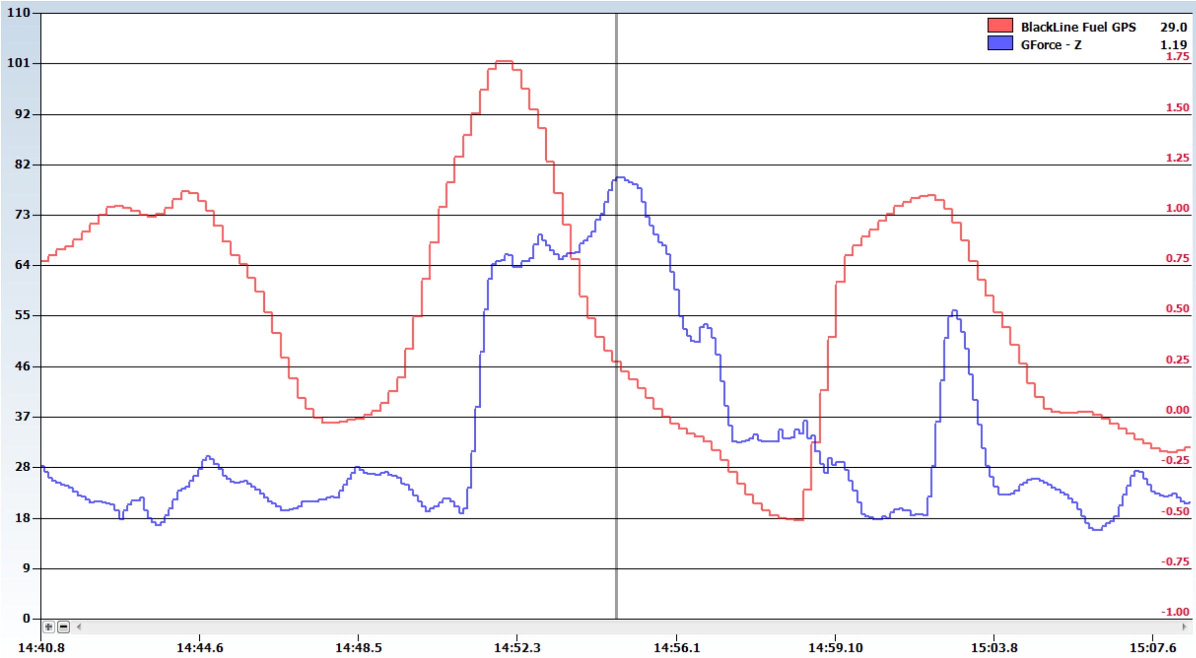

Fig. 7.7.4 Mt Cotton- Max Positive Z Direction G Force

Driver Evaluation

Even in the wet conditions the modifications were found to make a remarkable difference. The understeering nature of the car has been greatly reduced and the oversteer that the car now has is much easier to control in the wet conditions. The car responded well to the wet weather and was more consistent than was previously found in such conditions. The main area that the benefits were noticed was across the top of the hills. In this section the front griped extremely well. Even in the wet conditions it was found that the front end reacted quickly and positively to steering inputs. The car felt very good across the top of the hills, an area which resulted in slight understeer before the modifications were performed. Turn one exit also provided a predictable oversteer nature in the wet conditions, a practice which would have felt uncontrolled without the Watts link's additional lateral support.

7.8 Warwick Test Day 6th September 2010 – Circuit D, 2100m – After Modifications

Many different modifications were made during the test day to find the effects that they had on the handling of the car. During the testing phases of the race car the relevant theory of the modifications and the predicted handling characteristic changes could be evaluated. Many different changes were made over the day and a final setup was achieved that created a more neutral handling car and hence a reduction in lap times was achieved. Note that the silencer was run on all testing runs to limit the strain on the engine. The silencer was then removed once the final setup was achieved and a final performance run performed for later comparison to other cars. The lap times mentioned in this section are those achieved by the GPS data logger and not those normally gained from other timing equipment.

7.8.1 First Run – Standard Tyres – 1.21.994 lap time

The first run on the track was to ensure that the modifications performed were behaving as expected, that no unknown issues were found and to familiarise the driver with the extended track layout. Once the car was found to be performing as expected the race tyres were placed on the car for a baseline run. The laps performed with the standard tyres showed the difference that is achieved by the overall increased grip level.

Tyre Temperatures

No tyre temperature readings were taken with the standard tyres.

GPS Data Logger

X – left (negative) and right (positive), cornering force

Maximum negative g force of -0.90 at turn 7, flat corner

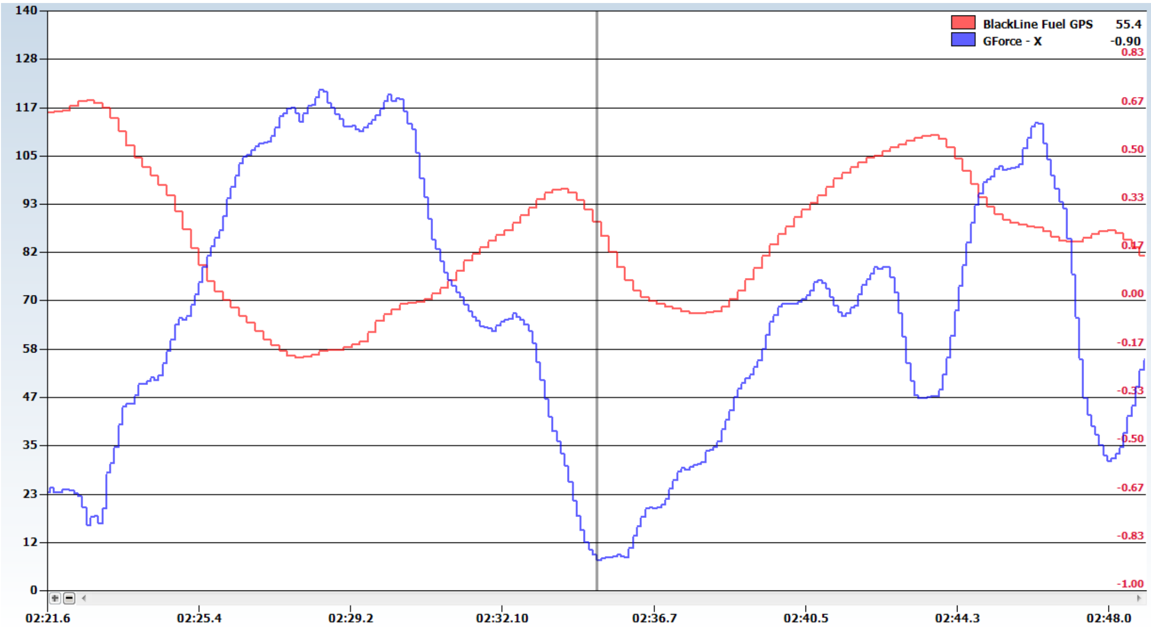

Fig. 7.8.1.1 Warwick Testing – Standard Tyres – Max Negative X Direction G Force

Maximum positive g force of 0.78 at turn 2, slight uphill flattening out

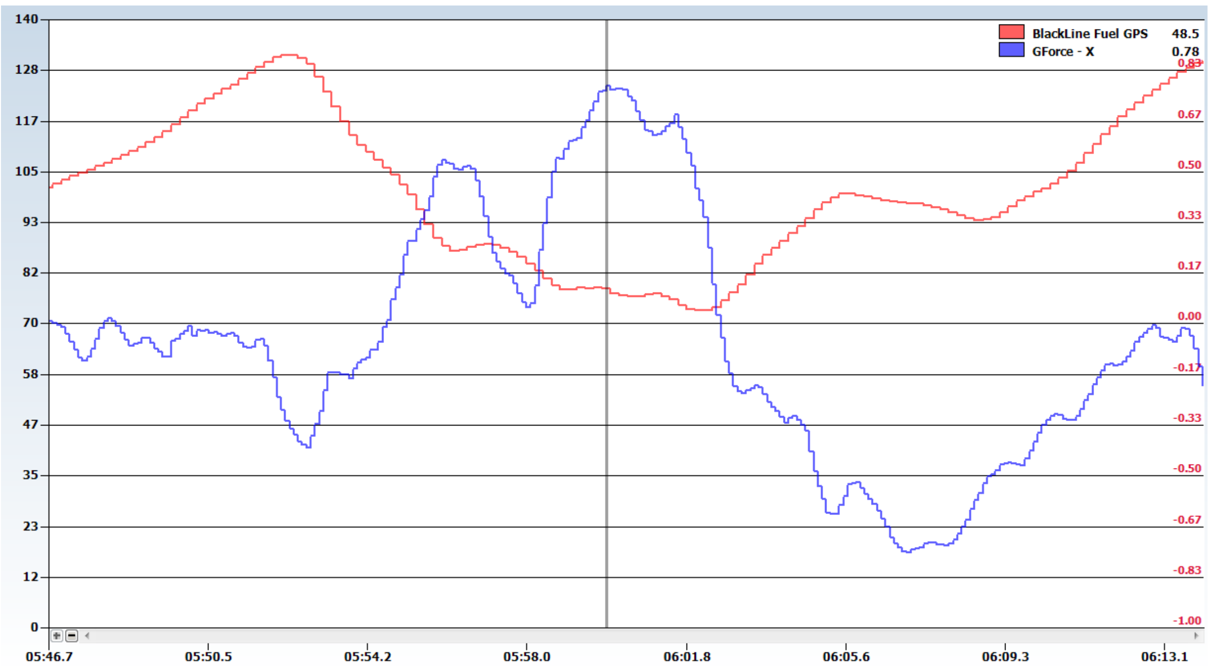

Fig. 7.8.1.2 Warwick Testing – Standard Tyres – Max Positive X Direction G Force

Y – up and down (negative), gravity

Maximum negative g force of -1.26, between turn 9 and 10

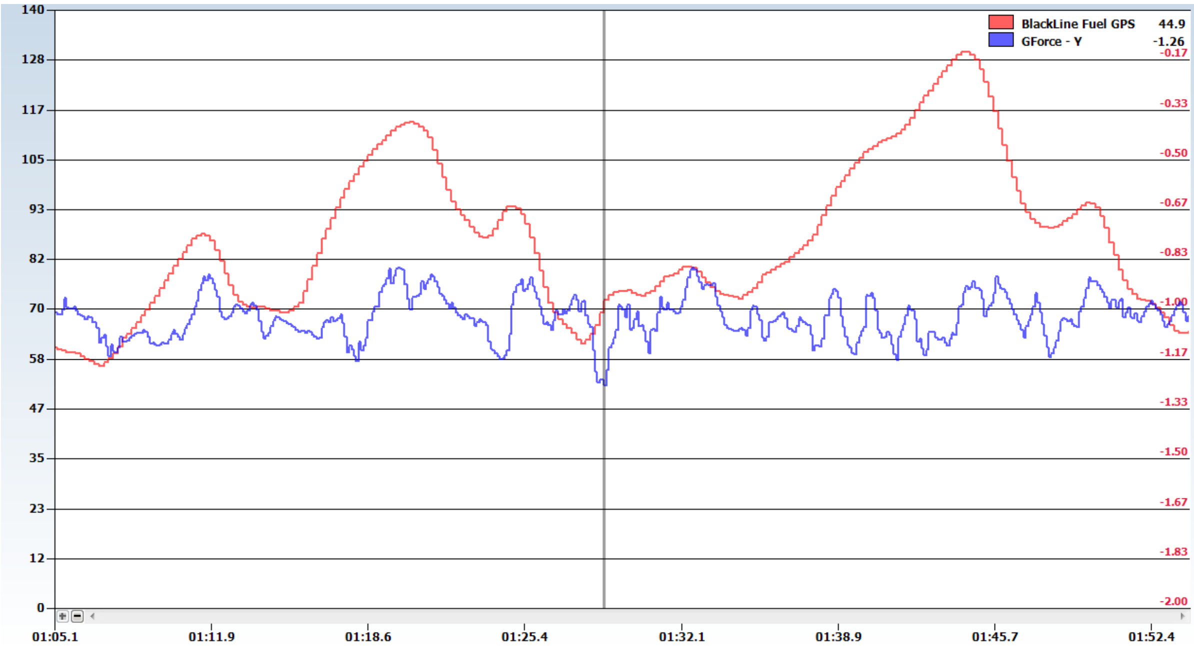

Fig. 7.8.1.3 Warwick Testing – Standard Tyres – Max Negative Y Direction G Force

Z – acceleration (negative) and braking (positive)

Maximum negative g force of -0.56 at turn 7 apex

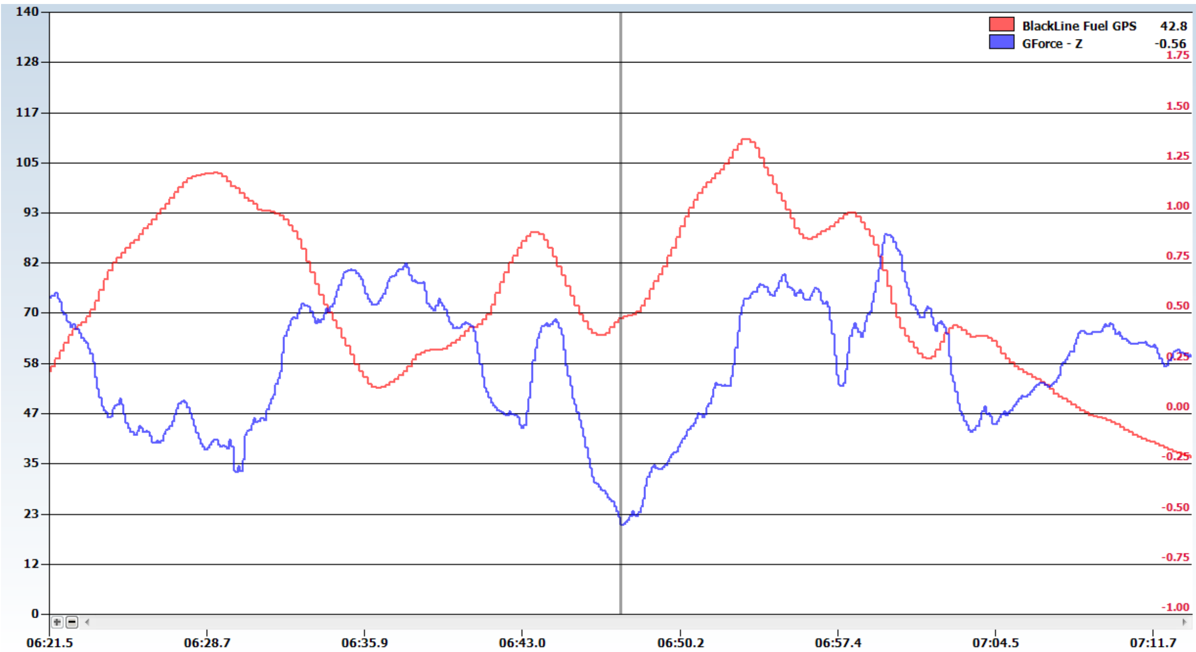

Fig. 7.8.1.4 Warwick Testing – Standard Tyres – Max Negative Z Direction G Force

Maximum positive g force of 1.10 at turn 4 braking area

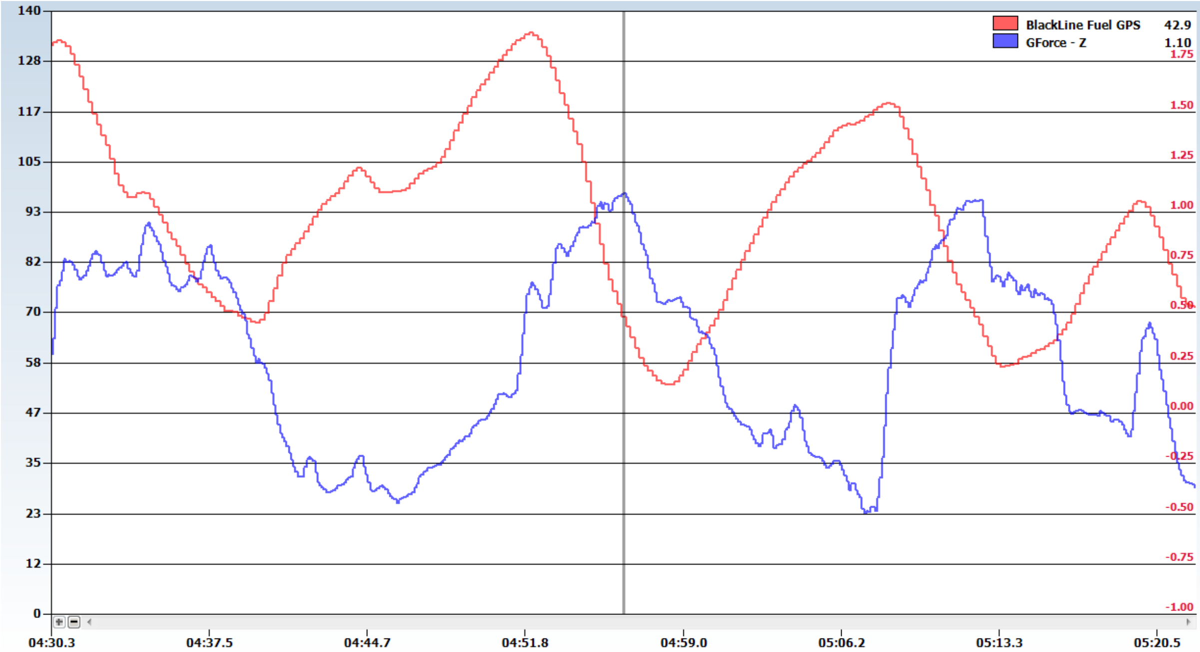

Fig. 7.8.1.1 Warwick Testing – Standard Tyres – Max Negative X Direction G Force

Driver Evaluation

The overall car grip level was lowered greatly when operating with standard road tyres. The car had a tendency to understeer in most instances, although the tyre size difference would have added to this effect. The front did struggle for grip in most instances and heavy trail braking was required to increase turn in.

7.8.2 Second Run – Baseline with Race Tyres – 1.18.542 lap time

Tyre Temperatures

Table 7.8.1.1 Warwick Testing – Baseline Run – Tyre Temperatures

48.7	47.7	53.1		52.8	45.7	44.2
34psi	Pressure low	Camber less negative		34psi	Pressure good	Camber less negative
61.1	58.2	58.0		52.3	48.2	46.5
29psi	Pressure good	Camber irrelevant		28psi	Pressure good	Camber irrelevant

The tyre temperatures show that the car is working the rear tyres harder than the front. This is a direct result of the oversteer which is now evident in the suspension package.

GPS Data Logger

X – left (negative) and right (positive), cornering force

Maximum negative g force of -0.89 at turn 5, flat corner

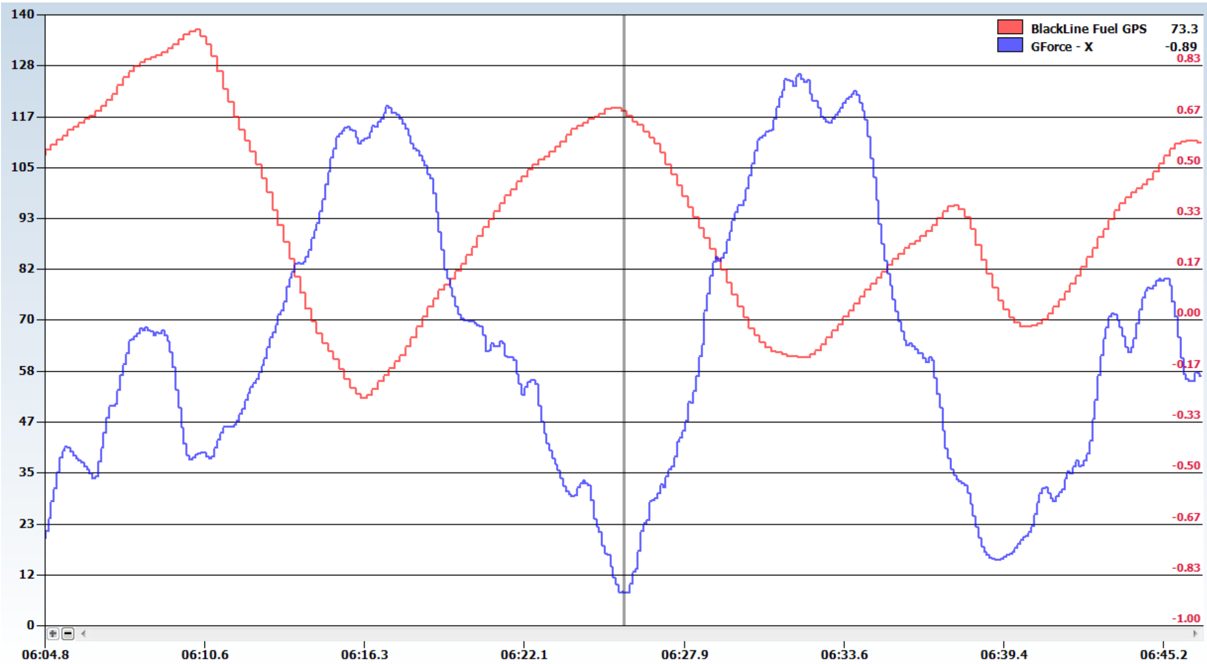

Fig. 7.8.2.1 Warwick Testing – Baseline Run – Max Negative X Direction G Force

Maximum positive g force of 0.90 at turn 2, slight uphill flattening out

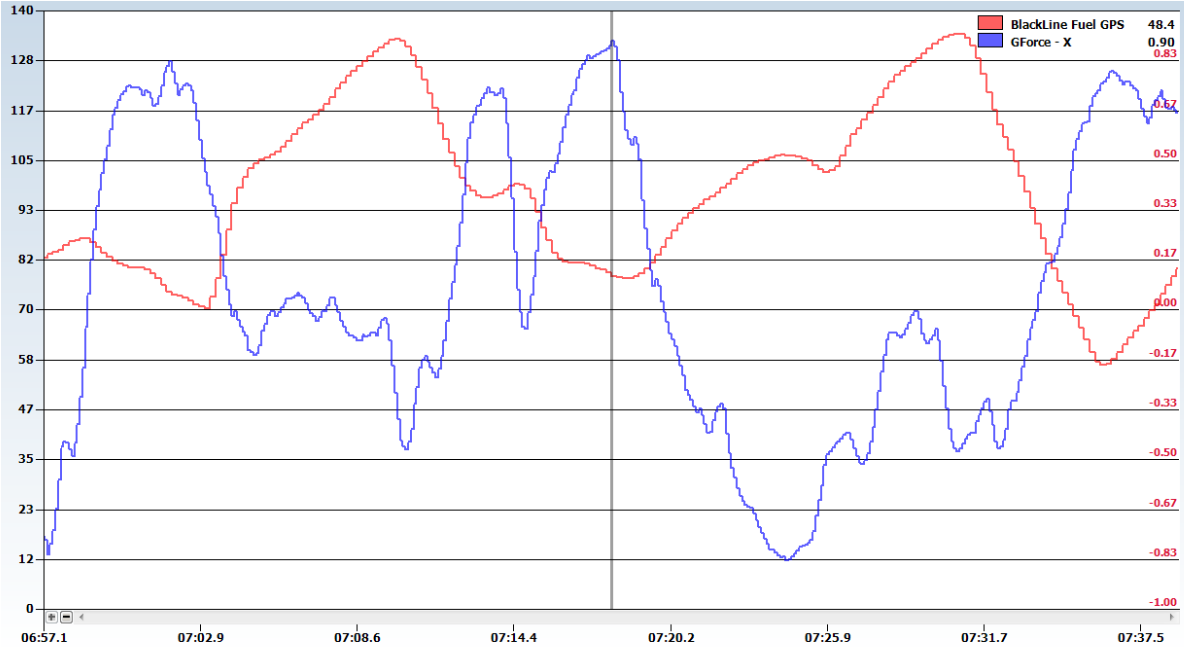

Fig. 7.8.2.2 Warwick Testing – Baseline Run – Max Positive X Direction G Force

Y – up and down (negative), gravity

Maximum negative g force of -1.24 at turn 7 exit

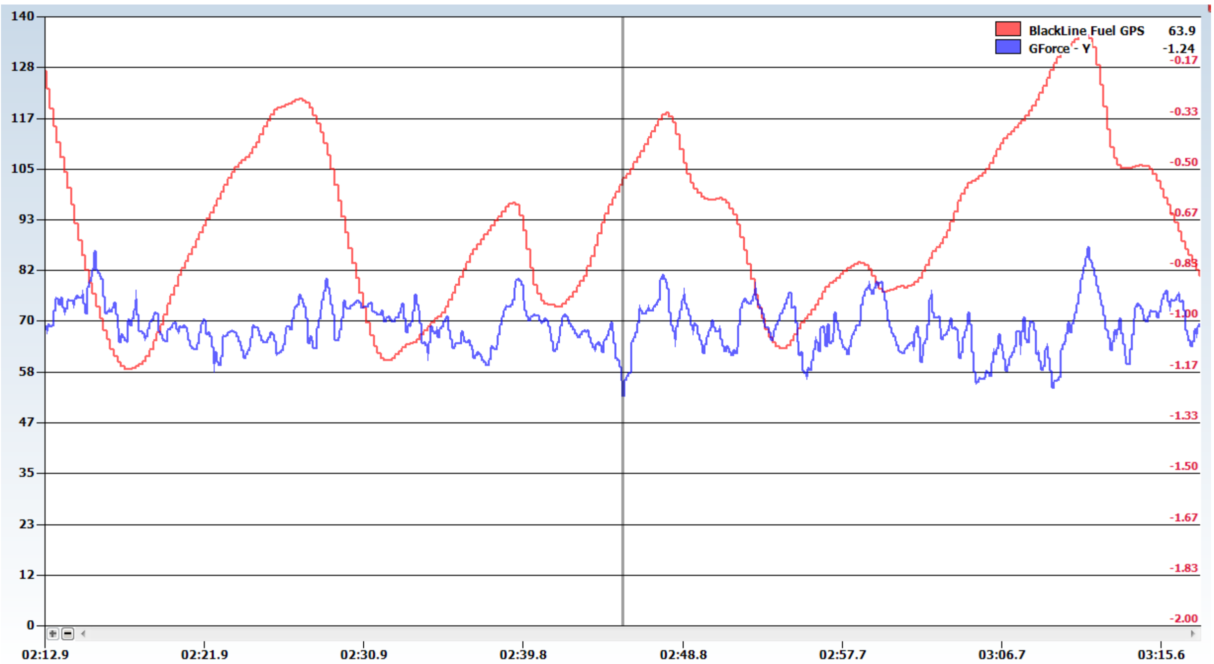

Fig. 7.8.2.3 Warwick Testing – Baseline Run – Max Negative Y Direction G Force

Z – acceleration (negative) and braking (positive)

Maximum negative g force of 0.64 at turn 7 apex, flat corner

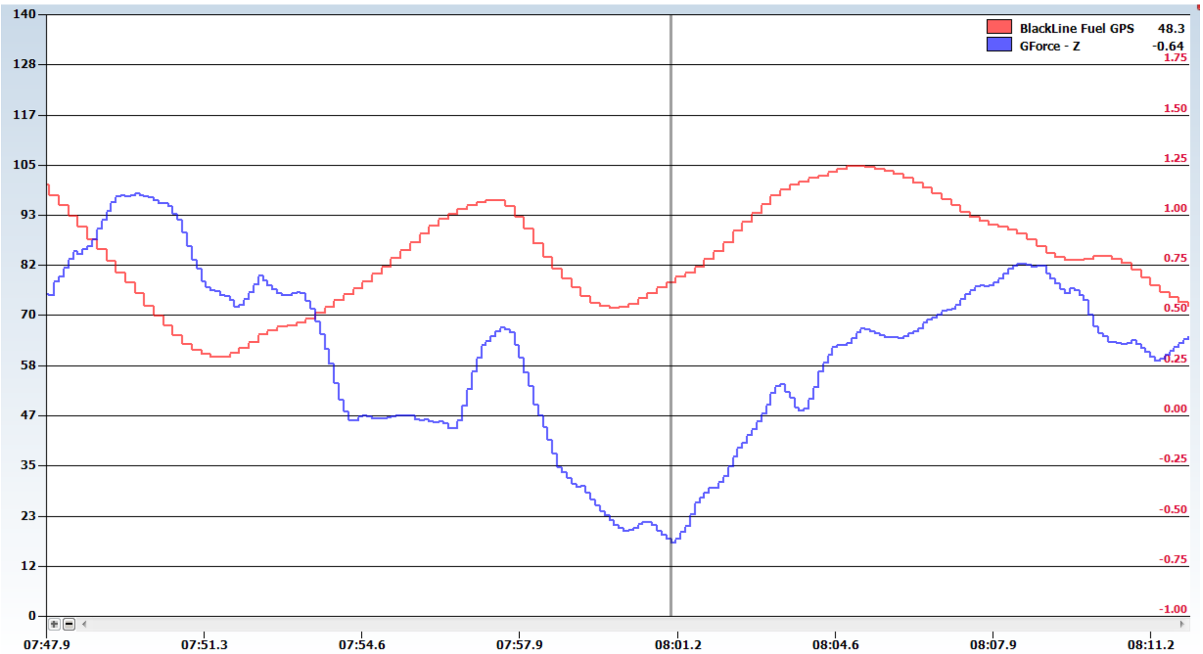

Fig. 7.8.2.4 Warwick Testing – Baseline Run – Max Negative Z Direction G Force

Maximum positive g force of 1.12 at turn 6 braking area, slight bumps, flat gradient

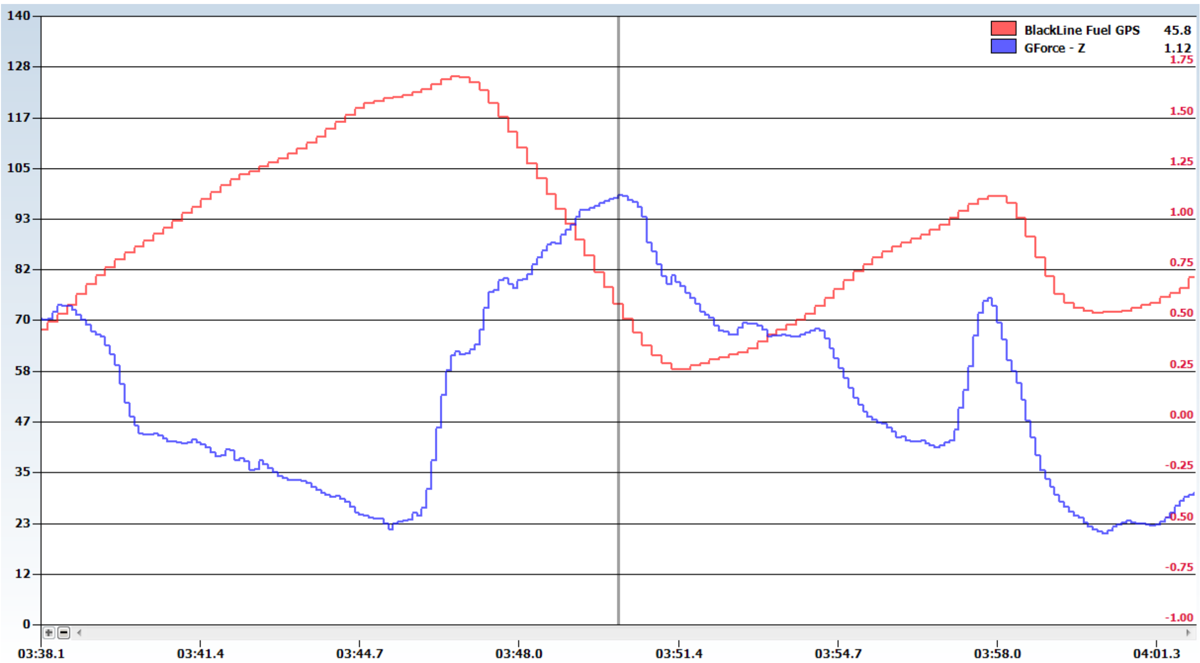

Fig. 7.8.2.5 Warwick Testing – Baseline Run – Max Positive Z Direction G Force

Driver Evaluation

The overall grip levels were much higher than the laps performed with road tyres. The tendency was towards oversteer in most instances. The car turned in well and most of the oversteer was occurring from mid corner onwards. Under braking there was slight understeer although this was greatly reduced in comparison to the amounts found before the modifications were performed. The oversteer on corner exit was the main concern that needed tuning. Although good drive was still found the maximum value was not being achieved. The race tyres also resulted in an increased peak in the cornering force data as shown in figures 7.8.2.1 and 7.8.2.2 The larger peaks show that the race tyres have a less progressive break away (compared to road tyres) although perform with a higher corner force.

7.8.3 Third Run – No Front Anti-Roll Bar – 1.19.346 lap time

Tyre Temperatures

Table 7.8.3.1 Warwick Testing – No Front Anti-Roll Bar – Tyre Temperatures

54.3	54.3	57.5		56.4	49.9	49.4
35 psi	Pressure good	Camber less negative		34 psi	Pressure good	Camber less negative
63.4	57.1	56.1		47.9	51.3	49.4
30 psi	Pressure good			29 psi	Pressure high	

The tyre temperatures again show that oversteer is the result with higher temperatures being achieved from the rear tyres. The front tyres were more evenly heated as a result of the decreased lateral weight transfer in the front suspension.

GPS Data Logger

X – left (negative) and right (positive), cornering force

Maximum negative g force of -0.82 at turn 7, flat corner

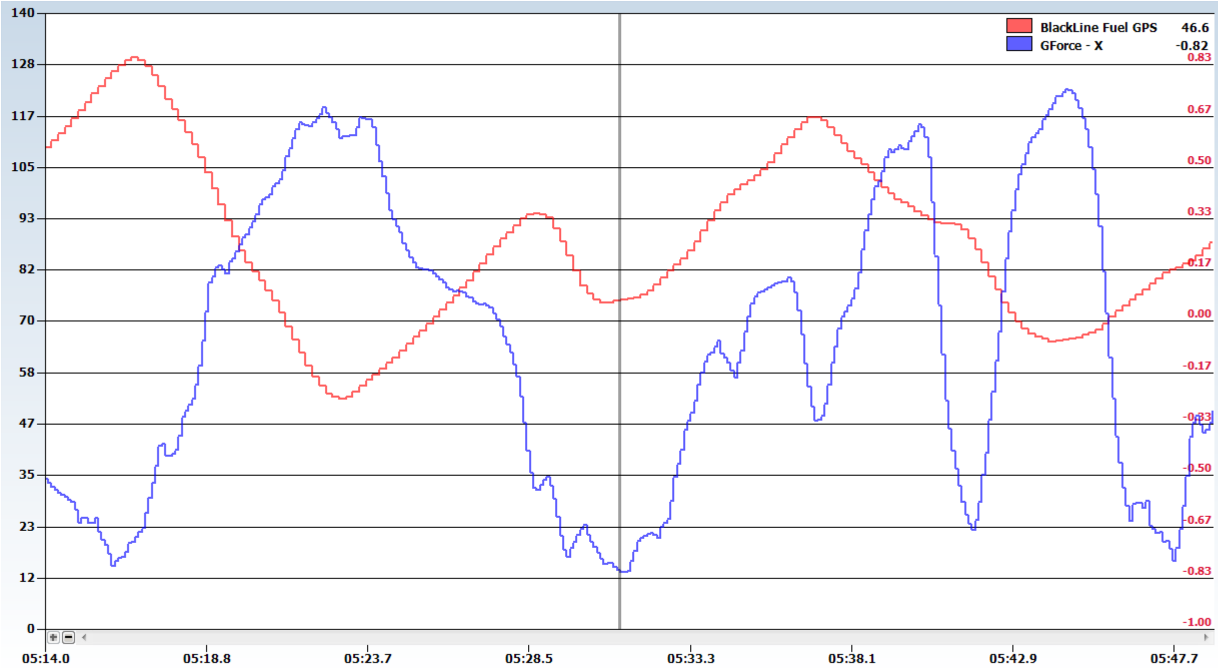

Fig. 7.8.3.1 Warwick Testing – No Front Ant-Roll Bar – Max Negative X Direction G Force

Maximum positive g force of 0.88 at turn 2, slight uphill flattening out

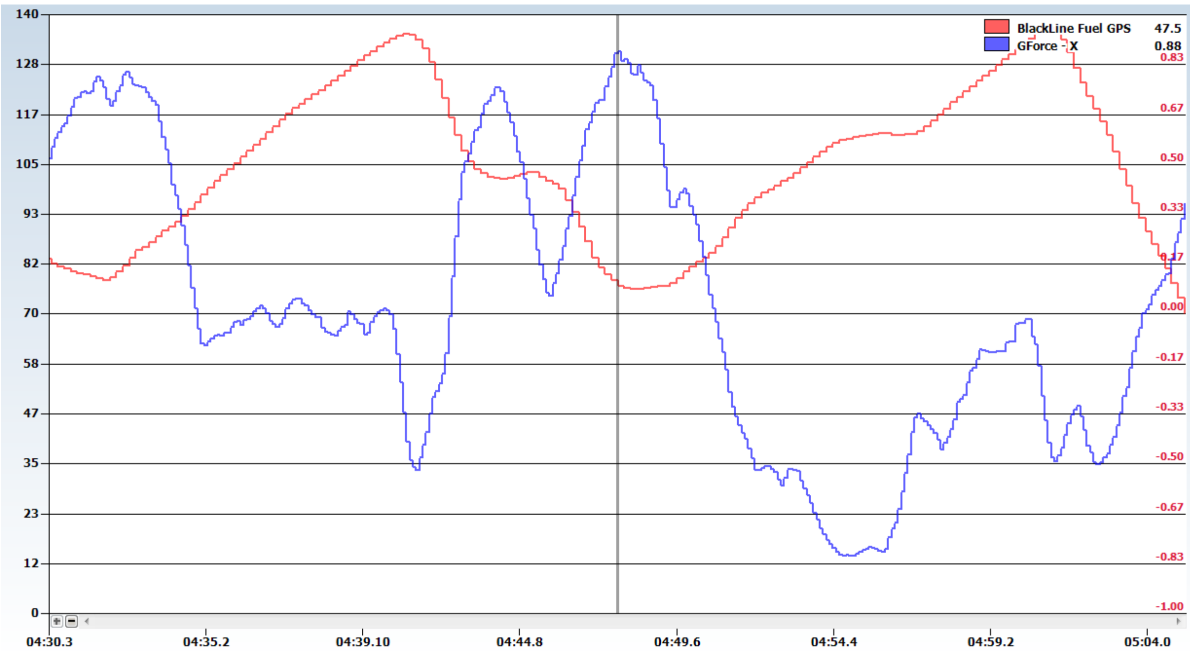

Fig. 7.8.3.2 Warwick Testing – No Front Ant-Roll Bar – Max Positive X Direction G Force

Y – up and down (negative), gravity

Maximum negative g force of -1.22 at bump on straight

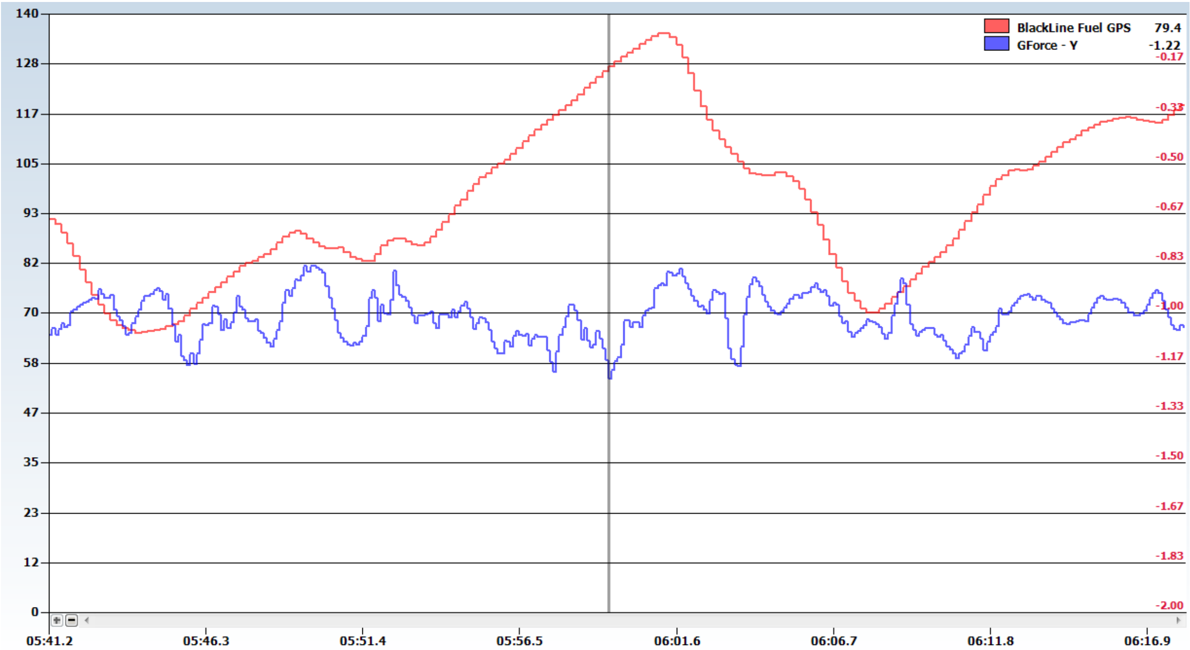

Fig. 7.8.3.3 Warwick Testing – No Front Ant-Roll Bar – Max Negative Y Direction G Force

Z – acceleration (negative) and braking (positive)

Maximum negative g force of -0.63 at turn 7 apex-exit

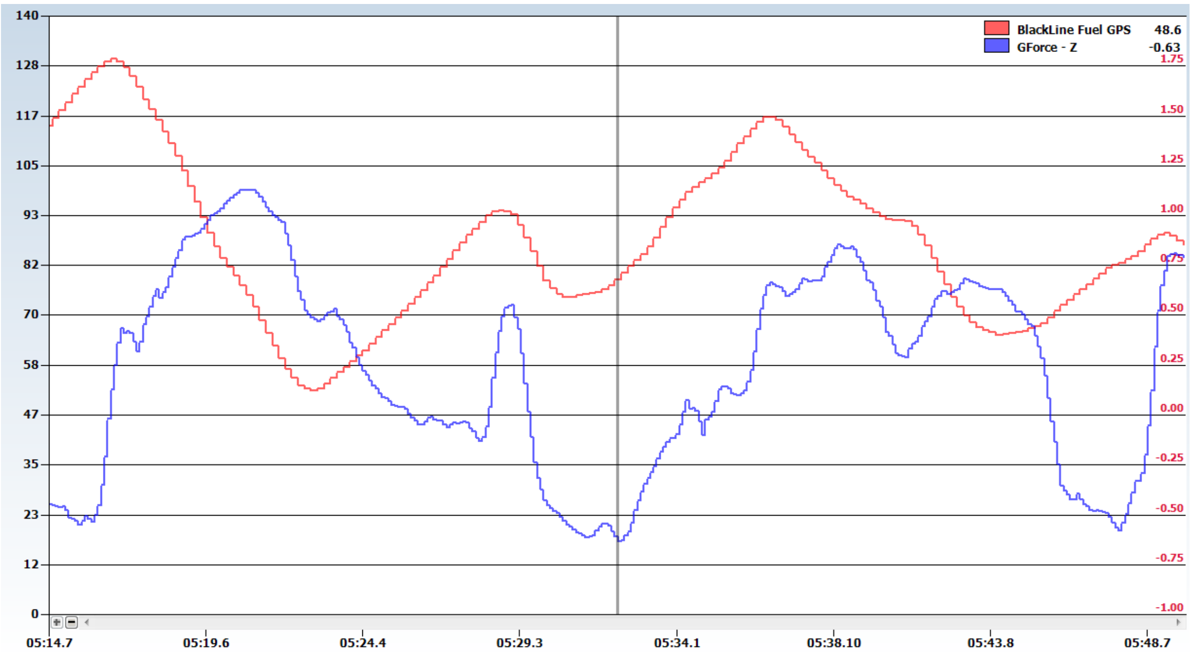

Fig. 7.8.3.4 Warwick Testing – No Front Ant-Roll Bar – Max Negative Z Direction G Force

Maximum positive g force of 1.13 at turn 6 braking area, slight bumps, flat gradient

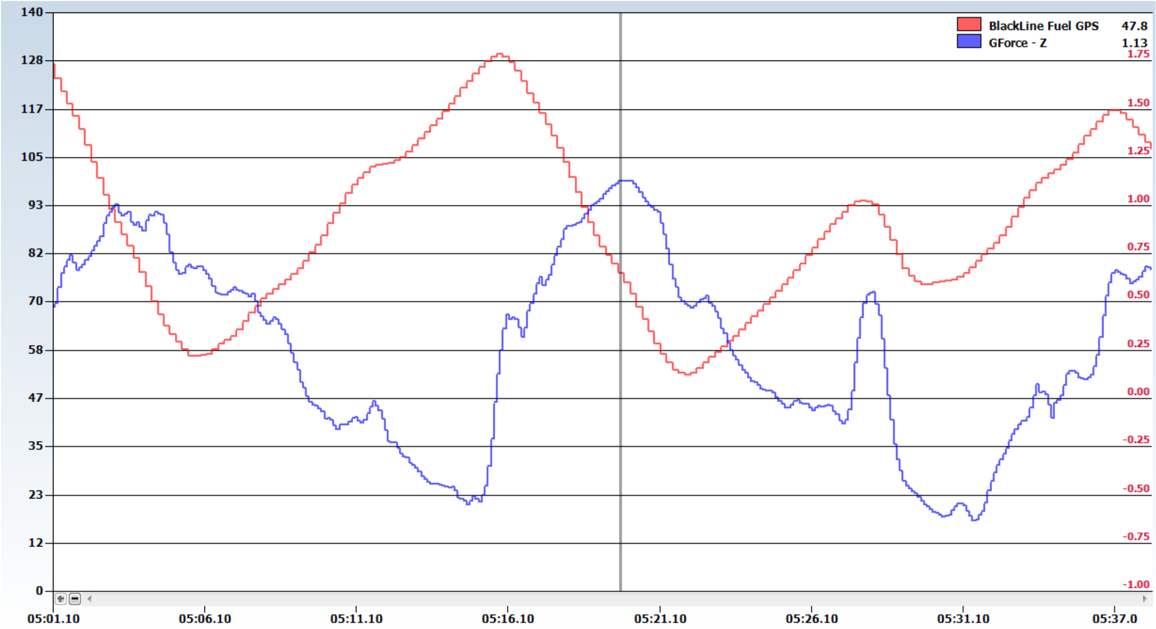

Fig. 7.8.3.5 Warwick Testing – No Front Ant-Roll Bar – Max Positive Z Direction G Force

Driver Evaluation

The car felt very unreliable through the corners. It was slow to respond to changes and generally gave the driver very little confidence in the driving experience. The front did grip well, although it always felt unpredictable and generally rolled too much and felt unconstrained. The front of the car lost its response that was always evident at initial turn in. The purpose of this modification is to simulate to some degree the affect of a smaller front anti-roll bar being used.

7.8.4 Fourth Run – Rear Roll Understeer Reduction – 1.18.253 lap time

Before rear shackle height change – height change over spring length = 45mm

After rear shackle height change – height change over spring length = 30 mm (lowest setting)

Tyre Temperatures

Table 7.8.4.1 Warwick Testing – Rear Roll Understeer Reduction – Tyre Temperatures

53.9	56.3	62.8		59.2	50.7	49.7
35psi	Pressure good	Camber less negative		35psi	Pressure good	Camber less negative
62.9	59.5	56.8		53.5	52.4	51.3
29 psi	Pressure good			29 psi	Pressure good	

The tyre temperatures have shown that the reduction of rear roll understeer has reduced understeer. The front tyres have increased in temperature, showing that an increased level of front grip is evident. The rear temperatures are still dominant however and the result is an oversteering car beyond the limit.

GPS Data Logger

X – left (negative) and right (positive), cornering force

Maximum negative g force of -0.90 at turn 5, flat corner

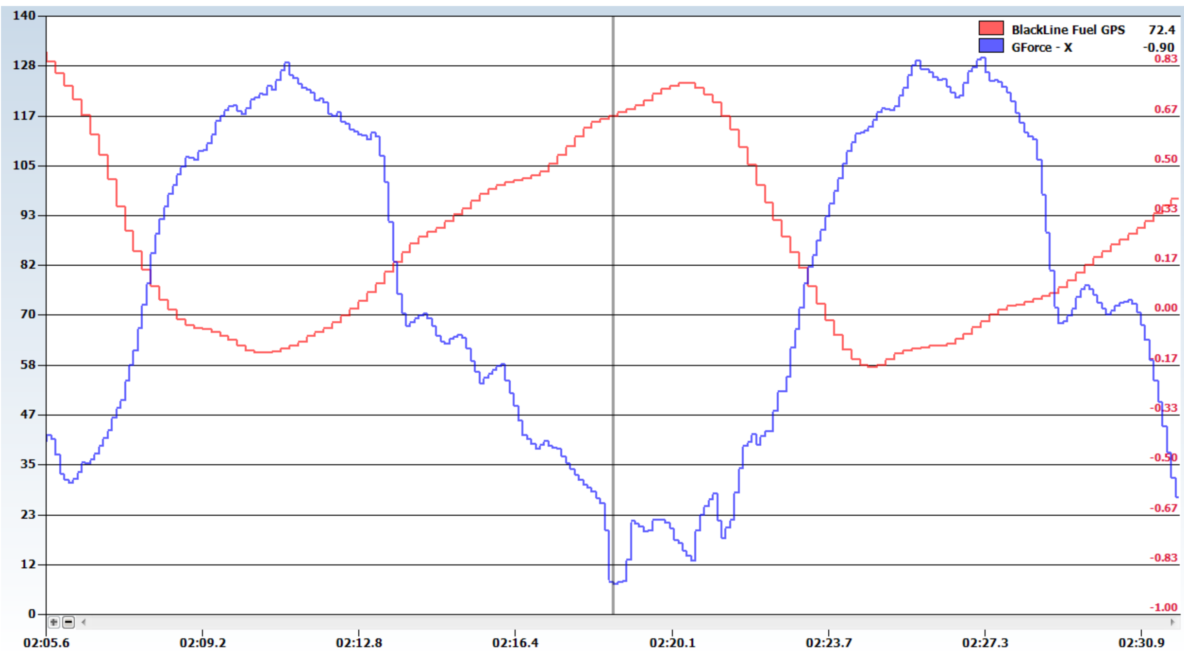

Fig. 7.8.4.1 Warwick Testing – Rear Roll Understeer Reduction – Max Negative X Direction G Force

Maximum positive g force of 0.90 at turn 6, flat corner

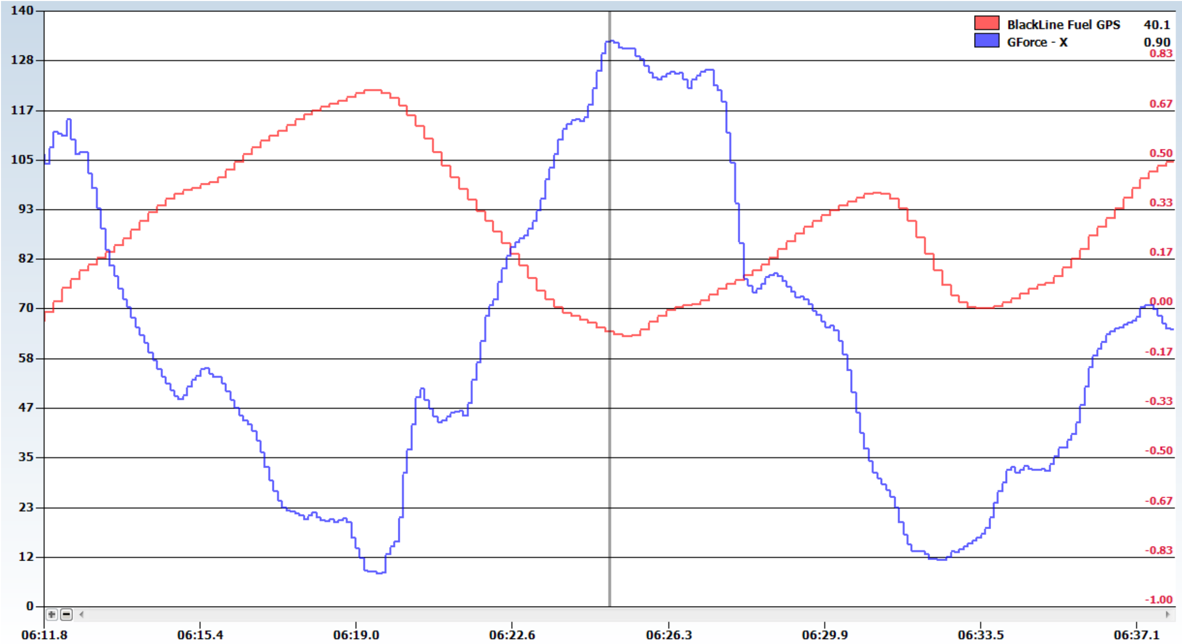

Fig. 7.8.4.2 Warwick Testing – Rear Roll Understeer Reduction – Max Positive X Direction G Force

Y – up and down (negative), gravity

Maximum negative g force of -1.27 between turn 9 and 10

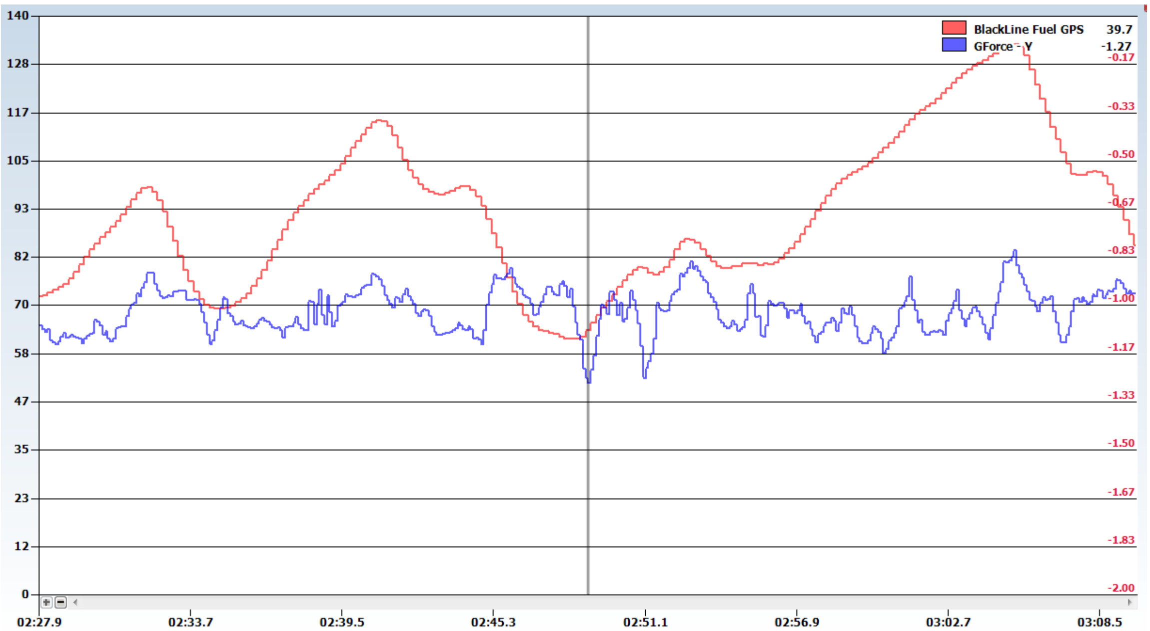

Fig. 7.8.4.3 Warwick Testing – Rear Roll Understeer Reduction – Max Negative Y Direction G Force

Z – acceleration (negative) and braking (positive)

Maximum negative g force of -0.59 at turn 7, apex-exit

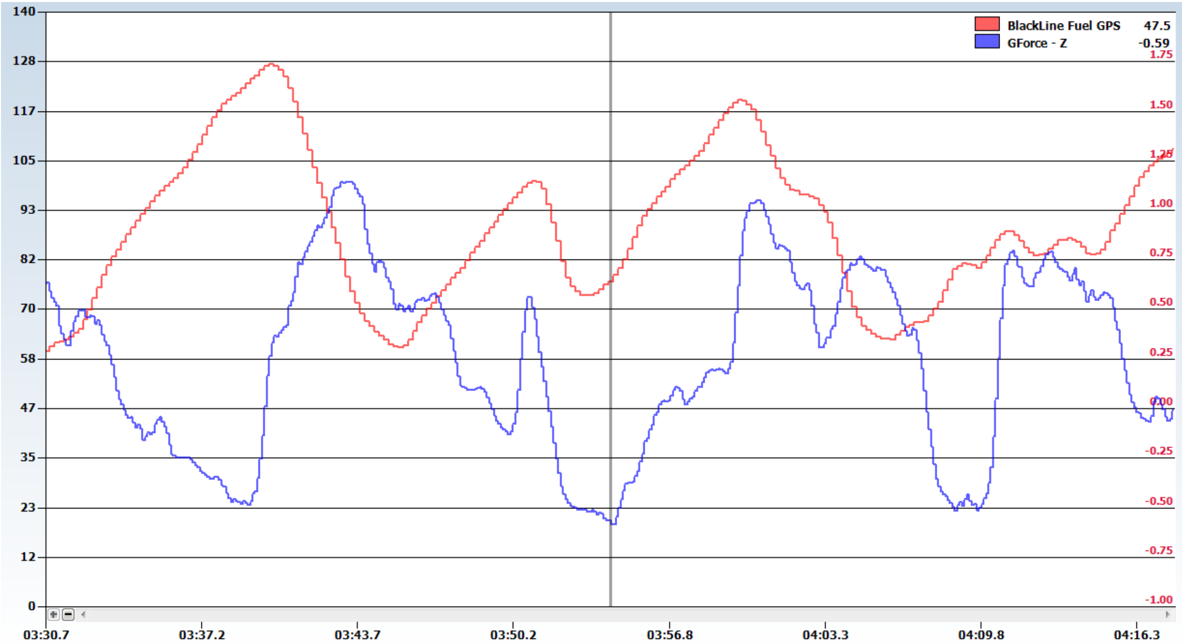

Fig. 7.8.4.4 Warwick Testing – Rear Roll Understeer Reduction – Max Negative Z Direction G Force

Maximum positive g force of 1.10 at turn 4 braking area

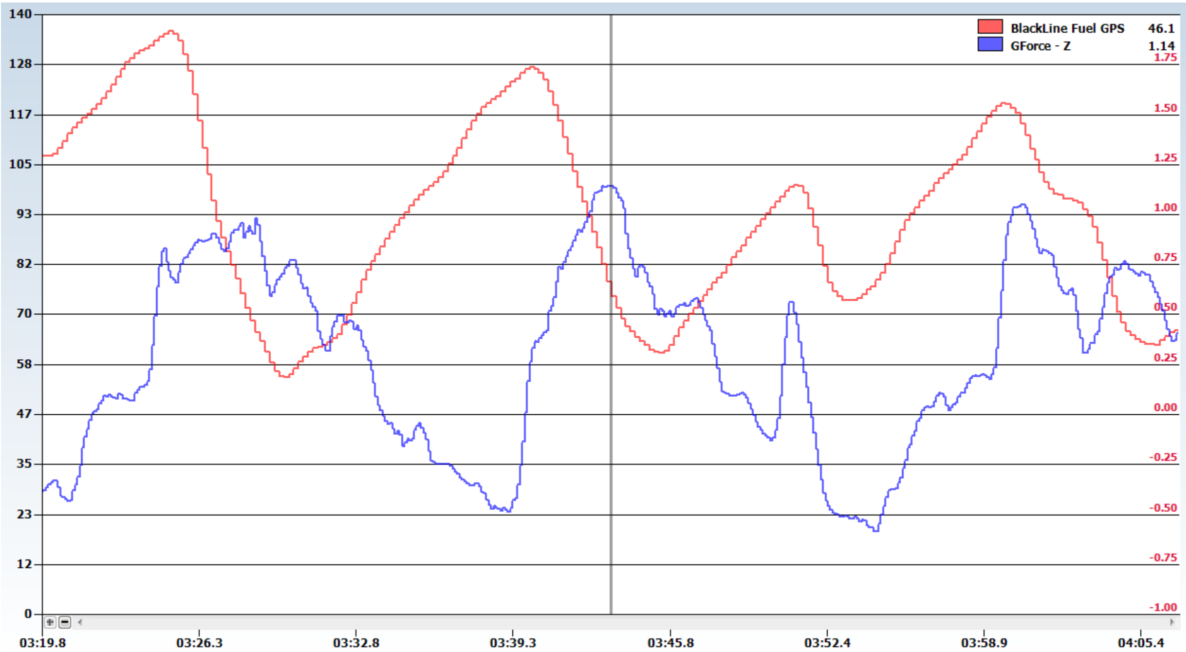

Fig. 7.8.4.5 Warwick Testing – Rear Roll Understeer Reduction – Max Positive Z Direction G Force

Driver Evaluation

The oversteer characteristics of the car have increased. There is more mid corner oversteer which follows on through the corner and results in a net total amount of oversteer. This tuning change also resulted in a rear ride height change that raised the back of the car a reasonable amount. The ride height change was only approximately 30-50mm although the extra weight that this would place on the front wheels cannot be ignored. For this reason the roll steer was left at the normal height until larger lowering blocks are made to correct the ride height change. Figure 7.8.4.1 also shows that at turn five the car was being trimmed around the corner, a means of controlling the oversteering tendency of the car. The dips in cornering force shows when the rear of the car was sliding and trimming of the variables was being performed to realign the car and return cornering force.

7.8.5 Fifth Run – Rear Roll Centre, Height Reduction – 1.17.690 lap time

Before rear roll centre height changed = 293mm above ground level

After rear roll centre height changed = 185mm above ground level (lowest setting)

Note: centre of differential carrier is 285mm above ground level

Tyre Temperatures

Table 7.8.5.1 Warwick Testing – Lowered Rear Roll Centre – Tyre Temperatures

65.9	61.4	67.5		64.4	56.8	53.8
35.5 psi	Pressure low	Camber less negative		35 psi	Pressure good	Camber less negative
60.7	58.8	58.7		52.6	54.0	49.3
30 psi	Pressure good			29 psi	Pressure high	

Tyre temperatures have shown that the handling has been greatly altered. The car is again working the front tyres harder and higher temperatures are the result. The understeer which has again been experienced is supported by lower rear tyre temperatures in comparison to the highly increased front temperatures.

GPS Data Logger

X – left (negative) and right (positive), cornering force

Maximum negative g force of -0.96 at turn 7 approach, flat corner

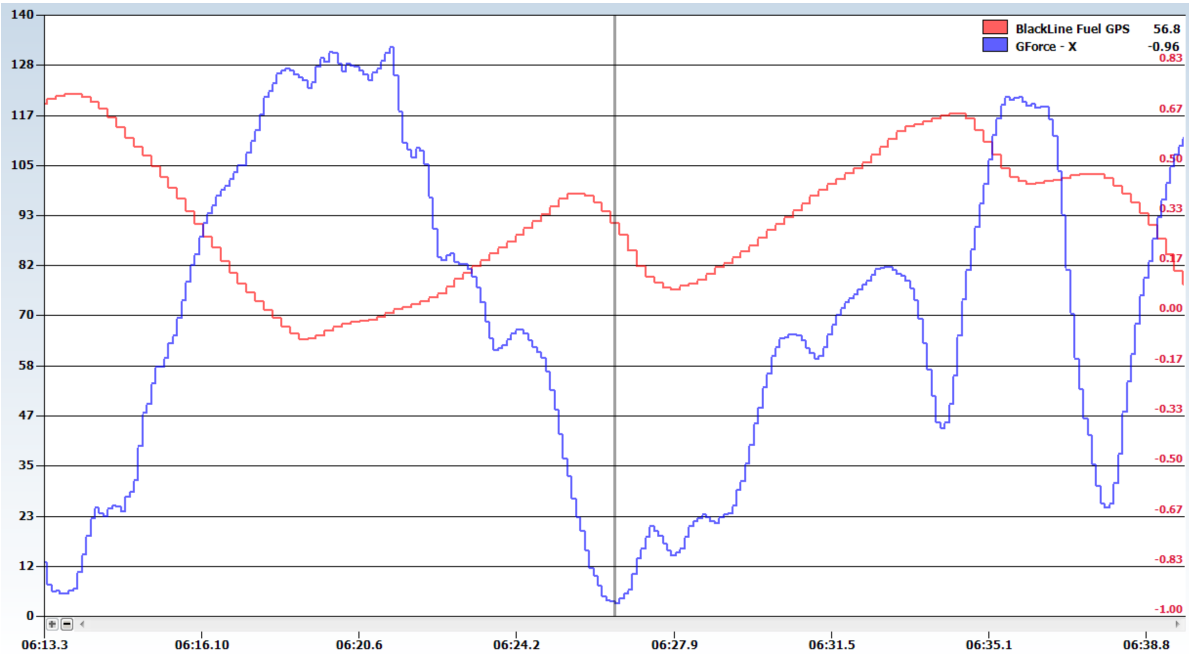

Fig. 7.8.5.1 Warwick Testing – Lowered Rear Roll Centre – Max Negative X Direction G Force

Maximum positive g force of 0.94 at turn 6, flat corner

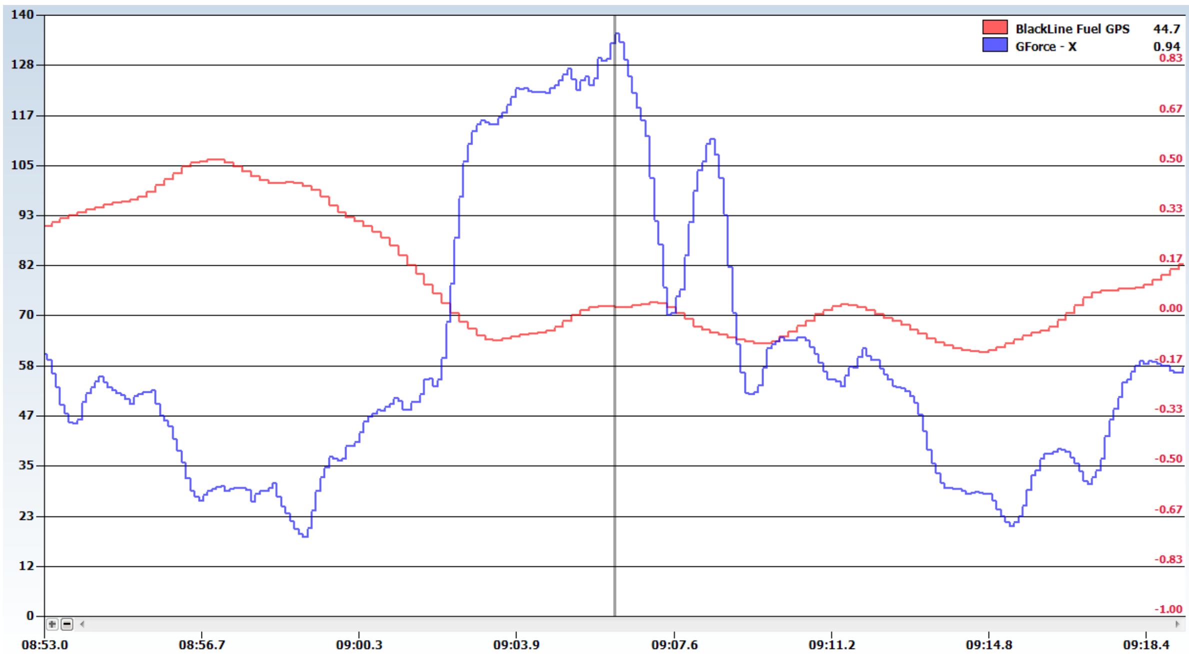

Fig. 7.8.5.2 Warwick Testing – Lowered Rear Roll Centre – Max Positive X Direction G Force

Y – up and down (negative), gravity

Maximum negative g force of -1.28, between turn 9 and 10

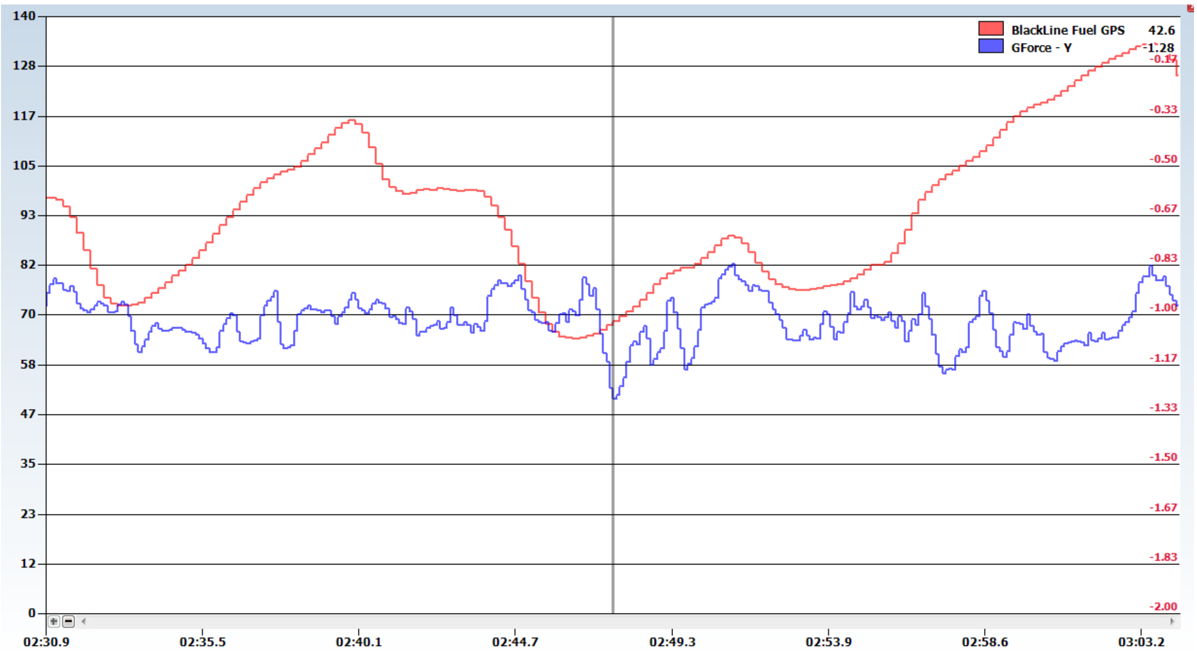

Fig. 7.8.5.3 Warwick Testing – Lowered Rear Roll Centre – Max Negative Y Direction G Force

Z – acceleration (negative) and braking (positive)

Maximum negative g force of -0.65 at turn 7 apex-exit

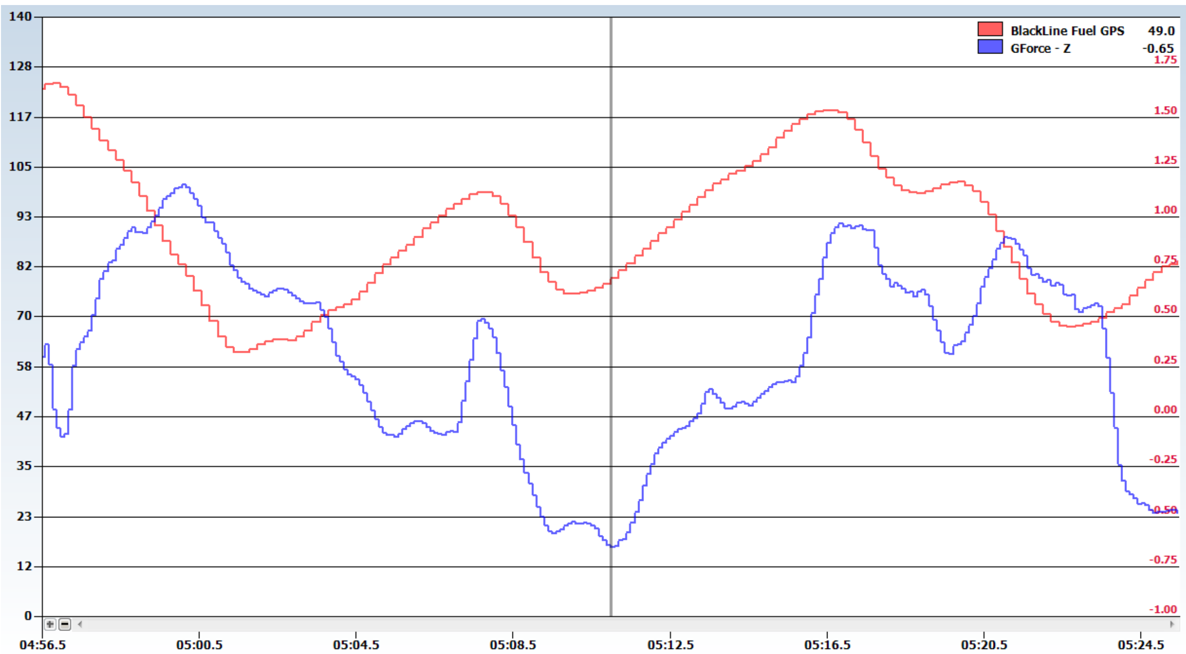

Fig. 7.8.5.1 Warwick Testing – Lowered Rear Roll Centre – Max Negative Z Direction G Force

Maximum positive g force of 1.16 at turn 6 braking area, slight bumps flat gradient

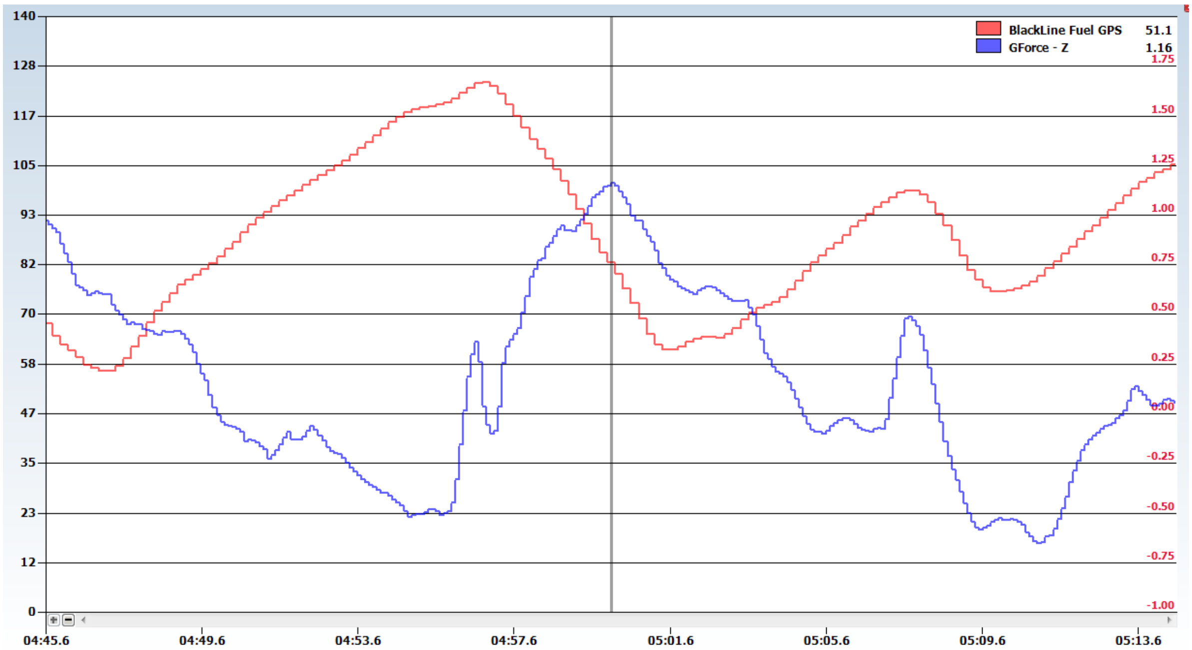

Fig. 7.8.5.1 Warwick Testing – Lowered Rear Roll Centre – Max Negative X Direction G Force

Drivers Evaluation

The extremely low rear roll centre resulted in an increased amount of understeer. The turn in was still evident and responsive although the cur tended to run wide on the approach to the corner apex. The understeer was severe enough to follow right through to the corner exit. The effect of the understeer had the car running wide on corner exit on many occasions. Figure 7.8.5.2 shows that understeer has occurred. The spike of decreased cornering force was the result of the front end losing grip and failing to turn. The limit of the cars current corning power was found just before this point. The understeer had reduced the total cornering power potential.

7.8.6 Sixth Run – Rear Roll Centre, Middle Location – 1.16.388 lap time

Tyre Temperatures

Table 7.8.6.1 Warwick Testing – Middle Rear Roll Centre – Tyre Temperatures

69.4	64.4	67.4		62.9	57.0	57.4
36psi	Pressure low	Camber more negative		35 psi	Pressure low	Camber less negative
71.2	69.8	67.0		55.5	60.2	58.2
30 psi	Pressure good			29 psi	Pressure high	

Higher rear tyre temperatures show that the car has a slight oversteering tendency. The difference between front and rear values is extremely close and supports that a relatively neutral setup with slight oversteer should be the result.

GPS Data Logger

X – left (negative) and right (positive), cornering force

Maximum negative g force of -1.00 at turn 7, flat corner

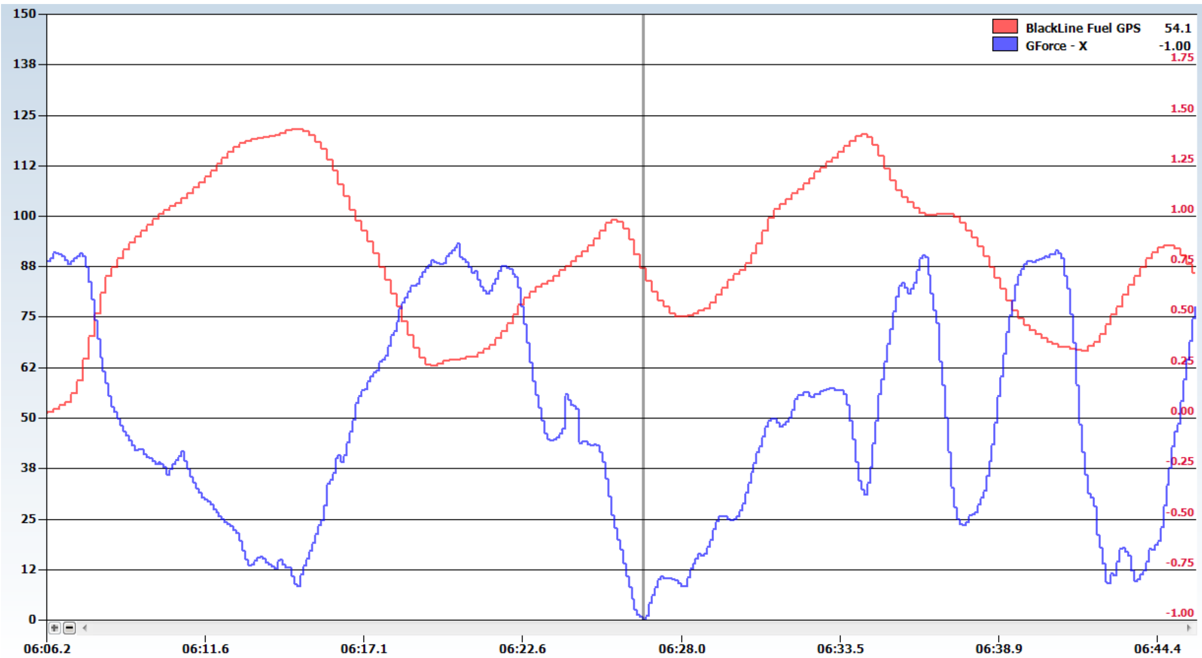

Fig. 7.8.6.1 Warwick Testing – Middle Rear Roll Centre – Max Negative X Direction G Force

Maximum positive g force of 1.02 at turn 11, flat corner – corner before straight

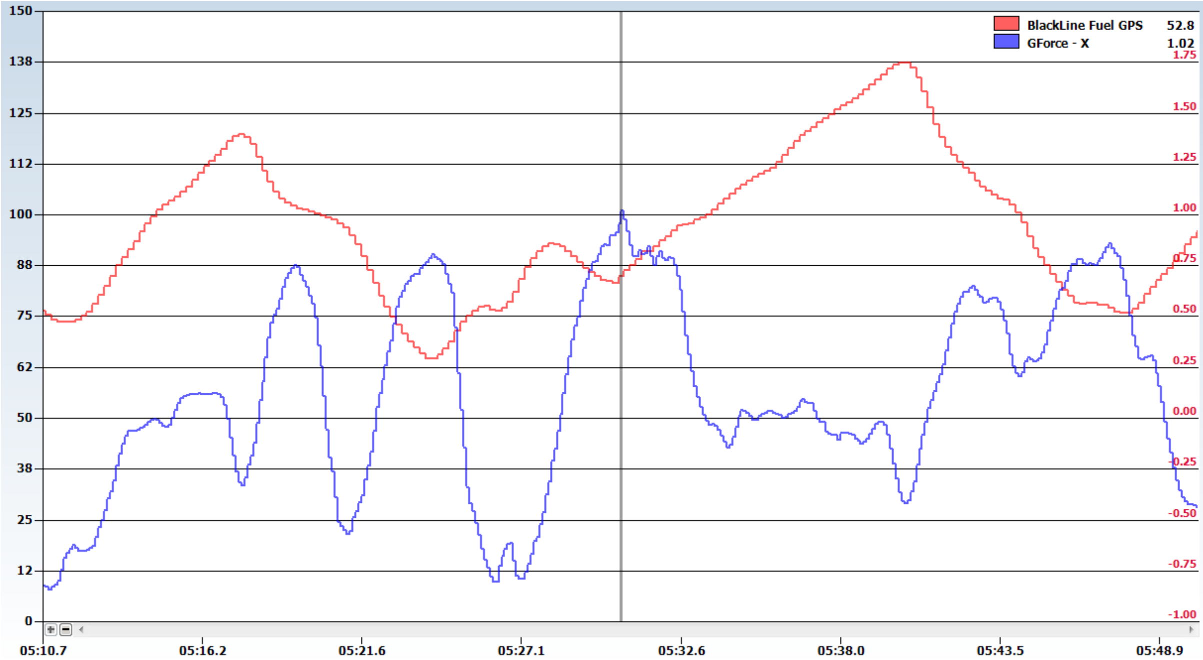

Fig. 7.8.6.2 Warwick Testing – Middle Rear Roll Centre – Max Positive X Direction G Force

Y – up and down (negative), gravity

Maximum negative g force of -1.26, between turn 9 and 10

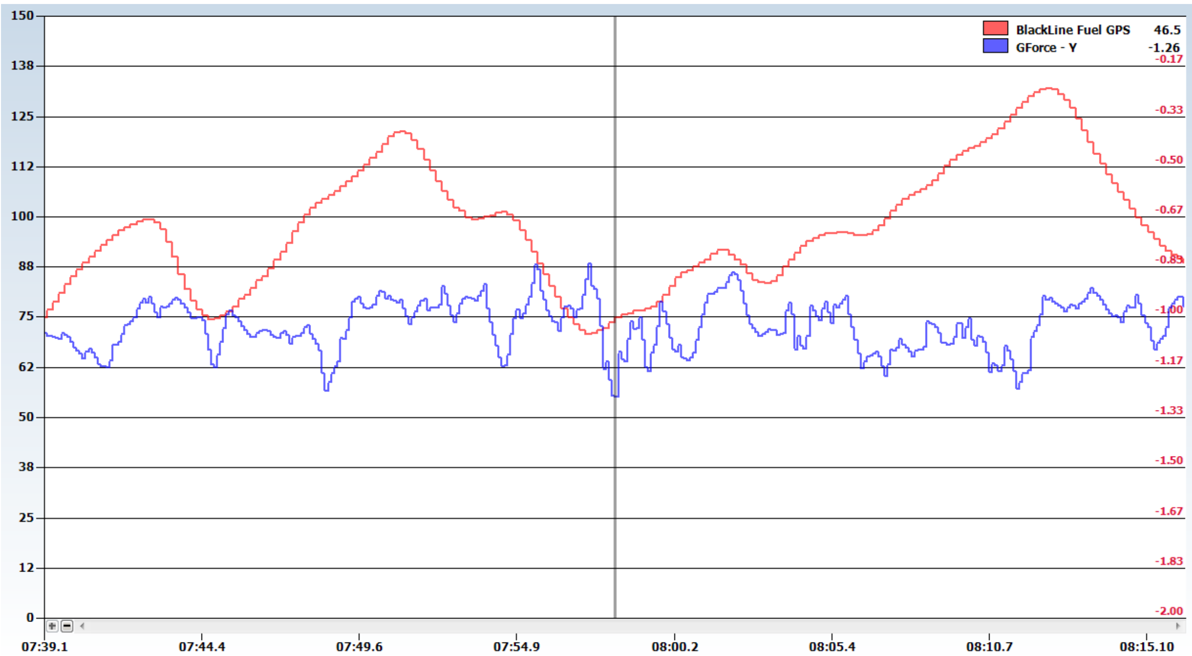

Fig. 7.8.6.3 Warwick Testing – Middle Rear Roll Centre – Max Negative Y Direction G Force

Z – acceleration (negative) and braking (positive)

Maximum negative g force of -0.71 at turn 10 apex-exit

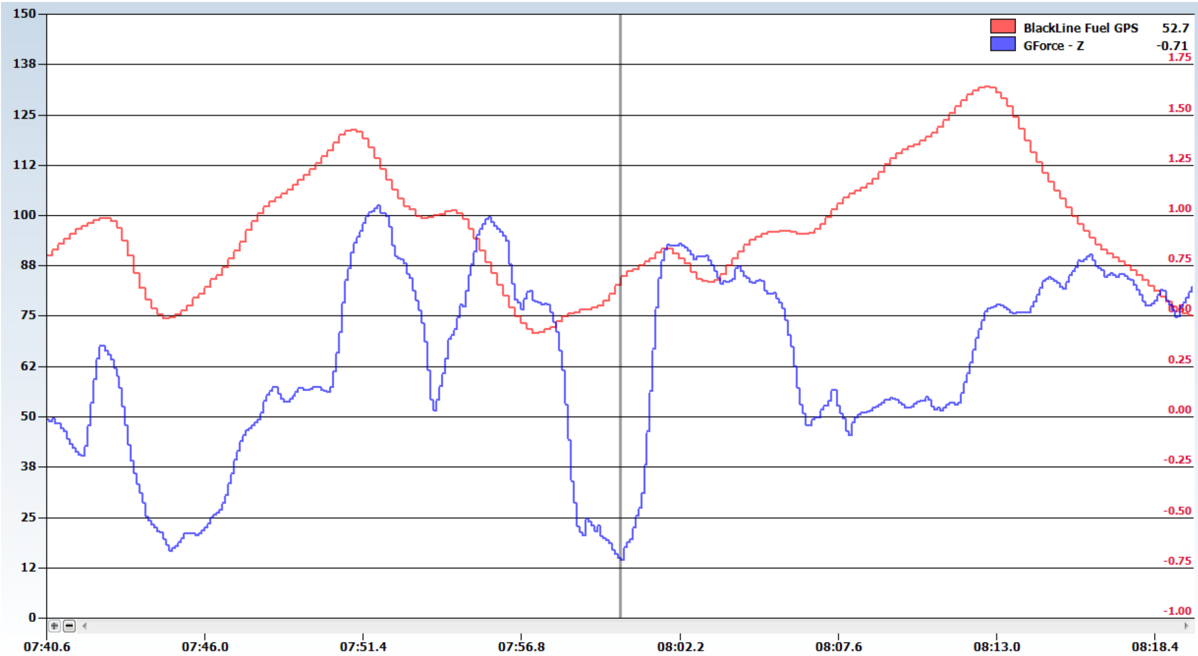

Fig. 7.8.6.4 Warwick Testing – Middle Rear Roll Centre – Max Negative Z Direction G Force

Maximum positive g force of 1.12 at turn 4 braking area

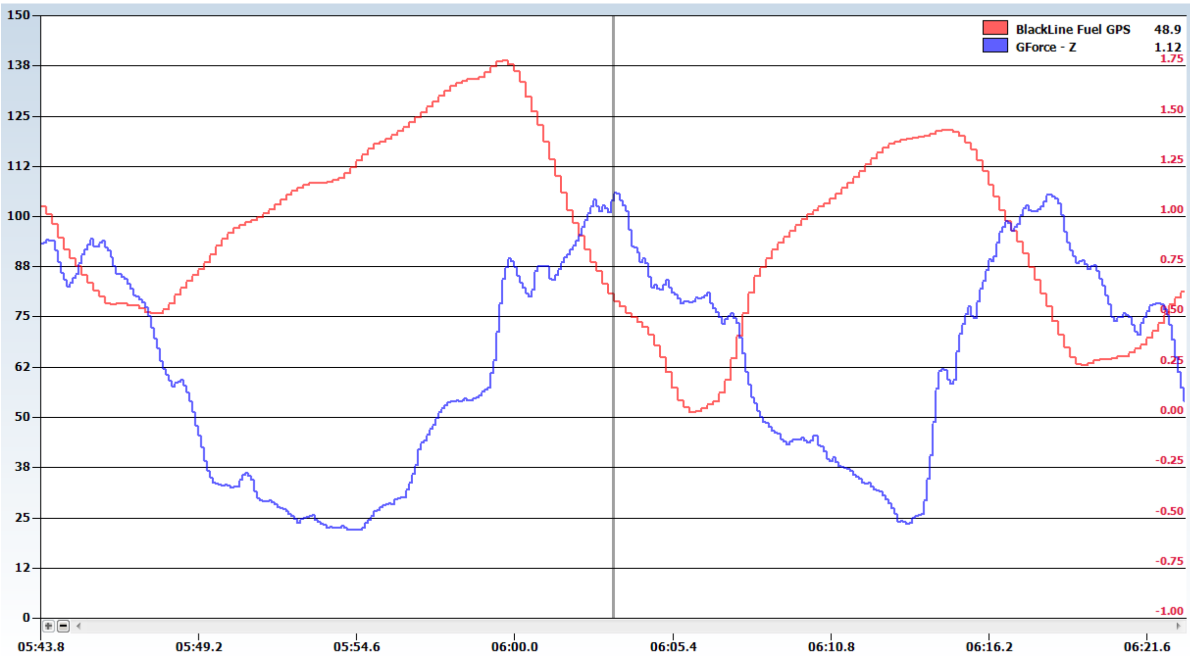

Fig. 7.8.6.5 Warwick Testing – Middle Rear Roll Centre – Max Positive Z Direction G Force

Driver Evaluation

The car handling was a lot more neutral than it had ever been. Turn in was good and mid corner loss of traction was fairly even with all four wheels breaking away at the same point in some instances. The tendency was towards very slight corner exit oversteer although this only achieved through the hard application of throttle. The car could be safely pushed hard, and the results show this with the fastest lap being achieved. Figure 7.8.6.1 shows the slight oversteer nature of the car on corner exit. The small bumps in cornering force refer to the car's gripping and releasing tendency on high throttle corner exit. The variance is only slight and presents little problems in overall handling.

7.8.7 Seventh Run – Exhaust Silencer Removed – 1.15.121 lap time

The exhaust silencer was then removed and a final run was made to ensure the end result was within the required handling parameters. The slight increase in power presents the car as it normally runs in sprint events.

Tyre Temperatures

Table 7.8.7.1 Warwick Testing – Final Sprint Setup – Tyre Temperatures

67.3	64.2	68.7		59.1	59.6	59.2
36psi	Pressure low	Camber less negative		36psi	Pressure high	Camber good
71.1	68.6	65.7		58.7	59.0	59.9
31psi	Pressure good			30psi	Pressure good	

The additional power has resulted in an increased amount of oversteer, as shown by the tyre temperatures. The front tyres have had to work slightly less to control the car while the rears have increased their input slightly. The oversteering nature of the car has been slightly increased by the removal of the silencer.

GPS Data Logger

X – left (negative) and right (positive), cornering force

Maximum negative g force of -1.04 at turn 5, flat corner

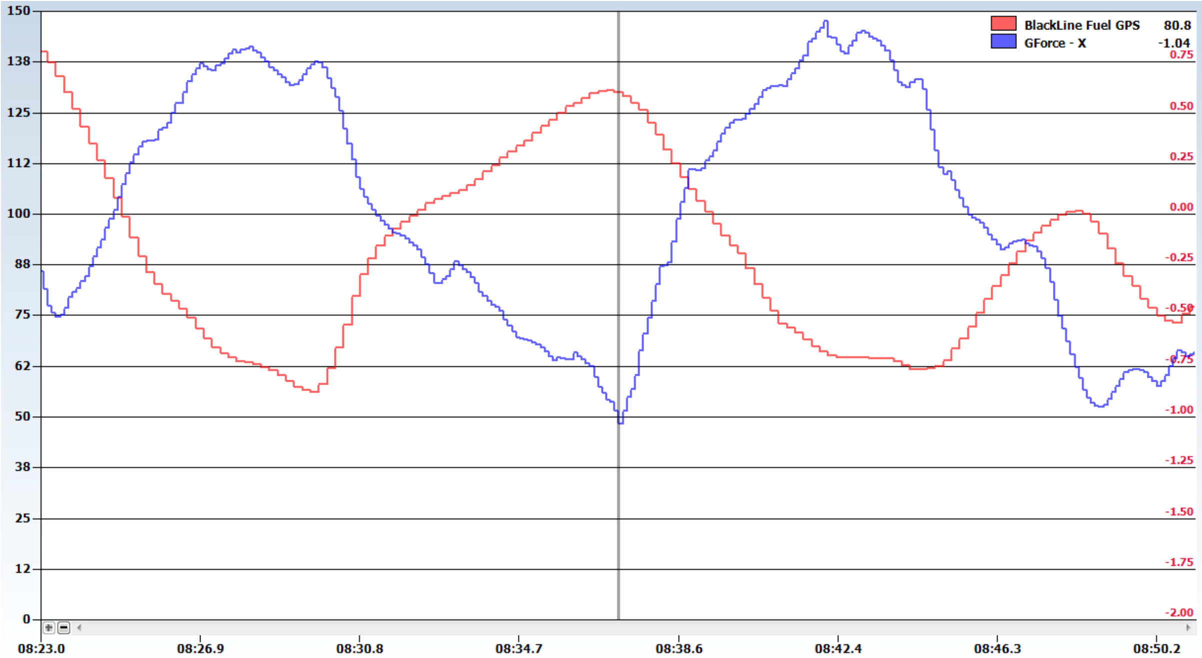

Fig 7.8.7.1 Warwick Testing – Final Sprint Setup – Max Negative X Direction G Force

Maximum positive g force of 0.95 at turn 2, slight uphill flattening out

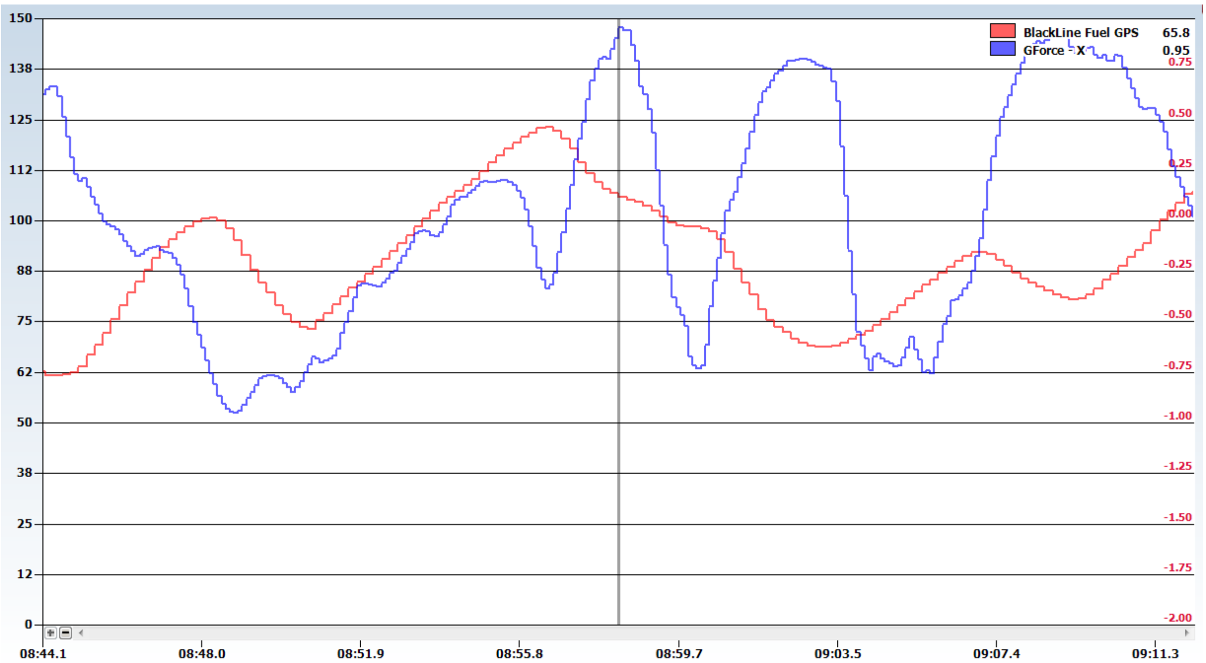

Fig 7.8.7.2 Warwick Testing – Final Sprint Setup – Max Positive X Direction G Force

Y – up and down (negative), gravity

Maximum negative g force of -1.32, turn 8 exit

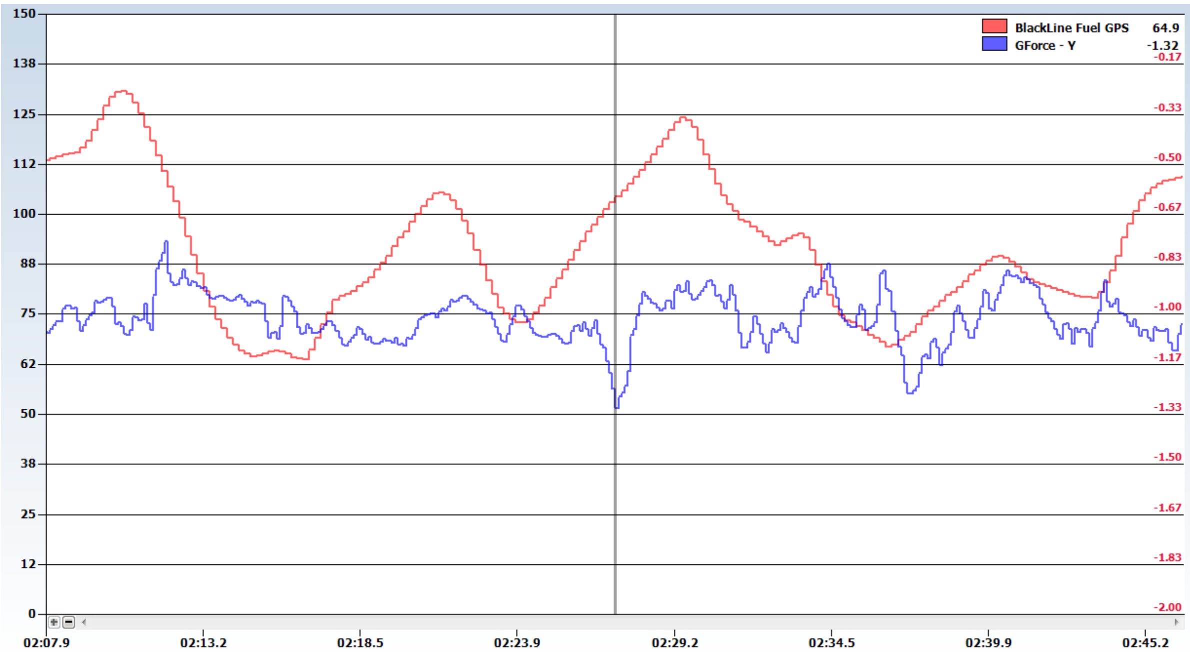

Fig 7.8.7.3 Warwick Testing – Final Sprint Setup – Max Negative Y Direction G Force

Z – acceleration (negative) and braking (positive)

Maximum negative g force of -0.71 at turn 7 apex, flat corner

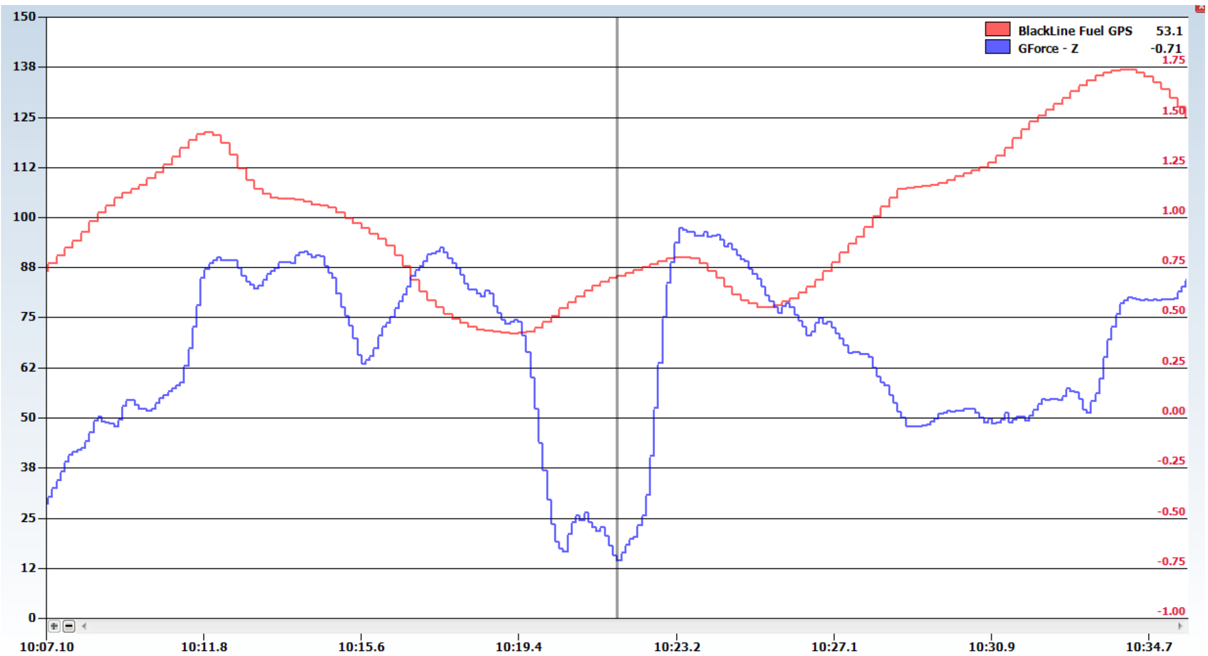

Fig 7.8.7.3 Warwick Testing – Final Sprint Setup – Max Negative Z Direction G Force

Maximum positive g force of 1.20 at turn 6 braking area, slight bumps, flat gradient

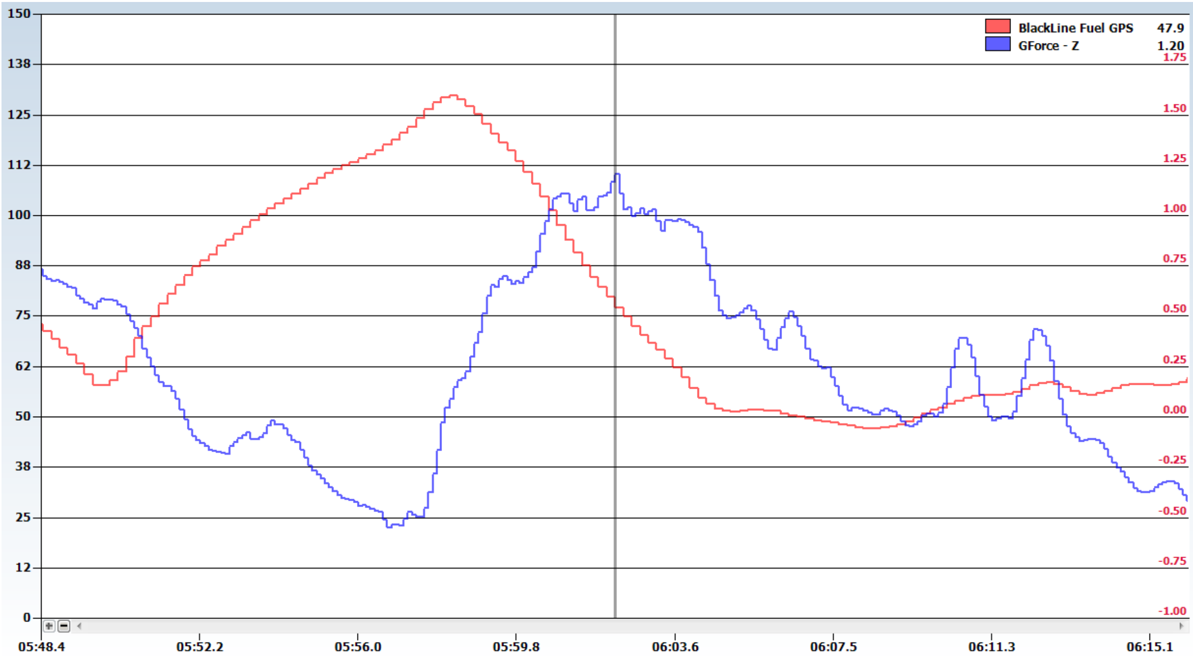

Fig 7.8.7.5 Warwick Testing – Final Sprint Setup – Max Positive Z Direction G Force

Driver Evaluation

The slight oversteer nature that was evident before the silencer was removed is still evident. The slight power increase has resulted in a slight increase in corner exit oversteer. The oversteer only increased slightly due to the nature of the silencer. The restriction it places in the system only dramatically affects the horsepower output, with the torque production being highly unaffected. As a result the cornering phase is highly unaffected by the silencers restriction. The handling was left with slight power oversteer in preparation for Stanthorpe, a track which normally results in understeer for most competitors.

7.9 Stanthorpe 23rd/24th October 2010 – After Modifications

Tyre Temperatures

Table 7.9.1 Stanthorpe Tyre Temperatures – 2st run Saturday

53.7	47.2	53.3		50.6	41.8	42.4
32psi	Pressure low	Camber good		31psi	Pressure low	Camber less negative
58.2	54.9	55.7		49.5	49.5	49.6
26.5psi	Pressure low			26psi	Pressure good	

Table 7.9.2 Stanthorpe Tyre Temperatures – 4th run Saturday

56.1	55.0	47.8		52.5	46.7	42.9
38psi	Pressure slightly high	Camber more negative		36psi	Pressure good	Camber less negative
61.5	59.5	56.0		54.2	56.4	56.0
29.5psi	Pressure good			29psi	Pressure high	

Changes made from this run are:-

Front pressures changed to 36psi

Rear pressures changed to 28psi

Rear roll centre lowered 30mm

Table 7.9.3 Stanthorpe Tyre Temperatures – 1st run Sunday

61.4	58.7	55.8		56.1	41.5	43.5
36psi	Pressure good	Camber more negative		34psi	Pressure low	Camber less negative
56.2	55.2	54.2		48.6	52.1	46.7
27psi	Pressure good			27psi	Pressure high	

Rear roll centre raised 15 mm

Table 7.9.4 Stanthorpe Tyre Temperatures – 2nd run Sunday

58.0	56.1	58.3		50.2	44.4	45.9
35psi	Pressure low	Camber good		34psi	Pressure low	Camber less negative
55.0	54.4	50.2		43.4	42.6	43.8
27psi	Pressure good			27psi	Pressure low	

Rear roll centre raised 10 mm

Table 7.9.5 Stanthorpe Tyre Temperatures – 3rd run Sunday

66.9	60.1	57.7		56.4	42.8	42.4
37psi	Pressure good	Camber more negative		36psi	Pressure good	Camber less negative
73.4	68.9	60.9		60.6	59.7	50.9
29psi	Pressure good			29psi	Pressure good	

GPS Data Logger

The GPS data logger was used for all runs. The data gained from the logger will be used to compare g force values with those achieved before the modifications were implemented.

X – left (negative) and right (positive), cornering force

Maximum negative g force of -1.14 at the initial turn of turn 2 – flat to uphill after apex

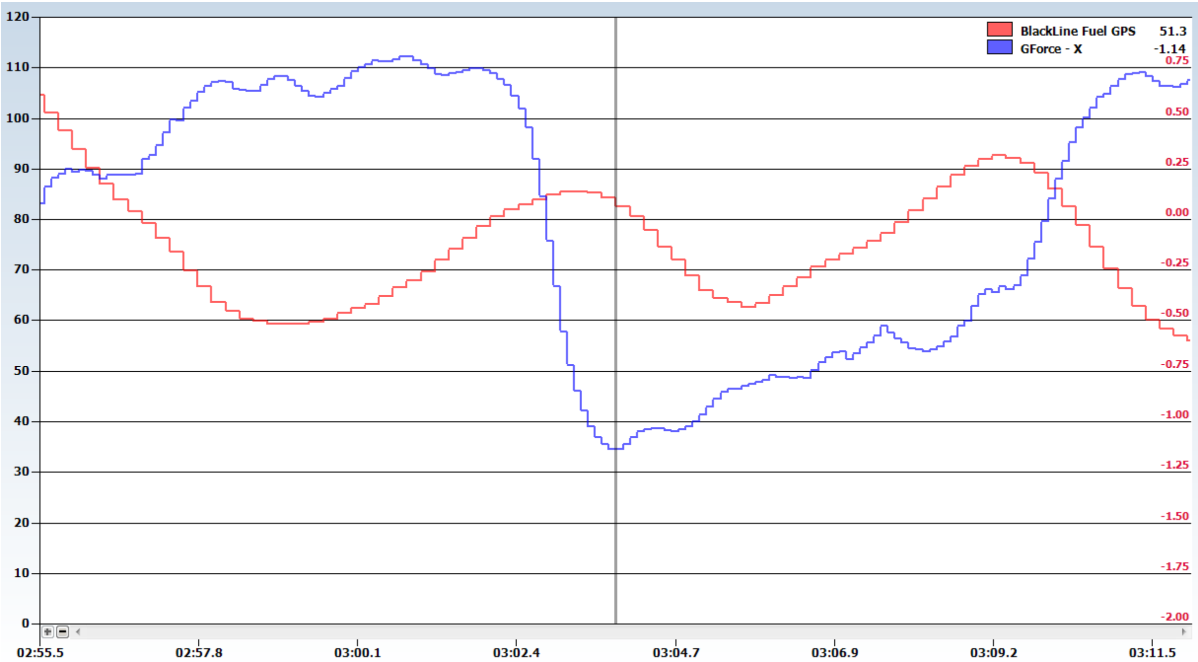

Fig. 7.9.1 Stanthorpe – Max Negative X Direction G Force

Maximum positive g force of 0.93 at turn 1 exit – slight camber

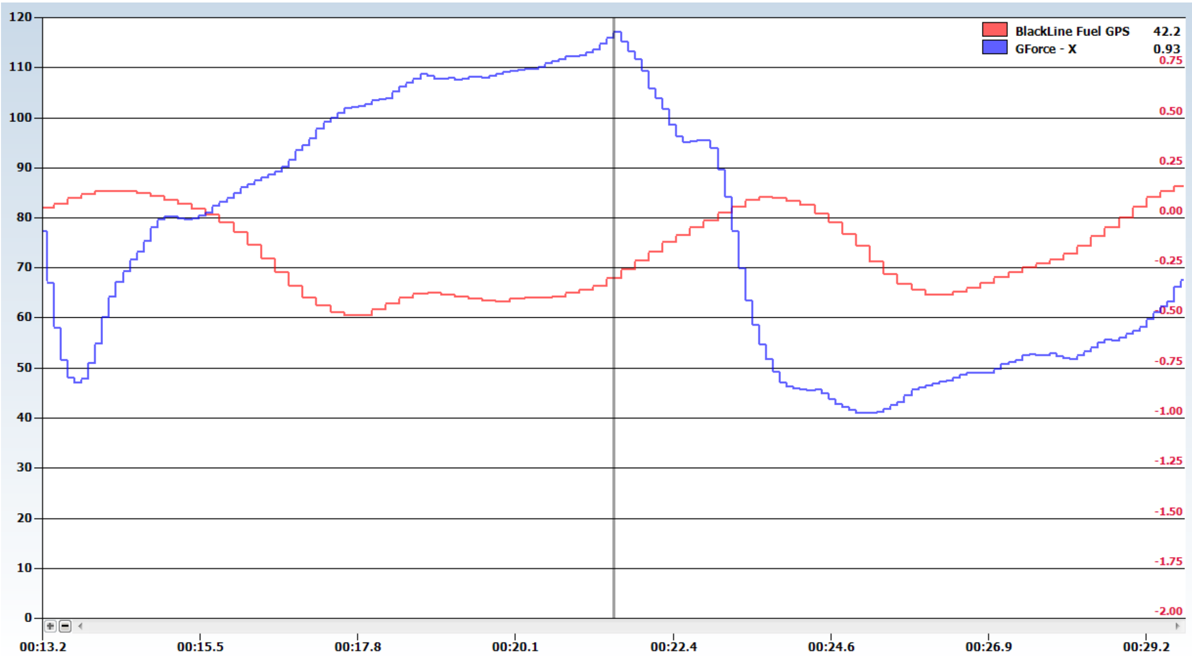

Fig. 7.9.2 Stanthorpe – Max Positive X Direction G Force

Y – up and down (negative), gravity

Maximum negative g force of -1.21 at straight between turns 3 and 4

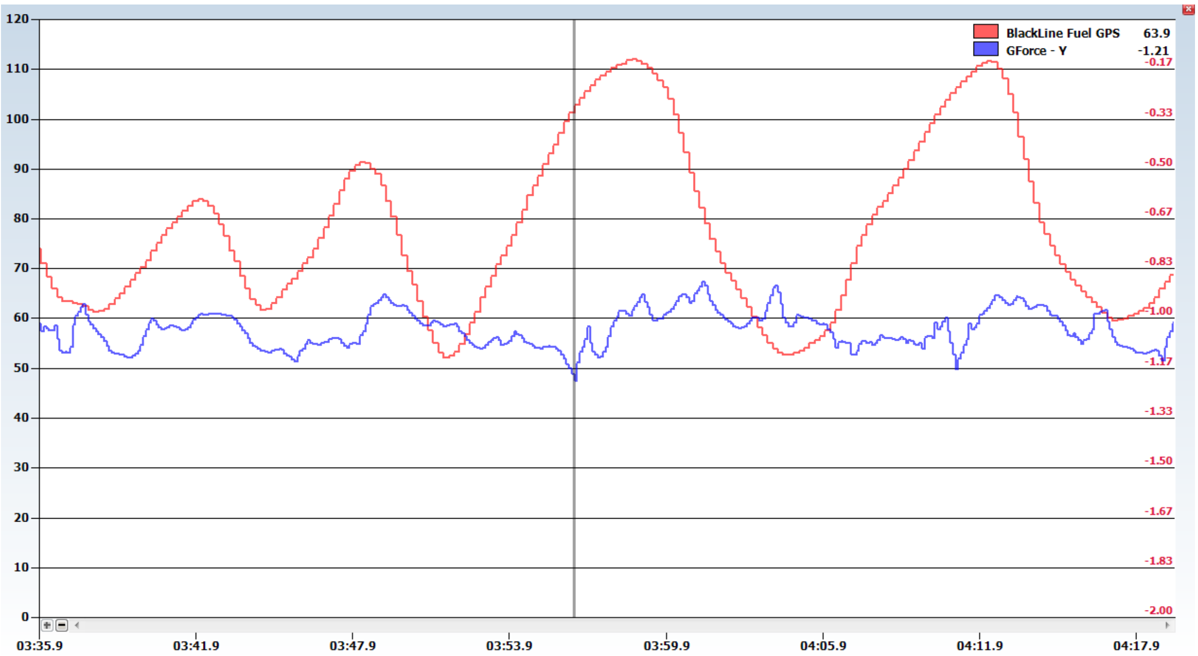

Fig. 7.9.3 Stanthorpe – Max Negative Y Direction G Force

Z – acceleration (negative) and braking (positive)

Maximum negative g force of -0.64 at turn 2 exit – uphill exit

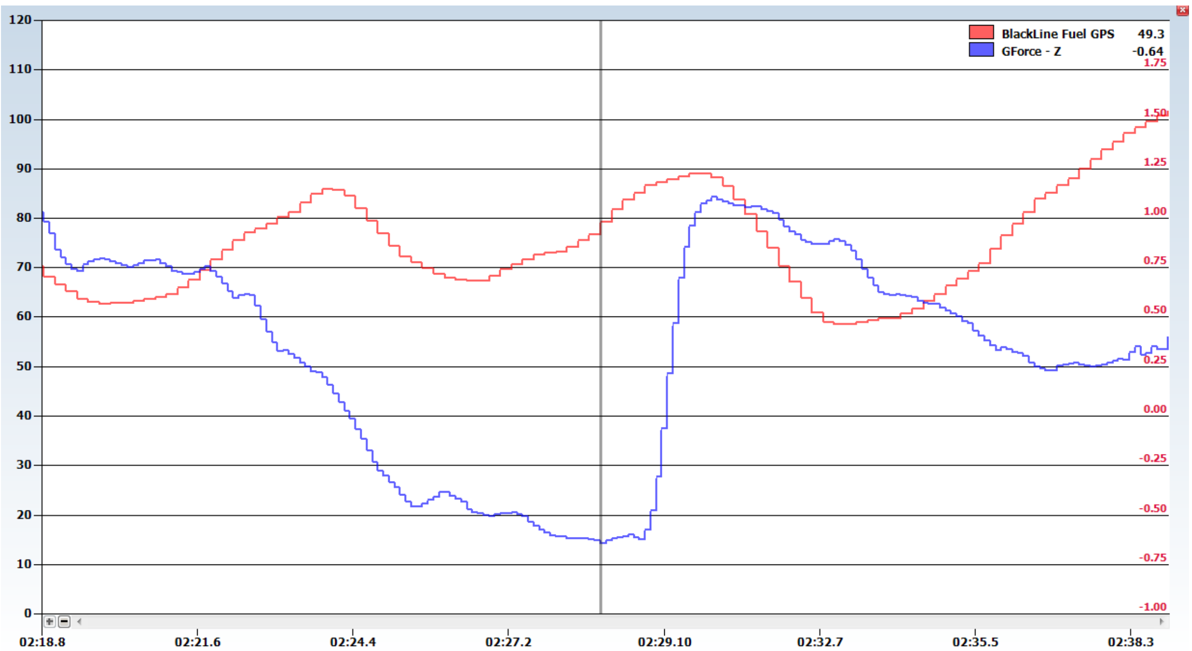

Fig. 7.9.4 Stanthorpe – Max Negative Z Direction G Force

Maximum positive g force of 1.15 at turn 1 entry – flat surface

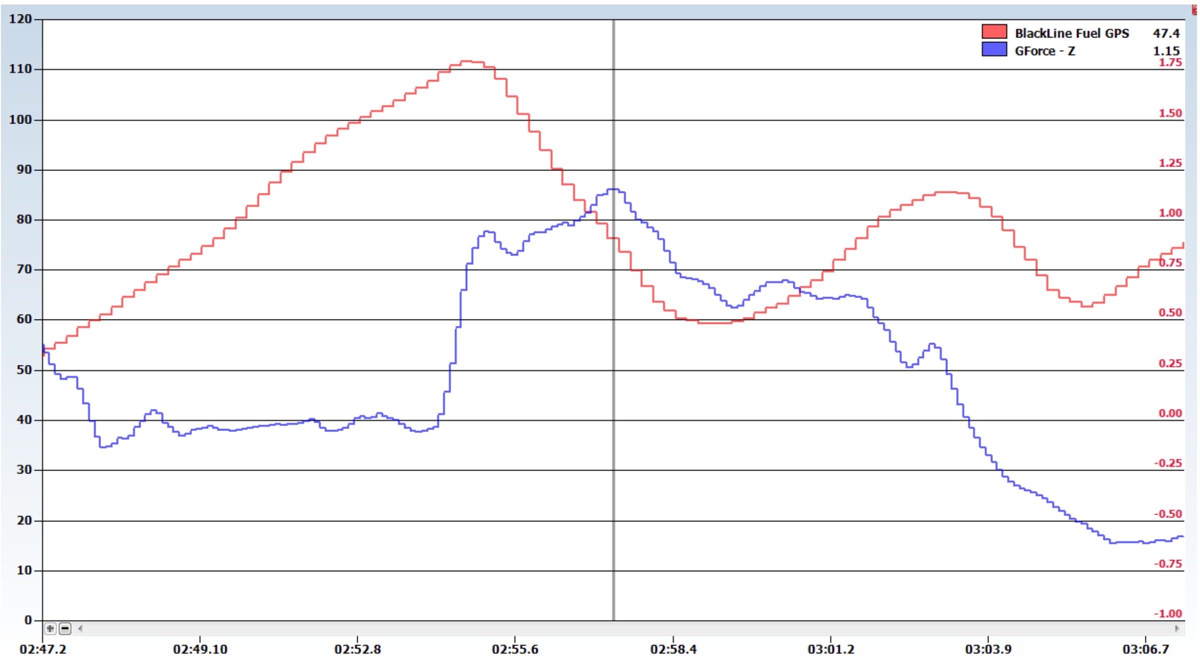

Fig. 7.9.5 Stanthorpe – Max Positive Z Direction G Force

Drivers Evaluation

The potential of the car at this track could not be fully reached. Driving and maintenance errors have affected the results, and limited the overall potential of the car. The rear brakes had worn considerably at the test day at Warwick, resulting in rear brakes which were out of adjustment. This decreased their effectiveness and resulted in premature front wheel locking. The result was a severely flat spotted front tyre which limited the progression of the lap times. The unbalanced car resulted in lap times which were not in fitting with the increasingly ideal setup. The fastest time was set earlier in the weekend before adjustments could be made, showing that good potential does exist given the correct parameters. A total time was only given, meaning that individual lap times were not achieved. The consistency of the driver is therefore more important, a factor which given the flat spotted tyre, had caused many problems at this event in particular. The cars balance was however being increasingly based towards a more neutral setting with each rear roll centre adjustment increasing the optimization of the setup.

Turn one turn in was always strong followed by the applicable setting of rear roll centre attitude which was predominately slight oversteer. Turn two was similar although the drive out of this corner felt much improved compared to the last event. Turn three resulted in very similar turn in, with a fraction more oversteer than most other corners. Turn four turn in felt strong and mid corner attitude depended greatly on the rear roll centre height. Full power could be used upon exit with no wheel spin once the roll centre was setup correctly.

The tyres have also had a fair workout, with them generally being worn and slightly harder. The test day at Warwick did run the tyres through a lot of heat cycles, a contributing factor in the performance life of the tyres. The tyres have well and truly passed the life that most competitors have considered to be ideal. The tyres will progressively increase lap times as tyre age contribution factors increase.

8. Results and Discussion

In accordance with the methodology set out in section three, and the data portrayed in section seven, the following results were achieved.

8.1 Tyre Temperatures

The difference between front and rear tyre temperatures has many uses in gauging the overall balance of the car. If one end of the car is heating the tyres more quickly than the other, that end is likely to wear tyres more quickly. The optimum setup contains all tyres being heated evenly, a process which in this case can only be altered by changing the suspension setup. Table 8.1.1 shows the difference between the front and rear tyre temperatures to show how the balance of the car has been altered.

Note: the values of front and rear temperatures are taken as an average of all the temperature readings taken from that run.

Table 8.1.1 Tyre Temperature Comparisons – Car Balance

Event/Run	Tyre Temperatures (Front then Rear) °C	Difference	Handling
Before Modifications			
Stanthorpe - July 2nd run Sat	30.483 36.133	5.65	Oversteer
Stanthorpe – July 3rd run Sat	39.633 43.117	3.484	Oversteer
Stanthorpe – July 4th run Sat	43.483 46.767	3.284	Oversteer
Stanthorpe – July 2st run Sun	49.3 44.1	5.2	Understeer
Stanthorpe – July 3rd run Sun	50.883 43.983	6.9	Understeer
Stanthorpe – July 4th run Sun	50.05 43	7.05	Understeer
After modifications			
Warwick testing baseline	48.7 54.05	5.35	Oversteer
Warwick testing No front anti-roll bar	53.633 54.2	0.567	Oversteer
Continued next page			

Warwick testing Rear roll steer reduction	55.433 56.067	0.637	Oversteer
Warwick testing Low rear roll centre	61.633 55.683	5.95	Understeer
Warwick testing Middle rear roll centre	63.083 63.65	0.567	Oversteer
Warwick testing Baffle out	63.017 63.833	0.816	Oversteer
Stanthorpe – October 2nd run Sat	48.167 52.9	4.733	Oversteer
Stanthorpe – October 4rd run Sat	50.167 57.267	7.1	Oversteer
Stanthorpe – October 1st run Sun - rear roll centre 30 mm lower	52.833 52.167	0.666	Understeer
Stanthorpe – October 2nd run Sun – rear roll centre raised 15 mm	52.15 48.233	3.917	Understeer
Stanthorpe – October 3rd run Sun – rear roll centre raised 10 mm	54.383 62.4	8.017	Oversteer

The tyre temperatures show how much difference the roll centre makes to the overall balance of the car. At Stanthorpe before the modifications were made there was no means of adjusting the balance of the car. It was found that the harder the car was pushed throughout the weekend the more it developed understeering tendencies. The increasing understeering values show that understeer was becoming a major problem that needed addressing with the suspension modifications.

The test day at Warwick resulted in many different situations due to the high amount of changes made throughout the day. The baseline contained a large percentage of oversteer, a similar situation to that found at Pittsworth and Mt Cotton. The removal of the front anti-roll bar did result in a more neutral handling car with only slight oversteer being seen through an average temperature difference of 0.567 °C. The rear roll steer reduction also had a similar affect with a reduction in the difference between average tyre temperatures now residing at 0.637 °C. The difference these variables make to the balance as measure by the tyre temperatures have shown that good potential exists for modifications in this area.

The rear roll centre location did make a remarkable difference to the balance of the car. The lowest setting results in extreme understeer as shown by a temperature difference of 5.95 °C. The middle location for the roll centre then brought the temperatures closer to optimum. The difference of 0.567 and 0.816 °C has shown that good progress has been made to increasing the balance of the car.

Stanthorpe presented a largely oversteering car. This shows that each track is unique and requires different settings. The issues associated with setting a race car up are highlighted by the large difference that was found between Stanthorpe and Warwick.

The initial temperature readings showing oversteer were rectified by the rear roll centre being lowered 30 mm. The result was a reduction of the oversteering temperature difference from 7.1 °C to an understeering temperature difference of 0.666 °C. The rear roll centre was then raised 15 mm and an increase in understeer was the result. The increase in understeer was found by the tyre temperatures, although the readings here had been altered by the reduced speed of the return lap to the pits, an issue which arose due to other cars on the track. The slower return speeds allowed the rear tyres to cool while the fronts were still being used to steer the car at the reduced speed.

The rear roll centre was raised and oversteer again returned. The extremely high difference of 8.017 °C was a result of a spin which occurred on the final lap. The rear tyres spun excessively in returning the car to the racing line, heating them to well above their normal operating temperature.

The tyre temperatures have shown that the rear roll centre requires different locations for different tracks. The neutral setting of the rear roll centre resulted in many test runs with each run fine tuning the variable. Warwick generally used a rear roll centre 10 mm higher than Stanthorpe to achieve a more neutral handling race car.

8.2 G force Comparisons

The g force data contained from the events is used to gauge the cornering potential of the race car. All values presented are maximums and this in itself presents a limitation. The overall average cornering g force would more closely resemble the cars potential to reach and maintain the tyres at their limits through all parts of the corner. The maximum g force achieved is also a good indication of the cornering potential given the slight averaging that the data contains.

Table 8.2.1 G Force Values – Before Modifications

Event/force	negative	Description/validity	positive	Description/validity
Gatton				
X	-1.26	dropping corner over camber change	0.81	flat corner
Y	-1.52	dropping corner over camber change		
Z	-0.70	dropping corner over camber change	0.89	approach and heavy trail braking into the first chichane. Flat surface
Stanthorpe July				
X	-1.07	flat to uphill after apex	0.96	flat corner on top of hill
Y	-1.32	as begins to climb uphill		
Z	-0.63	uphill exit	1.14	flat surface
Noosa				
X	-1.14	cambered corner	0.95	bottom of hill
Y	-1.48	across flip flop		
Z	-0.76	cambered corner, rolls over for exit	1.16	flat surface
Warwick				
X	-0.95	bottom of hill	0.99	flat surface
Y	-1.32	bump on straight		
Z	-0.64	flat surface	1.22	down hill
Mt Cotton August				
X	-1.28 (-1.10) = -1.19	flat corner	0.86 (0.95) =0.905	rolling to off camber
Y	-1.54 (-1.46) =-1.5	drop between corners		
Z	-0.76 (-0.73) =-0.745	between turn 1 and 2	1.22 (1.22) =1.22	turn 3 entry

Table 8.2.2 Average G force Comparison Values – All Events – Before Modifications

Force	negative	positive
X	-1.122	0.923
Y	-1.428	
Z	-0.695	1.126
Cornering average = 1.0225		

The modifications were made and events were attended to find the potential cornering power increase. The cars handling could then be tuned to alter the balance of the vehicle.

Table 8.2.3 G Force Values – After Modifications

Event/force	negative	Description/validity	positive	Description/validity
Mt Cotton October		WET EVENT		
X	-1.11	Slightly cambered	0.77	Cambered – uphill exit
Y	-1.39	Drops before corner		
Z	-0.69	Cambered corner, flattens out	1.19	Flat
Warwick Test day (best)				
X	-1.04	Flat corner	0.95	Slight uphill
Y	-1.32	Drop between corners		
Z	-0.71	Flat corner	1.20	Slight bumps, flat gradient
Stanthorpe October		Flat spotted front tyre		
X	-1.14	Flat to uphill after apex	0.93	Slight camber
Y	-1.21	Straight between turns 3 and 4		
Z	-0.64	Uphill exit	1.15	Flat surface

Table 8.2.4 Average G force Comparison Values – All Event - After Modifications

Force	negative	Positive
X	-1.097	0.883
Y	-1.307	
Z	-0.68	1.18
Cornering average = 0.99		

The average g force comparisons show that cornering potential has increased slightly. Some of the values presented in the tables do not support this; however upon further inspection the reason for such discrepancies is apparent. The high cornering forces shown in table

8.2.2 include all events that were attended before the modifications were made. The inclusion of superior data from tracks such as Noosa and Mt Cotton has resulted in a high average cornering g force of 1.0225. The measurements taken after the modifications are limited in that Gatton and Noosa were not attended. Mt Cotton was included in the post modifications cornering force of 0.99, although its values are reduced due to the wet track in which racing was undertaken.

The decision was then made to compare only the values achieved at Stanthorpe and Warwick in an attempt to remove variables that may alter the outcome unnecessarily.

Table 8.2.5 Average G force Comparison Values – Stanthorpe and Warwick - Before Modifications

Force	Negative	Positive
X	-1.01	0.975
Y	-1.32	
Z	-0.635	1.18
Cornering average = 0.9925		

Table 8.2.6 Average G Force Comparison Values – Stanthorpe and Warwick – After Modifications

Force	Negative	Positive
X	-1.09	0.94
Y	-1.265	
Z	-0.675	1.175
Cornering average = 1.015		

The cornering potential of the car has increased from 0.9925 to 1.015 g force. The slight increase has been as a result of the increased balance that the car now possesses. 0.0225 g force is a small margin, although any increase in cornering force is beneficial in reducing lap times. The lower Y force variance has shown that the suspension can now follow the road more closely, a factor which is mainly attributed to the rear bump stop modification. The available tractive force (negative Z force) has increased considerably. A gain of 0.04 g force is quite large, especially considering the power and torque output has remained unchanged. The benefit in exiting corners is likely to result in faster lap times as faster speeds will be reached between corners. The drive out of corners is of major concern and the increase achieved in this section is highly desired. The braking force (positive Z direction) has decreased slightly. The difference of 0.005 is negligible, especially given the flat spotted tyre at Stanthorpe after the test day at Warwick. The braking force is therefore highly unaffected by the modifications performed.

8.3 Drivers Evaluation

The overall balance of the car has been changed considerably. All events which were attended before the modifications resulted in understeer in all cases. Oversteer was very rarely discovered and even under full power exits, oversteer was seldom the result. The modifications greatly altered the balance of the car.

The turn in has perhaps been slightly reduced. With the front roll centre being higher, the front is slower to turn in although the smaller moment arm has meant that less weight transfer is taking place. The mid corner understeer has been greatly reduced at all tracks and slight oversteer has been the generally tendency with the modifications. The corner exit understeer has been eliminated. The oversteer upon corner exit can effectively be controlled by the throttle application and faster corner exits have been the result.

The car is much more balanced due to the reduction of understeer. The car can now be driven to the limits of all four tyres and a more even breakaway occurs in terms of front to rear balance. The driver can now steer the car with the throttle, a situation which was never possible before. The modifications have improved the driver's perception of the cars handling.

9. Conclusions

The modifications have successful altered the cars handling characteristics. The required parameters have been altered in order to reduce the understeering nature of the race car. The cornering force has been increased and lap time reductions have occurred without modifying the engine or braking systems. These parameters have however presented a new situation from which suspension development should be continued from.

The modifications which were performed due to the literature review have resulted in the required handling changes. The results of the changes were as expected and the design and manufacture of the components has been sufficient to remain in a serviceable condition. The modifications have currently been implemented for three events and have performed faultlessly since their fitment.

The physical testing of the race car has shown the required handling variables have been altered. The roll centre location responded in the largest difference to car balance and this was highlighted through tyre temperatures. The rear roll centre height has been optimized for the different tracks with the current suspension setup. The tyre temperature difference between the front and rear has decreased with the optimization of the roll centre. A more neutral handling car has been the result of the increased front and decreased rear roll centre heights. The jacking forces have also been found to make the most difference to cornering potential when the roll centre is located above the ground. The lowering of the rear roll centre and subsequent jacking forces has resulted in a more balanced car by promoting rear grip and results in increasing amounts of understeer to tune to chassis.

The cornering force has also increased slightly as a result of the decreased weight transfer at the front and deceased jacking forces in the rear suspension. The net gain of 0.0225 g force is achieved due to the improved management of the roll centre and subsequent weight transfer and jacking forces. The acceleration force has also been increased by 0.04 g force, a direct result of the rear suspensions ability to better follow the roads profile. The car now possesses an increased potential to use the tyres to develop useful tractive forces. The forces achieved by the dynamic race car have improved and should result in further lap time reductions at the remaining events.

The driver's evaluation of the modifications has supported the data gained through testing and the general outlook has been positive. The driver can now alter the cars balance to support their own driving style, an attribute which was never able to be attained at the commencement of the project. The overall increase in cornering power and driver control has resulted in reduced lap times and a more competitive race car as shown in Appendix L.

The altered suspension parameters have resulted in a new situation from which the development process will be based. The overall traction capacity of the race car can be further improved through continued development and further investigation is required.

10. Further Work

Suspension development is never completed. The optimum setup will only be achieved through further testing and modifications. The results of the modifications have generally been positive, although more changes are required to gain the full potential of the modifications. The following is a brief list of variables that may require modifications to further develop the cars handling package.

An increased castor value would have benefits in increasing the front end communication to the driver. In order to increase this value the lower control arms would require spherical bearings to allow an increased misalignment and shorter castor arms to provide the necessary changes. The top strut mount could also be moved towards the rear of the car more, a modification which would result in the adaptation of the camber tops to allow castor changes also. The dynamic camber values are also affected by the roll centre location and their optimization should be prioritised to ensure efficient tyre contact is maintained with the new roll angle.

The bump steer and Ackerman characteristics have been greatly altered and their effects are now reduced to result in more consistent handling. The steering does however require a faster ratio and a modified steering arm setup may result in additional clearance between the steering arm bolt and front wheels to allow further bump steer reduction.

The testing showed that the rear roll steer reduction was favourable to the cars balance and lap times. The reduction of roll steer also resulted in the raising of the rear suspension and resultantly transferred more weight to the front. The application of modified differential mounts, or lowering blocks between the differential and the springs, will lower the car and allow the rear roll steer to be reduced. The application of larger spacers is therefore recommended to reduce the roll steer while maintaining the current ride height.

The spring rates have been found to be too stiff for the application. This will result in tyre compliance issues and the correct setting of softer spring rates is likely to result in faster lap times and an increase in potential cornering force through higher tyre contact. Stiffer dampers should be used, especially in the front to promote increased initial turn in. The rear springs and dampers also require optimization to increase their effectiveness in the current application.

Anti-roll bar rates require modifying now that the roll centres have been altered. The roll angle of the front has been reduced and resultantly the front anti-roll bar stiffness can be reduced to achieve the same roll angle. The reduction of the weight transfer through the anti-roll bar may return some of the mid corner turn that the car has felt as though it has lost through the fitting of the front roll centre adjusters.

The rear roll angle has greatly increased with the reduction of the rear roll centre height. The rear roll angle has now become noticeable, although the solid rear axle will be maintained parallel to the ground throughout the cornering process. The rear suspension has been left without an anti-roll bar as no camber change will result from the increased roll angle.

The tyres on the car are due for replacement and this represents another variable which will again alter the handling characteristics of the car. New tyres will increase the overall grip level and different roll angles will be achieved. The suspension parameters will again require modifications to ensure the optimum value of roll angle is achieved in all circumstances.

The overall balance of the car has been improved and tuning of the characteristics can now be performed easily and quickly. The other factors which are affected by the changes will now require attention to ensure continual development of the car is achieved.

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Appendices

Appendix A - Project Specifications

University Of Southern Queensland

Faculty of Engineering and Surveying

ENG4111/4112 Research Project

Project Specifications

FOR: Guy Nawratzki

TOPIC: SUSPENSION DEVELOPMENT FOR A SHORT CIRCUIT RACING CAR

SUPERVISOR: Chris Snook

ENROLMENT: ENG 4111 - S1, 2010; ENG4112 – S2, 2010

PROJECT AIM: This project seeks to decrease the current lap time of the racing car. This is to be achieved through suspension modifications and tuning to increase corner speed and driver control.

PROGRAMME: Issue A, 18th March 2010

1. Literature review of the conditions and needs for suspension systems in short circuit race cars.
2. Develop and validate the use of appropriate data acquisition systems.
3. Analyse the current suspension system to find methods of increasing cornering power.
4. Determine baseline parameters in car operations.
5. Analyse existing data to find areas in which to improve the handling.
6. Determine the specific modifications necessary to improve handling.
7. Further test the race car with the modifications and adjust accordingly.

As time and resources permit:

1. Make more suspension modifications and continue the development cycle.
2. Change the handling requirements by increasing engine power and trying to compete with more highly powered front runners.

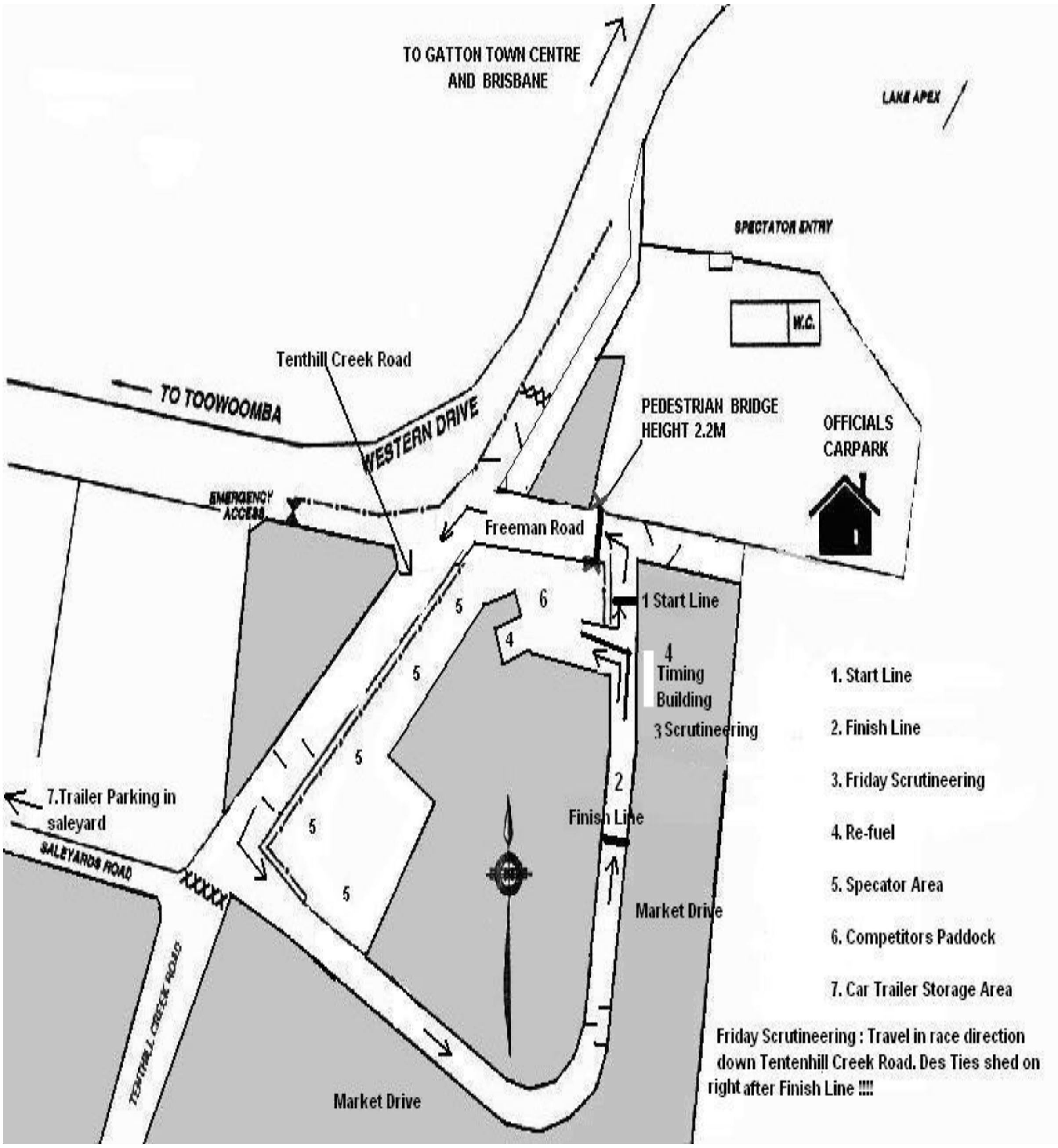
AGREED:

__________________(student) ______________________(supervisor)

Date: / / 2010 Date: / / 2010

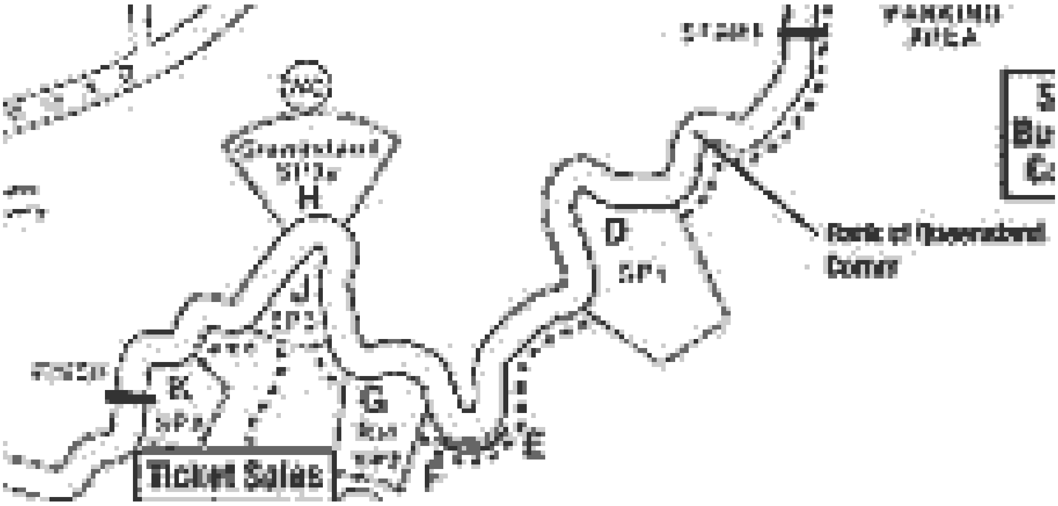
Examiner/Co-examiner:____

Appendix B - Gatton Track Layout



Appendix C - Stanthorpe Track Layout

Appendix D - Noosa Hill Climb Track Layout



Appendix E - Warwick Track Layout – 1200m

Appendix F - Warwick Track Layout – 2100m

Appendix G - Mt Cotton Hill Climb Track Layout

Appendix H - Pittsworth Track Layout

Appendix I - Suspension Modifications before Project Start

1st event Pittsworth September 2007

1st event with the new engine, gearbox and diff.

Front suspension-

Stanza struts with standard springs of 90 lbs/in spring rate

Standard stanza oil filled shocks

No front sway bar due to engine sump clearance issues.

Standard stanza brakes, 250mm non vented, single piston callipers

Unknown Alignment, toe set to zero using a homemade trammel bar.

Rear suspension-

Standard springs. 3 leaves. 2 small lowering blocks of 5mm height each.

Standard Hilux drum brakes

Tyres-

Front- Globe 14x6 alloy wheels, 195/65/14 madison street tyres, new

Rear- Standard 15x6 steel wheels, 205/50/15 madison street tyres, nearly worn out, mainly due to camber on previous car.

Lap Time – 51.61 sec

Observations-

Axle tramp fairly bad off the line

Way too much roll in the front, also lifts and dives way too much.

Rear end seems fairly stiff, rolling nowhere near as much through the turns

Understeer a major issue due to rolling and generally feeling bad through turns

2nd event Gatton March 2008

Front Suspension-

Springs from the rear of an independent rear suspension Datsun 180B, 550 lbs/in spring rate

Pedders gas sports riders shock absorbers, cut and rewelded the strut housing 30mm shorter to fit short insert. Increased the amount of travel before full bump. Shocks for stanza.

Lengthened lower control arms by 30mm and moved ball joint 5mm forward to increase positive castor.

Fitted larger stronger LJ torana radius rods to give adjustable castor

Modified and fitted a large sway bar from a VL commodore, 26 mm in diameter. To fit, the bar had to be cut, shortened, reshaped and rewelded. A collar was then placed over the welded area.

Custom made and fitted a strut bar, to tie the two front top suspension mounts together.

Lowered a considerable amount over previous race

Wheel alignment –

Castor - +0.80 degrees

Camber- Left -3.80 degrees, Right -2.10 degrees. The error here cannot be fixed as don't have adjustable suspension parts yet.

Toe - -2.0 mm total toe, -1.0 degree per side. To provide increased turn it.

Rear Suspension-

Standard springs. 3 leaves. 2 small lowering blocks of 5mm height each.

Fitted caltracs traction bars. Setup in the middle hole on the differential mount, middle hole on the pivot plate, and the furthest hole on the spring connection bolt. Preload was setup that it was only just touching the spring with no weight on the axle, full droop

Tyres-

Front – Nissan Bluebird TRX alloy wheels 15x6, 185/55/15 new street tyres, last minute rims and tyres that we had for another car, were required due to clearance issues with the outer guard

Rear- Standard 15x6 steel wheels, 205/50/15 madison street tyres, nearly worn out, mainly due to camber on previous car.

Lap time- 43.718 sec , 107th out of 134

Observations- Handling was a lot more neutral, although being the first outing with the new setup the car was not pushed to get good times. The lack of front roll stiffness was fixed and the front was feeling quite stiff and direct. The rear had gained a considerable amount of grip and was handling well, although required lowering due to feeling light through turns and looking too high. Slight understeer was felt in the front although nothing overly concerning as not being pushed yet.

Stock motor 44.96 sec. Lap – 3:05.11 heat (4 laps)

3rd event Stanthorpe May 2008

Front suspension – unchanged

Rear Suspension - unchanged

Tyres-

Front – unchanged

Rear- unchanged

Lap time- 40.20 sec. Lap – 2:46.22 heat (4 laps)

Observations-

First thing that was noticed is extreme understeer through the long turns. Turn in is good, but mid to late corner understeer is quite severe. The rear followed extremely well and had good grip on exit also.

4rd event Pittsworth September 2008

Front suspension – unchanged

Rear Suspension – unchanged

Tyres -

Front – unchanged

Rear - unchanged

Lap time - 48.47 sec

Observations-

Handled well. Although still not pushing hard due to lack of driver confidence in car. Slight understeer through tight turns such as last turn after chicane. Rear grips and follows well.

5th event Stanthorpe October 2008

Front suspension– unchanged

Rear suspension – Standard springs. 3 leaves, middle spring reversed. 2 small lowering blocks of 5mm height each.

Caltracs traction bars. Same setup and preload

Tyres-

Front – unchanged

Rear- unchanged

Lap time- 40.02 sec lap time – 2:44.06 sec heat (4 laps) – 59th out of 84 for fastest heat

Observations-

Same as last time. Good turn in but understeer at mid and corner exit. Found increasing rear pressure and decreasing front increased turn. Front pressure was found to reach a point in which maximum grip was obtained, and any lower pressure resulted in too much tyre squirm. Also driving style needed to be altered to try and slide the rear to turn out of the corner, use the throttle to steer the car. The effectiveness of this was limited however due to high level of rear grip.

6th event Gatton March 2009

Front suspension – unchanged

Rear suspension – unchanged

Tyres-

Front – unchanged

Rear - Nissan R32 Skyline alloy wheels 16x6.5, 195/50/16 Toyo Proxes R888 semi-slicks

Lap time- 40.408 sec. Average of 41.209 = 4th in class out of 9. 63rd out of 136 going off best lap time

Observations-

Handled excellent. Nowhere near as much understeer was present like before. Front turned in well and even mid corner and exit were controlled. Rear grip was excellent. Only slight spin off the line, and no loss of traction during acceleration out of corners. Handled great through the chicanes and found I could really push it now, and resultantly the lap time represents this.

7th event Stanthorpe May 2009

Front suspension – unchanged

Rear suspension – unchanged

Tyres-

Front – unchanged

Rear- unchanged

Lap time- 39.46 sec lap time – 2:42.41 heat time (4 laps)

Observations-

First thing that was noticed is extreme understeer. Turn in is good with mid to exit understeer. The rear grip is excellent with it always following into the corner and driving well out of the corner. This may have been too well and caused the understeer. The problem was found to be alleviated a little by trail braking further into the corner and then powering out. This was found after trying to slide the rear around the corner, which didn't really work due to so much rear grip. The rear pressures where raised to help the situation, which did slightly improve handling allowing the rear to slide a little more.

8th event Pittsworth September 2009

Front suspension – unchanged

Rear suspension – unchanged

Tyres-

Front – unchanged

Rear - unchanged

Lap time - 47.69 sec

Observations-

Much the same as at Gatton earlier that year. Gripped well off the line, and through corners. Understeer was more of an issue here than at Gatton with mainly the 3rd corner resulting in slight understeer and the last corner often resulting in more severe understeer, due to being so close after the chicane.

9th **event Oakey September 2009**

Front suspension – unchanged

Rear suspension – unchanged

Tyres-

Front – unchanged

Rear- unchanged

Lap time- 47.413 sec. Average lap time of 49.382. 4th in class off average out of 9. 37th out of 127 on average lap time.

Observations-

Much the same as at Gatton earlier that year. Gripped well off the line, and through corners. Understeer was slightly more of an issue here than at Gatton with mainly the 3rd corner resulting in slight understeer and the last corner often resulting in more severe understeer, mainly due to being over a crest and dropping off to an off camber exit. Chicane handling is good although the front does slide if pushed hard.

10th event Stanthorpe October 2009

Front suspension – unchanged

Rear suspension – unchanged

Tyres-

Front – unchanged

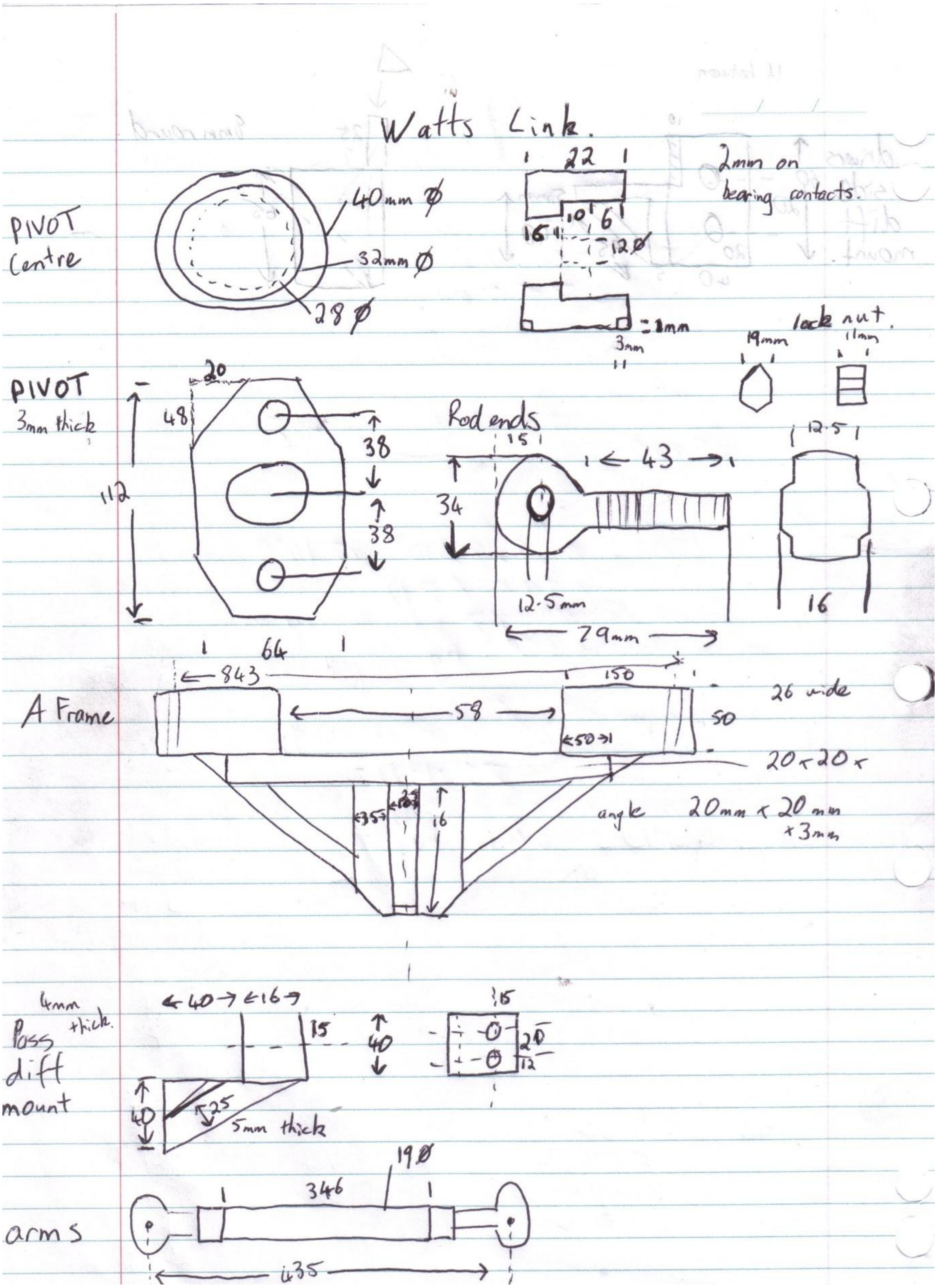
Rear - unchanged

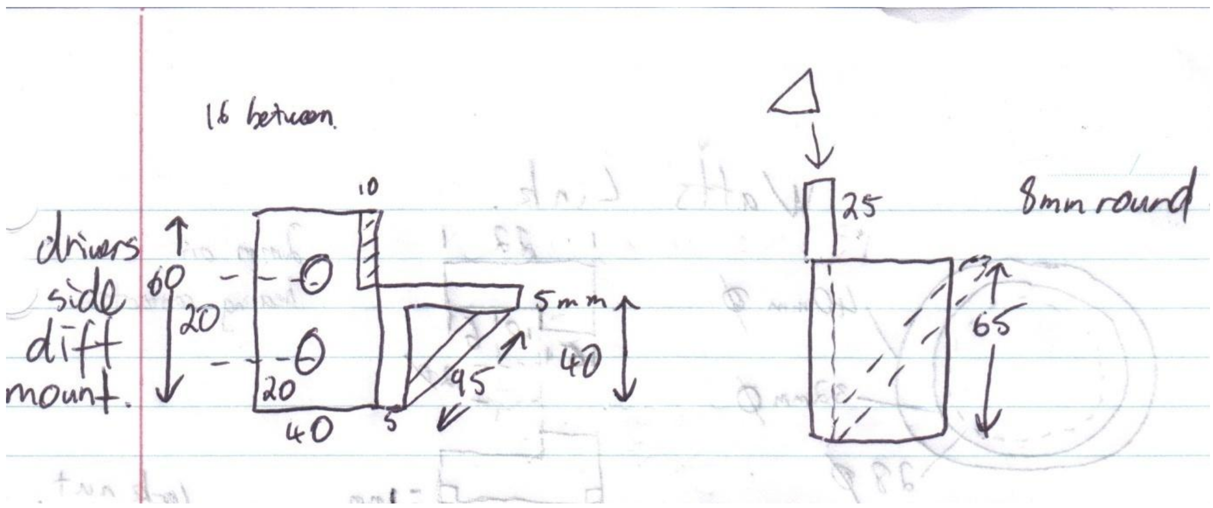
Lap time- 39.35 sec lap time – 2:41.50 heat time (4 laps) – 51st out of 92

Observations-

First thing that was noticed is extreme understeer. Turn in is good with mid to exit understeer. The rear grip is excellent with it always following into the corner and driving well out of the corner. This may have been too well and caused the understeer. The problem was found to be alleviated a little by trail braking further into the corner and then powering out. This was found after trying to slide the rear around the corner, which didn't really work due to so much rear grip. The rear pressures where raised to help the situation, which did slightly improve handling allowing the rear to slide a little more.

Appendix J - Watts Link Measurements and Design Parameters




16 between
drivers side diff mount.
10
60
20
20
40
5
5mm
95
40
25
8mm round
3
65

Appendix K - Win Geo 3 Measurement

Before Roll Centre Adjusters Fitted

Design Panel:

Kingpin axis 15.362 degrees

Scrub radius 30.703

Caster 3.772 degrees

Caster trail 12.791

Upper A-arm 0.000

Lower A-arm 322.724

Suspension

Lower A-arm forward A -266.000 277.000 194.000 A to B= 409.240

Lower ball joint B 0.000 588.000 194.000 C to B= 323.342

Lower A-arm rearward C 5.000 265.000 180.000 A to C= 271.627

Upper A-arm forward D 36.000 438.000 740.000 D to E= 0.000

MacPherson strut upper E 36.000 438.000 740.000 F to E= 0.000

Upper A-arm rearward F 36.000 438.000 740.000 D to F= 0.000

Wheelbase, track, tire diameter2355.0001344.000 590.000 Rollout1853.540

Camber, Toesteer, Toe span -4.500 -0.121 711.201 HubTrak 648.855

Tire contact patch 0.000 672.000 0.000 B to E= 567.373

Steering

Steering tie-rod on hub S 128.000 574.000 180.000 S to T= 344.372

Steering tie-rod inboard T 112.000 230.000 180.000 B to S= 129.522

Idler arm upper-axis P 0.000 0.000 0.000

Idler arm lower-axis Q 0.000 0.000 0.000

Drag-link to Idler arm X 0.000 0.000 0.000

Drag-link to Idler arm low Y 0.000 0.000 0.000

Drag-link tie-rod attach Z 0.000 0.000 0.000

Toe-inch span, Steering Box 711.201 50.000not used mm / 360 degrees

Clearance

Chassis clearance point 1 L -90.000 90.000 120.000

Chassis clearance point 2 0.000 0.000 0.000

Chassis clearance point 3 0.000 0.000 0.000

Chassis clearance point 4 0.000 0.000 0.000

Center of Gravity 785.000 20.000 550.000

Sketch

Rim width, wheel offset, spacer 210.000 40.000 120.000

Aspect ratio, Spacer Diam 30.000 300.000

Hub length, Diameter 100.000 150.000

Upright, Lower, Upper size 25.000 18.750 15.000

Steering Tie-rod, Rack 12.500 18.750

Frame rails, DriveShaft 50.000 50.000

After Roll Centre Adjusters Fitted (Variations Only)

Design Panel:

Kingpin axis 14.719 degrees

Scrub radius 39.604

Caster 3.608 degrees

Caster trail 10.655

Upper A-arm -99.000

Lower A-arm 322.685

Suspension

Lower A-arm forward A -266.000 277.000 194.000 A to B= 410.002

Lower ball joint B 0.000 588.000 169.000 C to B= 323.226

Lower A-arm rearward C 5.000 265.000 180.000 A to C= 271.627

Upper A-arm forward D 36.000 438.000 740.000 D to E= 0.000

MacPherson strut upper E 36.000 438.000 740.000 F to E= 0.000

Upper A-arm rearward F 36.000 438.000 740.000 D to F= 0.000

Wheelbase, track, tire diameter2355.0001344.000 590.000 Rollout1853.540

Camber, Toesteer, Toe span -4.500 -0.121 711.201 HubTrak 648.855

Tire contact patch 0.000 672.000 0.000 B to E= 591.470

Steering

Steering tie-rod on hub S 128.000 574.000 169.000 S to T= 344.548

Steering tie-rod inboard T 112.000 230.000 180.000 B to S= 128.763

Idler arm upper-axis P 0.000 0.000 0.000

Idler arm lower-axis Q 0.000 0.000 0.000

Drag-link to Idler arm X 0.000 0.000 0.000

Drag-link to Idler arm low Y 0.000 0.000 0.000

Drag-link tie-rod attach Z 0.000 0.000 0.000

Toe-inch span, Steering Box 711.201 50.000not used mm / 360 degrees

Appendix L - Lap Time Comparisons

Event /time	Gatton	Pittsworth	Oakey	Stanthorpe	Details
2005		57.75		44.96 lap 3:05.11 heat	Standard motor
2007		51.61			Stock suspension
2008	43.718 107th/134	48.47		40.20 lap 2:46.22 heat	1st modifications Front struts (1)
				40.02 lap 2:44.06 sec 59th/84 heat	
2009	40.408 lap 41.209 average 4th/9 class 63rd /136 lap	47.69	47.413 lap 49.382 average 4th/9 class 37th/127 average	39.46 lap 2:42.41 heat	Semi slicks on rear
				39.35 lap 2:41.50 heat 51st/92 heat	
2010	Changed track 36.773 5th/6 class 41st/141			37.90 lap 2.36.71 heat 5th/8 class 25th/73	2nd modifications Front struts (2) Semi slicks front and rear
2010		Changed track 57.59 1st/7 class 32nd/163			Watts Link
2010				2.36.28 8th/10 class 31st/95	All Project Modifications

Event/ time	Noosa Hill climb	Warwick	Mt Cotton Hill climb		
2010	67.16 sec	1200m track	50.70		2nd

	2^{nd}/4 class 51^{st}/117	44.915 lap 3.03.00 heat 4^{th}/7 class 24^{th}/62	4^{th}/8 class 47^{th}/99		modifications Front struts (2) Semi slicks front and rear
2010		2100m track 1.15.121	WET EVENT 51.96 1^{st}/5 class 28^{th}/80		All Project Modifications