

CHASSIS STIFFNESS INFLUENCE ON WHEEL LOAD DISTRIBUTION

An Investigation on the UWTSD MCR S2000 Race car.



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Declaration of Authenticity

‘This thesis is the result of the author’s original research. It has been composed by the author and has not been previously submitted for examination which has led to the award of a degree.’

Signed:

Date:

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Abstract

The torsional stiffness of a vehicle chassis is a key property in the overall handling balance and dynamics of a vehicle. While the effects of chassis stiffness on a race car are known, traditional hand calculations neglect any possibility of the chassis twisting in its longitudinal axis. During cornering, wheel load is transferred laterally from the inside to outside wheels. This imbalance from inside to outside creates a turning moment about the chassis longitudinal axis. In the real world, even the stiffest of chassis will experience some deflection. This creates an air of unpredictability from the hand calculations which can only be fine-tuned with simulation and real world testing. This project compares real world load transfer data to traditional hand calculations with a final aim of identifying the responsiveness to adjustments in roll stiffness of the UWTSD MCR S2000 chassis.

1 Introduction

The UWTSD MCR was designed and used to race in the sports 2000 category conducted by the sports racing car club, a highly competitive low budget series chosen to push the boundaries of the student run team allowing innovation and experience in a competitive environment. Chassis stiffness is known to have an effect on how the lateral load transfer at the front and rear of the vehicle changes with roll stiffness alterations. It is also known that small changes to load transfer can have large enough effects to be noticed by the driver and ultimately the overall handling performance of the car. As with any race car, minimising the weight of the car can greatly increase performance. Chassis stiffness can easily be altered within regulations with a trade-off of an increase in weight.

1.1 Project Aims

- Identify the influence of the MCR chassis stiffness on wheel load distribution by comparison of chassis weight transfer calculations with current track data.
- Validate results by comparing findings with current literature on chassis stiffness behaviour.
- Assess external factors influencing wheel load distribution and discuss how to manage and control these factors.

1.2 The UWTSD MCR S2000 Race Car



Figure 1 – The UWTSD MCR S2000 car on track. (Anthony, L. Photography. 2014).

The UWTSD MCR was designed and built to be entered into the sports racing car club (SRCC) sports 2000 duratec championship (S2000). The S2000 duratec championship prides itself on being a highly competitive, diverse, and affordable. There are a number of different chassis manufacturers of all ages and guises as the championship first started in 1977, with ford pinto engines. The current regulation engine is the ford 2000cc, 16v duratec DHE 420 engine, producing 145PS, at a cost brand new of £1,650. (Morris, 2016).

MCR race cars are a small manufacturing, design, and race team based in Wales, founded in 2007. MCR specialise in the design, manufacture, setup, and customisation of their tubular spaceframe chassis for a wide range of circuit, hill climb, or track day applications. The most popular choice for their chassis is the S2000 duratec championship. (Cars, 2017).

The UWTSD MCR has a static weight of 663.5Kg with a 78Kg driver on board with 20 Litres of fuel. It has a 39:61 front/rear weight distribution. The un-sprung mass for each front corner 22.5Kg and each rear corner was 24Kg.

Stationary							
				39%			
				LF + RF			
				2530.98			
Front Split, N		18.7%	LF			RF	20.2%
-98.10		1216.44					1314.54
LF + LR		49.3%	3207.87	6508.94	3301.07	50.7%	RF + RR
Rear Split - Front split, N	Rear Split, N	30.6%					30.5%
103.01	4.90	1991.43	LR			RR	1986.53
LR - LF Split, N		774.99					RR - RF Split, N
							671.99
				3977.96			
				LR + RR			
				61%			
				Wedge			
				50.8%			

Figure 2 –Static mass distribution map for UWTSD MCR.

1.3 Pembrey Circuit

Pembrey circuit was chosen to perform the testing of the race car due to its close proximity to the university and part of a relationship between the two parties. Pembrey is a 1.456 mile circuit with a total of 8 turns. Located in Carmarthenshire, Wales, and has been named the home of Welsh motorsport of all shapes and sizes from karting to truck racing and even rallying. Its first event was held in 1989. Honda curve was chosen as the datum point for analysis due to its relative length and speed for optimum steady state data.

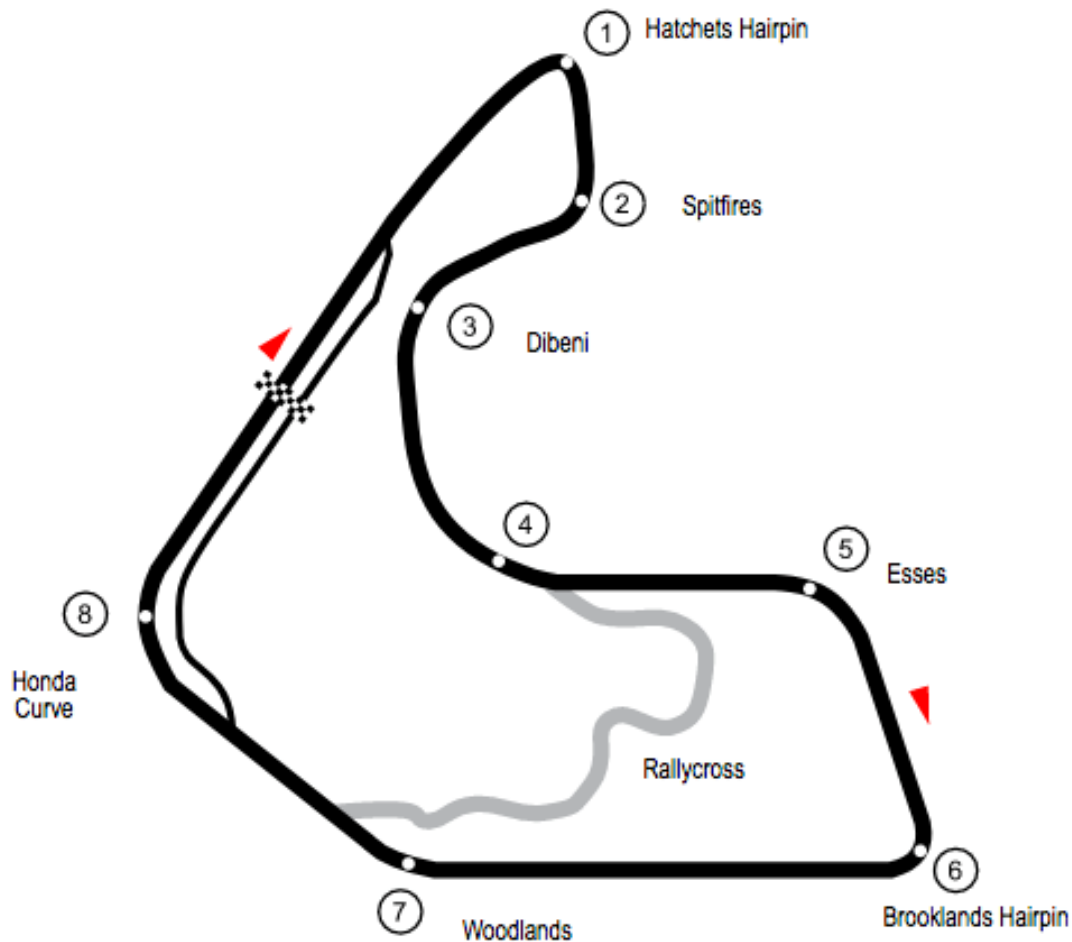


Figure 3 – Pembrey track map (roostertruckracing, 2018)

2 Research

When performing simple hand calculation for load distribution on a race car, it is important to recognise the limitations these calculations have. Currently, all simple hand calculations work on the assumption that the chassis of the car is rigid, where it does not twist under any driving condition. Torsional stiffness is the resistance to twisting along the wheelbase of a car, or X-axis. This idea can be simplified as to imagine a torsion spring connecting the front and rear axles of the car. It is known that if the chassis torsional stiffness is weak, the ability to control the lateral load transfer distribution, and handling balance, of the car is near impossible. According to Milliken and Milliken, Predictable handling can best be achieved if the chassis is stiff enough to be safely ignored. (Milliken & Milliken, 1995).

Table 1 - Average ranges of different vehicle types chassis stiffness.

Vehicle	Torsional Stiffness (Nm/Degree)
BMW E90 3-series (Family Saloon)	22,500
BMW E90 Convertible (Convertible)	15,750
Mazda RX8 (Sports Car)	30,000
Lamborghini Aventador (Supercar)	35,000
Formula 1 Cars	25,000+

All data regarding road cars was obtained online. (Carr, 2016). Formula 1 figure obtained from A, Deakin. (Deakin A, 2000).

As can be seen from table 1, the shear range of values given for each class of vehicle underlines a certain ambiguity as to how stiff a vehicle chassis should be. A number of reports and papers have been created to try and highlight a useable design process to create a chassis stiff enough to perform well and also handle predictably without having a 'sloppy' feeling sensation to the driver.

A paper written by researchers at the University of Leeds investigated the effect of chassis stiffness variation on handling balance for a formula SAE car. The paper looked at identifying the required chassis stiffness to achieve the best resultant handling balance and adjustment for the car. It is said that a good handling car is one which can respond well and predictably to changes in suspension setup. (Deakin A, 2000).

Adams flex was used to simulate the chassis cornering to its optimum capability while adjustments to the chassis stiffness were implemented along with changes to roll stiffness distribution. It was found that chassis with a softer torsional stiffness required a larger adjustment in roll stiffness distribution to achieve the same front to rear handling balance, while a stiffer chassis was able to translate the changes to roll stiffness distribution far more easily. (Deakin A, 2000).

There are two main reasons why a race car chassis has to be sufficiently stiff, the first being the ability to tune the handling balance of the car i.e. controlling the amount of understeer/oversteer during cornering. This is controlled by controlling the amount of weight transfer at either the front or rear axles. By increasing the roll stiffness on the front axle and reducing it on the rear axle, more weight transfer will then be seen at the front axle compared to the rear, total weight transfer will stay the same. By increasing the weight transfer on the front of the car, the outer tyre will gain a small amount of grip whereas the inner tyre will lose significantly more grip due to normal tyre characteristics, thus reducing the overall lateral grip across the front axle and inducing oversteer. It can be said that a chassis with a low torsional stiffness can act like a spring in series with the front suspension which in turn will be reducing the roll stiffness of the vehicle. It was suggested that for a

chassis to be 80% responsive to tuning, the chassis torsional stiffness must be close to the total roll stiffness of the vehicle. A stiffer chassis will perform and respond better as long as there is not too much of a weight penalty. (Seward, 2014).

The second reason a race car chassis should be sufficiently stiff is that it can store large amounts of strain energy during cornering. This energy stored by the chassis can unsettle the car at critical points during cornering such as when the driver straightens up and accelerates out of the corner. This can cause unpredictable and even dangerous handling characteristics. Although springs in a vehicle work in just the same way, their behaviour is controlled by dampers in the spring system, there is currently no way to dampen a chassis in torsion and thus can oscillate when the energy is released. (Seward, 2014).

2.1 Simulation Study on Chassis Behaviour

A study conducted by the University of Michigan was conducted to create a design target for their future formula SAE cars. Research was conducted using their 2016 car using Vi-Grade CarRealTime to simulate both steady state and transient cornering effects to determine chassis stiffness design goals. For the simulations, a series of three turns were theorised as to simulate different corner types found on race circuits. A wide corner, taken at 25m/s where the steering input was increased from 0° to 10° over a second, transient cornering, and then held for a further 10 seconds, steady state cornering. A mid corner, taken at 20m/s where the process was the same for a 20° steering input. And a tight corner taken at 15m/s with a total steering input of 30°. (Velie, 2016).

Each test mentioned above was used to identify the effects of the “actual” chassis, of stiffness 2100Nm/deg. An extremely “sloppy” chassis, of stiffness 210Nm/Deg. And an extremely “stiff” chassis, of stiffness 21000Nm/deg.

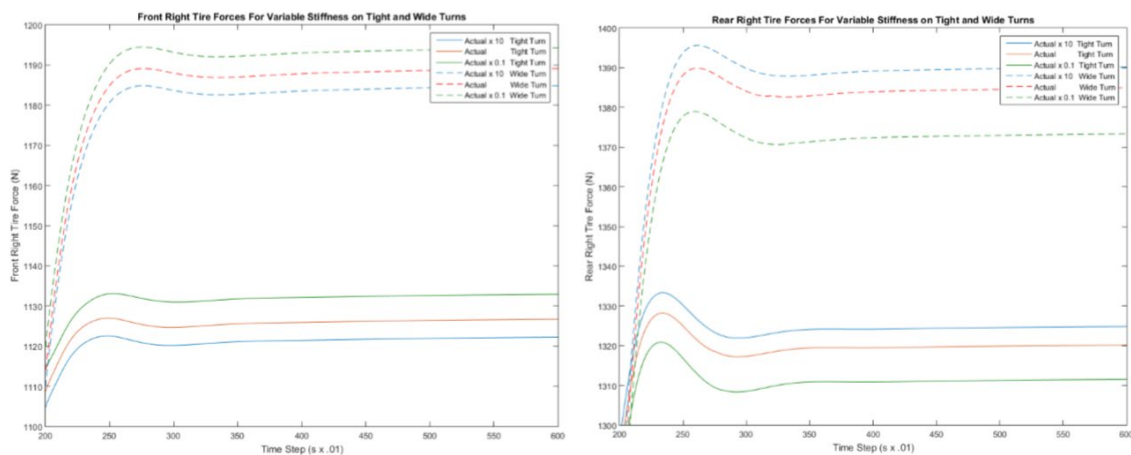


Figure 4 - (Velie, 2016)

The figure above shows the front right tire force (left), and the rear right tyre force (right), during cornering for the tight and wide turns simulated. It can be seen that with a soft chassis (green lines), the front vertical force had increased from their actual chassis design, and on the rear of the car, the force had decreased. This was consistently found across all tests in their report and clearly demonstrates the known effects of chassis stiffness on wheel loads, where the front of the vehicle will see larger load transfer and the rear will see less. (Velie, 2016).

It was then suggested that these graphs only demonstrated the way that chassis stiffness effects the way load transfer was distributed between the front and rear axles. It was then theorised that the soft 210Nm/degree chassis could attain the same front and rear load splits by softening up the front roll stiffness and increasing the rear to reduce the weight transfer on the front and increase it on the rear whilst maintaining the same overall roll stiffness. The following graph was produced.

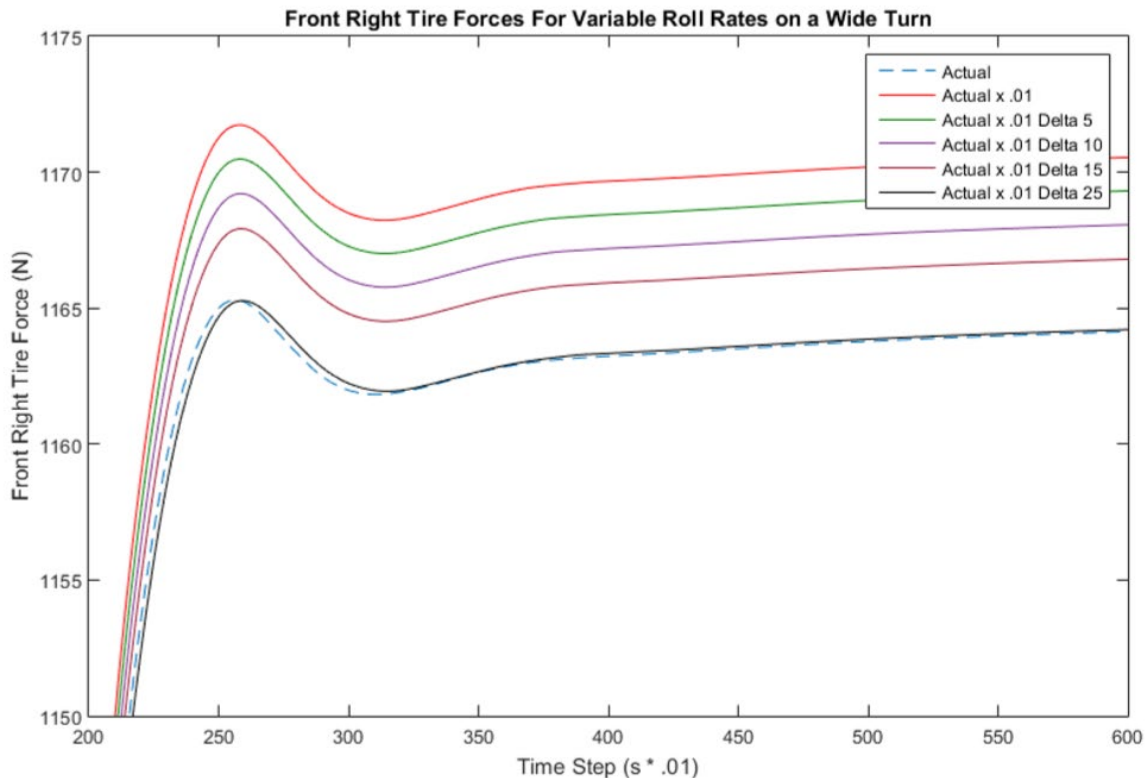


Figure 5 – Front outer tyre force with varying roll stiffness distribution, transient. (Velie, 2016).

As can be seen in the figure above, the outer tyre force for the soft chassis was able to be made to match that of the actual chassis. By matching the tyre load, it can be said that the chassis would have the same handling balance as the original chassis during steady state cornering. However, during the transient phase of cornering, it can be seen that the softer chassis tyre load was found to have a slight delayed reaction in forces being transferred. This could be said to be a sloppy feel to the car overall. (Velie, 2016).

This research has identified a number of ways in which the chassis stiffness effects the handling balance of the car. As the current data for the MCR was taken during steady state for a number of laps, it is possible to compare how, for a fixed chassis stiffness, changes in roll stiffness distribution can effect vertical loads. Being real world data, it is only possible to compare sprung mass weight transfer, however, it is assumed to follow the same trends as mentioned above.

2.2 Effects of Chassis Stiffness with Roll Stiffness Distribution

Another study conducted by the University of Leeds looked at how chassis stiffness effects the handling balance of a race car. The study was conducted using their template of a formula SAE race car. The report represents a static model for simple idea analysis and also a dynamic model used in Adams Flex simulations.

The static model for chassis analysis can initially be considered as two mass point loads, one at the front axle, and one at the rear axle, M_f and M_r respectively. Both axles are connected by a torsional spring, with a suspension at each axle to represent the vehicles roll stiffness. As can be seen in the figure below.

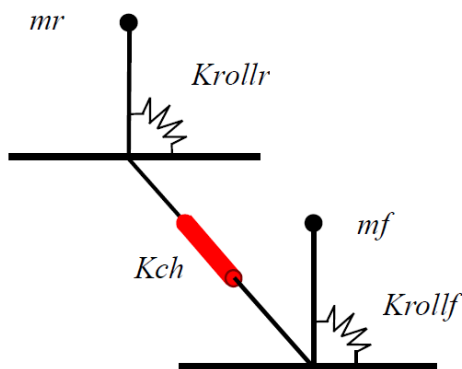


Figure 6 – Simplified static chassis model. (Deakin A, 2000).

It can be said that this model of the vehicle is not analytically correct as a real world vehicles mass is distributed unevenly along its wheelbase. The chassis also does not have an even torsional spring rate along the wheelbase as areas of denser material and structure prove to be stiffer than open cabin areas. A more accurate model was then devised seen below.

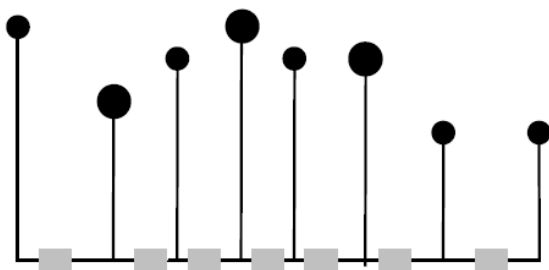


Figure 7 – Advanced static chassis model. (Deakin A, 2000).

The static analysis conducted looked at the way lateral load transfer changed with varying vehicle roll stiffness distribution for a number of different chassis stiffness's, across a number of different roll stiffness's commonly seen in racing. The chassis stiffness's calculated were; 500, 1500, 5000, and 15,000 Nm/degree respectively. It was assumed that the car had a 50/50 static weight distribution.

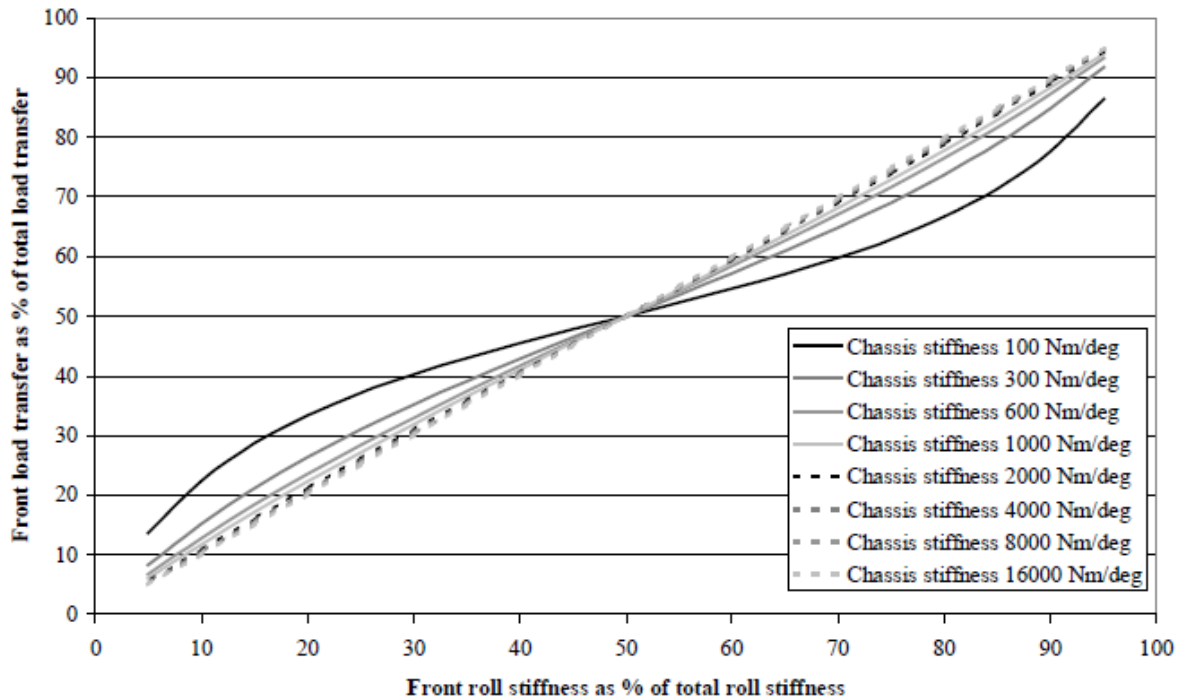


Figure 8 – Lateral load transfer of a racing car with a roll stiffness, 500Nm/degree. (Deakin A, 2000).

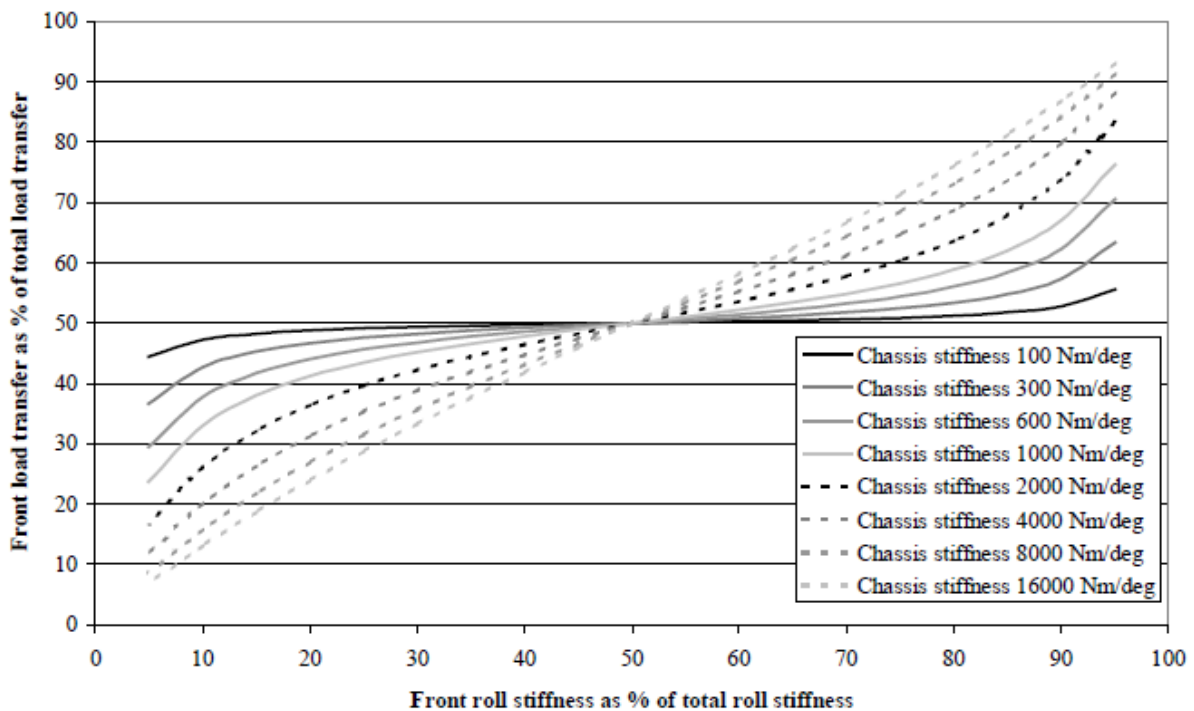


Figure 9 – Lateral load transfer of a racing car with a roll stiffness, 15,000Nm/degree. (Deakin A, 2000).

As can be seen by figures 8 and 9 above, the general findings from the static models were that cars with low total roll stiffness's can still be easily tuned with a low chassis torsional stiffness. As the roll stiffness was increased, chassis with softer torsional stiffness require more adjustment to roll stiffness distribution to achieve the same percentage of load transfer across an axle. In essence, a higher overall roll stiffness requires higher levels of chassis torsional stiffness to achieve the same level of adjustability in load transfer across each axle.

For the dynamic analysis used by the University of Leeds, the steady state analysis feature was used in Adams where the car was forced to follow a constant path radius, from this, the steering wheel angle and lateral acceleration were generated for a number of different vehicle speeds. The chassis being tested were; a fully stiff chassis, 250, 1300, and 1500 Nm/degree, along with the chassis generated from the Adams flex model.

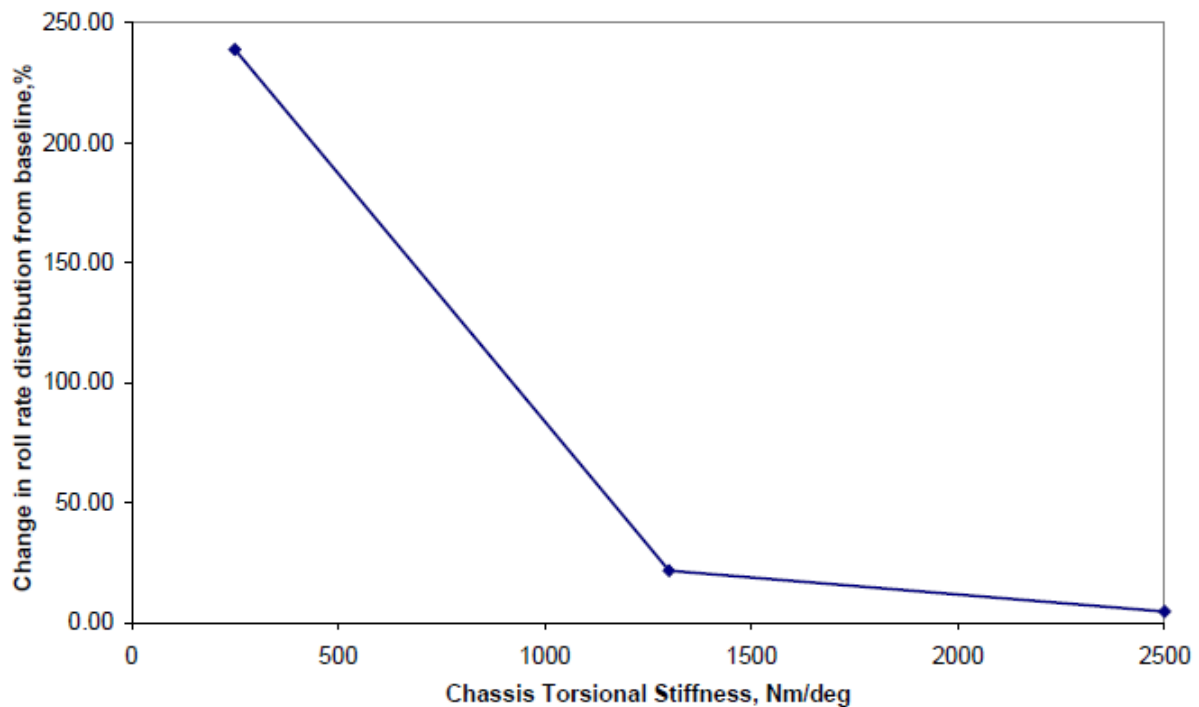


Figure 10 – Difference in roll stiffness for flexible chassis to give the same handling balance compared to a stiff chassis. (Deakin A, 2000).

Figure 10 was produced to compare how roll stiffness distribution had to be changed for a flexible chassis to achieve the same handling balance as a completely stiff chassis from a baseline. As it can be seen from the graph, the stiffer the chassis, the less change in roll stiffness distribution is required. Whereas, a very flexible chassis required a much larger adjustment to roll stiffness distribution to achieve the same handling balance.

From the results of the static analysis, it was found that, to translate a certain percentage of suspension roll stiffness distribution into lateral load transfer distribution, the chassis torsional stiffness to total suspension roll stiffness must be a certain ratio. (Deakin A, 2000).

3 Methodology

3.1 Excel

Microsoft excel was used as a basis to perform traditional hand calculations whilst adding the ability to make changes to certain values and instantly work out the end results with the use of formulas. Key measurements of the vehicle were taken in the workshop and on test beds and input to excel as the set of baseline requirements for the calculations. This included; total vehicle mass, individual static corner mass, un-sprung corner mass, spring rates, track width front and rear, wheelbase, rolling radius front and rear. Other parameters such as centre of gravity height and roll centres were extrapolated from measurements entered into WinGeo3.

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D3

	A	B	C	D	E	F	G	H	I	J	K	L	M	
1		Lateral Acceleration, M/s²		0										
2		Lateral Acceleration, G		0										
3		Measured Suspension Data						Calculated Suspension Data						
4		MR fl	From WinGeo3	0.81	Ms fl	Front Left corner sprung mass [kg]	101.25	Kgf	Front Roll Stiffness (Nm/rad)	53324.2	930.682			
5		MR fr	From WinGeo3	0.81	Ms fr	Front Right corner sprung mass [kg]	116.54	Kgf	Rear Roll Stiffness (Nm/rad)	39771.1	694.136			
6		MR rl	From WinGeo3	0.888	Ms rl	Rear Left corner sprung mass [kg]	178.3	Kgf	Total Roll Stiffness (Nm/rad)	93095.2	1624.82			
7		MR rr	From WinGeo3	0.888	Ms rr	Rear Right corner sprung mass [kg]	179	Mot	Overtuning moment (Nm)	0				
8		M fl	Front Left total measured static corner mass [kg]	123.75	Ma f	Total front sprung mass [kg]	217.79	a	Roll angle (Radians)	0	0	0 Degrees		
9		M fr	Front Right total measured static corner mass [kg]	139.04	Ms r	Total rear sprung mass [kg]	357.5	WTsf	Front weight transfer due to sprung mass [N]	0	0	MotEC		
10		M rl	Rear Left total measured static corner mass [kg]	225.28	Matot	Total sprung mass	575.29	WTAr	Rear weight transfer due to sprung mass [N]	0	0	MotEC		
11		M rr	Rear Right total measured static corner mass [kg]	210.19	Mtot	Total Mass	668.29							
12		Mus fl	Front Left unsprung mass [kg]	22.5	Mustot	Total unsprung mass	93	WTsrcf	Front WT due to sprung mass at RC Height	0				
13		Mus fr	Front Right total unsprung mass [kg]	22.5	Ka fl	Front Left Spring rate (N/mm)	71.3883	WTsrcr	Rear WT due to sprung mass at RC Height	0				
14		Mus rl	Rear Left unsprung mass [kg]	24	Ka fr	Front Right spring rate (N/mm)	72.1412	WTusf	Front WT due to Unsprung mass at wheel centre height	0				
15		Mus rr	Rear Right unsprung mass [kg]	24	Ks rl	Rear Left Spring rate (N/mm)	80.1958	WTusr	Rear WT due to Unsprung mass at wheel centre height	0				
16		Ks fl	Front Left Spring rate (lb/in)	407.7	Ks rr	Rear Right Spring Rate (N/mm)	80.3184		Total front WT	0				
17		Ks fr	Front Right Spring rate (lb/in)	412	Kw fl	Front Left wheel rate (N/mm)	46.8378		Total Rear WT	0				
18		Ks rl	Rear Left Spring rate (lb/in)	438	Kw fr	Front Right wheel rate (N/mm)	47.3318		Total WT calculated by adding all Weight Transfers	0				
19		Ks rr	Rear Right Spring rate (lb/in)	458.7	Kes rl	Rear Left Wheel Rate (N/mm)	37.9602	Fz fl	Left front vertical force [N]	1213.99				
20		Tf	Front Track 1.505	1.505	Kes rr	Rear Right wheel rate (N/mm)	38.0182	Fz fr	Rear Right vertical force [N]	1363.98				
21		Tr	Rear track	1.447	b	distance of cg of sprung mass from front axle [m]	1.60825	Fz rl	Rear Left vertical force [N]	1986.53				
22		L	Wheel base	2.388	hr-hf	Rear -Front RC Height [m]	0.10536	Fz rr	Rear Right vertical force [N]	1991.43				
23		Rrf	Rolling Radius front	0.25783	z	Depth of triangle at CG [m]	0.0636		Total wheel loads by individual wheel load check	6555.92				
24		Rrr	Rolling Radius rear	0.27662	z+hf	Height of roll axis at CG [m]	0.09933		Total wheel load by static corner weight check	6849.93				
25		Htot	Height of CG of total mass above ground	0.468	hsra	Height of CG above roll axis [m]	0.39876							
26		Hs	Height of CG of sprung mass above ground	0.49808				WT	Total WT calculated for 'solid' car	0				
27		Hf	Height of Front roll centre above ground from Win Geo	0.03373				Musf	Total Front Unsprung Mass [Kg]	45				
28		hr	Height of Rear roll centre above ground from Win Geo	0.13929				Musr	Total Rear Unsprung Mass [Kg]	48				

READY

Sheet1Load Split CalculatorLive CalculationsData PointsSheet3

Figure 11 – The complete excel spreadsheet calculator

Figure 12 above shows the completed excel spreadsheet used to calculate roll angles and vertical wheel loads. Traditional weight transfer calculations are fairly simple calculations which use basic suspension geometry along with vehicle weights, heights and positions to predict how the main body of the vehicle will behave during cornering. This excel calculator can predict load changes more accurately than can be seen by testing a real world car. Highlighted in red on the right hand side of the page is the weight transfer due to the sprung masses front and rear, this is highlighted as this is the only weight transfer that can be compared to real world test data. It is currently not possible to measure any weight transfer caused by the un-sprung mass or even the sprung mass acting at roll centre height without a way of measuring the vertical force seen by the contact patch of the tyre.

The spreadsheet calculator is split into three columns; measured suspension data, calculated suspension data, and the results section. The measured suspension data column features information directly measured from the static vehicle. Measured data included; motion ratios for each corner of the car, total individual corner masses, un-sprung corner masses, spring rates, wheelbase, track widths front and rear, wheel rolling radius', centre of gravity heights, and roll centre heights. Some of this information was calculated by a program called WinGeo3, mentioned in more detail later on. Note, the anti-roll bar rates are included just off screen below the measured column as they were disconnected for a number of test sessions and were therefore un-needed.

The calculated suspension data column was used to convert the static measured suspension features to useable information later required for the weight transfer calculations. This included using the motion ratios to convert the spring rates into the wheel rates, identifying the sprung masses per corner, and finding the roll axis of the car at the centre of gravity.

Finally, the results column was created. To begin with, roll stiffness's for the front and rear of the car were calculated using the equation;

$$K_{\phi f} = \frac{K_{wf} \cdot T_f^2}{2}$$

Equation 1

Where,

K_{ϕ} is the roll stiffness [Nm/Rad].

K_w is the wheel rate [N/m].

T is the track width [m].

And f identifies the front of the vehicle.

This equation is repeated to find the roll stiffness of the rear of the vehicle using the rear track and wheel rates. The total roll stiffness of the vehicle was then found by simply adding the front and rear roll stiffness's together.

Next, the overturning moment of the vehicle. The overturning moment is defined as the moment created about the roll axis of a vehicle caused by a lateral acceleration acting on the sprung mass centre of gravity. This can be calculated by;

$$M_{ot} = M_{Stot} \cdot a_y \cdot H_{sra}$$

Equation 2

Where,

M_{ot} is the overturning moment [Nm].

M_{stot} is the total sprung mass of the vehicle [Kg].

H_{sra} is the height of the centre of gravity above the roll axis [m].

And a_y is the lateral acceleration of the vehicle [m/s^2].

The roll angle of the car then needed to be found by using the equation;

$$\phi = M_{ot} / K_{\phi t}$$

Equation 3

Where,

Φ is the roll angle [Radians].

And, $K_{\phi t}$ is the total roll stiffness of the car [Nm/Rad].

Finally for the comparison to track data, the front and rear weight transfer due to sprung mass could be found using the equation;

$$WT_s = \frac{\varphi \cdot T \cdot Kw}{2}$$

Equation 4

Where,

WTs is the weight transfer due to sprung mass [N].

This equation is used to calculate the front and rear weight transfers using the individual tracks and wheel rates.

This was all the information that was needed to calculate the weight transfers due to sprung mass which is all that is able to be analysed alongside the Motec track data. The final calculations being used for the analysis were the individual vertical corner loads. This step is used in Motec by taking the static sprung corner mass and adding or subtracting the weight transfer due to sprung mass depending on the co-ordinate system used to define the vehicle axis.

It was possible then to find the front and rear load splits along with the front-rear load split distribution by simply subtracting the right vertical force from the left vertical force. From previous research and simple static testing, the front and rear load splits are where discrepancies can highlight any chassis behaviour during cornering loads.

3.2 MoTeC

Motec is a world renowned company specialising in data acquisition in most all forms of motorsport and every class imaginable. Motec began in the late 1980's initially being used by sports sedans and even road going vehicles, it was quickly adapted to all forms of motorsport and even to commercial and military applications.

Motec allowed the collection of data from a live test session in the MCR race car. The car was equipped with a number of sensors such as a GPS position, lateral and longitudinal accelerometers, suspension forces, and much more. The car was then sent out to perform a number of steady state and fast paced laps around Pembrey circuit, south wales, with a number of adjustments made to the setup of the car.

The GPS positioning on the car allows a map of the circuit to be used in Motec. Using this, a small section or corner of the track was singled out for comparison between setups of the vehicle. Identifying a long corner where the vehicle maintained a steady state was needed to allow the best comparison. The long steady state corner allows the forces to be averaged more accurately over a longer period of time. This is necessary as with live track data, there is often a large amount of "noise" caused by a number of uncontrollable factors such as engine vibration, roughness of the track surface, and even movements by the driver such as turning the steering wheel or changing gears.

The data acquired had no recorded lap beacon for the start finish line. This resulted in each test session counting as a single lap. Reducing the test data into laps was required to identify steady state cornering to be used for analysis. The same corner must be used for consistent analysis during steady state. By using the same corner, inputs to and from the car could then be measured to describe how the car was behaving during the corner. A GPS map of the track was created in Motec and a lap beacon placed on the map on the start finish straight. The co-ordinates of the start/finish beacon were recorded for reference to add to each test session accurately.

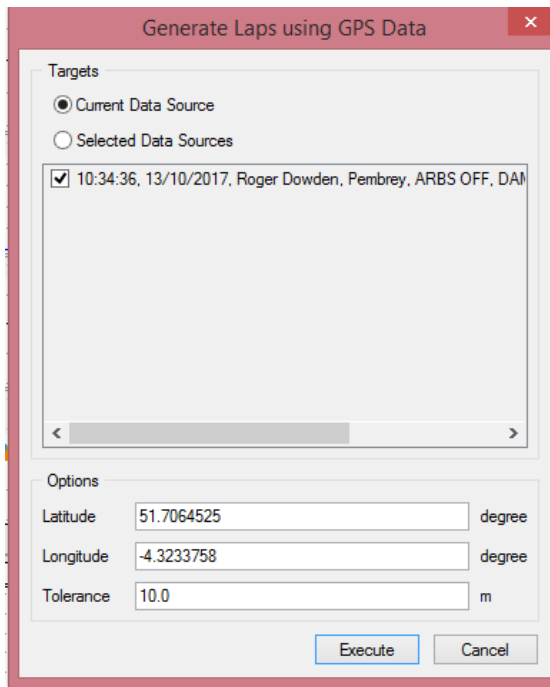


Figure 12 – GPS co-ordinates for the start finish line.

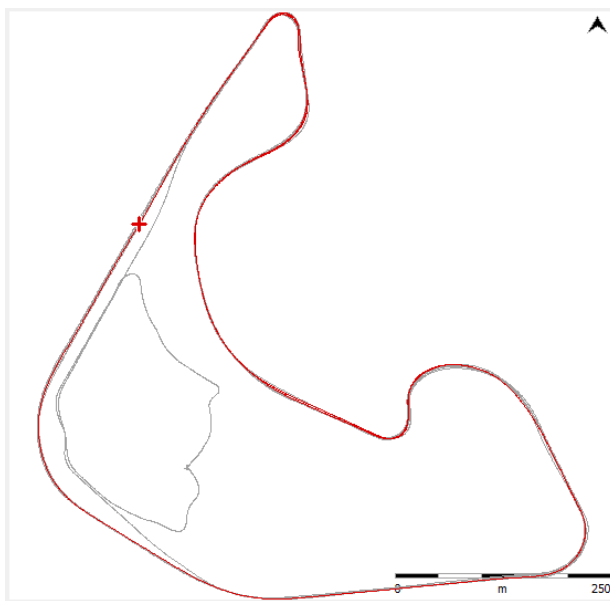


Figure 13 – GPS Track map of Pembrey circuit.

Another factor to account for with Motec data is that there are limitations in what calculations can be compared. As the suspension force sensors are directly connected to the suspension, only weight transfer due to sprung mass can be compared as there is no direct way to measure any load transfer from the un-sprung masses. As of yet it is also impossible to measure the direct tyre loads from the contact patch to the road. This would be needed to then break down the total weight transfer into its constituent parameters.

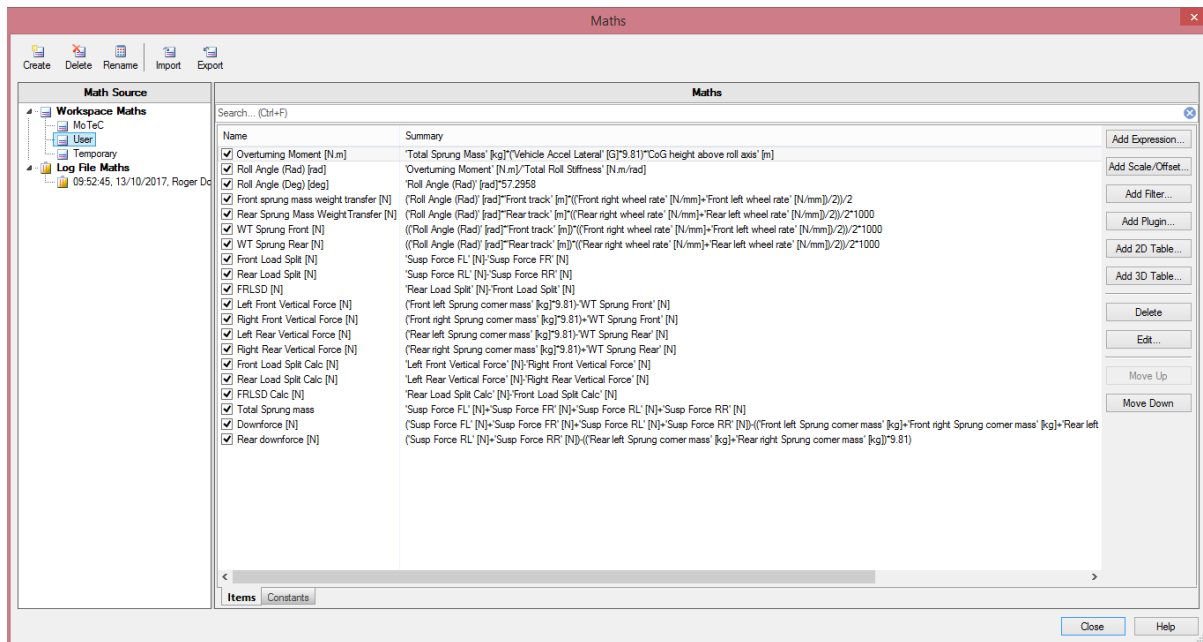


Figure 14 – Motec maths channels

Name	Value
Centre of gravity height [m]	0.466
CoG distance from front axle [m]	1.65676
CoG height above roll axis [m]	0.39385
Front left spring rate [N/mm]	71.3883
Front left Sprung corner mass [kg]	101.25
Front left Unsprung mass [kg]	22.5
Front left wheel rate [N/mm]	46.8378
Front right spring rate [N/mm]	72.1412
Front right Sprung corner mass [kg]	116.54
Front right Unsprung mass [kg]	22.5
Front right wheel rate [N/mm]	47.3318
Front roll centre height [m]	0.03373
Front Roll Stiffness [N.m/rad]	53324.2
Front track [m]	1.505
Motion ratio front	0.81
Motion ratio rear	0.688
Rear left spring rate [N/mm]	80.1958
Rear left Sprung corner mass [kg]	178.5
Rear left Unsprung mass [kg]	24
Rear left wheel rate [N/mm]	37.9602
Rear right spring rate [N/mm]	80.3184
Rear right Sprung corner mass [kg]	179
Rear right Unsprung mass [kg]	24
Rear right wheel rate [N/mm]	38.0182
Rear roll centre height [m]	0.13929
Rear Roll Stiffness [N.m/rad]	39771.1
Rear track [m]	1.447
Roll axis height at CoG [m]	0.10131
sprung mass centre of gravity height [m]	0.49516

Items Constants

Figure 15 – Constant parameters of the vehicle in Motec (1/2)

Name	Value
Front right Unsprung mass [kg]	22.5
Front right wheel rate [N/mm]	47.3318
Front roll centre height [m]	0.03373
Front Roll Stiffness [N·m/rad]	53324.2
Front track [m]	1.505
Motion ratio front	0.81
Motion ratio rear	0.688
Rear left spring rate [N/mm]	80.1958
Rear left Sprung corner mass [kg]	178.5
Rear left Unsprung mass [kg]	24
Rear left wheel rate [N/mm]	37.9602
Rear right spring rate [N/mm]	80.3184
Rear right Sprung corner mass [kg]	179
Rear right Unsprung mass [kg]	24
Rear right wheel rate [N/mm]	38.0182
Rear roll centre height [m]	0.13929
Rear Roll Stiffness [N·m/rad]	39771.1
Rear track [m]	1.447
Roll axis height at CoG [m]	0.10131
sprung mass centre of gravity height [m]	0.49516
Static front left corner mass [kg]	123.75
Static front right corner mass [kg]	139.04
Static rear left corner mass [kg]	225.28
Static rear right corner mass [kg]	210.19
Total Front Sprung Mass [kg]	217.79
Total Rear Sprung Mass [kg]	387.47
Total Roll Stiffness [N·m/rad]	93095.2
Total Sprung Mass [kg]	605.26
Wheelbase [m]	2.588

Items
Constants

Figure 16 – Constant parameters of vehicle in Motec (2/2).

As can be seen above, a number of measured constants defining the car were entered into Motec as “constants”. These constants are also seen in excel under the measured suspension data and calculated suspension data sections, required for to replicate the weight transfer calculations from excel in Motec.

For final analysis of data in Motec, a number of equations were used to compare the hand calculations to measured data.

3.3 Wan 32

Wan 32 is an evolved excel worksheet created by Warren Rowley of Rowley Race Dynamics. Wan 32 uses the same classic hand calculations as in excel combined with other calculations with more intricate parameters such as anti-jacking geometry and even co-ordinate positioning of the driver, fuel, and aero loads acting on the car. This system is then used to derive the individual loads seen by the wheels and therefore load splits.

Wan 32 was chosen as another tool as a comparison between real world data and classic hand calculations. Being able to compare excel calculations with the real world data and then again to another set of calculations can help to identify any disparities between the predictions and real world testing and validate findings associated to chassis stiffness.

A1 Chassis						A3 Other loads									
		(1) LF	(2) RF	(3) LR	(4) RR			x	y	z					
	W Base	L	1294.00	1294.00	1294.00	1294.00	mm	Ballast Kg	Ba	0	0	0	mm		
	Track	t	752.500	752.500	723.500	723.500		Driver Kg	Dr	0	0	0	mm		
	Radius	R	257.830	257.830	276.620	276.620	mm	Fuel Kg	Fu	0	0	0	mm		
	Rad Chge	Rc	0	0	0	0	mm	Aero - N	Ae	0	0	0	mm		
Unspr Mass		Mus	22.5000	22.5000	24.0000	24.0000	kg								
A1.1	Springs		(1) LF	(2) RF	(3) LR	(4) RR		A4 Anti/Jacking							
	M Ratio	m	0.810	0.810	0.688	0.688		A4.1	Indep	0/1	1	1	1	1	
	Rate	N/mm	71.3883	72.1412	80.1958	80.3184	N/mm		IC	ay	69.1	69.1	287.6	287.6	mm
									e/a	-0.014	0.014	-0.076	0.076	mm	
PreLoad		D	-7.62	-7.62	0	0	mm		Force	0.0	0.0	0.0	0.0	N	
A1.2	Third Spring		(1) LF	(2) RF	(3) LR	(4) RR		A4.2	TrackBar	0/1	0	0	0	0	
	M Ratio	m ₃	1.000	1.000	1.000	1.000				Axle y	Axle z	Frame y	Frame z		
	Rate	N/mm	0		0			Bar Geom		558.8	228.6	609.6	355.6	mm	
	PreLoad	mm	0.000		0.000			L A Spring Span		508	508	1447	1016	S Span	
A1.3	Roll Bars		(1) LF	(2) RF	(3) LR	(4) RR				e	a	angle	Track Bar Length		
	M Ratio	mR	0.751	0.707	1	1				48.020	164.700	6.203	deg	1168.40	
	Rate	N/mm	0		0					h3	h4	AA	BB		
	PreLoad	DR	0	0	0	0	mm			IC	210.70	367.98	1294	1294	
Dampers		(1) LF	(2) RF	(3) LR	(4) RR		A4.3		rT	0.5	Fce =	0.0	0.0	N	
	M Ratio	ms	0.752	0.752	1.036	1.036		Accel	0/1	0	0	1	1		
	Preload	fsp	0	0	0	0			e	474.80	474.80	326.57	326.57	mm	
	Load	N	0	0	0	0		IC	ax	9421.85	9421.85	6074.14	6074.14	mm	
	Total	N	0	0	0	0			e/a	0.0	0.0	0.1	0.1		
Initial Condi								A4.4		ri	0.5	0.5	0.5	0.5	
Scale Readings		(1) LF	(2) RF	(3) LR	(4) RR	Tot		rA Force	0.0	0.0	0.0	0.0	0.0	N	
Mass - Kg	P	123.8	139.0	225.3	210.2	698.3	kg	Brake	0/1	1.0	1.0	1.0	1.0		
Load - N	P	1213.6	1363.5	2209.2	2061.3	6847.59	N			e	474.80	474.80	326.57	326.57	mm
Bias	Bx	Left	Rear	Wedge				IC	ax	9421.85	9421.85	6074.14	6074.14	mm	
		50.0%	62.4%	52.2%					e/a	0.1	0.1	0.1	0.1		
CG		x		y		z			q	0.2	0.3	0.2	0.3		
T-Mass kg	WT	698.26	-320.007	-0.842	466.00	mm	A4.5	qB Force	0.0	0.0	0.0	0.0	0.0	N Total	
U/S -Mass kg	Wu	93	-41.742	0.000	267.53		Total Antis	Wheel Rates (Tires not in calculation)							
S -Mass kg	Ws	605.26	-362.763	-0.972	496.50			(1) LF (2) RF (3) LR (4) RR							
			-362.763	-0.972	50.42	Total		N/mm							
					186.2	CG Z		Combined Springs & Tire Rates							
Sprung Mass in kg		101.3	116.5	201.3	186.2		(1) LF (2) RF (3) LR (4) RR								
Tire Spring Rates		K _t N/mm	350	350	350	350	N/mm								
			41.31	41.69	34.25	34.29									

Figure 17 – Geometry and measurements entered into Wan 32A.

A5

Inertial Forces and CG location

		698.26	x	y	z	
Wt (g)	W	1	-320.01	-0.84	466.00	mm
Left(g)	TL	0	-320.01	-0.84	466.00	
Right(g)	TR	0	-320.01	-0.84	466.00	
Bank	α	0	(Deg)	0	Change from Pad	
Accel(g)	A	0	-320.01	-0.84	466.00	
Brake(g)	B	0	-320.01	-0.84	466.00	
	pF =	50%	pR =	50%		

A6

Solution

	Y Tran.	0.3	X Tran.	0.0			
	(1) LF	(2) RF	(3) LR	(4) RR	Total		
Wheel	P	1284.8	1292.3	2135.2	2135.3	6847.6	N
Wheel P		131.01	131.78	217.73	217.74	698.26	kg
Load Change		71.18	-71.18	-74.04	74.04		
Spring	F	1313.71	1323.07	2761.40	2761.54		N
RollBar	FR	0.00	N	0.00	N		
Damper	Fs	0.00	0.00	0.00			N

A6.2

Motion

Wheel	dP	32.13	32.05	50.05	49.97	mm
Spring	d	18.40	18.34	34.43	34.38	mm
RollBar	dR	0.08			0.00	= dRo
Ride	z	140.19	140.26	132.36	132.43	
		Left	Rear	Wedge	Kin. x	Kin. Y
Bias		49.9%	62.4%	50.1%	-320.01	-0.8
Chassis mm		Squat	Pitch	Roll		
Pad Squat	15.882	41.049	-0.007	0.000	mm. rad	
Dynamic Squat	25.167	-0.397	0.003	deg Ch.		
Squat	Change	NoseUp	Even			

A6.3

Ride

		(1) LF	(2) RF	(3) LR	(4) RR	
Initial	zo	177.8	177.8	177.8	177.8	
Coord	x	501.65	501.65	628.65	628.65	
	y	711.2	711.2	711.2	711.2	
Chge	z	37.612	37.539	45.440	45.367	mm
Final		140.188	140.261	132.360	132.433	mm
	ω					
Pad Ride		161.470	161.470	166.599	166.599	mm
Change in Ride		-21.282	-21.209	-34.239	-34.166	mm

Roll Gradient

RG0.000

Not Calculated

Tire Spring Rate

N/mm	350	LF
	350	RF
	350	LR
	350	RR

Figure 18 – Results section in Wan 32A.

3.4 WinGeo3

WinGeo3 is a software program designed to define calculable parameters of suspension geometry from co-ordinate based geometry measurements. By making accurate measurements of fixed and free connections in the suspension system in relation to the chassis of the vehicle. From these base measurements, parameters such as the ride height and roll angle can be adjusted and the behaviour of the rest of the suspension geometry is then predicted as it would be seen to change in a dynamic situation.

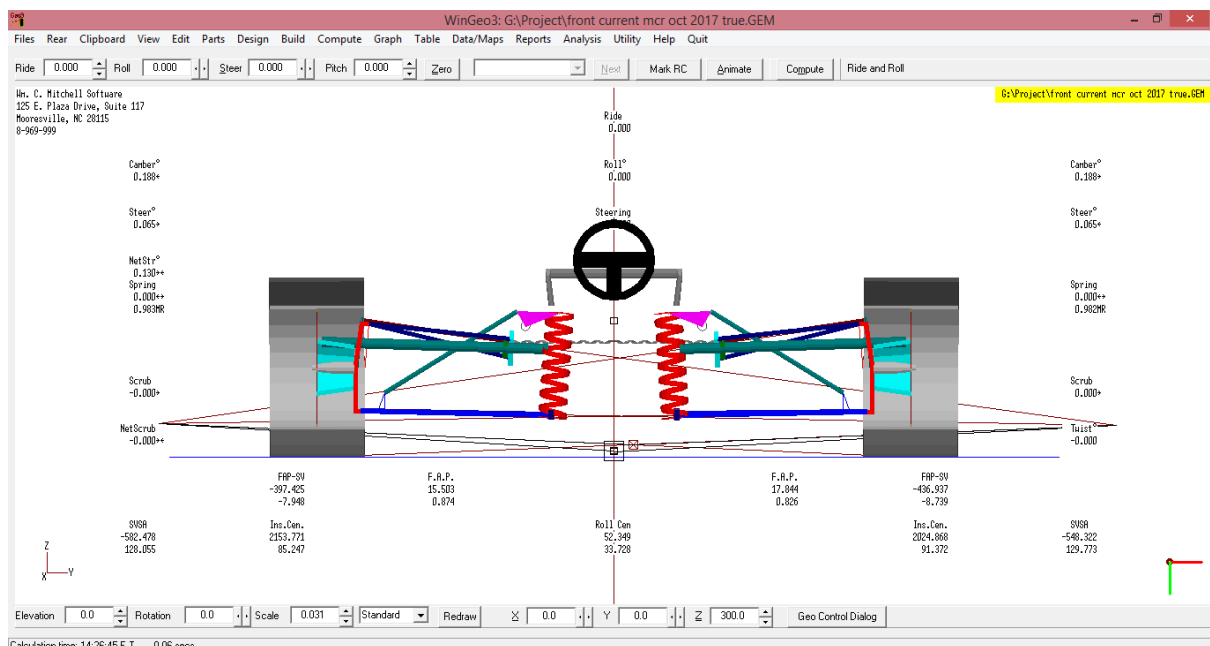


Figure 19 – Current MCR Geometry, Front.

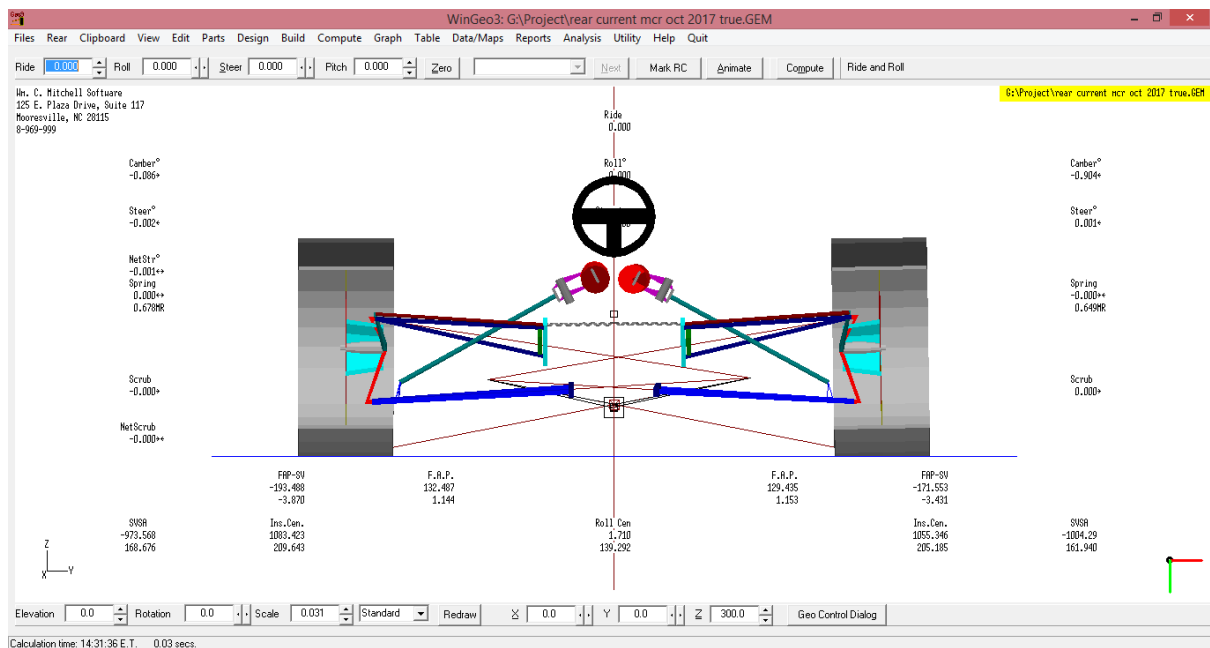


Figure 20 – Current MCR Geometry, Rear.

Recreating the current geometry for the front and rear of the MCR in WinGeo3 provided useful information needed for some of the excel spreadsheet calculations. Front and rear motion ratios were taken from these geometry models along with the roll centre heights for the front and rear axles.

3.5 Twist Rig Testing

A fellow student, Sudqi El-Alami, was also working on a similar project looking at chassis stiffness of the MCR chassis. For the purpose of their project and also to benefit mine, the MCR chassis was mounted to a torsional twist rig. The torsional twist rig mounted the chassis to a fixed point at the rear where the shear plate would be mounted. This provided a fixed end to the rig for testing. The front end of the chassis was then supported by a steel box section tube which extended left and right of the chassis. The steel box section was supported by a pivot point relative to the centre line of the chassis which allowed a torsional rotation. A jack placed on a weighing scale with a spherical bearing placed under the steel box section allowed a point load to be applied and measured at an exact point. Increasing the load in increments of 5N created a moment which twisted the free end of the chassis. A series of DTI gages placed along each length of the car allowed the deflection of the chassis to be measured at a number of points. With the twisting moment known, and the deflection measured, it was then possible to identify the torsional stiffness of the chassis overall.



Figure 21 – MCR Chassis mounted to a twist rig, front view.



Figure 22 – MCR Chassis mounted to a twist rig, rear view.

4 Analysis

During live testing, a number of test sessions were performed in steady state with adjustments made to suspension setup.

Test session 1: Anti-roll bars off, Dampers mid.

Test session 2: Anti-roll bars off, Dampers soft.

Test session 3: Anti-roll bars off, Dampers hard.

Test session 4: Anti-roll bars hard, Dampers hard.

Test session 5: Anti-roll bars hard front and soft rear, Dampers soft.

Test session 6: Anti-roll bars soft front and hard rear, Dampers soft.

Specifically for this report, test's 2, 5, and 6 will be compared and analysed. From research, it is known that the chassis stiffness' main effects are on how the handling balance of the car reacts to changes in roll stiffness. For each of these test runs, the dampers were set to their softest setting meaning the transient cornering phase was consistent and the damper characteristics can be eliminated from analysis.

4.1 Test Session 2

For test session 2, the car was sent out with the anti-roll bars disconnected and the dampers set to their softest setting. By having the dampers set to their softest setting, any forces being generated by their movement will be kept to a minimal for more accurate result analysis. With the anti-roll bars disconnected, the overall roll stiffness of the car will be lower than in other tests being analysed, however, this will act as a good baseline investigation for calculation validation.

To begin with, the circuit data was compared to identify the best corner and lap of the circuit would be suitable for further analysis. For test session 2, this was Honda curve on lap 2, with a total lap time of 1(m).44(s).064. The weight transfer due to the sprung mass of the vehicle was calculated in Motec using equation 4. The results can be seen below.

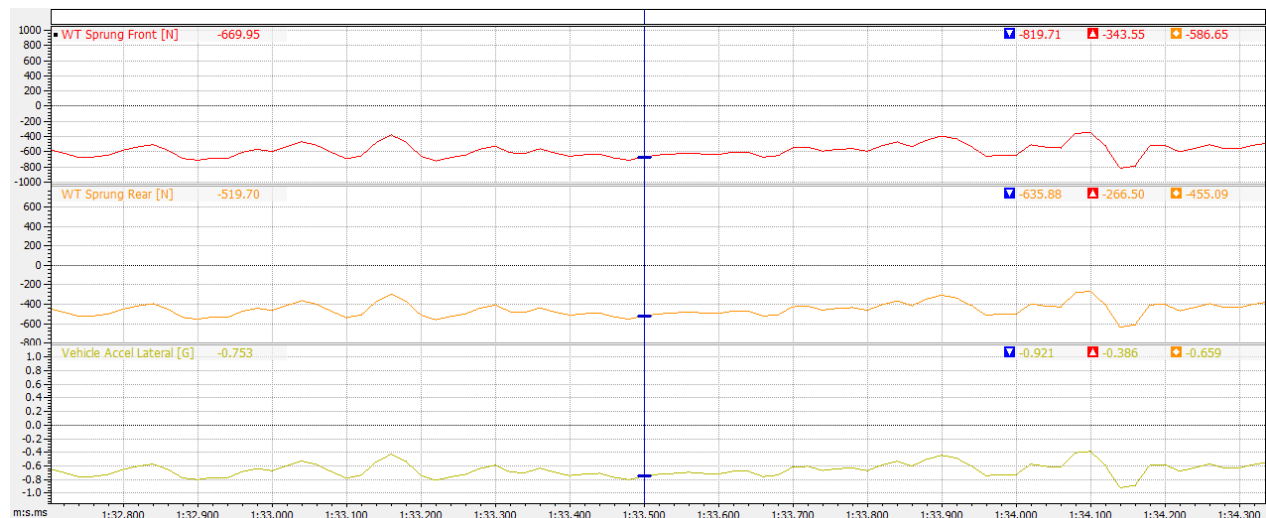


Figure 23 – Motec data map of weight transfer front, rear, and vehicle lateral acceleration.

The data acquired spans a total of 1.6 seconds of steady state cornering, with an average lateral acceleration of -0.659G. During the course of this period, the front weight transfer can be seen to be -586.12N and the rear weight transfer was -454.67N. When -0.659G was entered into the excel weight transfer calculator, the front weight transfer was predicted to be 564.43N and the rear 437.85N.

Table 2 - Front and rear weight transfer due to sprung mass comparison.

	Motec (Average)	Excel Calculator	Difference [%]
Front Weight Transfer [N]	-586.12	-564.43	3.7
Rear Weight Transfer [N]	-454.67	-437.85	3.7

Table 2 shows the comparison of weight transfers calculated in Motec using live track data, and feeding information into excel and calculating the weight transfer traditionally. This was done as a validation technique to compare how Motec would compare initially to calculated data. Overall, the calculated front and rear weight transfer values were 3.7% from the live data providing a strong correlation between hand calculations and live data.

Some of the variation seen by the data can be down to a number of uncontrollable factors such as bumps in the road. Downforce can also be a consideration as Honda curve is a large radius turn good for steady state cornering analysis, however taken at higher speeds. Another graph was created to show how downforce effected the suspension loads with speed.

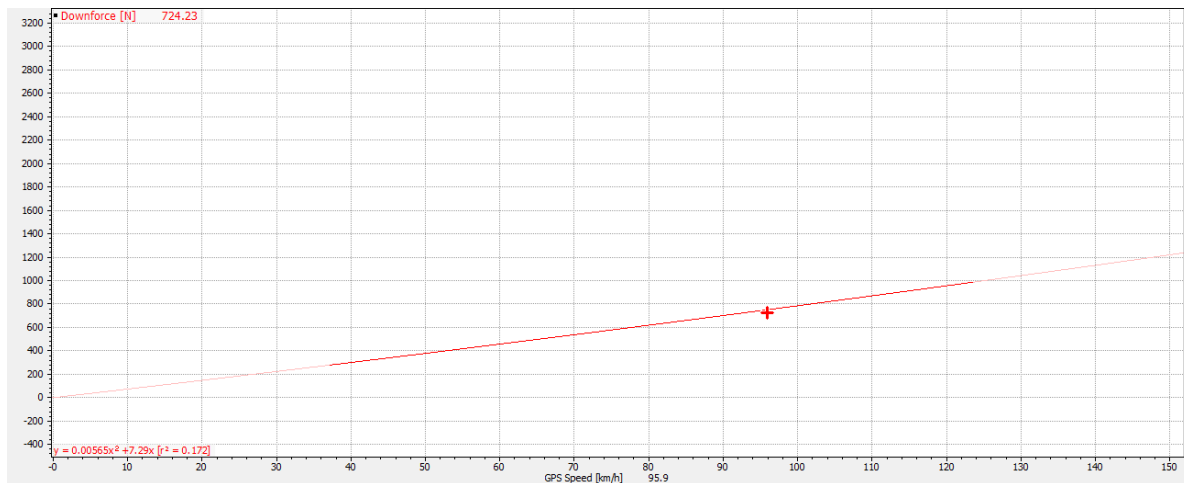


Figure 24 – Graph showing how downforce changes in relation to vehicle speed.

Vehicle downforce was calculated simply by combining all suspension forces and subtracting the static suspension forces. The resultant figures can be assumed to be attributed to the downforce of the car. An XY scatter plot was created to represent how the suspension forces changed with vehicle speed. It is known that, the faster a car can go, the more downforce/lift can be created resulting in the graph figure 24.

Using the live graph pictured in figure 24, it was calculated that the car was producing 492.94N of downforce at 94.1 kilometres an hour. There is some ambiguity again in this calculation as results could be affected by other influences such as bumps in the road or fuel usage. Downforce acts across the whole vehicle at all times and doesn't transfer with weight so for now will be omitted as the force on the left and right sides of the vehicle due to downforce should be equal therefore the only load splits will be due to weight transfer.

To find the front and rear load splits, vertical suspension forces had to be calculated by taking the sprung corner force and adding or subtracting the weight transfer depending on the definition of the car positioning. The equations used in Motec were as follows;

$$FL\ Vertical\ Force = FL_{SF} - WT_{SF}$$

Equation 5

Where;

FL, is Front left.

FLsf, is Front left sprung force.

WTsf, is Weight transfer due to sprung mass front.

$$FR_{VF} = FR_{SF} + WT_{SF}$$

Equation 6

Where FR refers to front right of the vehicle. The process was the same for the rear of the car providing equations 7 and 8 respectively.

$$RL_{VF} = RL_{SF} - WT_{SR}$$

Equation 7

$$RR_{VF} = RR_{SF} + WT_{SR}$$

Equation 8

☒ Left Front Vertical Force [N]	('Front left Sprung corner mass' [kg]*9.81)-WT Sprung Front' [N]
☒ Right Front Vertical Force [N]	('Front right Sprung corner mass' [kg]*9.81)+WT Sprung Front' [N]
☒ Left Rear Vertical Force [N]	('Rear left Sprung corner mass' [kg]*9.81)-WT Sprung Rear' [N]
☒ Right Rear Vertical Force [N]	('Rear right Sprung corner mass' [kg]*9.81)+WT Sprung Rear' [N]

Figure 25 – Equations 5, 6, 7, and 8 as used in Motec.

From this, the front and rear load splits were calculated by using the following equations;

$$\text{Front Load Split (Calc)} = \text{Left front vertical force} - \text{Right front vertical force}$$

Equation 9

$$\text{Rear Load Split (Calc)} = \text{Left rear vertical force} - \text{Right rear vertical force}$$

Equation 10

☒ Front Load Split Calc [N]	'Left Front Vertical Force' [N]-'Right Front Vertical Force' [N]
☒ Rear Load Split Calc [N]	'Left Rear Vertical Force' [N]-'Right Rear Vertical Force' [N]

Figure 26 – Equations 9 and 10 as used in Motec.

Another way of working out weight transfer due to sprung mass and therefore load splits is to take the measured vertical suspension force readings and subtract one from the other in the same fashion. This is done to show the actual weight transfer and therefore load splits and is to be compared with classic hand calculations to identify any inaccuracy between the two and also validate findings with previous research on chassis stiffness.

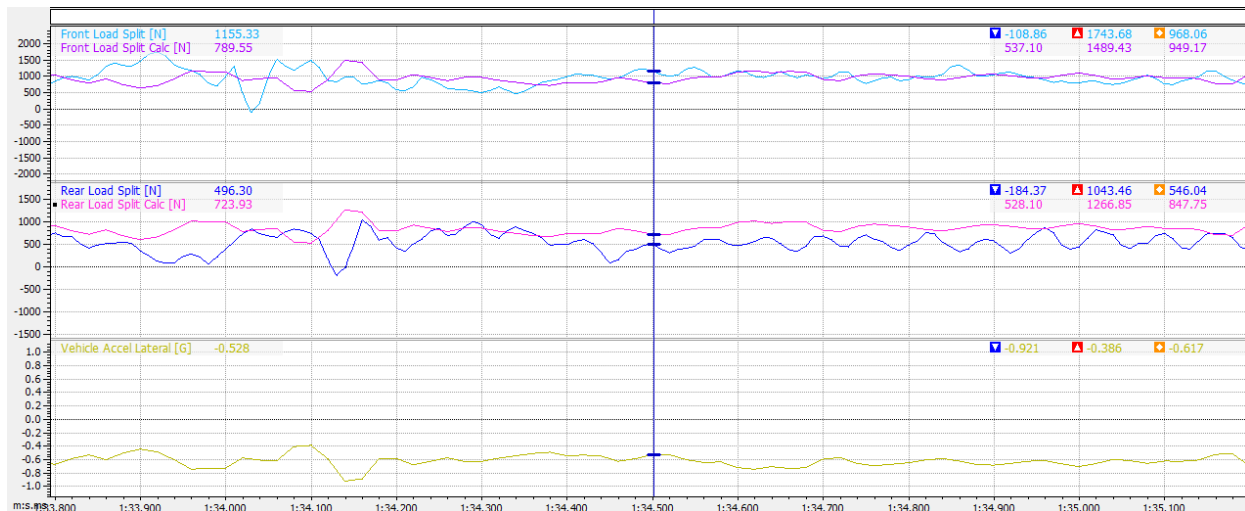


Figure 27 – Motec comparison of calculated and actual load splits front at rear.

Table 3 – Average Load splits as seen in figure 27.

	Actual	Calculated	Difference
Front Load Split [N]	968.06	949.17	18.89
Rear Load Split [N]	546.04	847.75	-301.71
Front Rear Load Split Distribution	63.94%	52.82%	+11.12%

The data seen in table 3 compares how the actual load split seen by the suspension forces compares to the calculated load splits assuming a 100% stiff chassis. From research, the general trend for a softer chassis than predicted, there should be a larger front load split than previous and a lower rear load split than expected. (Velie, 2016). The results do seem to follow this trend as the front load split is higher than calculated by 18.89N while the rear is lower than expected by 301.71N. It was also noted that the percentage of the load transfer seen by the front wheels was 11.12% larger than calculated which is significantly noticeable.

While the front load split was very close to as it was predicted, the rear load split was found to be much lower than expected. It is already known that the MCR chassis has a low torsional stiffness of around 2000Nm/degree which may be allowing a large amount of flex under the steady state loading condition. With more weight and a stiffer chassis at the rear of the car, the front could be said to be flexing with the chassis essentially being a spring in series with the front suspension and thus not transferring the weight transfer through the chassis as is calculated with a stiff chassis.

4.2 Test Session 5

Test session 5 consisted of the front anti-roll bars being reconnected to their hardest setting of 59.844N/mm, and the rear anti-roll bar set to its softest setting of 47.52N/mm whilst the dampers remained on their softest setting. By reconnecting the anti-roll bars, the total roll stiffness of the car was increased from 1624.84 Nm/Degree with a front roll stiffness proportion of %57.28, to a total roll stiffness of 3676.77 Nm/Degree with a front roll stiffness proportion of %57.51.

Table 4 – How roll stiffness varied between test 2 and 5.

	Test 2	Test 5	Difference %
Front Roll Stiffness [Nm/degree]	930.68	2114.35	127.18
Rear Roll Stiffness [Nm/degree]	694.14	1562.42	125.09
Total Roll Stiffness [Nm/degree]	1624.82	3676.77	126.29
Front Roll Stiffness %	57.28	57.51	0.23
Torsional/Roll Stiffness	1.43	0.63	-55.94

Table 4 shows how the roll stiffness of the car had changed with the addition of the anti-roll bars. From research, it was found that a low total roll stiffness could still be responsive to changes to roll stiffness distribution with a soft chassis with a low torsional stiffness. However, cars with higher total roll stiffness's require a stiffer chassis to be as responsive to those changes. (Deakin A, 2000). From this, it was predicted that the actual load splits would deviate more from the classic hand calculations than before.

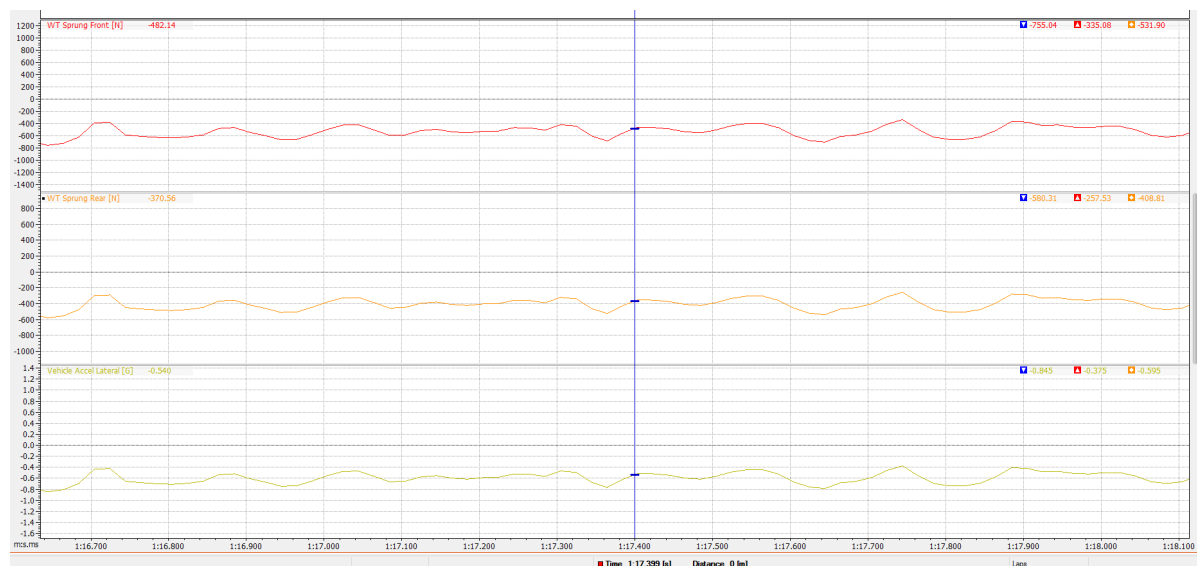


Figure 28 – Motec data graph of sprung mass weight transfer for test session 5.

Figure 28 was produced in Motec to represent the weight transfer seen by the front and the rear axle of the car in similar conditions to test session 2. The same corner was selected to minimise error in driver inputs from skewing the results, the lateral acceleration for this test session didn't reach the same average as test 2 at only -0.595G. For consistency, the average was taken over a period of 1.6 seconds of steady state cornering.

Table 5 – Front and rear weight transfer due to sprung mass comparison.

	Motec (Average)	Excel Calculator	Difference [%]
Front Load Transfer [N]	-531.9	-509.61	4.19
Rear Load Transfer [N]	-408.81	-395.32	3.30

Table 5 was generated to validate excel calculations with the ones used in Motec. During this test session, some amendments were made to the equations used in both Motec and Excel to account for the introduction to the anti-roll bars. Firstly, the roll stiffness of the had been increased, to include this into the roll stiffness calculation, equation 1 became;

$$K\varphi = \frac{(K_w + K_{arb}).T^2}{2}$$

Equation 11

Where;

Karb is the anti-roll bar rate, [N/m]

The inclusion of the anti-roll bars is included wherever roll stiffness is used. Previously without anti-roll bars being used, the roll stiffness was substituted by the wheel rates as the only conditioning factor. Jörg Segers states that;

$$K_{Roll} = K_{RollSprings} + K_{RollARB}$$

Equation 12 (Segers, 2014)

This equation is used for the front and rear axles of the car respectively to find individual front and rear roll stiffness's, and finally combined to get a total roll stiffness. This then gives a roll angle for a given lateral acceleration using equation 3. The weight transfers due to sprung mass also had to be adjusted to include the effects of the anti-roll bars as follows;

$$WT_s = \frac{(K_w + K_{arb}).T.\theta}{2}$$

Equation 13

The inclusion of the anti-roll bars reduces the roll angle seen by the car. However, the weight transfer remains the same. As can be seen in table 5, the front weight transfer was only 4.19% higher than calculated while the rear was closer at 3.30%.

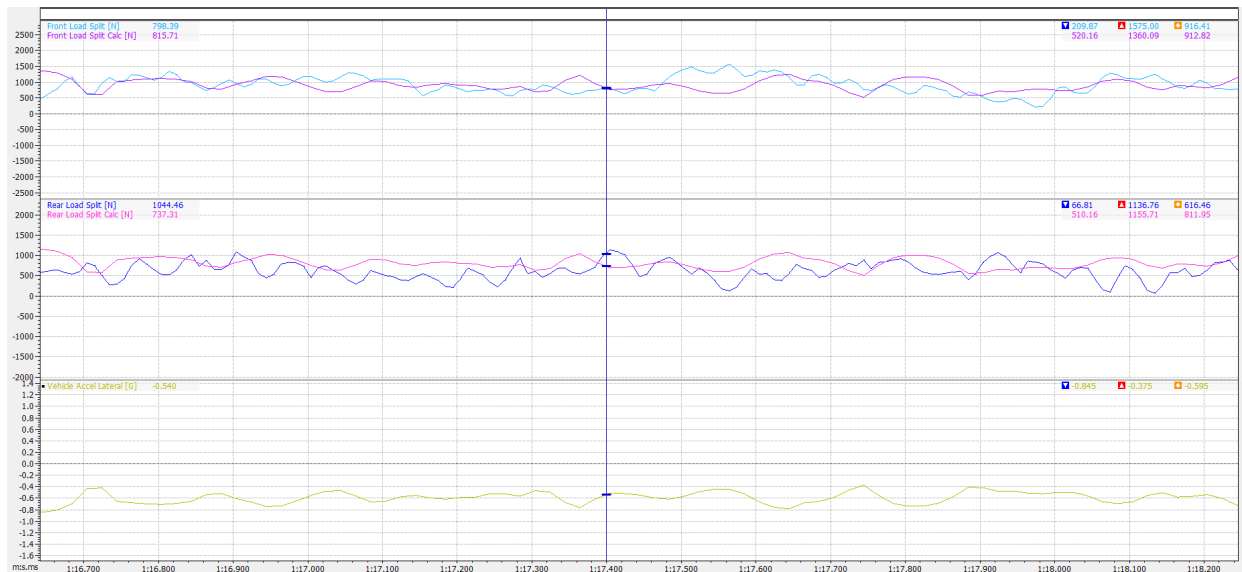


Figure 29 – Front and rear load split comparison of actual and calculated load splits.

Figure 29 above overlaps the actual and calculated data for the front and rear load splits respectively. With the front anti-roll bars set to their highest setting and the rears at their lowest, it could be suggested that the front of the vehicle would pick up more weight transfer and the rear axle should lose some respectively as the roll stiffness distribution of the car had shifted forward slightly.

Table 6 – Average load splits as seen in figure 29.

	Actual	Calculated	Difference
Front Load Split [N]	916.41	912.82	3.95
Rear Load Split [N]	616.46	811.95	195.49
Front Rear Load Split Distribution	59.78%	52.92%	+6.86%

Table 6 indicates the average load splits seen in figure 29. For the roll stiffness distribution of the car, the front load split is only seen to be 6.86% larger than was predicted. This may suggest that the predictions made from calculations were more accurate than before. However, it can also be suggested that with the increase in total roll stiffness, the chassis was less able to transfer the lateral load from the front to rear axles effectively. Even though the roll stiffness distribution shifted forward slightly from 57.28% to 57.51%, the load split distribution sifted rearward. This follows trends found by the University of Leeds shown in figures 7 and 8 mentioned earlier. (Deakin A, 2000).

4.3 Test Session 6

Test session 6 was the reverse of test session 5, where the front anti-roll bars were set to their softest setting, of 47.52 N/mm whilst the rear was set to its hardest setting of 59.884 N/mm. this change does not change the total roll stiffness of the car but does however shift the total roll stiffness distribution rearward. The front roll stiffness percentage was reduced to 51.12%, down from 57.51% in test 5. This adjustment to the car should see more weight transfer at the rear of the car previously and therefore a larger rear load split, in turn with a decrease in front load split.

Table 7 – Roll stiffness and distribution comparison, tests 5 and 6.

	Test 5	Test 6	Difference
Front Roll Stiffness [Nm/Degree]	2114.35	1869.97	-244.38
Rear Roll Stiffness [Nm/Degree]	1562.42	1788.33	225.91
Total Roll Stiffness [Nm/Degree]	3676.77	3658.30	-18.47
Front Roll Stiffness %	57.51%	51.12%	-6.39%
Torsional/Roll Stiffness	0.63	0.63	0

Table 7 represents the changes that were made to the car between tests 5 and 6. The front roll stiffness was reduced by changing the front anti-roll bar from the hardest to softest setting. The rear roll stiffness was increased in the same manner, increasing the anti-roll bar from the softest setting to the hardest. These changes shifted the overall roll stiffness distribution by 6.39% more to the rear. This should result in an increase in weight transfer across the rear axle and less on the front.

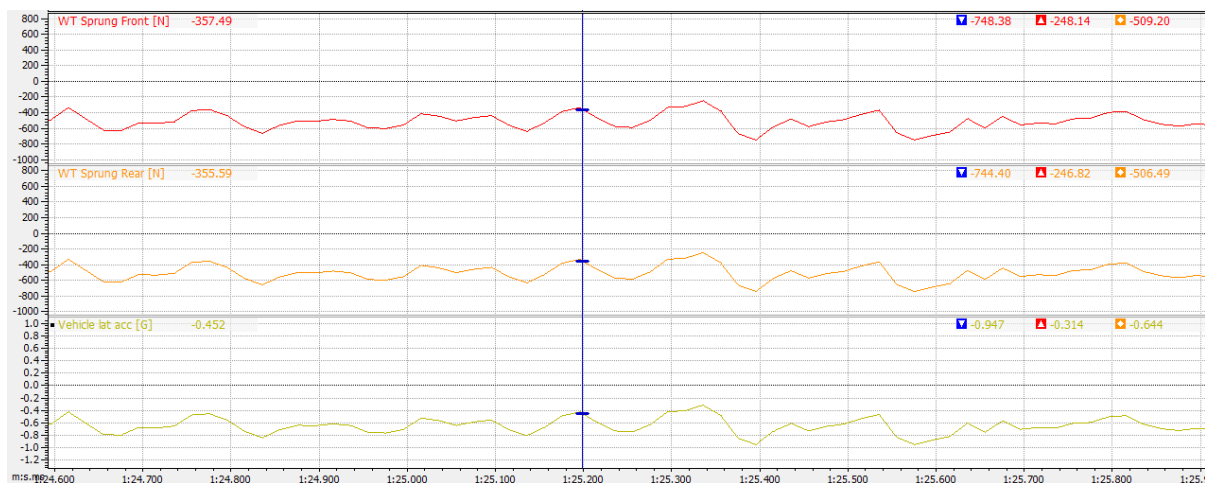


Figure 30 – Weight transfer graph as seen in Motec.

Table 8 – Comparison of Weight transfer seen in Motec compared to hand calculations.

	Actual	Calculated	Difference
Front Load Transfer [N]	-509.20	-492.23	3.33%
Rear Load Transfer [N]	-506.49	-489.62	3.33%

For this test data, a 1.6 second section was taken during steady state cornering on the same corner as previous tests. The lateral acceleration was found to be -0.644G average. The variation found between the calculated and actual values for lateral load transfer were only 3.33% indicating a good accuracy for the weight transfer seen by the car.

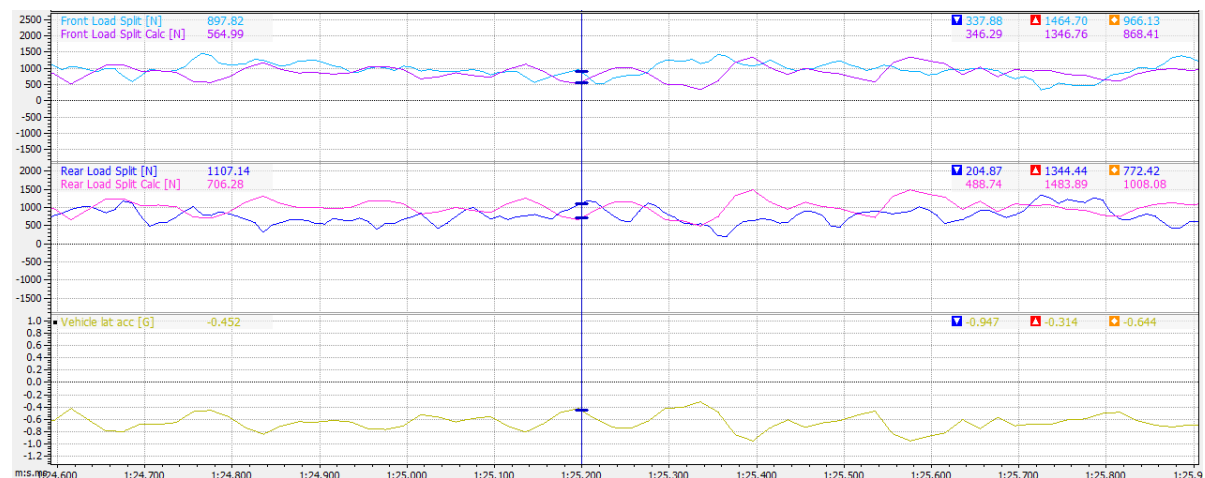


Figure 31 – Front and rear load splits calculated and actual for test session 6.

For test session 6, the lateral load splits seen by the car did not follow as strong of a trend as was expected. The front load splits were found to be lower than predicted, whilst the rear load split was larger than expected. The data can be seen in the table below.

Table 9 – Front and rear load split data as seen in figure 31.

	Actual	Calculated	Difference
Front Load Split [N]	966.13	868.51	97.62
Rear Load Split [N]	772.42	1008.08	-285.66
Front Rear Load Split Distribution	55.57%	46.28%	+9.29%

The figures in table 9 represent the load splits front and rear compared to the calculated load splits as seen in figure 28. From the data, it is clear to see that the front axle had a higher load split than predicted followed with a lower than predicted rear load split, consistent with findings from the University of Michigan and the University of Leeds.

Table 10 – Comparison of the front rear load split distributions of test 5 and 6, actual and calculated.

	Test 5	Test 6	Difference
Front Rear Load Split Distribution (Actual)	59.78%	55.57%	4.21%
Front Rear Load Split Distribution (Calculated)	52.92%	46.28%	6.64%

Table 10 was created to compare how the lateral load transfer was seen by the front axle for sessions 5 and 6, and comparing the actual distributions with the calculated predictions. The table allows a snapshot comparison of how the chassis reacted to a change in roll stiffness distribution. By increasing the rear anti-roll bar rate and reducing the front anti-roll bar rate, calculations predicted a 6.64% decrease in load split seen by the front axle, when in real world testing it was found that the

front rear load split distribution only reduced by 4.21%. This suggests that the chassis requires more adjustment to roll stiffness distribution, or an increased chassis torsional stiffness, to achieve the desired front rear load split distribution, as according to the University of Leeds. (Deakin A, 2000).

5 Discussion

5.1 Test Session Results

For test session 2, the anti-roll bars were disconnected on the front and rear, resulting in a lower roll stiffness for the car than test sessions 5 and 6. The lower roll stiffness has the adverse effect seen mostly in the transient phase of cornering, with a lower roll stiffness, the car and chassis will see higher roll angles and thus it takes time for the car to roll, delaying the weight transfer during cornering resulting in a slower turn in and delayed driver feedback. On the other hand, from research, the lower roll stiffness in relation to the chassis stiffness makes the car more responsive to tuning adjustments to the handling balance (Deakin A, 2000). Unfortunately, adjustable anti-roll bars are the easiest way to manipulate roll stiffness distribution, without them, a longer and more expensive process of replacing the springs is needed.

For test session 5, the car behaved as would be expected from research. The close results with slight deviation followed perfectly with research by the University of Michigan in where the front load splits are slightly higher than predicted and the rear load split slightly lower. (Velie, 2016). The inclusion of the anti-roll bars significantly increased the roll stiffness of the chassis, raising it from 1624.82 Nm/degree to 3676.77 Nm/degree. From research, this has the effect of reducing the cars sensitivity to adjustments in roll stiffness distribution, therefore requiring larger adjustments to achieve the desired handling balance. To achieve the same sensitivity to handling balance adjustments, it can be estimated that the chassis should have a torsional stiffness of at least 5257.78 Nm/degree.

Test session 6 provided an opportunity to analyse how the chassis reacted purely to a change in roll stiffness distribution. The front anti-roll bars were reduced to their softest setting whilst the rear was increased to its stiffest setting, maintaining the same overall roll stiffness whilst shifting the roll stiffness distribution rearward. This should increase the weight transfer at the rear axle and reduce the weight transfer at the front of the car. It was highlighted in table 10, that the changes made from tests 5 and 6 would result in the front load split to reduce, as a percentage of overall load transfer, by 6.64%. Motec data showed that the adjustments made only resulted in a decrease of 4.21%.

By comparing tests 5 and 6, it is possible to describe the behaviour of the MCR chassis in terms of the effects chassis stiffness have on the load split distribution. It was stated that the front rear load split distribution should decrease by 6.64% when in testing, it was found to only decrease by 4.21%. This indicates that the chassis was only 63.4% responsive to small changes in roll stiffness distribution.

The UWTSD MCR has been described by drivers as a very capable chassis, able to perform and handle well even with the anti-roll bars disconnected. This may be partly due to the relative light weight and low height of the car that the roll stiffness is not required to be large, as the overturning moment is relatively small. As the total roll stiffness of the car is low, the chassis torsional stiffness can be made low. A chassis with a low torsional stiffness can be assumed to be lighter than those with higher roll stiffness's when designed for the same application. This can be acceptable as long as the chassis stiffness is high enough to effectively translate suspension changes into predictable results, as with research by the University of Leeds, a chassis of torsional stiffness close to or exceeding total roll stiffness is required. (Deakin A, 2000).

5.2 Proposed Errors

The results from each test session once formatted, provided a good foot hold for further analysis. There are a number of different factors that could be effecting the final results creating an air of ambiguity from the findings. Most notably are the initial weight transfer calculations comparing the Motec data with the classic calculations, each test returned higher weight transfer for each axle by 3-4%. This can be caused by a number of unpredictable different factors such as fuel weight moving about in the tank, driver weight can change day to day, and any camber or bumps in the track.

Another consideration to make is how the aerodynamic loads on the car effect the weight transfer or roll angles. A scatter plot was created in Motec to show how downforce varied with vehicle speed. Without load sensors or aero rakes to measure exact air flow, a simple solution was created by using the suspension load sensors. In theory, a cars weight shouldn't increase and thus any increase in overall sprung loads can be assumed to be downforce being produced by the car. By subtracting the static corner suspension forces from the measured suspension forces and plotting the results against vehicle speed, a graph of downforce was created.

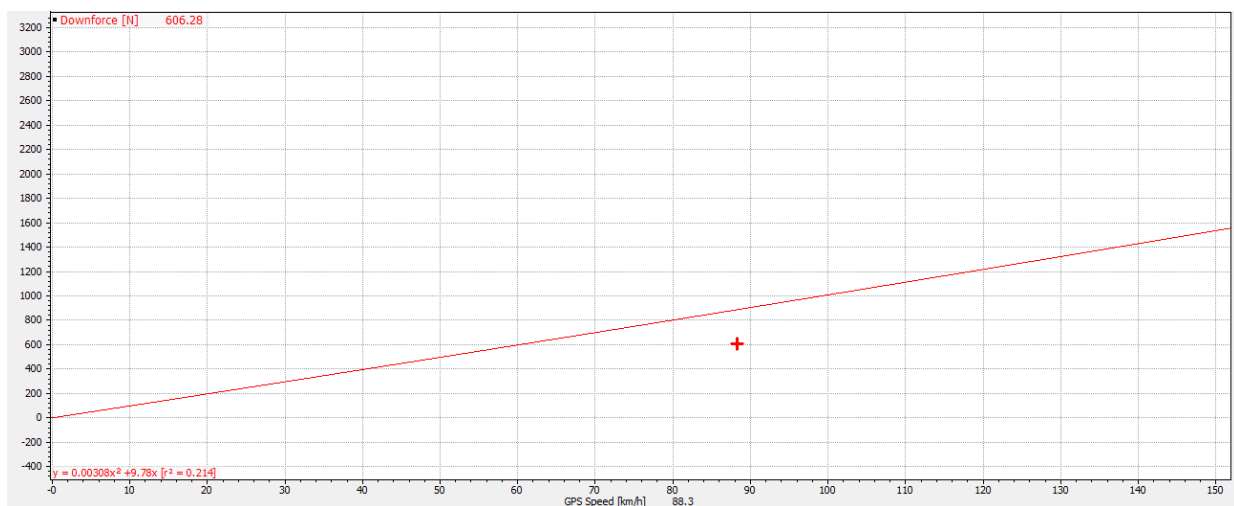


Figure 32 – Increase in suspension loads against vehicle speed.

Figure 32 was created by implementing a trend line to a scatter plot, an exponential equation was also implemented to define the trend line. This is only a crude estimation of possible forces as the original scatter plots show below in figure 33.

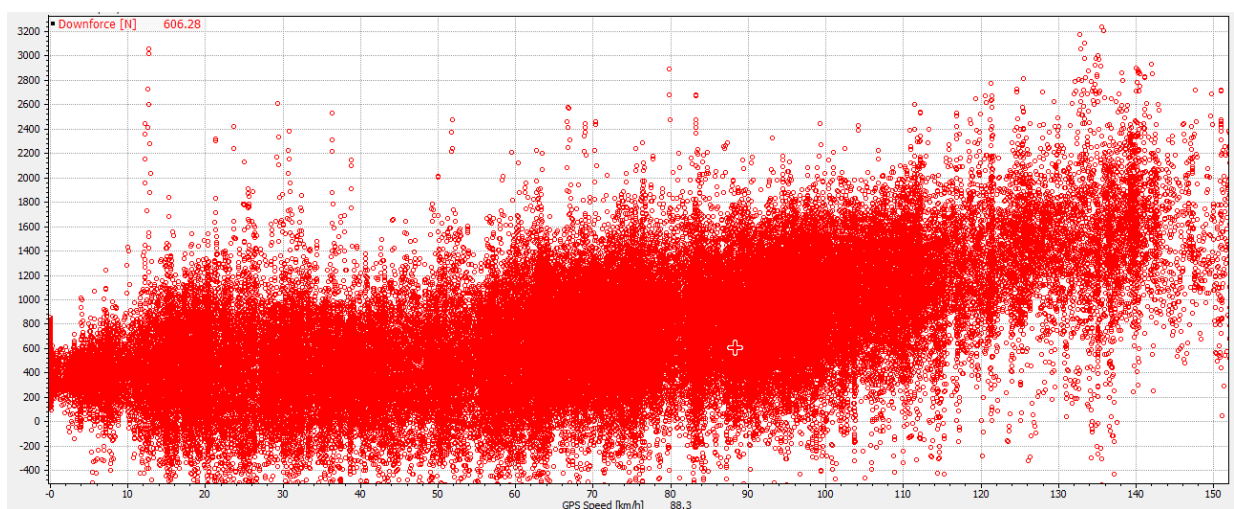


Figure 33 – Suspension force vs vehicle speed scatter plot.

6 Further Work

Due to the nature of the project, there were a number of additional ways to analyse and develop the understanding of the chassis behaviour. Unfortunately due to time restrictions and licensing factors, it was not possible to apply these methods to the analysis in this report. Even though these methods were not used in the final report, great consideration and research was conducted into how these methods could be applied to the analysis and what results and information could be acquired by the use of these methods.

6.1 Adams Flex

Adams flex is a program that allows standard FEA models to be integrated with the complete Adams simulation package, replacing previously assumed rigid bodies with flexible ones for more accurate simulation and prediction. Adams flex was previously mentioned in research as used by the University of Leeds in defining their chassis stiffness design targets for their formula SAE car.

Adams flex could be used to add a layer of analysis on how the MCR chassis is effected by torsional stiffness. Existing cad models of the chassis could be added to or adapted to investigate changes in chassis stiffness and mass. Adams flex allows easy transition from standard FEA packages such as Solidworks, meaning adjustments of the cad model can easily be uploaded back into Adams and analysed. Being able to validate original findings from Motec analysis with the Adams flex model could prove invaluable in predicting the handling characteristics of and modifications that could be made to the chassis in future.

Another project currently being run by the university is the adaption of the MCR chassis to accept Darrian cars bodywork. One of the main differences between the standard MCR and the Darrian MCR is the inclusion of a more rigid roll structure around the driver. It is safe to assume that the overall torsional stiffness of the chassis will be increased, and thus the chassis will behave differently with the same suspension setup as before and will also become more sensitive to changes in roll stiffness distribution.

6.2 Load Transfer Due to Non-Dynamic Steer Effects

A study conducted by Tim Tudor, UWTSU, was conducted to identify the effects of non-dynamic diagonal load transfer which occurs when the front wheels of a vehicle are steered about an inclined steer axis. It was stated that the inclined steer axis creates non symmetrical deformation of the contact patches of the front tyres, resulting in the inner suspension compressing and the outer suspension extending, creating a roll torque across the front axle. An opposing torque of equal and opposite magnitude (assuming a 100% stiff chassis) is created across the rear axle. (Tudor & Tudor, 2016).

7.0 Conclusion

Steady state handling data was used to gather data about the race car chassis and handling balance for a small range of configurations. Three test sessions were used to compare and validate the effect of chassis stiffness on wheel load distribution for the UIWTSD MCR S2000 car. The three test sessions were used to compare how the chassis reacted to an increase in roll stiffness and an alteration to roll stiffness distribution.

From the collected steady state data and accurate geometry measurements, weight transfer calculations were used to find load splits and thus load split distributions to analyse how the handling balance of the car reacted to changes. It is known from previous papers and books that a chassis has to be sufficiently stiff to react appropriately and predictably to changes in roll stiffness distribution. A race car is tuned by increasing the front roll stiffness to increase understeer and reduce oversteer, and vice versa to increase oversteer.

With the anti-roll bars disconnected, the roll stiffness of the car was lower than the torsional stiffness of the chassis which was suggested to be sufficiently stiff to react predictably to roll stiffness distribution changes.

Without anti-roll bars, it is difficult to change the roll stiffness distribution and therefore handling balance of the car. When reconnected, hard on the front and soft on the rear, the total roll stiffness of the car increased to 3676.77 Nm/Degree, while the chassis torsional stiffness remained at 2319.17 Nm/Degree. When the roll stiffness of a race car is increased in relation to the chassis torsional stiffness, the chassis will become less responsive to changes in handling balance.

The rear anti-roll bar was then set to its stiffest setting whilst the front anti-roll bar was decreased to its softest setting. This change shifted the roll stiffness distribution without changing the total roll stiffness. It was predicted that the front load split % of total load split distribution would reduce by 6.64% when in testing it was found to only reduce by 4.21%. This indicates that the chassis was only 63.4% responsive to changes in handling balance for the given ratio of chassis torsional stiffness/total roll stiffness equal to 0.63.

From the analysis, it can be said that the UWTSD MCR S2000 race car has a chassis stiffness lower than desired for responsive and effective changes to handling balance. It is however possible that the desired range of load split distribution is already available with the adjustment available to the front and rear roll stiffness'.

Whilst the data used for this report was sufficient enough to conduct analysis, it may have been more useful to have been present at the time of live testing to further understand how the data was collected and influence choice in the tests being performed, such as, providing more data in relation to roll stiffness and distribution, as well as, possibilities of changing the chassis torsional stiffness with the use of removable supports/braces. Access to simulation software and models would also have been of great analytical help to determine the full ranges of adjustability available as well as make changes to chassis stiffness. This could help find an optimum design specification for desired chassis stiffness for reduced weight and better predictability.

The data provided was enough to produce a simple answer which can be used as a baseline result for further investigations. As time progressed, more details and ideas became apparent. Unfortunately due to time constraints, it was not possible to follow up on many of them. More track data is needed to perform full analysis of the effects of chassis stiffness and should be the starting point for any future study on the UWTSD MCR S2000.

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