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Abstract

Race car damping is not discussed often at club level and is not widely reported as being utilised to improve handling balance or considered important enough to do more than to set it up after track testing and leave it. However it does contribute to handling balance and any and all methods of improving this should be understood and worked upon to gain an advantage over the competition. In this study, analysis is made of the contribution of springs and especially their dampers to handling balance, their role and involvement, and the set-up technique examined in a bid to improve lap times through better cornering stability. Both Koni and Penske have detailed methods of testing at the track and setting dampers by driver feel and if these methods are monitored through data acquisition then it will be possible to see what the driver feels through the sprung mass, shown on a data trace. With enough empirical data it may be possible to cut down on track test time through making simultaneous changes to damping or by making the adjustments via a simulation.

Introduction

Dampers to fit race car applications come in various brands, shapes and sizes and are fully adjustable. If money is available to buy the very best it would be of no help to a racer if they were set up incorrectly, and it has been reported by Staniforth (REF) that Lotus made money as race consultants on damping technology, proving racing gains can be made in understanding this area.

Recommendations on damper set-up are from Koni, a highly respected damper manufacturer. Their instruction is track testing and driver feel (see Koni damper set-up in the Appendix). This is both expensive and time consuming. If different spring and suspension configurations are to be trailed, optimised settings would vary with each track and conditions. It would not be practicable or possible to test all the possibilities and Segers (P100) suggests drivers have difficulty reliably determining high speed road inputs that cause tyre load variation and loss of grip. Due to these reasons, it is of benefit to race teams to match dampers to a vehicle using a numerical technique or simulation. There are guidelines provided by Kaz Technologies based on experience of what damper settings work and by visual analysis. These are a good method of gaining ball park figures, but final answers remain with the track test approach.

Better handling balance increases cornering speeds and decreases lap times. On a low down-force race car and neglecting tyres, changes to suspension dimensions, steering geometry, roll centre heights and centre of gravity position, weight and suspension springs alter handling balance. Therefore, this is of primary importance and the spring dampers are part of this, being adjusted to suit the best sprung set-up. Handling balance targets can be optimised through roll stiffness ratios

and change roll bar settings to meet it. Any spring change requires a change in damper settings to match the new spring values and control transition phases in cornering. In a bid to reduce track testing time, this paper aims to examine spring damping contribution to handling balance by analysing and assessing damper performance on a dynamometer test rig, utilising sampled values from track data to assess motion in real conditions and calculate the damping levels required to match springs used.

Literature Review / Background Research

Investigation of this subject has necessitated reading from a range of sources to develop an understanding of the engineering problem. This list represents the referenced material including books, papers and journal articles and some of the web sites which have provided suitable information on the workings of springs, spring dampers, their uses and the maths to understand their operation:

An Introduction to Racecar Engineering by Warren J. Rowley:

A discussion of chassis and tyre basics in great depth and their affect on vehicle handling characteristics. Additionally, aerodynamics and steering geometry are covered. There are explanations and formulae learned and test data from past race team projects, and there is also extensive coverage of the use of WinGeo3 by the late William C. Mitchell: a computer software package to demystify chassis geometry offering many set-up parameters including: the location of all roll centres, steering movement and wheel location relative to body movement. The description of tyre squish, spring formulae and difference between motion ratios and coil spring motion relative to wheel motion have been used for this paper among other ideas presented.

Competition Car Suspension by Allan Staniforth:

An exciting overview of the history and design of race car suspensions, but written mainly from a spectator's perspective. However, Chapter 8 – Weight Transfer contains a paper by David Gould and works through necessary mathematical formulae using the real dimensions of a self-built, race winning hill-climb car. Formulae and their explanations have been adapted for use in this paper.

Car Suspension and Handling by Donald Bastow & Geoffrey Howard:

This book is a very useful approach to the design of car suspension systems demonstrating the theory behind them and the first principles knowledge needed to apply calculations. It also describes and discusses spring and damper systems in detail, using worked examples. This book has been invaluable in providing formulae and calculation methods.

Analysis Techniques for Racecar Data Acquisition by Jorge Segers:

Both a description and application of data acquisition graphs to tuning a race car, and the information found in specific data traces explained, so a good companion for working with MoTeC. The advice on data available from spring dampers and their analysis has been used to influence spreadsheet calculations in this paper.

Race Car Vehicle Dynamics by William F. Milliken & Douglas L. Milliken:

The authoritative book on vehicle dynamics. Koni Damper Set-up technique is reproduced here as well as data regarding transmissivity of wheel motions to the sprung mass, and consequential relationship with tyre ground contact. An excellent background read for complete understanding of all motor engineering aspects and possibly the most cited book when examining sources used by other authors.

Race Car Engineering and Mechanics by Paul Van Valkenburgh:

A captivating book by a man who has been everything to everyone in his racing career, and shows the highest level of enthusiasm for applied science to a race car. A source of ideas, solutions and questions raised to give direction to investigative techniques, particularly useful in defining chassis and spring interaction. Also a useful reference in the application of telemetry data for tuning vehicles.

Damper Identification and Operation

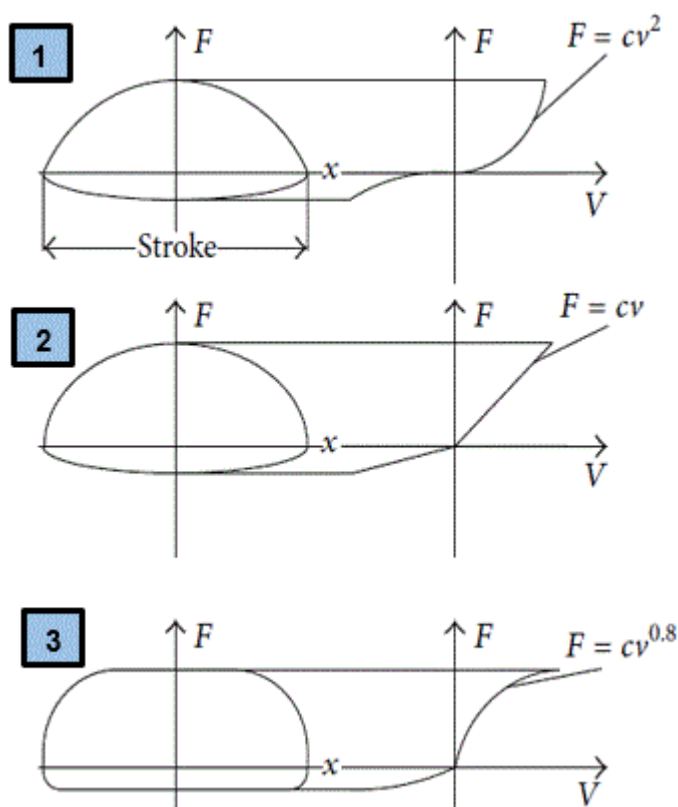
The spring damper type used for the M.C.R. is a Penske 8300 series which is a coil over monotube damper using both oil and nitrogen gas as damping media and having a separate canister connected via flexible hose to the main tube. The



separate canister contains mainly gas and enables the damper to have a shorter working length, which is advantageous when space is limited (KazTech – FSAE Guide). Gas is separated from the oil by a piston seal and is present to pressurise the oil, preventing cavitation from oil being forced through the small internal orifices of the damper in

turbulent flow. Bastow suggests (p66) the gas pressure improves Unsprung weight control but adds an increase in seal friction and increases the suspension spring rate and Rowley adds (Chap 14 -p30) that a force greater than the gas pre-load is required to set the damper in motion. These are negative effects of the design as Bastow states every effort must be made to eliminate friction (p55)

Damper inner workings are controlled by orifice sizing which, as the piston moves, only allows oil to pass through at a given speed before choking. Higher piston speed (Penske Racing Shocks, 2018) stack and springs system, opening valves allowing higher flow rates. The design has the advantages of a monotube damper in that it can dissipate heat well and the large piston diameter allows for low working pressures (BOSCH p616). However, limitations are discussed between users of the product concerning the flexible hose where the advantage gained by damper shortening adds uncertainty to its operation, which, owing to its construction, looks likely to occur in both low velocity bump and rebound. This means sprung mass control could be compromised by the design.



(Konieczny, 2016)

The damper design is double adjustable and regressive (3), which as the velocity of the damper increases, the force produced increases to a lesser degree as the indices are less than 1. Regressive damping allows high velocity, high amplitude chassis inputs, such as driving over curbs, to be resisted by little more force than road surface vibrations. This means less kinetic energy is transferred to the Unsprung mass, resulting in less fluctuation in Weight Transfers.

(Kasprzak, 2010)

Damper Role

The Spring Damper is required to:

1. Control resonant frequencies.
2. Control transient Weight Transfer rates (Kaz Tech)

The Spring Damper achieves these goals in the first instance by controlling the sprung and unsprung mass movements separately which requires a low and high speed damping rate. These movements need to be applied in both bump and rebound as there are differing requirements. The reason is that road disturbances add kinetic energy to a suspension only in bump, transferring energy to the spring system. In rebound no upward energy is present and the downward energy can be dissipated as heat from spring dampers as the wheel returns to position. Therefore less damping is needed in bump than rebound.

Simple Harmonic Motion affects wheels when they are displaced from equilibrium. They resonate with a natural frequency:

$$f = 1/T \quad (T = \text{Periodic Time for 1 cycle})$$

$$T = 2\pi\sqrt{M/S} \quad (M = \text{mass, } S = \text{spring rate}) \quad (\text{BASTOW P38})$$

The compound spring system includes springs, anti-roll bars and tyres and the sprung and unsprung masses can vibrate both in and out of phase. If the wheels are excited by a road disturbance at the natural frequency of the system, they may resonate resulting in loss of contact with the road surface. At system resonant frequencies, the sprung mass may also be provoked to vibrate, reducing ride quality. Bastow states the damping force to be directly proportional to the velocity difference between the sprung and unsprung masses.

Frequencies of motions must be controlled to keep a tyre in physical contact with the road, but if there is to be grip there must be minimal Weight Transfer and it should take place slowly. As the sprung mass contributes the largest amount to Weight Transfer, keeping it stable and level reduces fluctuation, maximising available grip. (Brisson, Unknown)

Bastow, D. & H. G., 1993. *Bastow, Donald & Howard, Geoffrey; Car Suspension and Handling – 3rd Edition 1993; Pentech Press Ltd, London UK & SAE International, Warrendale, Pennsylvania, USA*. 3rd ed. London & Warrendale: Pentech Press Ltd & SAE International.

Brayshaw D. L. & Harrison, M. F., 2005. Use of Numerical Optimisation to Determine the Effect of the Roll Stiffness Distribution on Race Car Performance. *Proceedings of ImechE*, 219(ProQuest Central p.1141), p. Part D: J. Automobile Engineering.

Brisson, M. G. & S., Unknown. *Tech Tip: Spring & Dampers, Episode Four*. [Online] Available at: http://www.optimumg.com/docs/Springs&Dampers_Tech_Tip_4.pdf [Accessed 29 April 2018].

Kasprzak, J., 2010. *Understanding Your Dampers - A guide from Jim Kasprzak*. [Online] Available at: <http://www.kaztechnologies.com/wp-content/uploads/2015/06/A-Guide-To-Your-Dampers-Chapter-from-FSAE-Book-by-Jim-Kasprzak-Updated.pdf> [Accessed 23 April 2018].

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Milliken, W. F. & M. D. L., 1995. *Race Car Vehicle Dynamics*. 5th ed. Warrendale, Pennsylvania: SAE International.

Penske Racing Shocks, 2018. *8900 Series SV 650*. [Online] Available at: <https://www.penskeshocks.com/product/8900-series-sv650/> [Accessed 23 April 2018].

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Staniforth, A., 1994. *Competition Car Suspension*. 2nd ed. Yeovil, Somerset, England: Haynes Publishing.

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Winkler C.B., C. K. a. M. C., 1990. *Variability of Centre of Gravity Height Measurement*, Warrendale, Pennsylvania: SAE International.

Appendices

Koni Damper set-up guidelines, annotated:

Bump Damping is to control the upward motion of the sprung mass and the motion of the unsprung weight from bumps on the track. It is not for the control of downward movement of the vehicle hitting dips and should not be used to control “Roll” or “Bottoming”. Ideal bump setting is vehicle dependant and could appear at any point in adjustment. The setting is found when the “side hopping” in a bumpy turn is minimised and the “Ride” is not unduly harsh. At any point different from the ideal setting, either the “side hopping” will be more pronounced or the “Ride” unduly harsh.

Bump Adjustment:

1. Commence with all dampers at full Soft.
2. Drive a lap or two to develop driver feel. Disregard “Lean” or “Roll” and concentrate on “Ride” harshness. Also note if the car “Walks” or “Side-Hops” on a rough turn.
3. Increase bump by three clicks on all dampers. Drive a lap or two. Repeat until car feels hard on bumpy surfaces. Back off two clicks to set.
4. The “Back-Off” point is most likely to be reached for one end of the vehicle first. If this occurs, keep increasing the bump on the soft end until it too feels hard. Then back it off two clicks.

Bump is now set.

Rebound damping is adjusted only after the ideal bump has been found. It controls corner entry transitional “Lean” - the time taken to reach a Roll-Angle. Too much rebound on either end of the vehicle will cause an initial loss of acceleration (cornering power) at that end – and will understeer excessively if this occurs at the front or oversteer excessively if at the rear upon corner entry. If too much rebound is applied in comparison to bump then “Jacking-Down” results, where successive bumps cause the sprung mass to lower on to its bump stops as the wheels have not had time to return to their neutral position before the next bump. Contact with the bump stops causes a large increase in roll stiffness – if this occurs on the front the car will understeer, if it occurs at the rear, the car will oversteer.

Rebound Adjustment:

1. Set bump. At full soft rebound drive one or two laps. Note how the vehicle “Rolls” when entering a turn.
2. Increase rebound damping by “Three Sweeps” ($\frac{1}{4}$ turn) at all four corners. Drive for one or two laps.
3. Repeat step 2 until car enters turns smoothly (no dramatic attitude changes) and without excessive “Lean”. To increase rebound beyond this point is unnecessary and may be detrimental.

EXCEPTION: It may be desirable to have a car with an over or understeering attitude at corner entry. This preference will vary between drivers and is to suit their driving style.

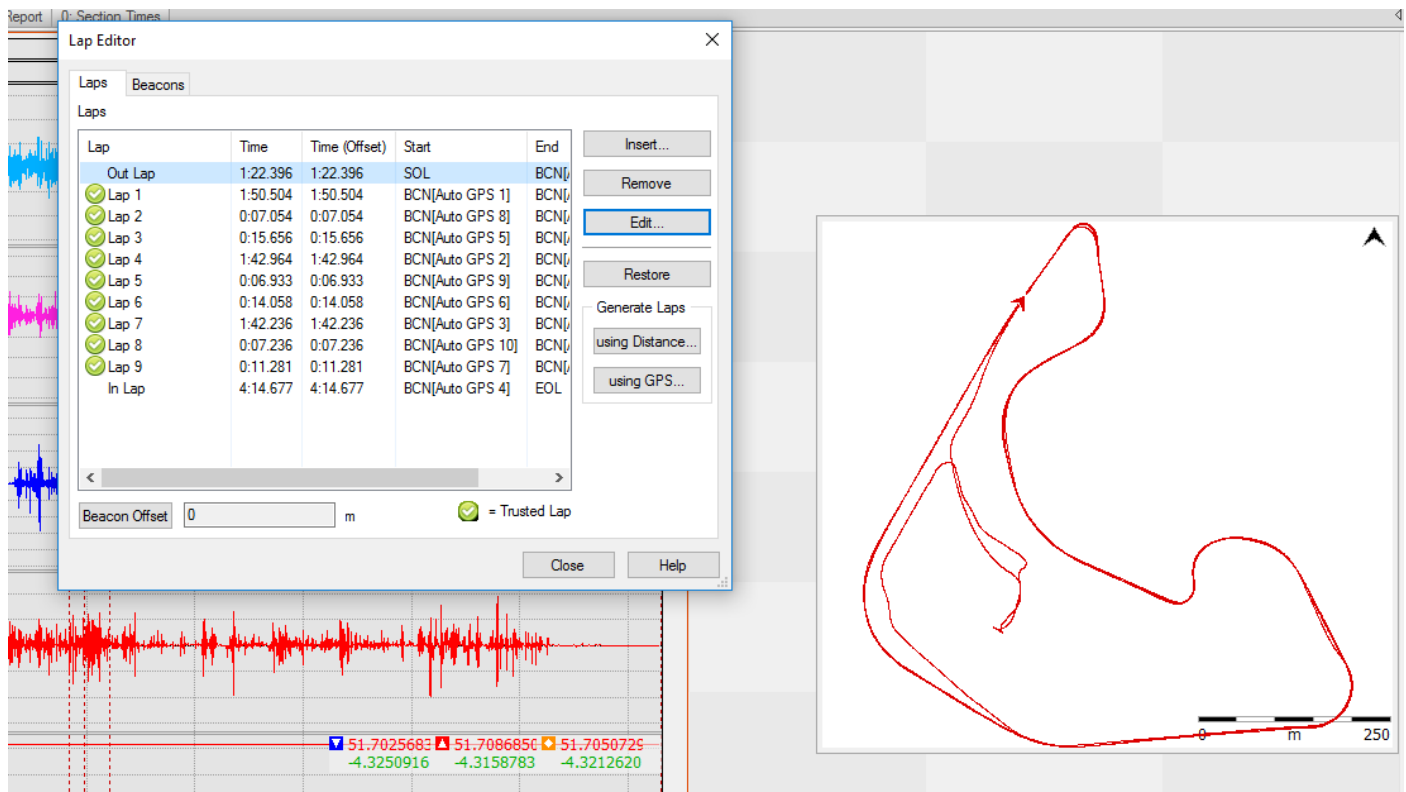
Installation of Reference Beacons on Pembrey Track in MoTeC

It is important to analyse data sets in a balanced way. When comparing samples of track data, the same piece of track is required in each data set to deliver representative samples. In MoTeC, this can be selected through using beacons to mark laps. A lap is then defined as any track length between two beacons.

To implement a beacon, GPS positioning can be used. A convenient location on the track is found in the traces of track data and the blue line cursor is placed on top. This enables a GPS map to be drawn in conjunction with latitude and longitude channels in a group on the “Graph” display tab.

Tools__ Lap Editor:

- With the “Blue Line Cursor” (the centre of a selected point of interest in the “Graph” tab) over start/finish of lap position, which is any point for reference and not necessarily the true position at Pembrey, a series of laps can be generated by selecting generate laps__using GPS. The tolerance due to accuracy can be changed. The default is to within 10m.



- Trusted Laps are generated which can be turned off to redefine laps. Beacons are defined by their geographical position, which is automatically selected on a separate drop down box if a selection is made for data to be generated from current position.

- Start/Finish Lap beacon selection:

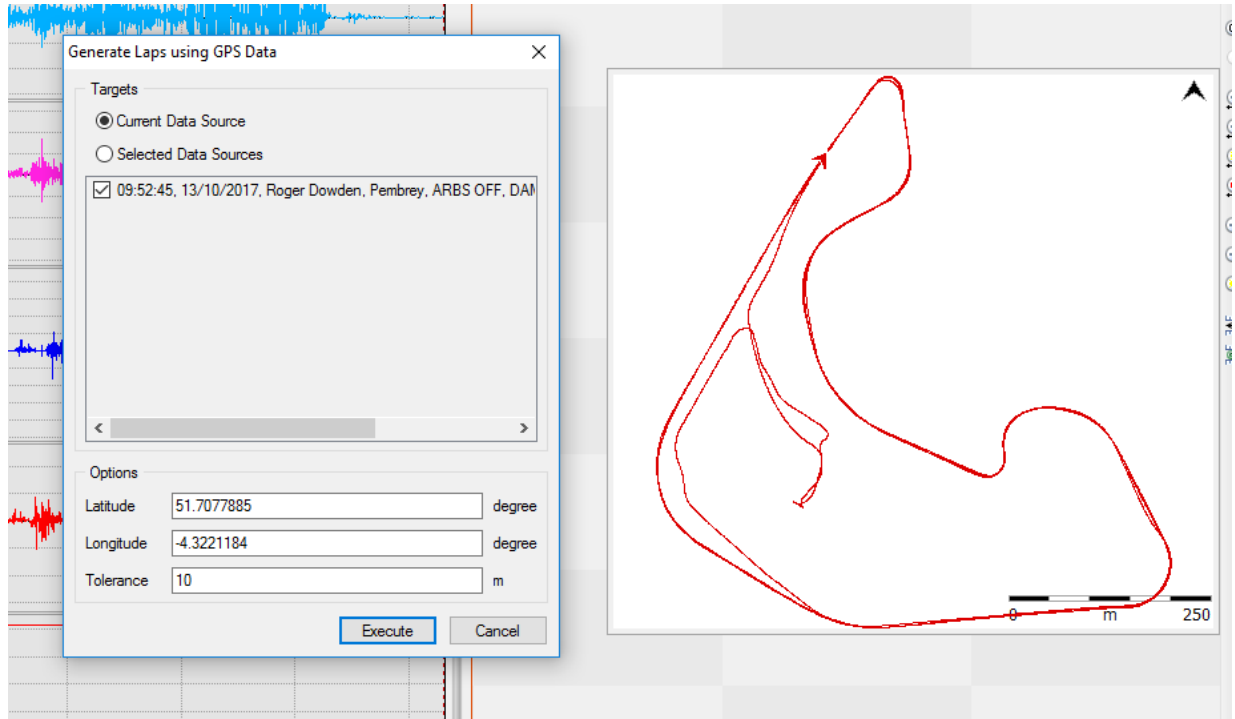
Latitude	Longitude
51.7077935	-4.3220926

The position must be the same every time to make the test fair.

- Restore button deletes all lap beacons.
- Remove deletes the lap beacon at the beginning of that lap e.g. select by highlighting a lap in the list. If the final lap beacon needs to be removed before the "In lap", then the "In lap" beacon must be selected. The result is the final lap is removed and the duration of "In lap" extended.

Tools__Lap Generator:

- This is useful for placing a beacon at the start/finish of a section of track for analysis. In this example case, beacons have been placed either side of “Honda Corner”.



- This gives the option of using the “Current Data Source” from the “Targets” section. It is important to position the “Blue Line Cursor” first to ensure the data source is correct as the track position can’t be manipulated with the box open. There is the option to move the red cursor track arrow on the track diagram but the application of this is limited to amount of data selected.

Methodology

This investigation examines the Weight transfer of an M.C.R. and how Spring Damping affects it at tyre contact patches. Weight Transfer is linked to the Roll Stiffness Distribution of the Sprung Mass which is dictated by the spring and anti-roll bar strength at each axle so these have to be calculated, and the forces transmitted to the damper by velocities of both the Sprung and Unsprung Mass have to be resolved. Both a hand calculated steady-state cornering model and MoTeC telemetry data have been used to analyse behaviour of the vehicle through Honda Corner at Pembrey under a variety of anti-roll bar strengths and changes to damper settings. Honda Corner has been selected for analysis as it is a high speed bend, allowing time to see the full extent of chassis movement in roll as well as reveal areas of transition. Damper rebound controls Roll rate (Milliken, 1995) total roll is determined by the point at which the springs Anti-Roll Moment resists the Roll couple of the Sprung mass Centre of Gravity acting about its roll centre.

The starting point for calculations is knowing the static masses and geometry of the M.C.R. so the Overturning moment can be calculated and Weight transfer quantified:

A list of calculations required for an ideal, steady-state cornering model of the M.C.R.. Ideas and formulae are presented by Staniforth (Race car Suspension p202 – 213) and Kaz Technologies FSAE Seminar (Kasprzak, 2010)

1. Axle masses i.e. Total front/rear masses, based on the longitudinal position of the Centre of gravity of the Sprung Mass.

2. Separate the sprung mass from unsprung mass. Find the Total sprung mass on each axle.

3. $\text{Unsprung mass} * \text{Cornering acceleration} * \text{Rolling radius of tyre} / \text{Track width}$

Calculate for each axle respecting difference in Track width and Wheel size at Front/Rear axles.

This is the Geometric Unsprung Weight Transfer.

4. $\text{Sprung Mass Weight Transfer via Roll Centres} = \text{Sprung Mass} * \text{Cornering Acceleration} * \text{Roll Centre Height} / \text{Track Width}$

Calculate for each axle. Data will be required from a chassis set-up program e.g. WinGeo3 for the Kinematic Roll Centres positions. The Roll Centre Height is lower on the M.C.R. at the front.

5. $\text{Sprung Mass at Rear Axle} / \text{Total Sprung Mass}$

This is the Ratio of Sprung Mass at the rear of the M.C.R. To get the front: $1 - \text{rear}$.

6. Calculate: $((\text{Rear track} - \text{Front track}) * \text{Ratio of Sprung Mass at rear}) + \text{Front Track}$

This is the Mean Track at the longitudinal position of Centre of Gravity, due to the front/rear axle Track difference. This is easiest to view from above.

7. Calculate: $((\text{rear Roll Centre} - \text{front Roll Centre}) * \text{ratio of Sprung Mass at rear}) + \text{front Roll Centre}$. This is the Mean Height of the Roll Centre above the ground at Centre of Gravity position and is easiest to see from the side.

8. If the Centre of Gravity height of the Sprung Mass is unavailable it can be located with reasonable accuracy by raising the height of front wheels of the M.C.R. as far as safe to do so with the rear wheels on scales. Calculate: $\text{Mean Centre of Gravity height} = (\text{Increase in weight on scales} * \text{Wheelbase Length} * (\text{Wheelbase Length}^2 - \text{The height difference of the front to rear wheels})^{0.5}) / (\text{Total Vehicle Mass} * \text{The height difference of the front to rear wheels})$ (**Van Valkenburgh, 2000**)

9. Calculate: $\text{Mean Roll Moment} = \text{Mean Centre of Gravity height} - \text{Mean Roll Centre height}$

10. Calculate: $\text{Sprung Mass} * \text{Mean Roll Moment} / \text{Mean Track for Centre of Gravity} = \text{Total Sprung Weight Transfer}$.

11. Calculate: $\text{Wheel Rate} = \text{Spring rate} * \text{Motion ratio}^2$

This is the hub centre rate without the tyre (**Rowley**). The roll-bar rate is added later and is affected by the motion ratio as its spring force is transmitted through the push-rod. The tyre is accounted for first.

12. Calculate: $1 / \text{Combined Spring Rate} = 1 / \text{Wheel rate} + 1 / \text{Tyre rate}$

Therefore: $\text{Combined Spring Rate} = (\text{Wheel rate} * \text{Tyre rate}) / (\text{Wheel rate} + \text{Tyre rate})$

13. The Roll-bar rate is added next:

$\text{Combined Spring \& anti-roll bar Rate} = (\text{Wheel rate} * \text{Tyre rate}) / (\text{Wheel rate} + \text{Tyre rate}) + \text{anti-roll bar Rate} * \text{Motion Ratio}^2$

14. Roll Rates are needed for front and rear axles.

The Roll Rate of an axle = $0.5 * \text{Combined Spring Rate} * \text{track}^2 + \text{anti-roll bar Rate} * \text{Motion Ratio}^2$

It is best to multiply the result by $\pi / 180$ to convert it from radians to degrees, giving a more visually acceptable answer as the M.C.R. produces small roll angles and degrees require fewer decimal places than radians.

15. Roll Stiffness Distribution at front axle = $\text{front axle Roll Stiffness} / (\text{front} + \text{rear axle Stiffness's})$

16. Roll Stiffness Distribution at rear axle = $1 - \text{Roll Stiffness Distribution at front axle}$

17. The amount of Sprung Mass Weight Transfer to each outer wheel in Roll can be calculated, hence the total amount of loading at the contact patch of each tyre known for steady state cornering. For a given axle:

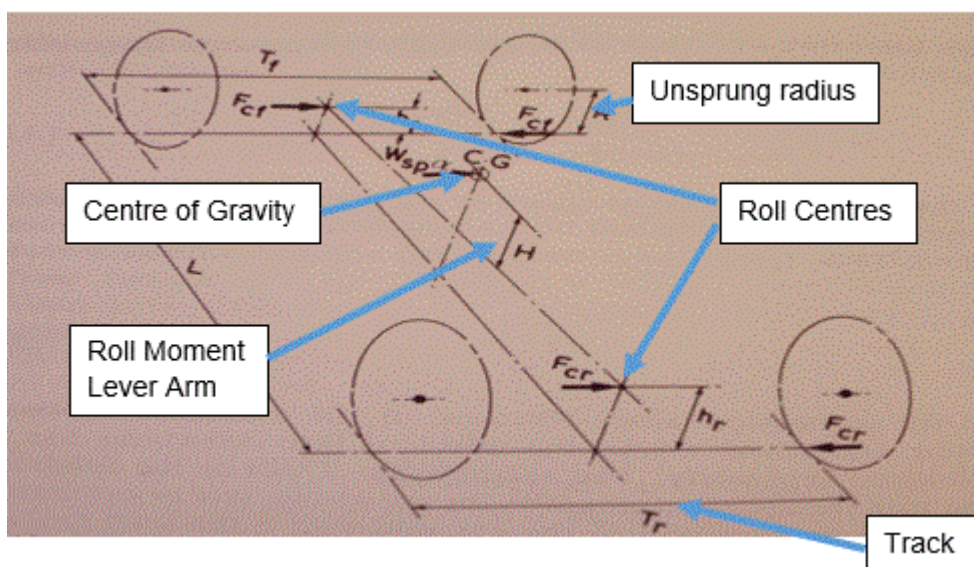
$(\text{Total Sprung Mass} * \text{Roll Stiffness Distribution}) + \text{Unsprung Weight Transfer} + \text{Sprung Mass Weight Transfer via Roll Centres} = \text{Weight Transfer to outer wheel}$

18. The individual tyre loads are calculated with respect to axles. The Weight Transfer comes off the inside and onto the outside wheels:

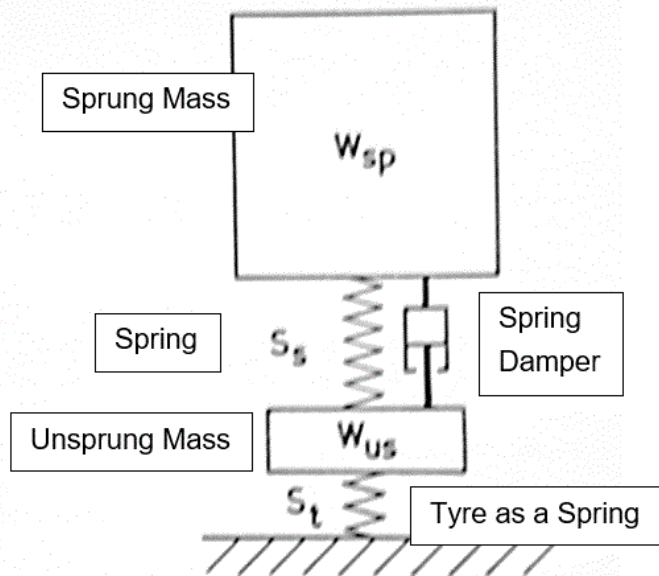
Outside wheel = $(\text{Total axle weight} / 2) + \text{Weight Transfer to outer wheel}$

Inside wheel = $(\text{Total axle weight} / 2) - \text{Weight Transfer to outer wheel}$

Assuming steady state conditions.



The spring, anti-roll bar and tyre need to be considered as a system with oscillations between the Sprung and Unsprung Masses managed by the Spring Damper. This can be represented by a quarter vehicle model.



(Bastow, 1993)

Roll Stiffness Distribution dictates Sprung Mass Weight Transfer at each axle, and as it provides the largest

contribution to Weight Transfer, control of it will change handling balance. Alterations may affect vehicle pitch attitude under braking and acceleration as well as yaw angle change from a tyre loads relationship with slip angles (Brayshaw D. L. & Harrison, 2005). The alteration of Roll Stiffness after set-up is easily controlled by adjustable anti-roll bars.

MoTeC telemetry:

The data traces are from true events, so they are able to give settings and the actual condition of the M.C.R. in Honda Corner. As Measurements for Sprung Mass Weight Transfer through push-rods are a given channel, the readings in the corner can be compared to static settings, showing the maximum dynamic Sprung Weight Transfer to the outside wheels. The ratio of Roll Stiffness Distribution is found for the Sprung Mass By comparing Transfers from front and rear axles, then Weight Transfers calculated using geometry and Lateral Acceleration are added. Wheel Displacement is calculated from total Weight Transfers appreciating the spring rates at the wheel which is inclusive of Tyre Squish, taken as a linear spring. Axle Roll is the difference in Wheel

Displacements across an axle over Track Length and the Roll Angle is Wheel Displacement difference per axle / Track. The average value is the Mean roll Angle.

The anti-roll bar strengths have been found through use of graphs which demonstrate spring deflection under loading. The graphs also show Tyre deflection. As the anti-roll bars could not be calculated, there needed to be a range of answers given, with the right answer between the extremes.

Anti-roll bars Calculation:

Weight Transfer and its locations are governed by the geometry of the M.C.R., its mass and centrifugal accelerations in Honda Corner. The maths used to find Roll-Bar strengths:

1. Roll Stiffness Distribution. This is calculated from the difference between static readings and push-rod forces when cornering in steady state and is the ratio between weights gained on the front and rear outside wheels when cornering (the push-rods only measure forces affecting the Wheel/Hub centre) from Roll motion of the Sprung Mass.
2. $(\text{Coil Springs} + \text{Roll-Bars}) * \text{Motion Ratio}^2$ give a Hub Rate.
3. $\text{Combined Rate} = (\text{Tyre Rate} * \text{Hub Rate}) / (\text{Tyre Rate} + \text{Hub Rate})$
4. $\text{Total suspension displacement} = (\text{Sprung Weight Transfers} + \text{Geometric Weight Transfers}) / \text{Combined Rate}$
5. $(\text{Spring} + \text{Anti-roll bar rate}) / \text{Damper movement} = \text{Force to compress spring and damper}$
6. $\text{Force to compress spring and damper} = \text{Force at hub} * \text{MR}^2$
7. $(\text{Sprung Weight Transfers} + \text{Geometric Weight Transfers}) - \text{Force at hub} = \text{Force reacted by tyre}$
8. $\text{Force reacted by tyre} / \text{tyre rate} = \text{tyre squish}$

As the anti-roll bar strength is unknown, a spreadsheet uses a range of values to plot a graph of calculated damper movement. For a given Weight Transfer, the hub and damper displacements can be calculated. A problem lies with the contribution of the anti-roll bar to a single wheel being based on its deflection at the opposite wheel on the axle. Its total deflection gives its total force absorption which should be split equally between each axle. This has caused some confusion with measurements as the deflection of the anti-roll bar may be required to be linked to the Weight Transfers at the opposite wheel to make calculations.

The maths above are calculations made in the Excel sheet "Roll-Bar Strength_MoTeC". Some data needs to be entered manually:

1. Firstly, data points from Push-rod Forces channels have to be selected by visual examination, to find the Static (the car at rest) and Dynamic Forces (when the M.C.R. is at steady state and at maximum Roll in Honda Corner). These need to be entered manually into the correct Wheel Position cells.
2. Lateral Acceleration must also be entered manually and has to be from the same instant of time the Push-rod Forces were recorded at Maximum Roll, to give accurate cornering readings.

The spreadsheet calculator gives no value for the Roll-Bars, but what it does do is aim to deliver an understanding of the amount of Spring Displacement to expect when the right amount of contribution from an anti-roll bar is added to the spring rate of a given wheel. This is completed by matching a calculated spring displacement to an actual one. Anti-roll bar strength cannot be calculated from the data, however the effects from a bar have been determined by modelling the suspension as a grouping of linear springs then using the Weight Transfer to determine reaction force. The deflections of each spring in the system have been predicted with the check being damper position. With a small modification to take into account wheel motions at the opposite side of a given wheel, it is possible anti-roll bar strengths could be predicted.

Damper Effects:

The damper motion and time can be observed in turns from data retrieved from Honda corner. It is shown along with Velocity, Steering angle and Weight transfer, as well as Lateral acceleration and Roll angles.

Damping level:

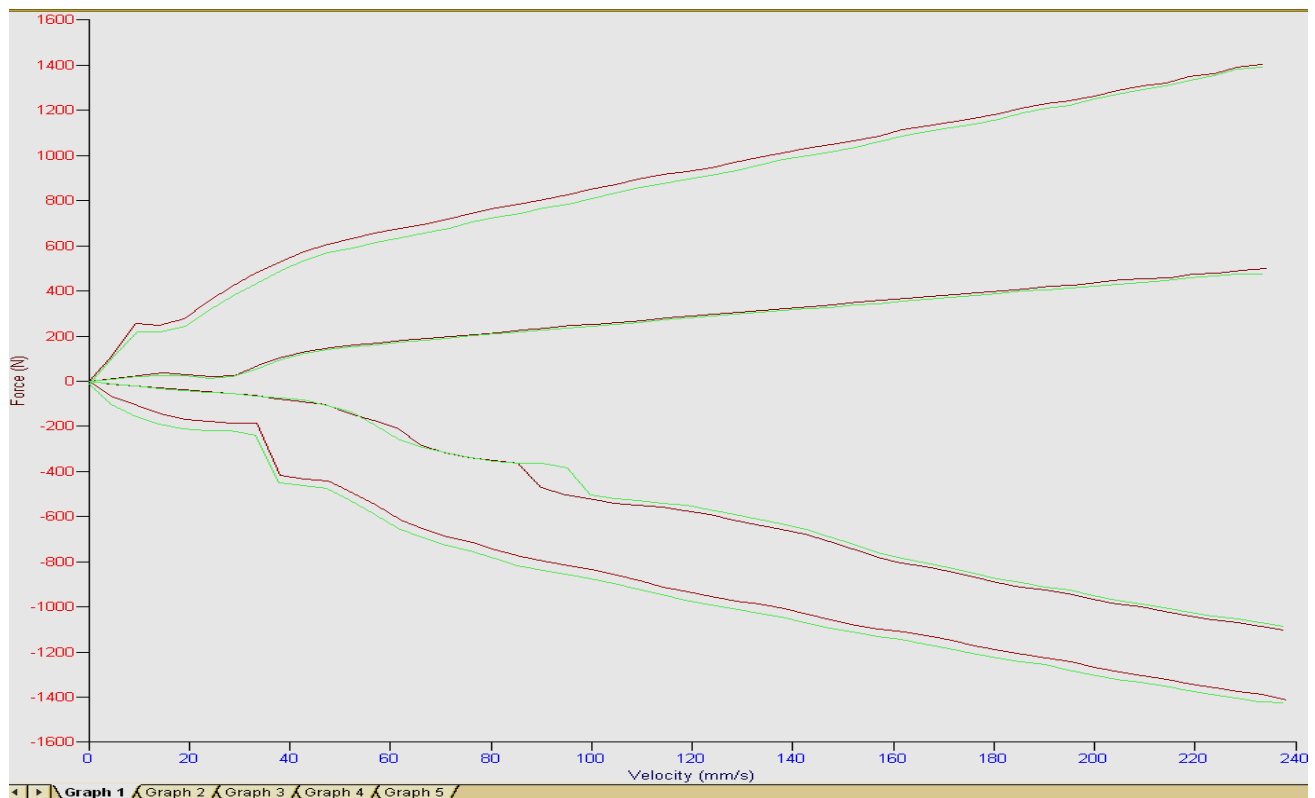
Dampers containing oil provide viscous damping and it is accepted as being proportional to the vertical velocity between the sprung and Unsprung masses and this works as a good approximation (Bastow, p55). Without careful valve control in opening and shaping, the piston speed vs damper force would be a squared function curve. The dampers used on the M.C.R. to retrieve telemetry data have been tested on a damper dynamometer rig to show the Force / Velocity and Force / Displacement at settings used on the track.

Results

Force / Velocity Graphs

All 4 of the M.C.R. spring dampers were tested at 3Hz on the same date by Paul Davies (experienced UWTSD Technician and ex. Koni Technician) so best quality and consistency is expected. An observation from any of the dynamometer test graphs at either soft or hard settings, reveals a very shaky line especially at low speed bump or rebound. However, any bias looks to be present in the machine or most likely, the mounting points, as the rear dampers are very nearly matched. Both knee points are in the region of 30mm/s and linear damping ratios are: Approximately 1:2.5 Bump to Rebound on Soft, & 0.7:1 on Hard settings.

The fronts are not as good as they showing signs of variation. Particularly the Left Front in bump on the soft setting, where the knee point is at a lower velocity. However, the appearance of the Force / Velocity graphs is that the settings for the front are almost tuned to a linear response for both low and high speed bump settings.



This example is of the left and right rear which are very well matched. There are some slight flaws in the mechanical transition between low to high speed settings.

There is no doubt that at low speed, there is a stepped reading appearing in all the graphs in almost the same place with exception to the left front which occurs sooner. This has no effect on the position of the knee point but in low speed rebound is almost a curve. Of the 4 dampers tested, the left front damper looks to exhibit the least resistance to motion. The Force / Displacement graphs show low speed stepped traces are common to all 4 dampers in the soft settings and one trend that is of concern: the high speed rebound varied and gave much greater resistance on the right than the left.

For the hand calculations, values of the springs and bars are taken from the M.C.R. Summary Data. When Roll stiffness distributions are calculated for similar bars the distribution is about 0.535 to front axle. The calculations performed are a close match to the method used in the Cranfield University paper (Brayshaw D. L. & Harrison, 2005) and Lecture Notes by Dowden, R from Year 1, Motorsport Engineering, UWTSU:

Roll Stiffness = $0.5 * \text{Combined Rate (No Bar)} * \text{Track}^2 * \text{Roll-Bar (inclusive of Motion Ratios)}$

Roll Stiffness Distribution:

Hard roll-bar at Front / (No roll-bar at rear + Hard roll-bar at Front) = 0.68

No roll-bar at Front / (Hard roll-bar at rear + No roll-bar at Front) = 0.384

Difference in Roll stiffness distribution across the range = 0.296

Analysis

Roll Stiffness Distribution

Using the M.C.R. Summary data it is of interest to note the widest range of Roll Stiffness Distribution possible in the set-up is under 30%. With reference to the Cranfield University test on a race car in Barcelona, their recorded data for variable Roll-Bars shows a range between 0.25 and 0.9. To be fair, theirs is a better environment: Barcelona G.P. track, a car of F1 dimensions where some speeds required calculating from 80m/s, and opportunity to participate in a potentially rule breaking experiment. Although they did suggest that “active suspension” meant electronically controlled which does not rule out a mechanical arrangement. A review of current rules affecting races in which the M.C.R. participates may be beneficial.

The Roll time at corner entry (Koni damper set-up guidelines) controlled by the rebound was the hardest to see on the telemetry data from MoTeC. It controls the rate of change of wheel loading due to sprung mass position, which affects Roll Stiffness Distribution at points in the turn differently due to the timing. So wheel load variation in this circumstance is best viewed from a distance measurement at the track, as when a section of corner or straight is analysed the position can be seen to have moved in relation to the sample length. Variations in track speed as well as adjustments to set-up and filtering applied during data analysis lead to uncertainty of results.

Reviewing hand calculations of a vehicle in Roll, they represent simplified, ideal versions of real world figures and so deviate from real results. This is due to rigid items bending and friction in joints

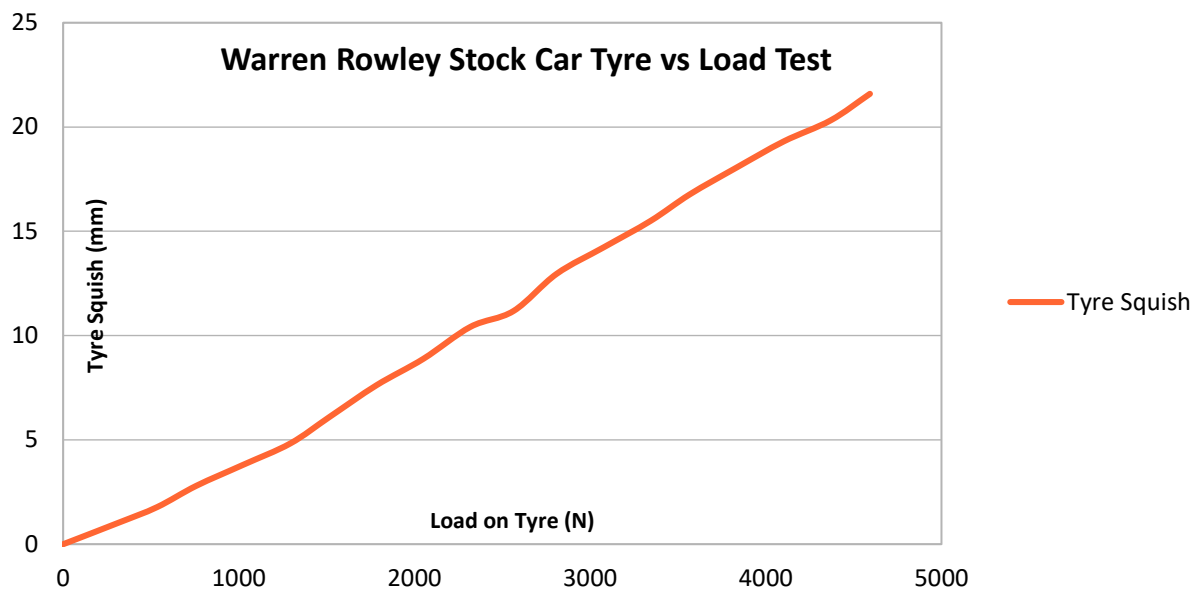
(Staniforth, 1994). They are also only applicable to steady state cornering which is not a true representation of driving (Valkenburgh p139). So the investigation to monitor and retrieve Roll-Bar data is flawed in many ways. However, the driver (Dowden, R) has owing to overlays and by his own admission, made best attempt to drive in a way closely representing steady-state, and as he stated: accelerated “normally” later.

Examining Rowley test data, radial tyres from a stock car were placed under an increasing load at 20psi and -2 degrees camber; results are tabulated. This was investigated initially to verify the Roll angle of the M.C.R. Sprung Mass.

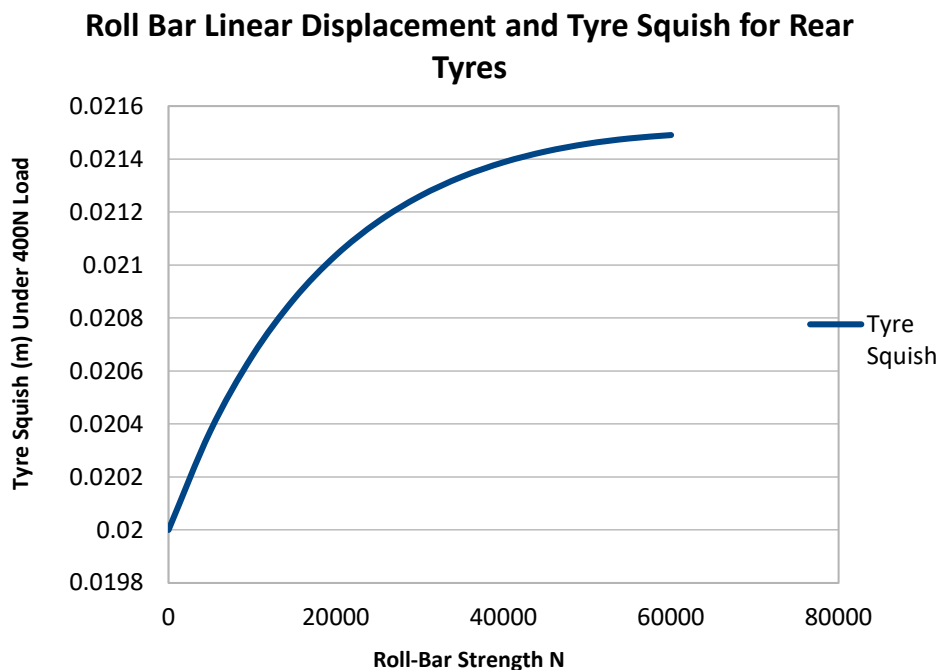
Rowley Table: Tyre Squish_20psi_-2 degrees Camber							
					Tyre Squish		
Load			Radius		Difference	Contact patch	
lb	kg	N	inches	mm	from original	Inch^2	mm^2
0	0	0	10.59	268.99	0	0	0
70	31.82	312.1542	10.55	267.97	1.02	3.5	2258
120	54.55	535.1355	10.52	267.21	1.78	6	3871
170	77.27	758.0187	10.48	266.19	2.8	8.5	5484
230	104.55	1025.6355	10.44	265.18	3.81	11.5	7419
290	131.82	1293.1542	10.4	264.16	4.83	14.5	9355
340	154.55	1516.1355	10.35	262.89	6.1	17	10968
400	181.82	1783.6542	10.29	261.37	7.62	20	12903
460	209.09	2051.1729	10.24	260.1	8.89	23	14839
520	236.36	2318.6916	10.18	258.57	10.42	26	16774
575	261.36	2563.9416	10.15	257.81	11.18	28.75	18548
630	286.36	2809.1916	10.08	256.03	12.96	31.5	20323
690	313.64	3076.8084	10.03	254.76	14.23	34.5	22258
750	340.91	3344.3271	9.98	253.49	15.5	37.5	24194
800	363.64	3567.3084	9.93	252.22	16.77	40	25806
859	390.45	3830.3145	9.88	250.95	18.04	42.95	27710
920	418.18	4102.3458	9.83	249.68	19.31	46	29677
980	445.45	4369.8645	9.79	248.67	20.32	49	31613
1030	468.18	4592.8458	9.74	247.4	21.59	51.5	33226
Total Squish:			0.85	21.59			

(Rowley, 2004, pp. Chapter 4 - page 35)

The first investigation made into suspension motion, produced very confusing results. Damper Displacements were analysed and considered as wheel position on a z-axis by multiplying by the suspension leverage. Using a solid wheel approach, the hub measurements allowed calculation of vehicle Roll-Angle in MoTeC to compare with the hand calculations at similar lateral accelerations. The outcome is a model with a less than rigid suspension, but does occasionally give meaningful results for a long bend like Honda Corner.

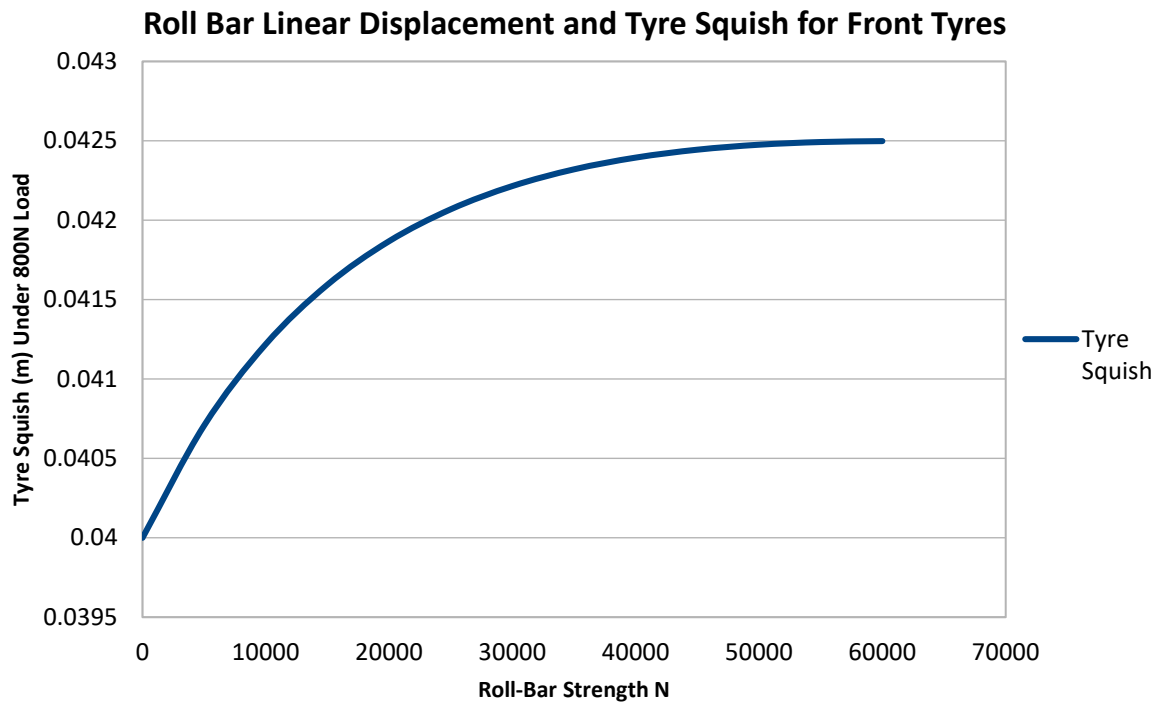


The Rowley investigation does at first glance seem to be useful for scaling but assumes only one spring of a single rate is compressed so cannot be used for comparison. It does however give validity to a tyre being considered as a linear spring.



Rowley's test uses Stock car wheels which are just over 21 inches in outer diameter so sized between M.C.R. front and rears, a -2 degree camber angle which is close to M.C.R. operating conditions and 20psi gauge pressure, like the M.C.R. Then loads it to nearly half a metric tonne and it does not squash any more than 22mm! The calculations made in this investigation show tyres deforming nearly twice as much for under 85kg loading. There was a significant flaw in the

first calculation method used, one of which was the spring rate which was misrepresented as 20,000N/m, when it should have been 256,000N/m. Hence, the M.C.R. tyres squash nearly 10 times as much.



From the Hand Calculations and Transferred MoTeC data as Tyre Squish at 1g, No Roll-bars:

$$76 / 20000 = 0.0038\text{m Front \& } 90 / 20000 = 0.0045\text{m Rear}$$

$$-29 / 20000 = -0.00145\text{m Front \& } 190 / 20000 = 0.0095\text{m Rear}$$

$$\text{Ride rate} = (\text{Spring} * \text{Motion ratio}^2 * \text{Tyre rate}) / ((\text{Spring} * \text{Motion ratio}^2) + \text{Tyre rate})$$

$$\text{Ride rate front} = (71765 * 0.81^2 * 20000) / ((71765 * 0.81^2) + 20000) = 14037.42$$

$$\text{Ride rate rear} = (80257 * 0.688^2 * 20000) / ((80257 * 0.688^2) + 20000) = 13102.16$$

Weight transfer via the Sprung mass through springs: 980N

Roll stiffness distribution for no Roll-bars: 0.537 Front, so 0.463 Rear

Weight transfer via the Sprung mass through springs: $0.537 * 980\text{N} = 526\text{N}$ Front, so 454N Rear

Weight transfer via the Sprung mass through springs / Ride rate:

$$526 / 14037.42 = 0.0374\text{m Front \& } 454 / 13102.16 = 0.0347\text{m Rear}$$

Total front outside suspension shortening: $0.0038 + -0.00145 + 0.0374 = 0.0398\text{m} = 39.8\text{mm}$

Total rear outside suspension shortening: $0.0045 + 0.0095 + 0.0347 = 0.0487\text{m} = 48.7\text{mm}$

This demonstrates how the decision was arrived at to quantify displacement in a combined springs system as shared spring deflections to find Tyre Squish. In this test case, if it was all a squashing tyre, the rear would have flattened nearly 2 inches. In Rowley's test example, he applies a load of 470kg (4611 N) directly to the wheel and tyre assembly, without a spring. The recorded Tyre squish is 22mm so it would imply his racing tyres are very rigid or the deflection expected should be very small. Sharing the Tyre Squish between the combined rear springs and the tyres as linear springs:

Hub Rate: 37989 N/m Tyre Rate: 20000 N/m This shows the Hub Rate to be nearly twice that of the tyre, and as a spring in series = 13102 N/m. So a force of 1000N would cause a deflection in the system of $1000 / 13102 =$ around 76mm. Assuming proportionality, this suggests approximately 25mm of movement should be taken up by the springs and 50mm by the tyre.

Reviewing the overall M.C.R. vehicle model, the way in which the chassis moves assumes total road contact even if the tyre and road surface are barely making contact, the road or surface is plane, the chassis is infinitely stiff. Taken with steady-state cornering, a fixed motion ratio and tyres that may be squashed or lifted from the floor, these are only a few of the simplifications causing erroneous results. Possibly the greatest factor was the smoothing of data streams to obtain usable data from the telemetry.

Discussion

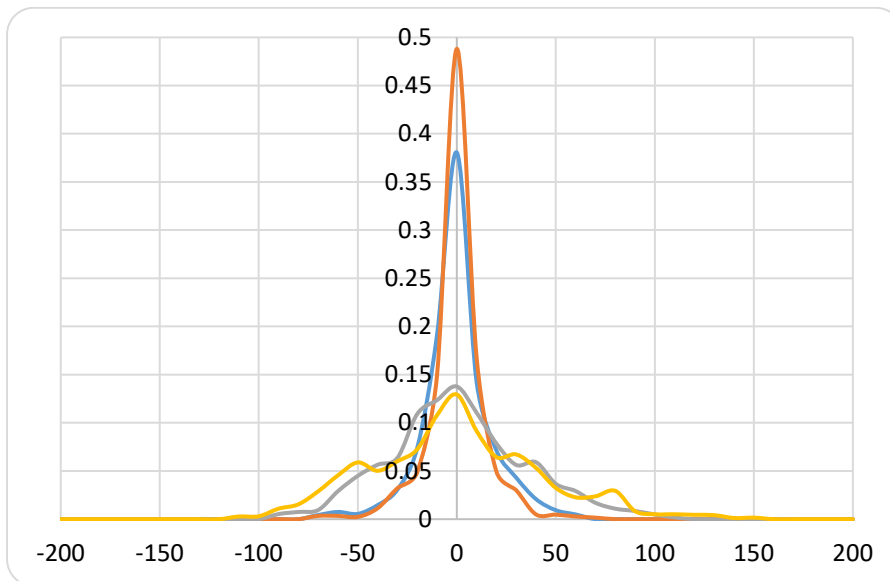
Although telemetry data was smoothed aggressively, it still yielded measurable results due to the length of Honda Corner, the principle track section analysed. If the track area examined been much smaller the higher resolution readings would be required. The recalculated Roll-Angle in maths channels is linked in part to Tyre Squish, which gives uncertainty to data and is likely to be increasing error. Many of the equations used rely on previous secondary calculations from raw data and the total number of calculation steps from raw data to answer is high.

Roll-bar values measured for the Summary Data were deemed unreliable so an attempt has been made to quantify their strengths at the different settings used. Bastow has given some guidance, stating anti-roll rate at the wheel should not exceed 50% of the suspension strength, as this will result in likely high roll excitation in Ride, if the car were to hit bumps or hollows on one side only. This would cause unnecessary discomfort by delivering high sideways accelerations to the drivers head. Armed with the upper limit values, the importance of knowing the correct strength of roll-bars has also been stressed in Use of Numerical Optimization to Determine the Effect of the Roll Stiffness Distribution on Race car Performance, in which an open wheeled race car, with similar

dimensions to an F1 machine, was fitted with a simple device/s which altered Roll resistance actively through the Roll-bars alone. The research was claiming improvement results of nearly 2.75 seconds per lap at Barcelona G.P. track. Even without active control and only a Quasi-Steady-State model, which does not come close to optimum driving techniques (Rowley and Valkenburgh concur), the advantages from using an "Optimisation" program, fed with minimal empirical data and working to find potential fastest times through track shape yielded significant improvements to lap times.

Spring damper rebound affects cornering transitions and the Koni track set-up guidance suggests that the driver should alter it only after setting the bump first and use the car handling as it enters turns to tune adjustment correctly. There are warnings against too stiff a setting: firstly too high for Weight transfer, where tyres are loaded too quickly so lose grip initially when entering a corner. This leads to initial understeer on the front or oversteer if on the rear axles. Secondly; in relation to the bump setting. If the ratio of bump to rebound is too low (small bump very large rebound) then "jacking-down" can occur, where the suspension springs do not extend before successive bumps cause the suspension to lower on to its bump stops. In the "jacked down" condition the suspension has a very high spring rate which on the front axle increases understeer by very high loading on front outside tyres, and on the rear results in a very unstable oversteer condition. However, a desirable effect occurs if used carefully on the front axle of a car having mild aerodynamic effects, where "jacking down" only occurs after specific speeds are reached, meaning that a car can have increased rake in straights with the right level of rebound at the front and recover by the time it is in the transition (Segers, 2008). So rebound tuning has a range of optimal settings which depend exclusively upon vehicle type and driver preference. Data for this set-up is purely qualitative in that the strength needs to be sufficient to prevent it from being worthless, and not so high that corner entry is more dangerous. Bastow states that in his worked examples of moment of inertia in roll, the sideways accelerations of a drivers head were calculated as 2.39 times that of forward and aft using representative figures (Bastow, 1993, p. page 306), so bearing this in mind it may be valuable to minimise sideways head motion through damper rebound control as this could also be a factor for reduced entry speed to a corner.

Results

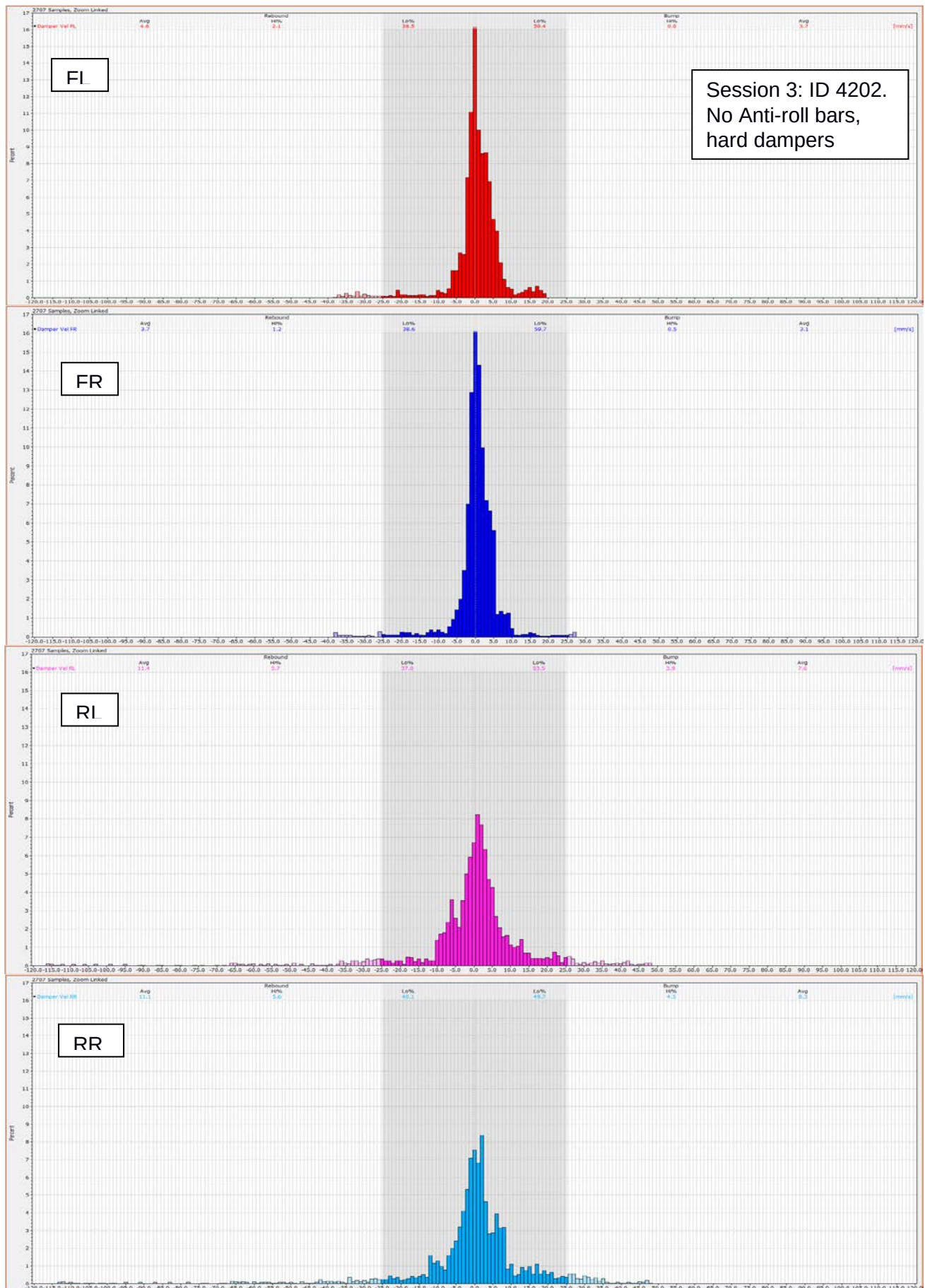


At present, it is possible to observe that the dampers were set up well when comparing their Bump against Rebound settings:

This sample was taken on the In lap leading out onto the track with dampers soft and no roll-bars.

The next page shows Session 3 I.D.- 4202: No roll bars with Hard Dampers. Although the scale is not easy to see, all are equal. Apart from a spike in rebound on the rear tyres (bottom of the page) all the dampers are slightly skewed towards bump being a softer setting than is necessary in comparison to the rebound. There are more low speed sprung mass motions there and fewer high speed vibrations. The rebound to the left has a better shape when referring to Segers recommendation of a normal distribution although it is still too tall and narrow. In comparison to bump it could be softened slightly to make the sides equal or the bump could be increased unless ride was becoming “unduly harsh” (Koni).

Looking at MoTeC histograms on one particular setting the spring rate at the rear axle is clearly too high and the peaks at zero movement have flattened. On the graphical data it will require a higher damping factor to control it, and from the Koni set-up guidelines, this is likely to take the car away from optimal settings. Roll Stiffness Distribution will be biased to the rear and Weight Transfer will happen there first – which shows on graphs of Front to Rear Load Split Difference, making them show curves as the inverse of a stiffer front axle.



Conclusion

From the data collected and damper set-up instructions, it is clear that “driver feel” can be measured through telemetry as a visual output. This means that a small amount of empirical data can be analysed and in theory, replace the need for extensive track testing, especially in the set-up of dampers. What is now required is some means of accurately quantifying the trends to determine exactly how much of an alteration is needed to get close to the “right” figures, so as Segers states: give confidence in the accuracy of the data.

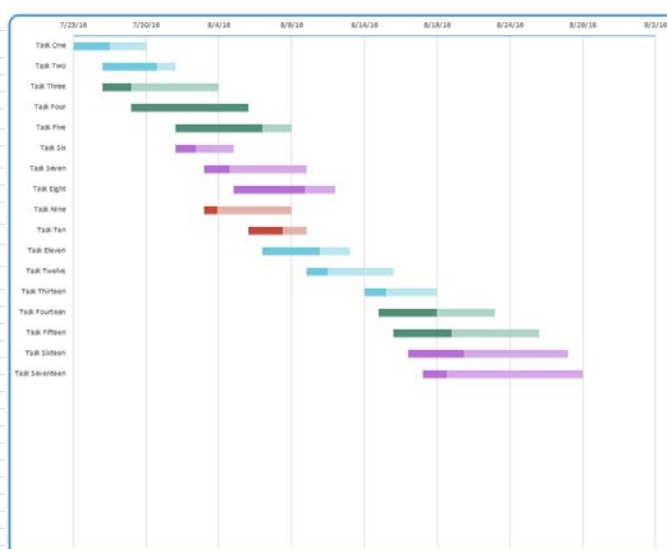
Evaluation

Looking at damping for telemetry: In “A Guide to Your Dampers” by Jim Kasprzak (Kaz Technologies) he states the need to have the damper displacement sensor set to 250Hz or 500Hz preferably. Also that the damper displacement should not be filtered before converting to damper velocity, warning that both practices result in substantial error. In light of this, a large amount of filtering was required to remove an 8Hz interference, which got progressively worse the more hard settings were used on either the roll-bar or the damper. As the channels were required for Weight transfer in a Honda corner sample it seemed reasonable to smooth them right down as the transitions and transfers had no requirement for instantaneous results in close proximity to each other and, excluding the Histograms, there has been no further need for pin point accuracy in that area. Most of the data requirements were to find a point of steady-state, where the Weight transfer was on the outside wheels alone. The method of modelling the M.C.R. from the contact patches was a corruption of ideas: a reverse of David Gould's calculations (Competition Car Suspension – Allan Staniforth), blended with the Tyre squish, spring formulae and Dr Juan Salinas ideas around a vehicle in contact with a plane surface (An Introduction to Race Car Engineering, Book 1 – Warren Rowley) and the ideas presented by Donald Bastow in Weight transfer where the Tyre squish was worked out partly by addition of Roll angles caused by the sum of the different types (Bastow, 1993). Results were within reason.

When considering the histogram data the dimensions of the area to measure to find a perfect Gaussian Distribution are dictated by the height and width of the area chosen for analysis. On one hand, data must be represented in a space of equal size at all times so variations between graphs can be seen, regardless of the inputs and outputs a standardisation and best fit approach is

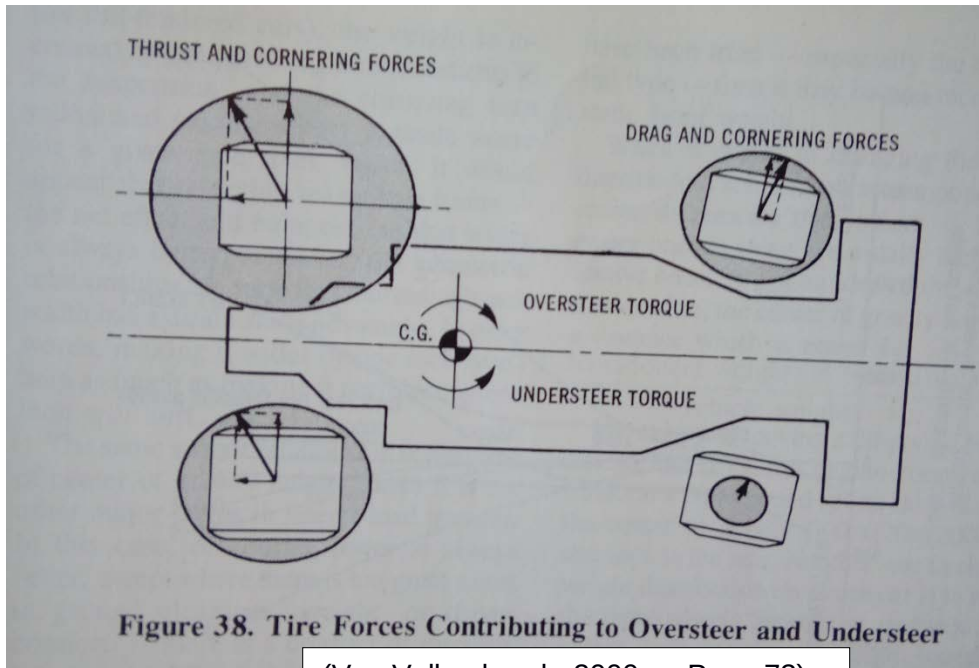
One of the hardest tasks, apart from writing the Math Channels for MoTeC then trying to remember which were no longer in use, was to find an acceptable calibration technique to start suspension motions from zero. The best way would be to possibly leave the numbers open at the top of the Maths Channel for altering to suit closest zero reading, giving the option for a user to find and change them quickly for any track session if the data appears skewed. Initially, it looked best to take the readings from when the car was stationary, but as the Damper Displacement Channels had so much noise, the readings could have been from any part of a fluctuating trace and although they were quite accurate that way, a more reasonable way seems to be to select numbers that can average zero movement for specific track straights and check it against Suspension Force to make sure it is balanced and any resulting increases from ride changes visible.

There are a number of significant findings, some of which, like the histograms, only give the relationship between bump and rebound to see if the movements are balanced. The vehicle model in Roll at Steady-state in MoTeC is a good start point for refinement and might give a reasonable view of anti-roll bar strength, even if the data traces have to be smoothed.

[illegible]

The Gantt chart shows the level of time devoted to each task having past. It does not account for the number of times processes were revisited subject to either personal displeasure with the output, or a new direction or idea being trialed.

Further Work



(Van Valkenburgh, 2000, p. Page 73)

Before detailed further analysis of dampers, Roll-Stiffness Distribution needs to be investigated with tyre loading behaviours, frictional drag resistances related to the tyres and yaw with tyre slip angles, to gain benefits from this type of chassis tuning. When considered

together, the best decisions can be made on how to optimise a vehicle in turns. From the evidence provided by Cranfield University's track testing the aim was clearly to develop a car to have best handling characteristics and ultimately be quicker in corners, which even worked well with the aerodynamically driven design of formula car that was used in testing. Personally, low hanging fruit would be tyre pressures, friction models and the change in rolling radius due to Tyre Squish and even to include optimisation of the tyres use as springs in circuit racing. The dampers would then be set-up to compliment gains made in other areas.

Dampers possess some resistance to motion and it may be useful to investigate different types of damper available for the M.C.R. and see if any are available for testing. Having the least motion resistant damper on the car may make damper tuning easier to investigate. There are sources of errors that cannot be located at both the damper test and the MoTeC data, and sensitivity in the MoTeC histograms suggesting a lack of low speed movement even when the settings are soft. This would benefit the university by achieving better quality modelling of weight transfer for their M.C.R. Darian race car, give a more accurate understanding of base set-ups and alterations, the more time can be spent utilising better data ranges in driver simulation or track testing. A working model of high accuracy for WinGeo3 would be useful to optimise as many parameters as possible in relation

to tyres. Then Wan31A (Rowley/Salinas) could be used to work through transient corner phases so all the sprung elements with Roll-bars could be seen and optimised at each section including the damper and then check it against telemetry data to see if improvements have been made and the correlation between modelled data in its different forms.

The obvious low hanging fruit delivering greatest benefit in the short term would be to model and do simulation work with the anti-roll bars. This would also include a program in Matlab to analyse the variables given by different track shapes, converting it to a series of curves and straights to give the answer of the best anti-roll bar set-up for Steady-state conditions. If the track is hard to reproduce, MoTeC has a very user friendly version of track analysis which may be able to be broken down and transferred as an image, which could be viewed in the section values given in MoTeC and colours given to the different sections so MatLab would be able to distinguish between corners and straights and recognise them. Grouped data would need to be analysed to match estimated cornering durations and accelerations for straights with lap times. Refinement could be made by minor modifications in chassis set-up for specific corners. In Brayshaw, D.L. & Harrison M.F.'s paper, vehicle attitude angle under braking was steeper and yaw accelerations higher as more of the tyres capabilities were realised through use of a roll-bar optimisation method. Although this may be inspired by the Brayshaw and Harrison investigation it will contain inventive steps in being applied to an M.C.R. and the track capable of being imported from MoTeC data.

Bibliography