Vehicle Dynamics Simulation with ChassisSim

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Introduction

Motorsport is a huge source of entertainment, and in one end, money. In 2019, Mercedes-AMG Petronas Formula One Team has spent almost half billion dollars on its path to win the F1 championship (Perez, 2020).

On the other end of the motorsport industry there are a bunch of categories with way lower budgets but equal, if not even higher, levels of entertainment (O´Leary, 2016). A shared characteristic among almost all the categories is the very competitive ambient where each tenth of second matters, and there is where the vehicle dynamics simulation software has a huge impact.

The vehicle dynamic simulation software has changed the way that the car is prepared for a race. Via a series of hundreds, or even thousands of interactions within the software, the car can hit the race track with a setup already very close to the optimum.

As an example of how the optimum car setup is key, on the 2nd round of the 2020 Austrian Grand Prix of Formula 2, the difference between the 1st place and 3rd place in qualify was 0.085s. In an environment where every single point matter, the use of a vehicle dynamics simulation tool can be the difference between winning or losing the championship.

One of the vehicle dynamics simulation software that stands out due to its relative low cost and high precision is the ChassisSim, which was the software used for the development of this work.

Definition of the Scenario

The starting point was based in a request from a racing team, the ChassisSim model of the car used as a baseline and a MoTeC log file, where the curvature line and bump profile could be extracted using ChassisSim. This is necessary because for the creation of a particular race track on ChassisSim, these two files are necessary. The complete request can be found on the Appendix.

On the request, the race team declares that the currently lap time is 1:22:685 and a target lap time is around 1:22:300. The path for lap time reduction should be solely based on the suspension geometry variation and analyses of the damper behavior, which was evaluated via the ChassisSim 7 post rig simulation tool and data comparison in the MoTec.

A 7 post rig test can reproduce all the bumps and vertical movement that a vehicle faces on a real lap with the benefit of also introducing the aero load that further compress the suspension.

The car modeled on this assignment is an open-wheel type. This model of car has no covers on its tyres, it is normally light on weight (when compared to a tourism car) and its engine is mounted on the back portion of the car. This basic configuration covers a wide range of racing series, that goes from Go-karting to Formula 1. More information about the car can be found on the appendix.

The track which this work is based is the Bruce Mclaren Motorsport Park, located in New Zeland. The record lap time on this track for a racing car was established by Niko Hülkenberg (2007) driving a Lola B05/52-Zytek (https://www.brucemclarenmotorsportpark.com/tracks/).

Initial Calculation

Before starting with the calculation, some forces and direction of displacement on different parts of the system that will be analyzed must be set.

Figure 1 shows a simplified model where the mass of the body (mass of the vehicle) and the mass of the tyre are connected to the ground. It also shows the damping rates and spring rates among them as well the direction of the movement (signalized by the red arrows). This investigation of one corner of the vehicle is known as ¼ Car Model.

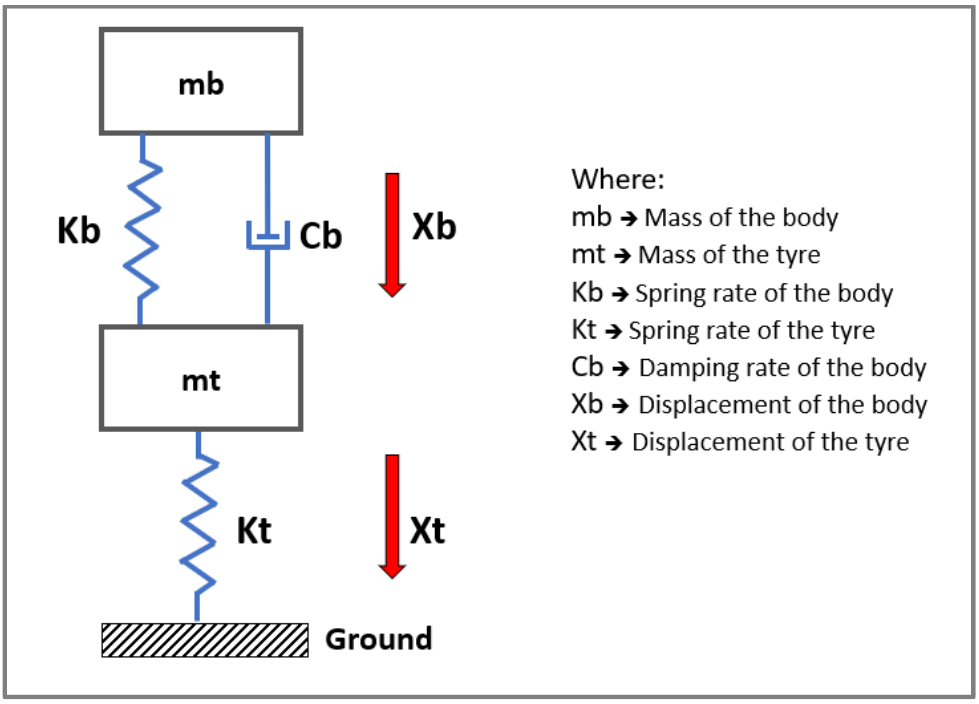


Figure 1. ¼ car model.

Xt and Xb are assumed negative when its displacement is towards the ground. The direction of the movement that will be considered as negative has no impact on the results, however once a direction is assumed, it must be consistent over the whole calculation.

Figure 2 shows the forces applied on both masses within the system:

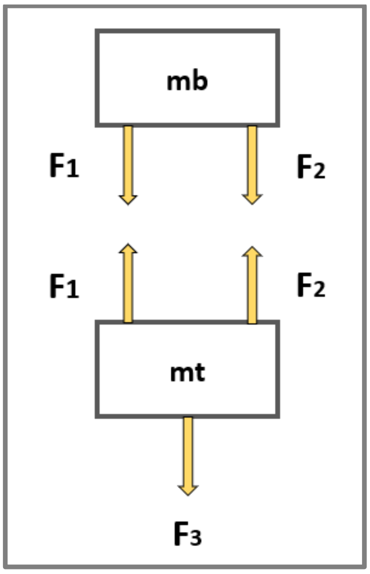


Figure 2. Forces applied within the system.

The masse defined as mass of body and mass of tyre can also be defined as sprung mass and unsprung mass. Sprung mass is the mass of the vehicle being suspended by the suspension (chassis, engine, gearbox, driver, fuel, etc.) while unsprung mass is the mass not suspended by the suspension (brakes, tyres, wheels, etc.).

The first step is the calculation of the spring force, given by equation 1:

$$Spring\ Force = K * X \qquad \text{(Eq. 1)}$$

Where:

K= Spring rate

X= Displacement

And the damper rate, defined by equation 2:

$$Dumper\ Rate = C * \dot{X} \qquad \text{(Eq. 2)}$$

Where:

K= Damper rate

$\dot{X}$= Velocity

Based on the figure 1 and 2, equation 3.1, 3.2 and 3.3 can be stated as:

$$F1 = Kb * (Xb - Xt) \quad \text{(Eq. 3.1)}$$

$$F2 = Cb * (\dot{X}b - \dot{X}t) \quad \text{(Eq. 3.2)}$$

$$F3 = Kt * (Xt - Ground) \quad \text{(Eq. 3.3)}$$

Nevertheless, whenever the ground can be considered static (most of racing series), equation 3.3 becomes equation 3.4:

$$F3 = Kt * Xt \quad \text{(Eq. 3.4)}$$

Equation 4 shows the well know Newton´s second law of motion:

$$F = m * \ddot{x} \quad \text{(Eq. 4)}$$

Where:

M= Mass

$\ddot{x}$= Acceleration

Rearranging equation 4 with the data presented on the figure 1 and 2, equation 5 states for the tyres that:

$$F = mt * \ddot{x}t = F1 + F2 - F3 \quad \text{(Eq. 5)}$$

On the equation 5 the term "F3" is negative as it was earlier defined that the movement towards the ground is negative.

While equation 5 is related to the tyres, equation 6 relates the force and acceleration on the vehicle body:

$$F = mb * \ddot{x}b = -F1 - F2 \quad \text{(Eq. 6)}$$

Combining equation 5 with equation 6, the equations of motion (for tyre and body) can be descried as equation 7.1 and 7.2:

$$mt * \ddot{X}t = Kb * (xb - xt) + Cb * (\dot{X}b - \dot{X}t) - Kt * (Xt) \quad \text{(Eq. 7.1)}$$

$$mb * \ddot{X}b = -Kb * (xb - xt) - Cb * (\dot{X}b - \dot{X}t) \quad \text{(Eq. 7.2)}$$

The effective spring rate is a useful approximation based on the ¼ Car Model, illustrated here by equation 8:

$$Keq = \frac{Kb * Kt}{Kb + Kt} \quad \therefore \quad \%Xb = 100 * \frac{Kt}{Kb + Kt} \qquad \text{(Eq. 8)}$$

Springs and Dampers Calculation

Based on the ¼ Car Model an initial suspension calculation regarding natural frequency, damping ratio and the weight distribution can be performed.

Equation 9.1 is the natural frequency calculation with results in Hz while the equation 9.2 has its result in rad/s

$$\omega = \frac{\sqrt{\frac{k}{m}}}{2\pi} \qquad \text{(Eq. 9.1)}$$

$$\omega = \sqrt{\frac{MR^2 * Kb}{mb}} \qquad \text{(Eq.9.2)}$$

Where:

ω= Natural frequency

k= Spring rate

m= Mass

MR= Matio ratio among wheel and damper

The necessary input data for the natural frequency calculation can be found on the provided ChassisSim model, shown by figure 3.

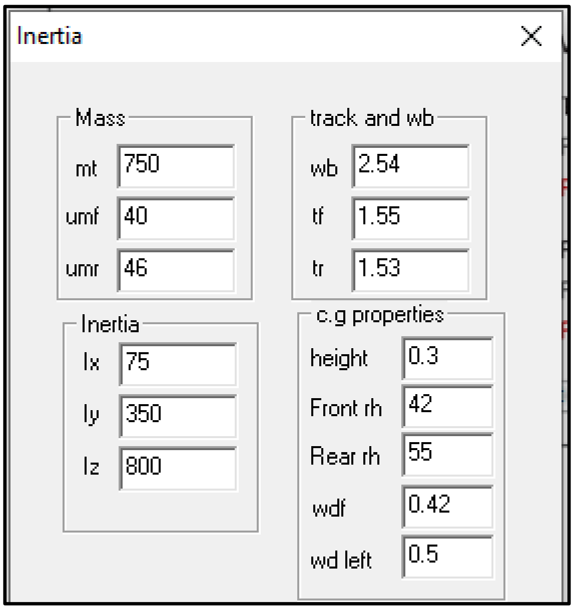


Figure 3. Vehicle mass distribution as modeled in ChassisSim.

The data found on the figure 3 can be summarized as:

mt= Total mass (kg)

umf= Unsprung mass front (kg)

umr= Unsprung mass rear (kg)

wdf= Weight distribution factor at front axle

wdl= Weight distribution factor at left side

Based on the data available on the figure 3, the weight distribution on each corner of the car can be calculated as shown in equation 10.1, 10.2, 10.3 and 10.4:

$$mfl = mt * wdf * wdl \qquad \text{(Eq.10.1)}$$

$$mfr = mt * wdf * (1 - wdl) \qquad \text{(Eq. 10.2)}$$

$$mrl = mt * (1 - wdf) * wdl \qquad \text{(Eq. 10.3)}$$

$$mrl = mt * (1 - wdf) * (1 - wdl) \qquad \text{(Eq. 10.4)}$$

Where:

mfl= Mass at front left corner

mfr= Mass at front right corner

mrl= Mass at rear left corner

mrr= Mass at rear right corner

Replacing the equations 10.1 to 10.4 with the numbers from the ChassisSim car model, the weight in each corner of the car can be found:

$mfl = 750 * 0.42 * 0.5 = 157.5\ kg$ (Eq.10.1)

$mfr = 750 * 0.42 * (1 - 0.5) = 157.5\ kg$ (Eq. 10.2)

$mrl = 750 * (1 - 0.42) * 0.5 = 217.5kg$ (Eq. 10.3)

$mrl = 750 * (1 - 0.42) * (1 - 0.5) = 217.5kg$ (Eq. 10.4)

With the clarification of the mass distribution around the corners of the vehicle, for the natural frequency calculation, it is important to subtract from each corner the unsprung mass as this mass will not play a hole on the natural frequency of the body. Equation 10.1.1, 10.2.1, 10.3.1 and 10.4.1 shows how it is calculated:

$smfl = mfl - usmfl$ (Eq. 10.1.1)

$smfr = mfr - usmfr$ (Eq. 10.2.1)

$smrl = mrl\ \ -usmrl$ (Eq. 10.3.1)

$smrl = mrl - usmrr$ (Eq. 10.4.1)

Where:

smfl= Sprung mass at front left corner	usmfl= Unsprung mass at front left corner
smfr= Sprung mass at front right corner	usmfl= Unsprung mass at front left corner
smrl= Sprung mass at rear left corner	usmfl= Unsprung mass at front left corner
smrr= Sprung mass at rear right corner	usmfl= Unsprung mass at front left corner

Considering that the unsprung mass is equally distributed on each axle (50% for each side), based on the data available on the figure 3, it is possible to calculate the unsprung mass as following:

$smfl = 157.5 - (40 * 0.5) = 137.5$ (Eq. 10.1.1)

$smfr = 157.5 - (40 * 0.5) = 137.5$ (Eq. 10.2.1)

$smrl = 217.5 - (46 * 0.5) = 194.5$ (Eq. 10.3.1)

$smrr = 217.5 - (46 * 0.5) = 194.5$ (Eq. 10.4.1)

With all the masses calculated and considering the motion ratio among wheel and damper from the modeled car (data available within ChassisSim model), the equation 9.2 can be calculated as following:

$$\omega f = \sqrt{\frac{MR^2 * Kfront}{smf}} \qquad \text{(Eq. 9.2)}$$

$$\omega f = \sqrt{\frac{0.854^2 * 192639}{137.5}} = 32.0 \text{ rad/s or } 5.1 \text{ Hz}$$

$$\omega r = \sqrt{\frac{MR^2 * Kr}{smr}}$$

$$\omega r = \sqrt{\frac{0.89^2 * 218908}{194.5}} = 29.9 \text{ rad/s or } 4.8 \text{ Hz}$$

Where:

kf= Spring rate front

kr= Spring rate rear

ωf= Natural frequency front

ωr= Natural frequency rear

When a change of the natural frequency in one or both axes becomes necessary, the needed spring rate can be calculated rearranging the equation 9. Considering a target of 4.5 Hz in the front axle and 4.1 Hz in the rear, the demanded spring rate is given by the formula 9.3:

$$k = (\omega * 2\pi)^2 * m \qquad \text{(Eq. 9.3)}$$

$$kf = (\omega f * 2\pi)^2 * mfront$$

$$kf = (4.5 * 2\pi)^2 * 137.5 = 109919 \text{ N/m}$$

And:

$$kr = (\omega r * 2\pi)^2 * mrear$$

$$kr = (4.1 * 2\pi)^2 * 194.5 = 129076 \text{ N/m}$$

Where:

kf= Spring rate front

kr= Spring rate rear

With the spring rate calculate, the critical damping can be calculate following the equation 10:

$$C = 2\sqrt{m * K} \qquad \text{(Eq. 10)}$$

Where:

C= Critical Damping

m= Mass

k= Spring rate

Solving equation 10 for each axle of the car with the baseline settings:

$$Cf = 2\sqrt{137.5 * 192639} = 10293.3\ Ns/m$$

$$Cr = 2\sqrt{194.5 * 218908} = 13050.3\ Ns/m$$

Where:

Cf= Critical damping front

Cr= Critical damping rear

k= Spring rate

Despite being an important value, the critical damp cannot be solely used for defining the damper. More important than that, is the relation between the critical damping and the actual damping, values too low would lead to a very long time until the body returns for an equilibrium state, while high value leads to a temperature increase in the tyres (Nowlan, 2010). Figure 4 shows how different damping ratios affects the damping.

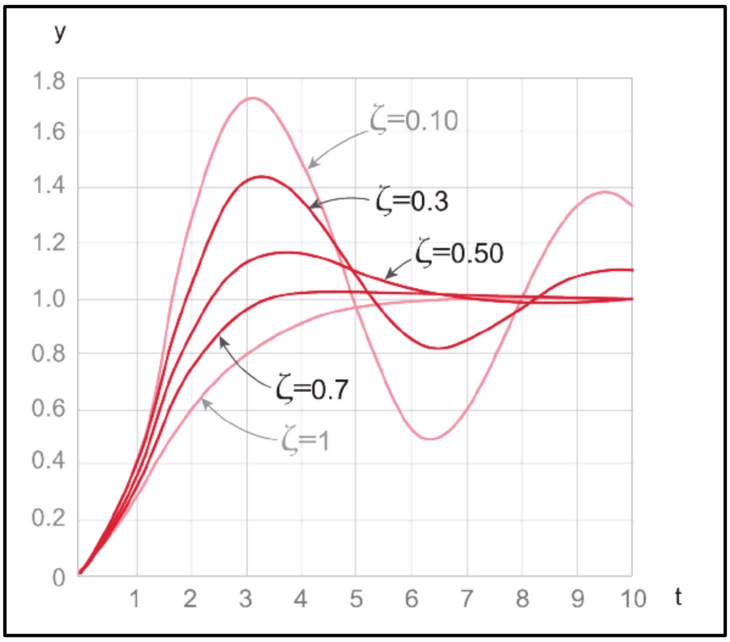


Figure 4: Second order system response to a step input (D´Azzo, J., and Houpis, C., 1988 as cited in Nowlan, 2010).

As an initial starting point, a damping below 0.5 is ideal for filtering bumps, values between 0.5 and 1.0 deals with body control and values higher than 1.0 can be used for buildup heat on the tyres.

Regarding the actual damping, it can be calculated as the delta on force and velocity of a damper as shown in figure 5 and equation 11.

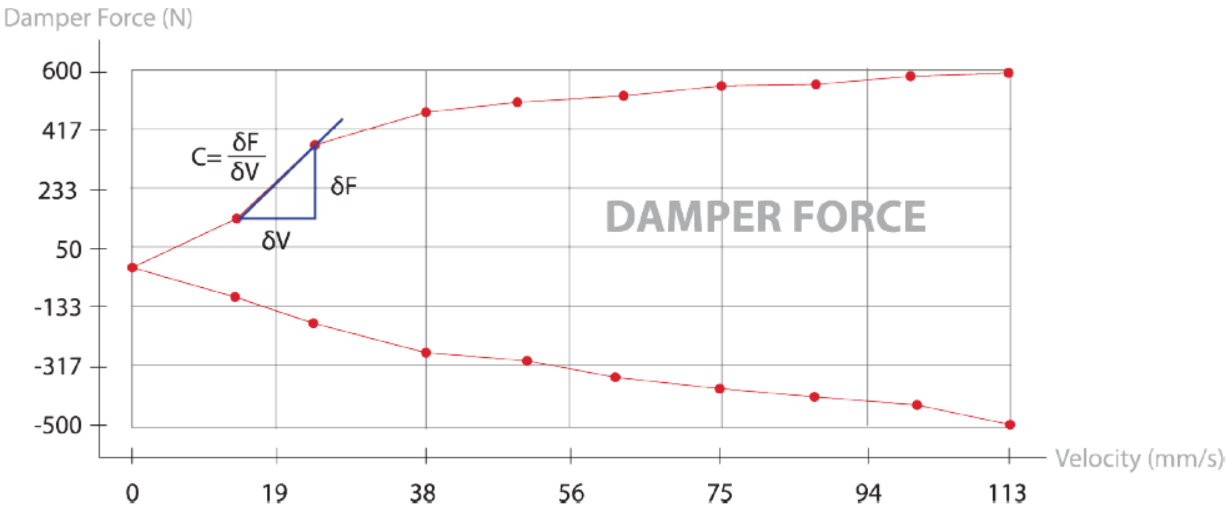


Figure 5: Determining damping rate (Nowlan, 2010).

$$C = \frac{\delta F}{\delta V} \quad \text{(Eq. 11)}$$

Where:

C= Damping rate

δF = Damper force

δV = Damper displacement velocity

7 post rig Simulation on ChassisSim

Through the use of the 7 post rig simulation of the ChassisSim, it is possible to evaluate the characterization of the damper based on its frequency and ratio of output to input (Nowlan, 2010.).

The output to input ratio can be described as a transmissibility ratio (Smith, 1978) and when the damping ratio is increased or the springs softened, the transmissibility ratio is reduced.

The process to determine the baseline damping behavior has involved the creation of a circuit and bump profile on the ChassisSim with data derived from a MoTec software.

Figure 6 to 11 show the behavior of the baseline damper regarding heave, pitch, front damper displacement, front tyre displacement, rear damper displacement and rear tyre displacement.

Heave: That is the amount of the movement of the car moving up and down. The highest transmissibility ratio is around the frequency of 4Hz.

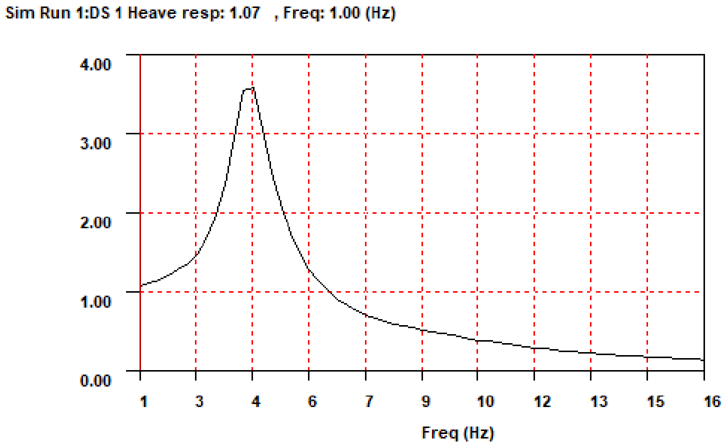


Figure 6. Heave behavior with the baseline suspension settings.

Pitch: That is the movement of the front of the car going up and down, while the back of the car follows the opposite direction of movement. The transmissibility ratio is high and the occurrence of two peaks (at 4Hz and 7Hz) contribute for higher instability of the vehicle.

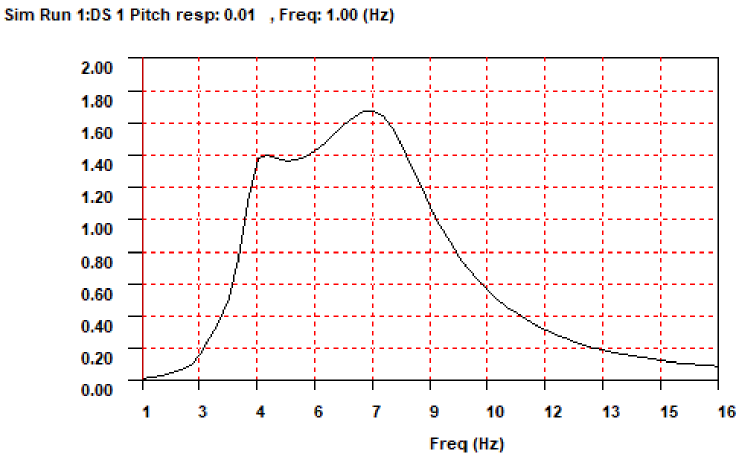


Figure 7. Pitch behavior with the baseline suspension settings.

Front damper: Amount of up and down movement on the front damper. As seen on the pitch characteristic, the existence of a second peak shows a too high transmissibility ratio.

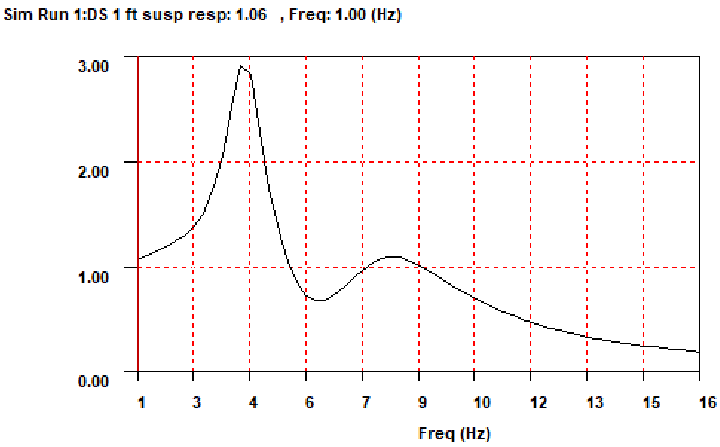


Figure 8. Front damper behavior with the baseline suspension settings.

Front tyre: Amount of up and down movement on the front unsprung mass. Despite the unsprung mass not being suspended by the dampers/springs it is also affected by those and its behavior has a similar shape than the one found on the damper.

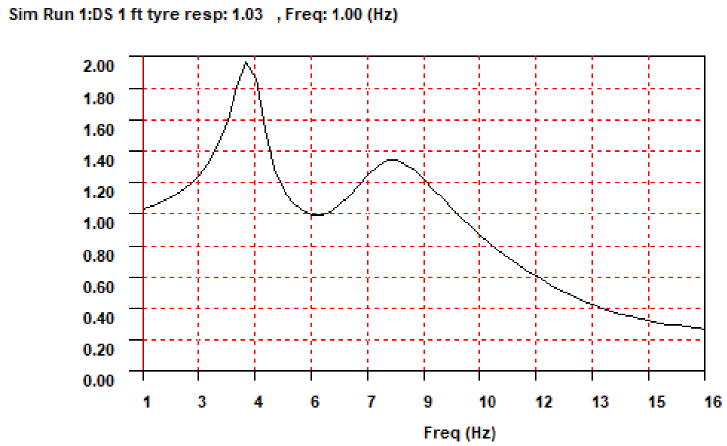


Figure 9. Front unsprung mass behavior with the baseline suspension settings.

Rear damper: Amount of up and down movement on the rear damper. Differently to the front one, the rear damper does not show a second peak, although its peak transmissibility value can be reduced as well the abrupt transmissibility reduction between 7.5Hz and 9Hz may be smoothed.

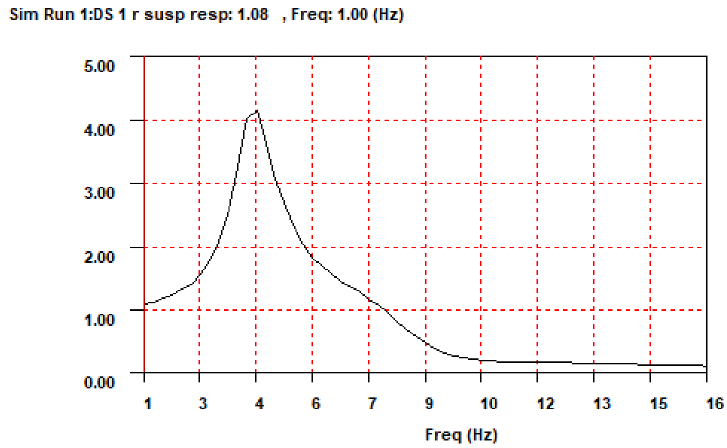


Figure 10: Rear damper mass behavior with the baseline suspension settings.

Rear tyre: Rear unsprung mass up and down movement. The "valley" between 8Hz and 10Hz should be avoided.

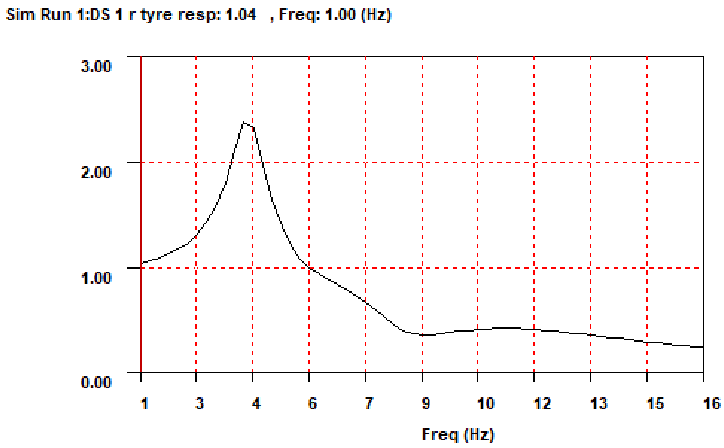


Figure 11: Rear unsprung mass behavior with the baseline suspension settings.

Based solely on the interpretation of the 7 post rig results, it seems that a possible path to reduce lap time is the reduction of the damping and a possibly reduction of the spring rate.

A first step to understand how the damping is behaving on the baseline car settings is the definition of the damping ratio for bump and rebound. It can be done applying the equations 9, 10 and 11.

In order to avoid the repetition of the formulas, an excel tool was develop for this automatic calculation and more information about it can be found on the Appendix. The starting damping ratio for bump and rebound (front and rear) can be found on the table 1 and 2.

Front		
Velocity (mm/s)	**Damping ratio in bump**	**Damping ratio in rebound**
0	1.16	1.22
20	0.26	0.46
50	0.26	0.46
100	0.26	0.46
150	0.26	0.46
200	0.26	0.46
250	0.26	0.46
300	0.26	0.46
350	0.26	0.46

Table 1. Damping ratio for bump and rebound at the front axle.

Rear		
Velocity (mm/s)	Damping ratio in bump	Damping ratio in rebound
0	0.92	0.92
20	0.26	0.21
50	0.26	0.21
100	0.26	0.21
150	0.26	0.21
200	0.26	0.21
250	0.26	0.21
300	0.26	0.21
350	0.26	0.21

Table 2. Damping ratio for bump and rebound at the rear axle.

Based on the results of the baseline settings on the 7 post rig and the calculated damping ratio, it seems that a reduced damping ratio could be beneficial, as it would reduce the output to input ratio on the dampers as well reduce the damping ratio value.

Vehicle Dynamics Optimization Process

In order to have a better understanding of how the car reacts to the changes, the priority was to change a single parameter in just one axle per simulation, but when it was judge that the car balance would be affected in a negative way, the other axle was also changed in the same magnitude in an attempt to keep the car balance. Below follows a short overview of the first steps.

1st Simulation

On this step, the ls (low speed) rate of bump and rebound were reduced aprox.10% in order to check if it would bring the results to the expected direction.

With the first set of changes, the lap time could be improved by 0.013s and a graph showing its tendency is shown on figure 12.

The output to input ratio of the front damper seems to not be affected in a significative way as compared on the figure 13 (front) and 14 (rear). For this whole investigation, the black line will represent the behavior with of the baseline settings (will always be the same) while the red curve will be the current car setting.

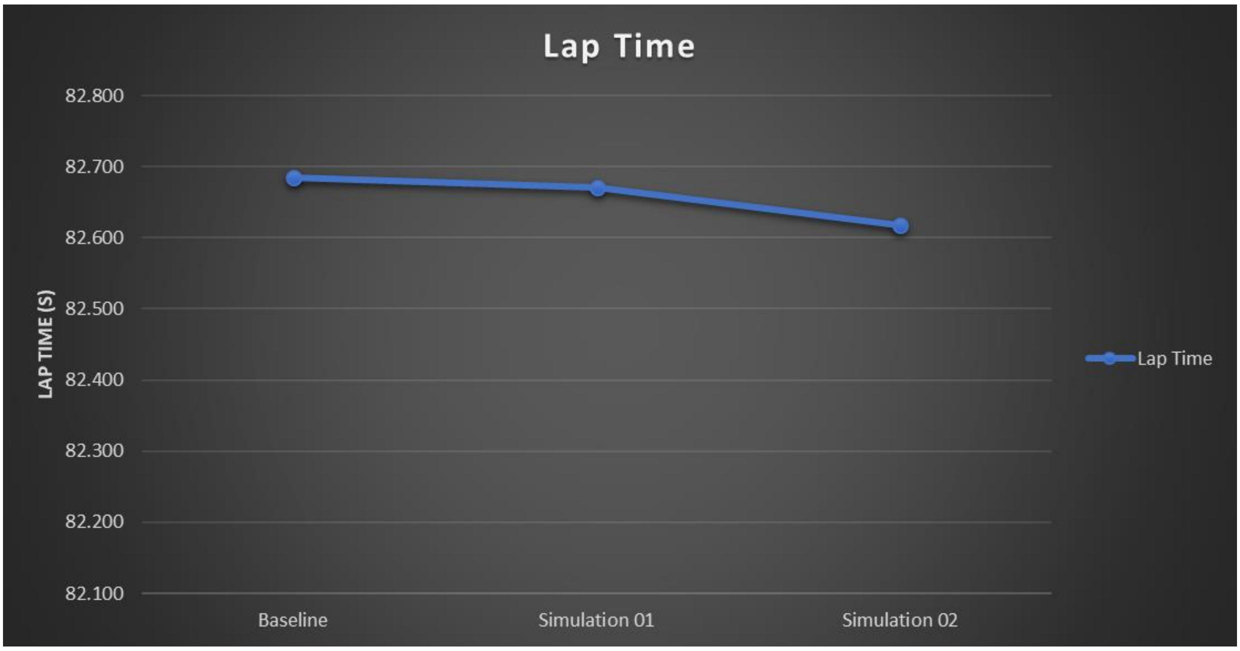


Figure 12. Lap time tracking.

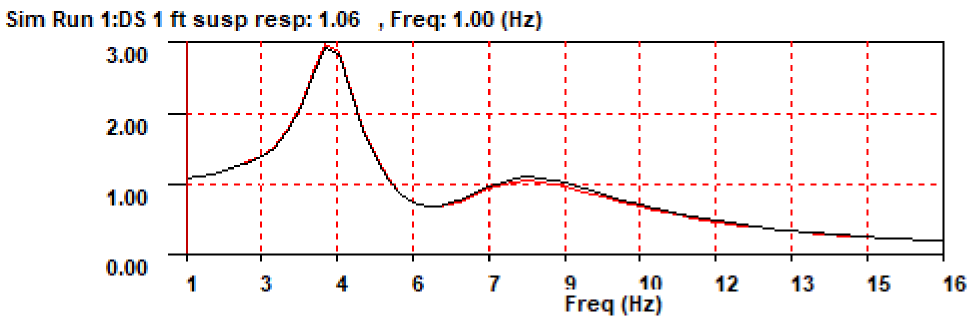


Figure 13. Result of the output to input ratio from the 7 post rig simulation for the front dampers. Black is reference and red is the “simulation 1” car settings.

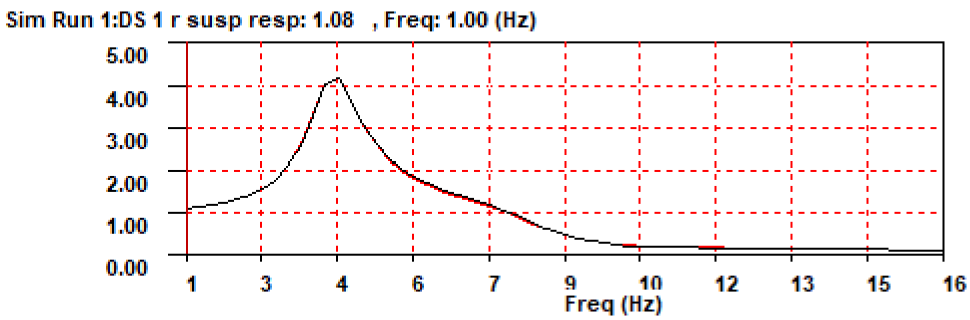


Figure 14. Result of the output to input ratio from the 7 post rig simulation for the rear dampers. Black is reference and red is the “simulation 1” car settings.

2^{nd} Simulation

Another factor that heavily affects the car behavior (and it is simple to have its sensibility evaluated) is the spring rate on both front and rear axles. In order to reduce the natural frequency of the front axle from 32rad/s to 30rad/s the front spring rate must be reduce from 192639N/m to 170000N/m.

Following the same trend on the rear axle for the purpose of not creating a too big contrast among the axles, the natural frequency of the rear was reduced from 29.9rad/s to 28.5rad/s by changing the spring rate from 218908N/m to 200000N/m. Figure 15, 16 and 17 shows the results of these changes.

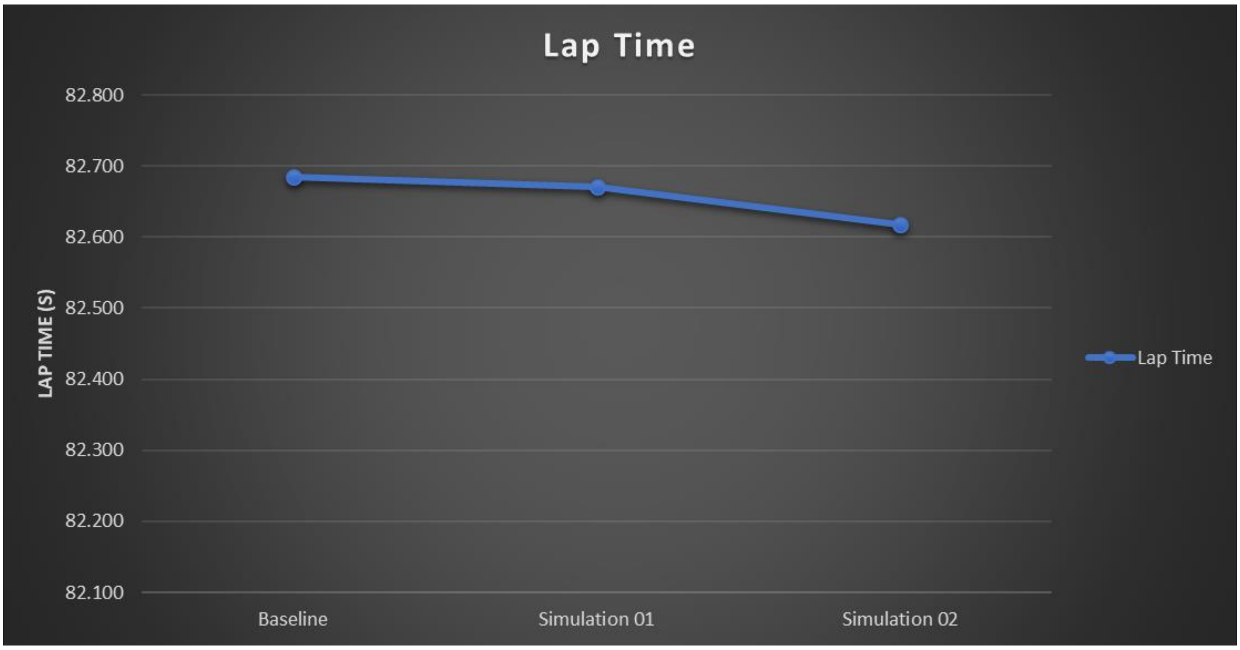


Figure 15. Lap time tracking updated for the 2^{nd} simulation.

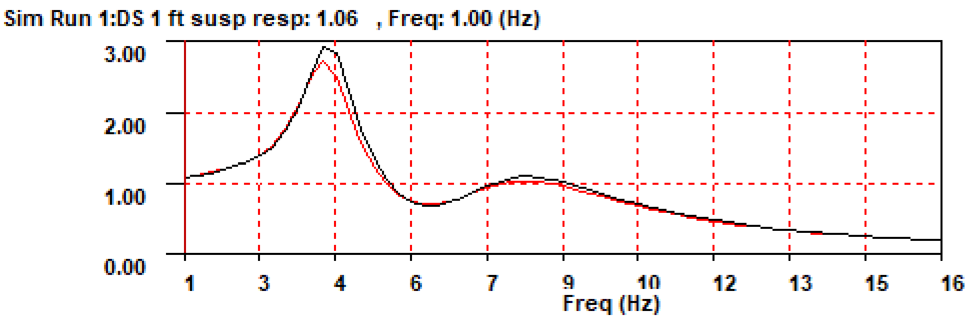


Figure 16. Result of the output to input ratio from the 7 post rig simulation for the front dampers. Black is reference and red is the "simulation 2" car settings.

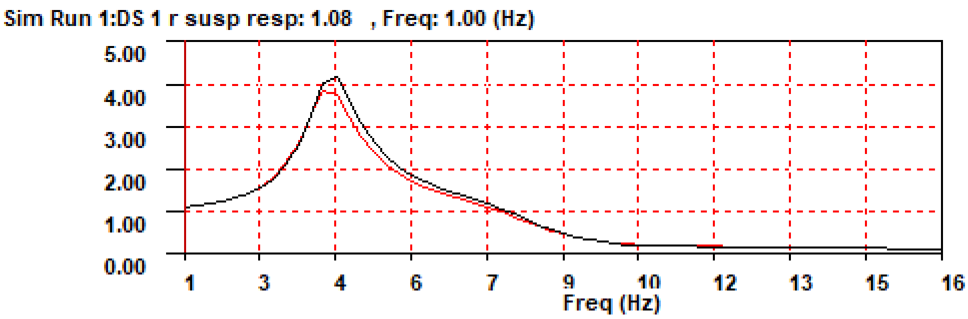


Figure 17. Result of the output to input ratio from the 7 post rig simulation for the rear dampers. Black is reference and red is the "simulation 2" car settings.

These changes have brought a positive impact on both lap time and damping behavior.

3rd Simulation

On the 3rd simulation, the damping ratio in bump and rebound for the front axle at low speed was around 1.2. In order to reduce it for approx. 0.7 the bypass (transition between the low speed to high speed rate of the damper) was reduced from 20mm to 10mm.

Table 3 shows the damping rate for the 3rd setting.

Front		
Velocity (mm/s)	Damping ratio in bump	Damping ratio in rebound
0	0.72	0.87
20	0.28	0.49
50	0.28	0.49
100	0.28	0.49
150	0.28	0.49
200	0.28	0.49
250	0.28	0.49
300	0.28	0.49
350	0.28	0.49

Table 3. Damping ratio with reduced bypass.

The results of the simulation performed with these settings can be seen on the figure 18, 19 and 20.

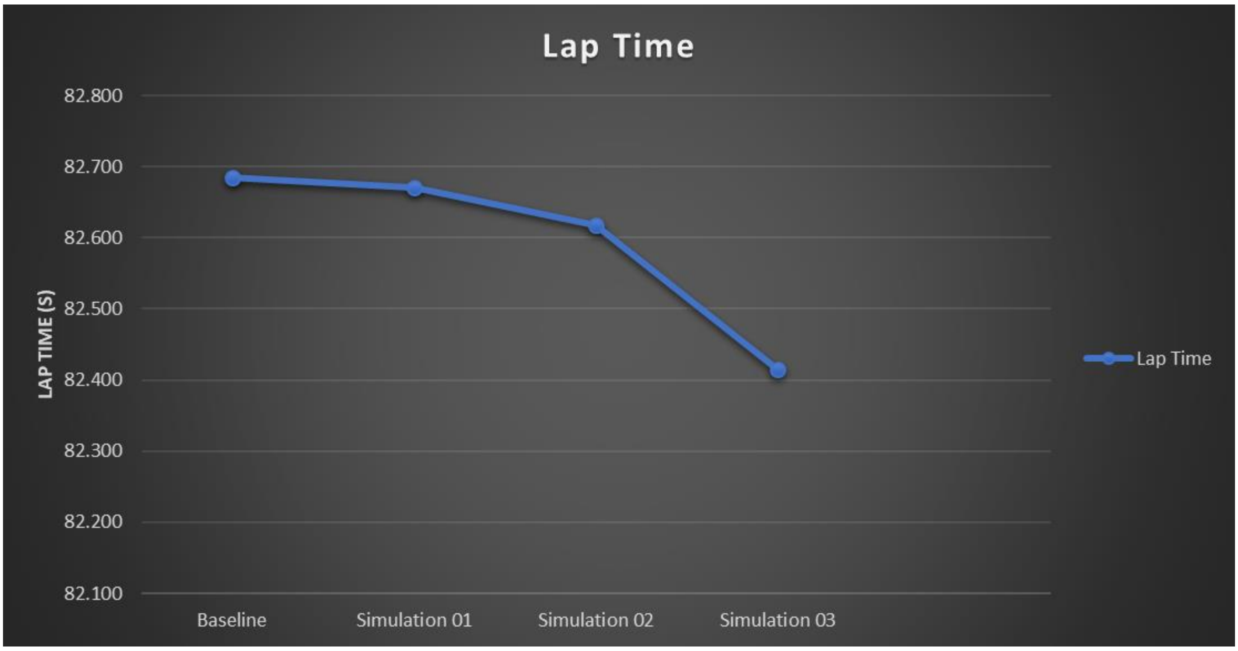


Figure 18. Lap time tracking updated for the 3rd simulation.

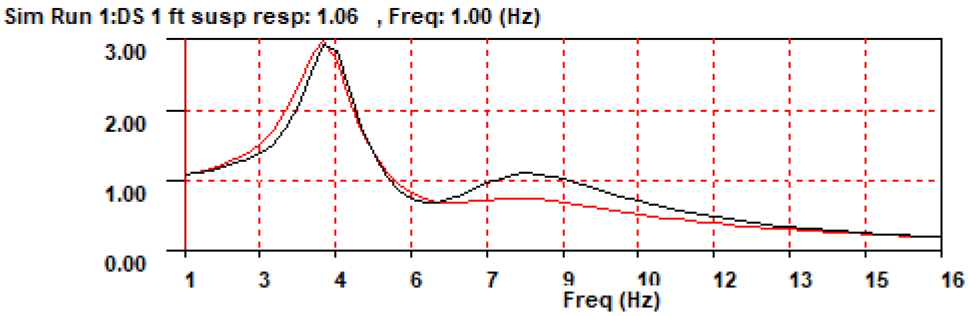


Figure 19. Result of the output to input ratio from the 7 post rig simulation for the front dampers. Black is reference and red is the "simulation 3" car settings.

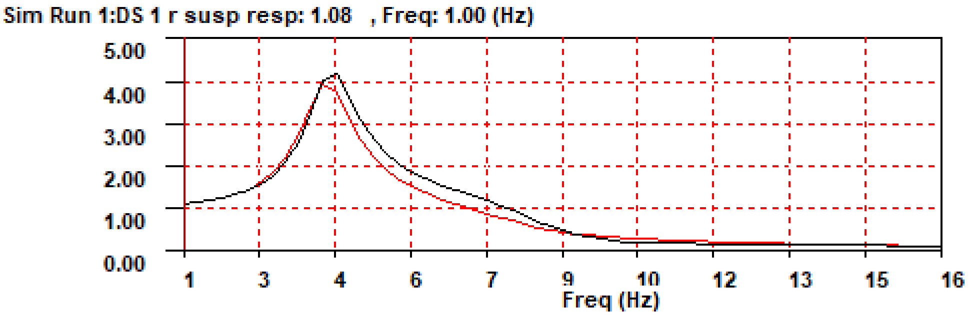


Figure 20. Result of the output to input ratio from the 7 post rig simulation for the rear dampers. Black is reference and red is the "simulation 3" car settings.

The change on the bypass of the front damper had a strong positive impact on lap time and dampers behavior. Just with this change, lap time was reduced in more than 0.2s.

4th Simulation

Due to the positive results of the bypass change on the front damper, the bypass on the back axle was also modified in the same way, however, on this case the lap time was increased by 0.1s (figure 21).

This means that a lower damping ratio on the back axle is not beneficial for the current car configuration.

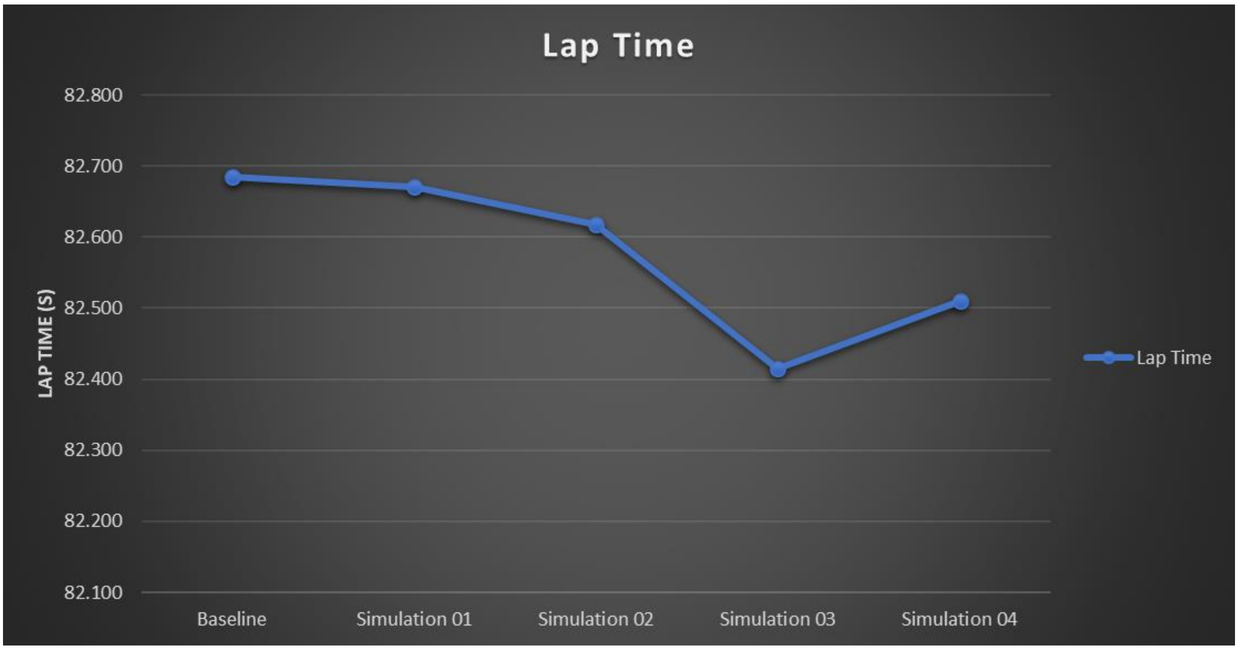


Figure 21. Lap time tracking updated for the 4th simulation.

With the identification of which parameters could affect the setup in the most significative way, the parameters were investigated, and several simulation sweeps were performed until the lap time of 1:22.197 was achieved, shown on figure 22. Table 4 and 5 show the final dumping ratio for front and rear.

The time of 1:22.197 is a reduction of 0.488s over the baseline time and approx. 0.1s faster than the target time (dashed line on figure 22).

The complete setup used on ChassisSim is described on the Appendix.

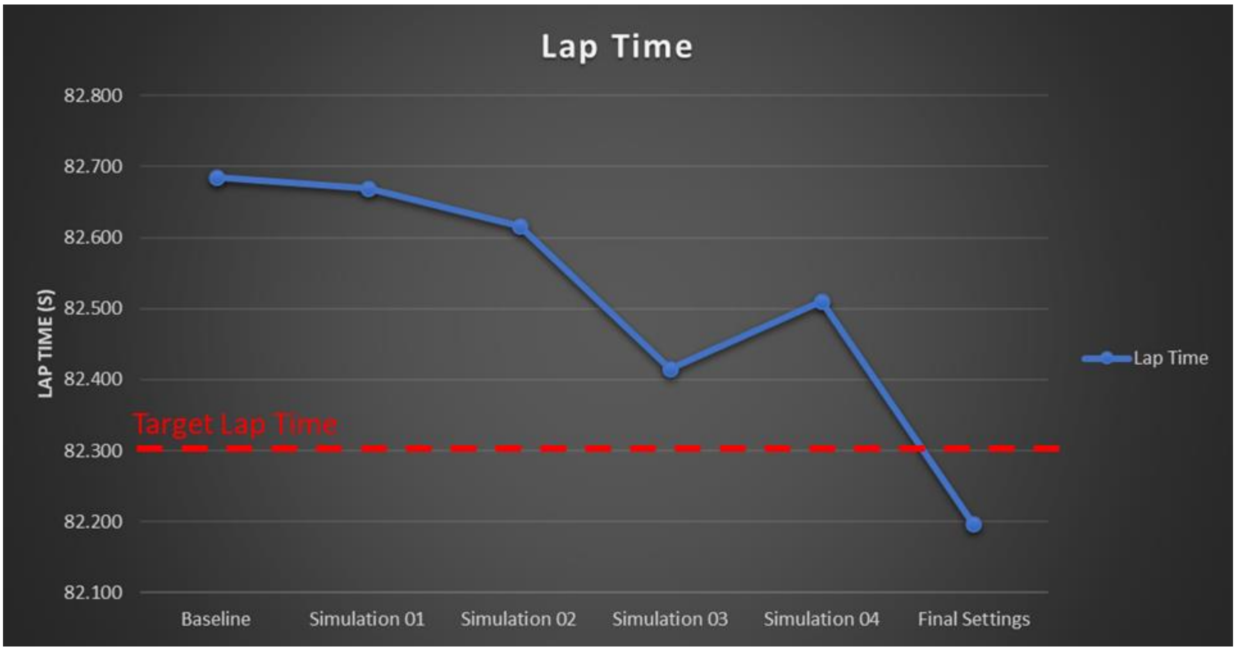


Figure 22. Lap time tracking with the final settings.

Front		
Velocity (mm/s)	**Damping ratio in bump**	**Damping ratio in rebound**
0	0.76	0.91
20	0.33	0.60
50	0.34	0.60
100	0.34	0.60
150	0.34	0.60
200	0.34	0.60
250	0.33	0.60
300	0.34	0.60

Table 04. Final damping ratio for the front axle.

Rear		
Velocity (mm/s)	Damping ratio in bump	Damping ratio in rebound
0	1.14	1.18
20	0.37	0.43
50	0.37	0.42
100	0.37	0.42
150	0.37	0.42
200	0.38	0.43
250	0.37	0.42
300	0.37	0.42
350	0.37	0.42

Table 05. Final damping ratio for the rear axle.

Analyze of Vehicle Dynamics

The analyze of the vehicle handling with the optimized settings has followed two different approaches: The first consists on the comparison of the damper behavior using the 7 post rig simulation and the second one being the direct comparison of both laps on the MoTec software.

7 Post Rig Results

The 7 post rig results gives a good overview of the difference among the two setups in several different conditions, which are shown on figure 23 to 26.

Heave: The final settings shows a reduction of ~20% in the up and down movement of the car body at 4Hz. Excessive car body movement can affect the maximum traction force between the tyre and the ground.

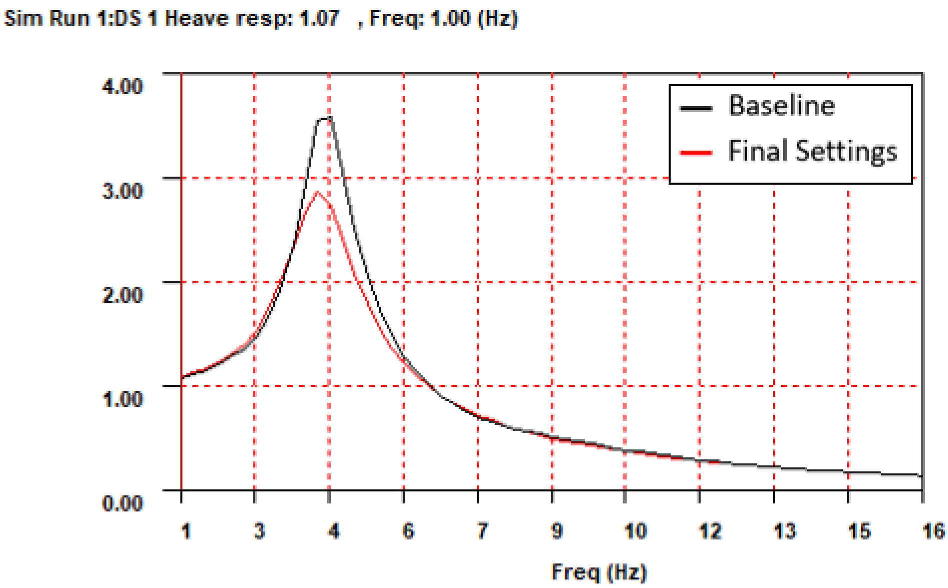


Figure 23. Heave comparison between baseline and final settings.

Pitch: The pitch was one of the most affected parameters of this investigation. The baseline settings had a significantly higher movement over a spread frequency range, which makes a car less predictable and harder to control, particularly when entering or leaving a turn.

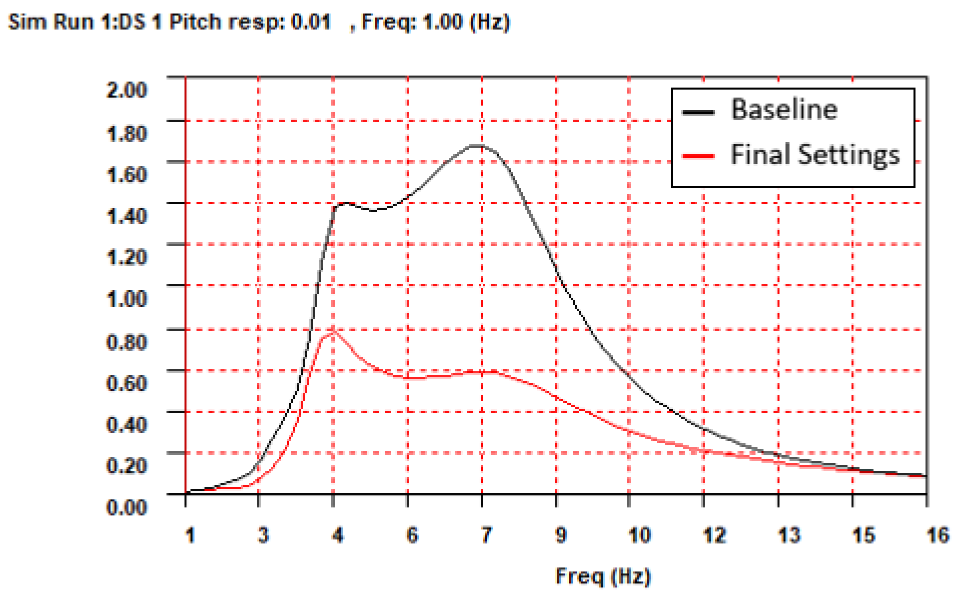


Figure 24. Pitch comparison between baseline and final settings.

Front Damper: A reduced peak output to input ratio shows that less vertical movement is being transferred to the body, which in this case was an improvement, however it depends on the track, type of car, kerbs height, etc. The mitigation of the second peak present at ~8Hz with the baseline setting had a significative positive impact.

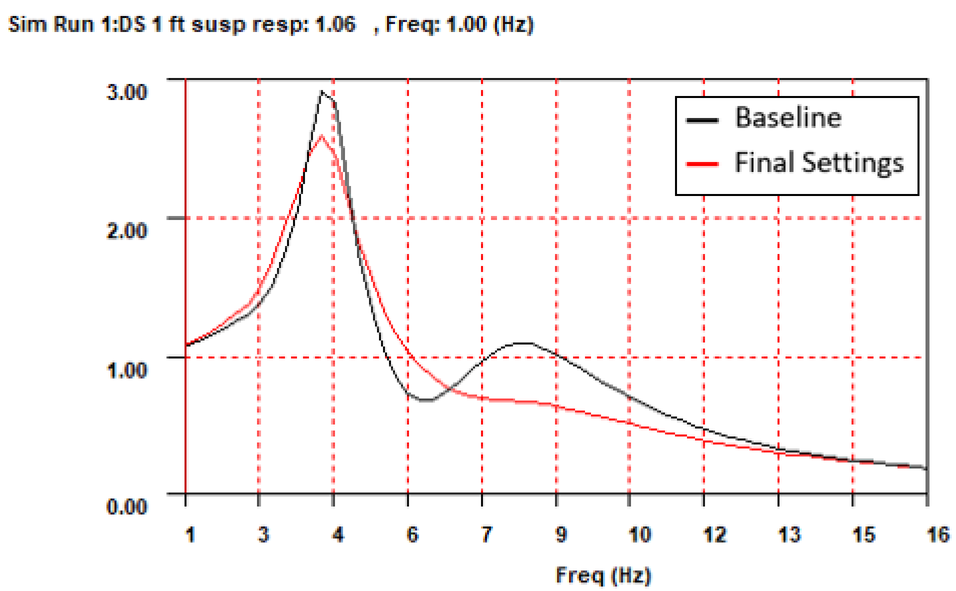


Figure 25. Front Damper comparison between baseline and final settings.

Rear Damper: The rear damper had a ~25% lower output to input ratio at 4Hz due to a lower spring rate. The output to input ratio is reduced more smoothly with the final settings, especially between 8Hz and 9.5Hz.

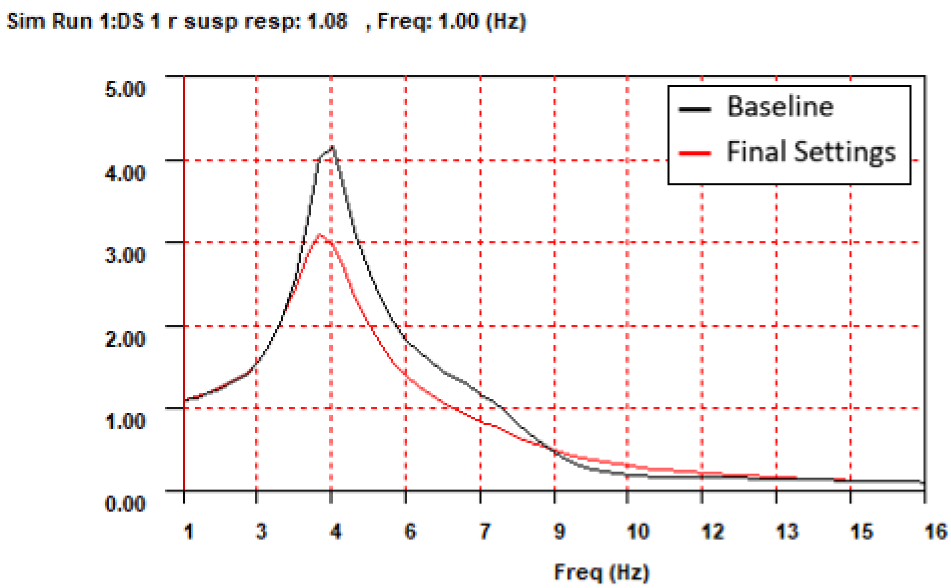
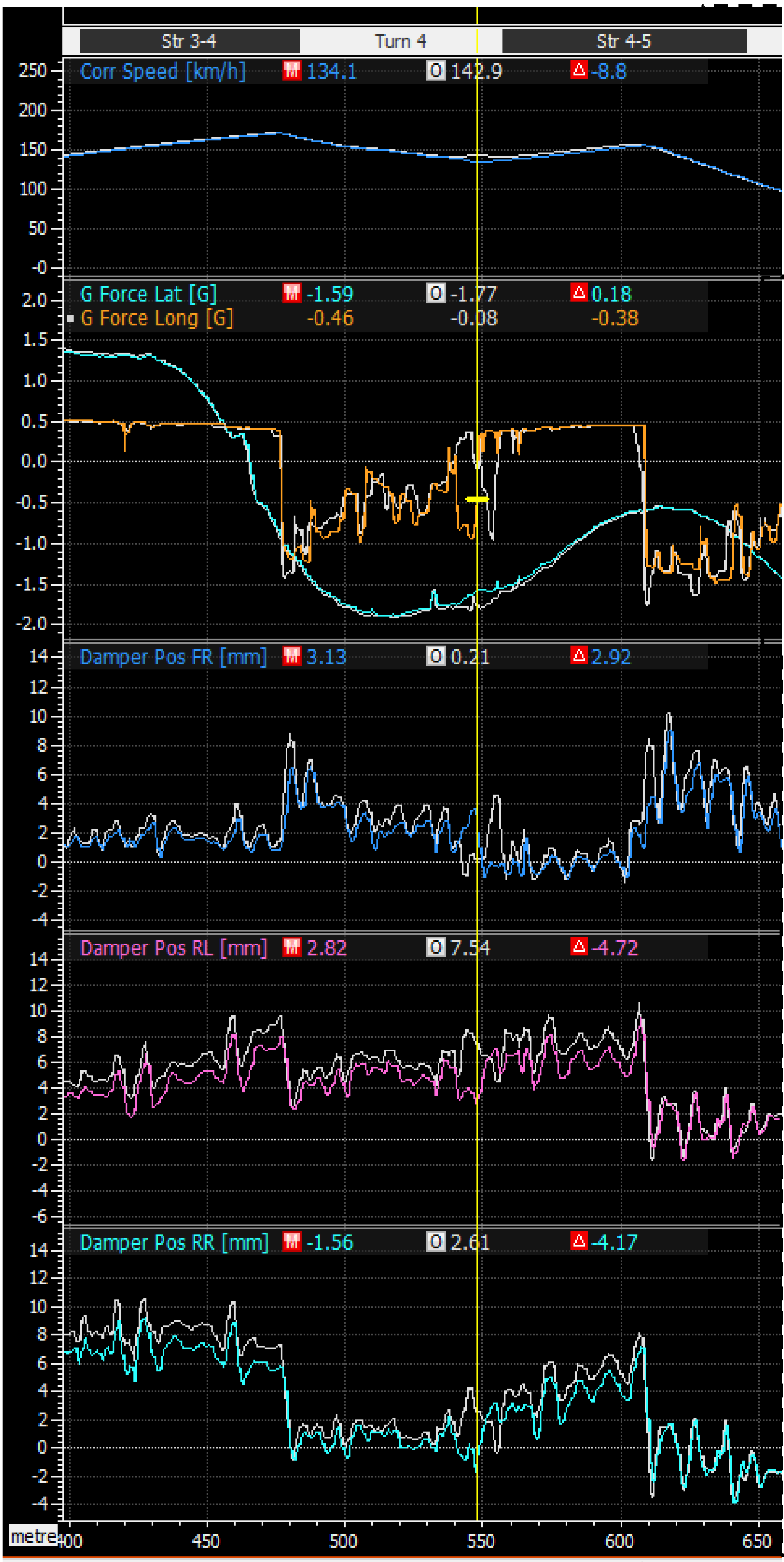


Figure 26. Rear Damper comparison between baseline and final settings.

Data comparison at MoTec

As an example of the difference on the car handling, the turn 4 was chosen and the car behavior detailed analyzed. Figure 27 shows the speed signal, G force and damper data with comments highlighting the main differences. The data shown in white relates to the final settings.



Higher speed out of the turn 4 leads to a higher speed over the str 4-5. Peak difference is 8.8km/h.

The increased lateral G force led to a higher cornering speed.

Longitudinal G force is closer to 0, so the driver is not breaking/coasting at this point.

The increased value on the damper position at the rear is an indication of improved mechanical grip due to a higher load that the rear dampers are being imposed to.

The change on height (due to the damper position change) has also an impact on the aero balance of the car, however it will not be further evaluated as it is out of scope.

Figure 27. Speed, G force and damper movement comparison within the MoTec software.

Figure 28 shows the time gain over the baseline setting for the same sector shown on figure 27, at the turn 4 (highlighted on the figure 28), the time improvement was 0.020s. The time difference over the whole lap is exploited on the figure 33.

Figure 28. ΔTime of the optimized setting over the baseline. Highlighted in green is the turn 04.

An useful information for dampers analysis that can be found in the MoTec is the shock speed histogram. This histogram shows how much time the shock absorber is spending in each speed interval.

In order to increase the mechanical grip in a car that the aerodynamic load is not very significative, the histogram must be as asymmetrical as possible, however in a car that the aerodynamic load plays a significative hole, it can change (Segers, 2014). Figure 29 to 32 shows the shock speed histogram for the baseline and for the final car settings, as shown before the data in white refers to the final settings results (x-axis is displacement speed in mm/s and y-axis is the percent of time spend in each condition).

Front Left: The original setup shows a very high percent of operation in rebound at low speeds (>25mm/s), which at the final settings was strongly reduced due to the bypass reduction. At low speeds, the damper is mainly dealing with the inertial chassis motion (roll, pitch and heave). The final settings show a better distribution with a higher operating time in Bump, which can be linked to the high aerodynamic load of the car.

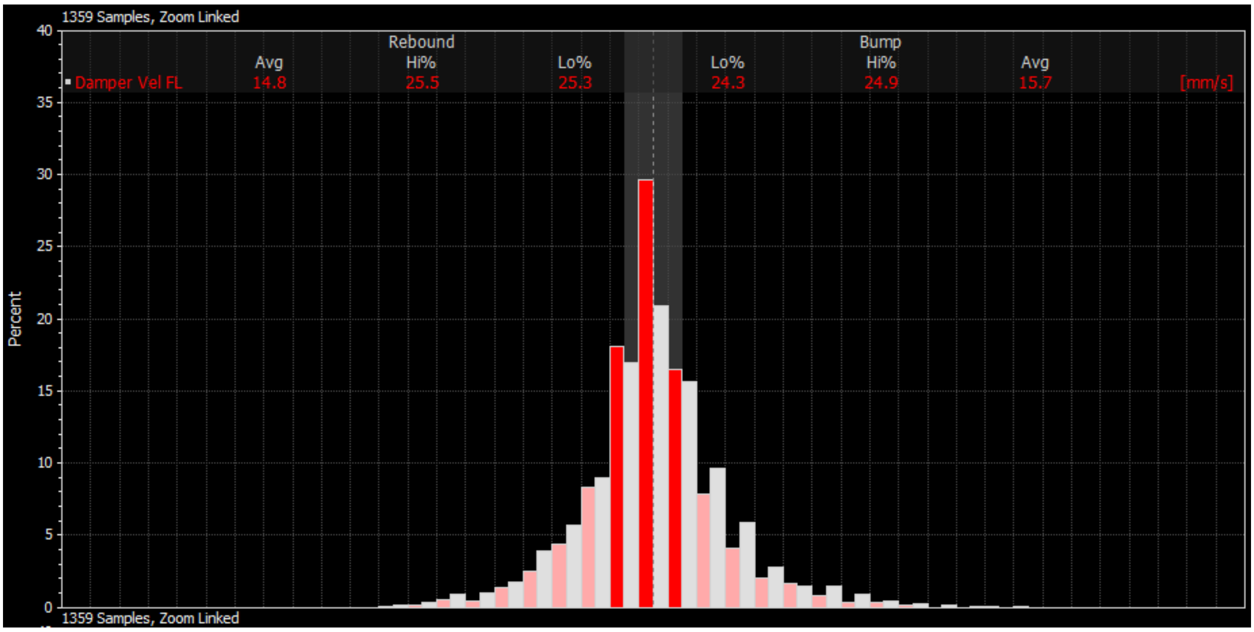


Figure 29. Shock speed histogram of the front left damper.

Front Right: As expected, the equal weight distribution between left and right side and symmetrical settings creates a very similar shape of the shock speed diagram on both sides. The difference among sides can be related to the amount of turns for each side, bumps and kerbs on the race track.

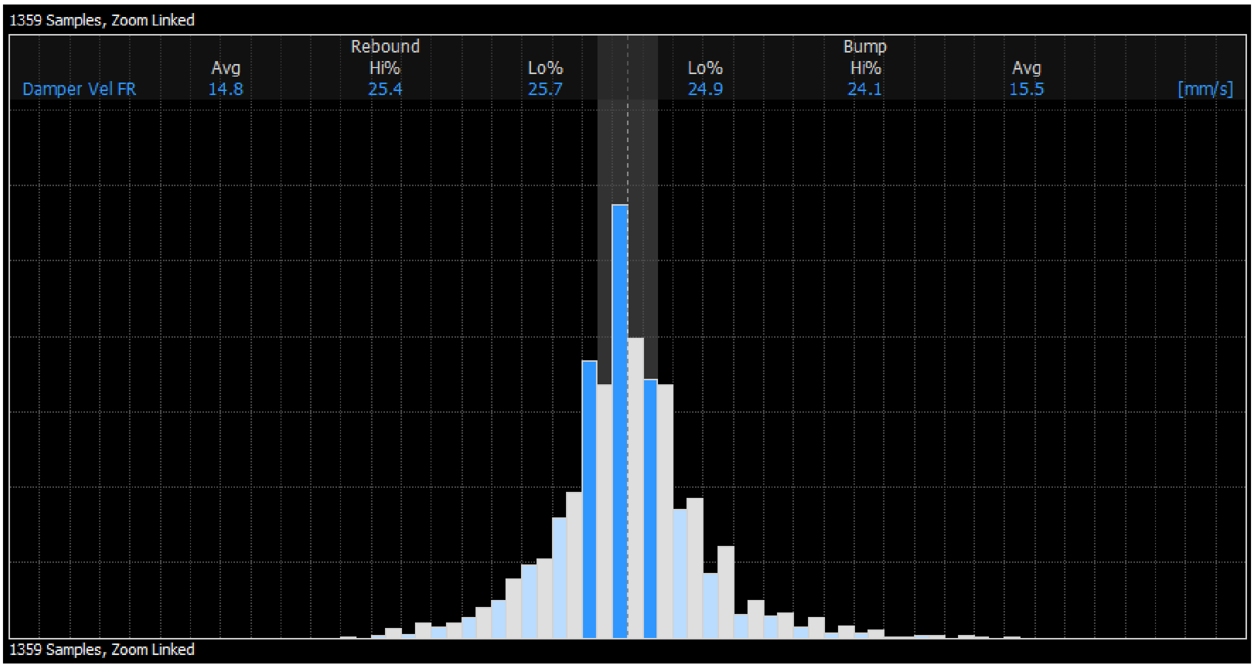


Figure 30. Shock speed histogram of the front right damper.

Rear Left: Different to the front damper, the rear component remains with a lot of its time displacing within the low speed area, as its bypass was kept the same as on the original setup. The final setup has a small tendency to work less on rebound, that can explain the improvement saw on the figure 27.

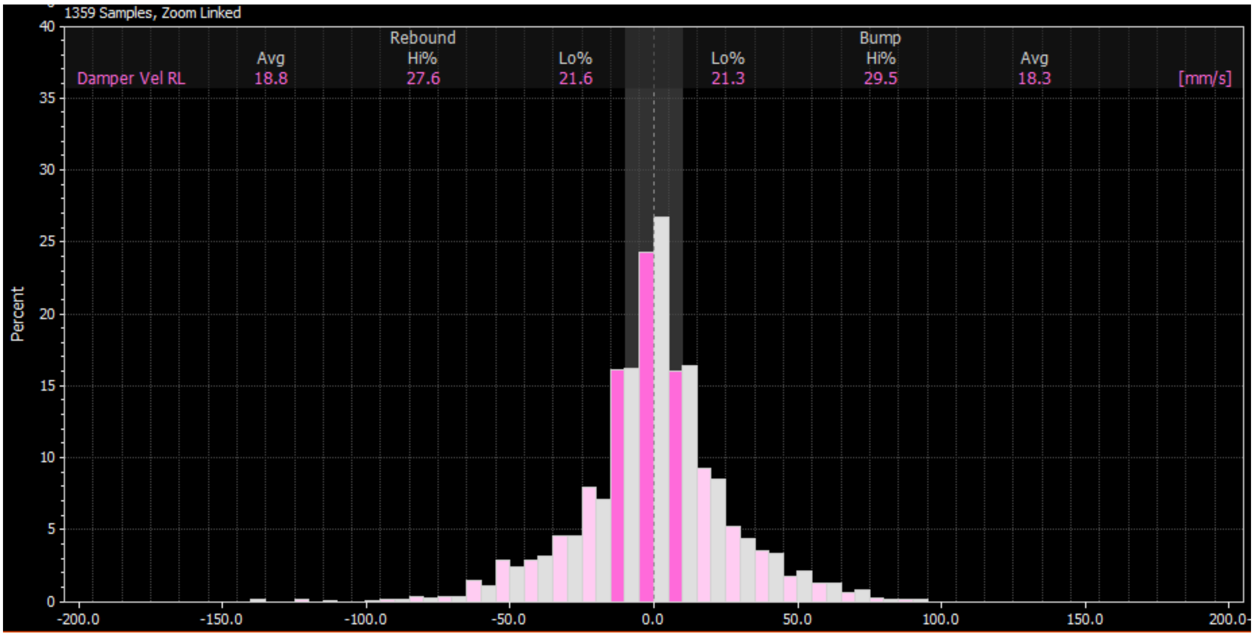


Figure 31. Shock speed histogram of the rear left damper.

Rear Right: As seen on the front damper, not many differences among the rear and left side.

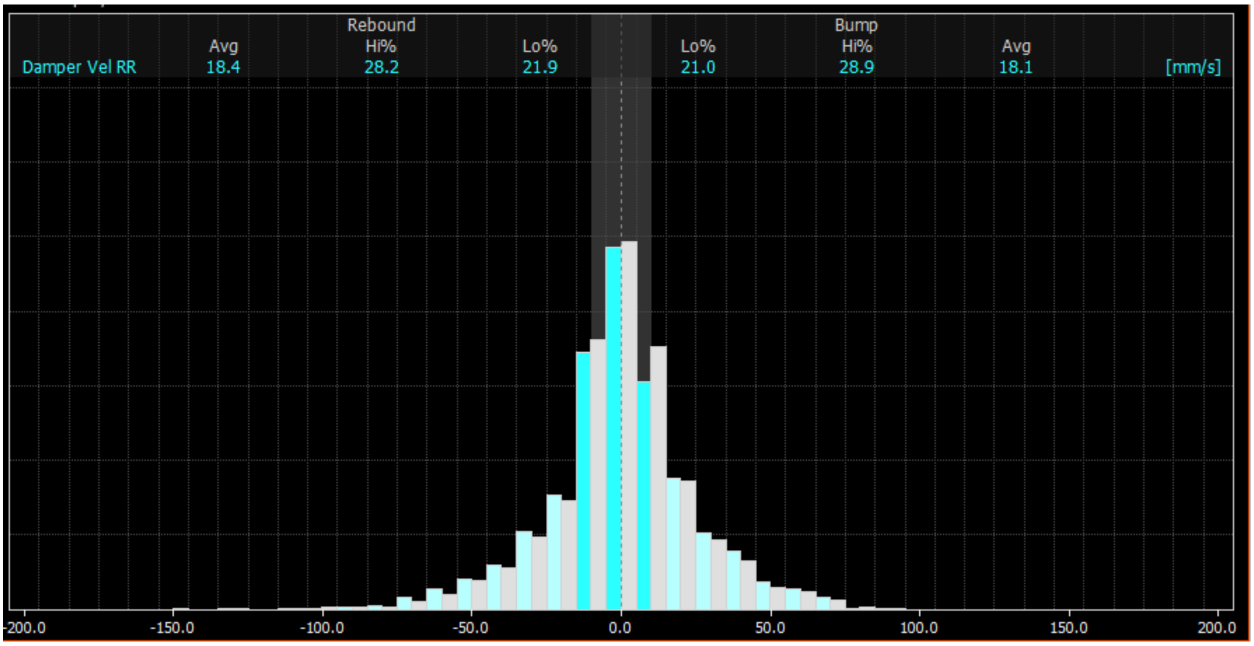


Figure 32. Shock speed histogram of the rear right damper.

Complete Lap Time Comparison

The complete lap time comparison carried on the MoTec software is shown on the top portion of the figure 33, as the white line is the representation of the delta time between both settings.

Is highlighted in red the only two sections of the lap where the car with optimized settings is slower than the baseline.

The time disadvantage occurs on the braking zone (negative longitudinal acceleration) of the straight 4-5/curve 5 and on the braking zone of the straight 5-6/curve 6, which indicates that the brake performance of the car was impaired. The reason for that can be the lower weight transfer to the front axle during the braking (Pitch reduction), which could be improved via the brake balance distribution among front and rear axle. As the scope of the work was limited to the suspension settings, the brake balance settings were no further investigated.

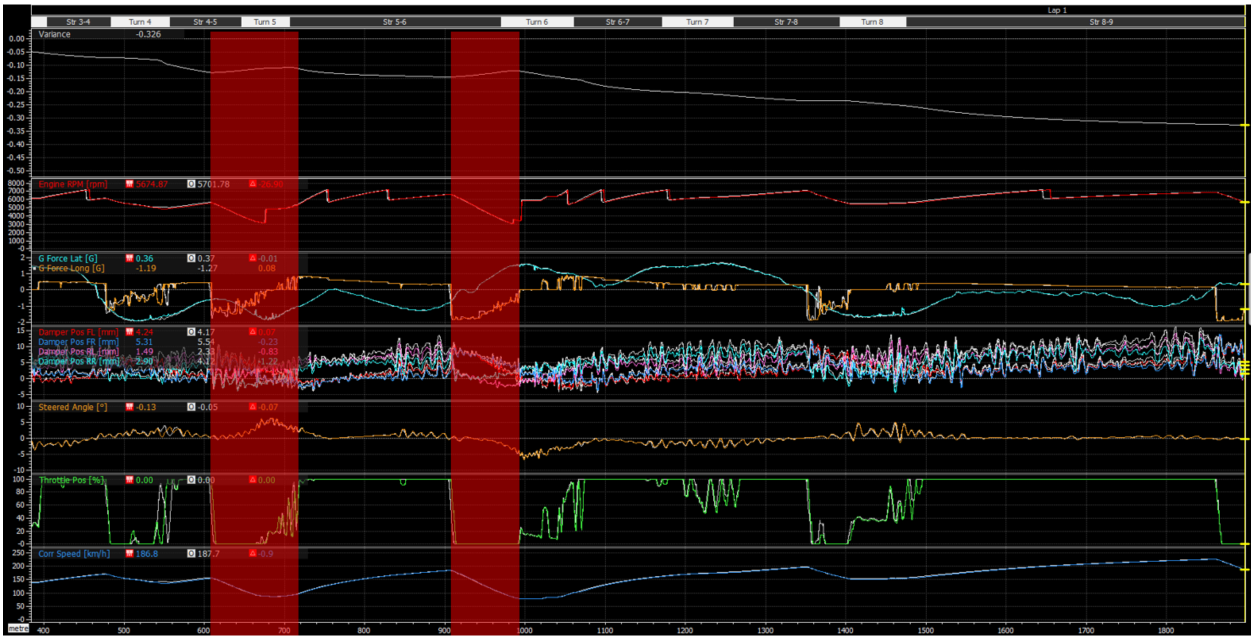


Figure 33. Complete ΔTime over the whole lap is shown by the first label in the top portion of the graph.

Stability Index

A tool that could be further exploited was the Stability Index, which is a reliable way to measure how the race car behaves. It can be applied based on data acquired during the lap (yaw rate sensor and accelerometers are necessary) or based solely on the vehicle model.

Stability Index values lower than 0 is an indication of understeer behavior and value above 0 is an indication of understeer. Based on that, during the set-up of the vehicle within ChassisSim, the expected behavior of the car can be tuned for the specific preference of the driver.

Conclusion

The use of a vehicle dynamic simulation software has changed how a race car is optimized even before hitting the race track. Its combination with a data acquisition software is a powerful tool when in the hand of a trained racing engineer.

On this work, a 7 post rig evaluation was performed many and many times. A race team could do it with a fraction of the cost of the real 7 post rig test, and there is where the main advantage of this kind of software relies – cost and time.

An initial calculation is very important as it gives a north of which path the car settings must follow, a process that can be quickly done with the use of excel tables after the first table is finished and validated. The math that calculates the damping ratio is clear and can be done without major challenges, the problem is (at least for me, an inexperienced racing engineer) to translate a given/desired damping ratio or natural frequency in car behavior or lap time impact. The math behind all the calculations must be fully understood, although experience and clear understanding of how the race car systems behaves and interacts is essentially.

Despite the lap time achieved being 0.1s faster than requested, the simulation process could be improved. One path for optimization could be the use of a tool called "Setup Sweep Options", available on the ChassisSim. Such tool permits an autonomously run through a set of setup options, which means a faster investigation and a sweep that covers the whole number of adjustments possibilities. Due to a lack of time for further investigation, this tool was not exploited, but it certainly would create a faster evaluation and maybe an even faster lap time. For settings that has a lot of variables, its use is a must.

As shown on figure 33, the same changes that made the car significantly faster in most of the circuit, had a penalty on the heavy braking zones. It is an example of how complex the task of setting the car is and every change must be evaluated on the whole picture, with the driver felling/preference included.

With the advance in computing capacity and modeling precision, the vehicle dynamics simulation tends to increase its precision over time - which is already very good and validated in many categories - replacing more and more the very costly and time demanding optimization process.

Appendix

Final Car Settings

After 35 simulation loops, the final settings that gave a lap time of 1:22:197 (on ChassisSim) had the following adjusts:

Front:

- **Spring Rate:** 160000N/m

  Bump

  - ls rate: 11000

  - hs rate: 3157

  - Bypass: 10mm

  Rebound

  - ls rate: 11500

  - hs rate: 5600

  - Bypass: 10mm

Rear:

- **Spring Rate:** 180000N/m

  Bump

  - ls rate: 13500

  - hs rate: 4424

  - Bypass: 20mm

  Rebound

  - ls rate: 14000

  - hs rate: 5024

  - Bypass: 20mm

Car characteristics

Weight: 750kg

Wheel base: 2.54m

CG priorities:

- Height: 0.3m
- Front: 0.3m
- Rear: 0.3m
- Wdf: 0.42m
- Wd left: 0.5

Excel tool for Damping Ratio Calculation

During the simulation process, it was necessary to create an automatic way to calculate the damping ratio of a determined spring rate or damper setting. Due to that, an excel file was created with all the formulas. Figure 34 illustrates it.

Spring Rate Front	160000		Input
Spring Rate Rear	180000		Calculation
Sprung Mass Front	137.5		Final Output
Sprung Mass Rear	194.5		
Natural Frequency Front	34.11		
Natural Frequency Rear	30.42		

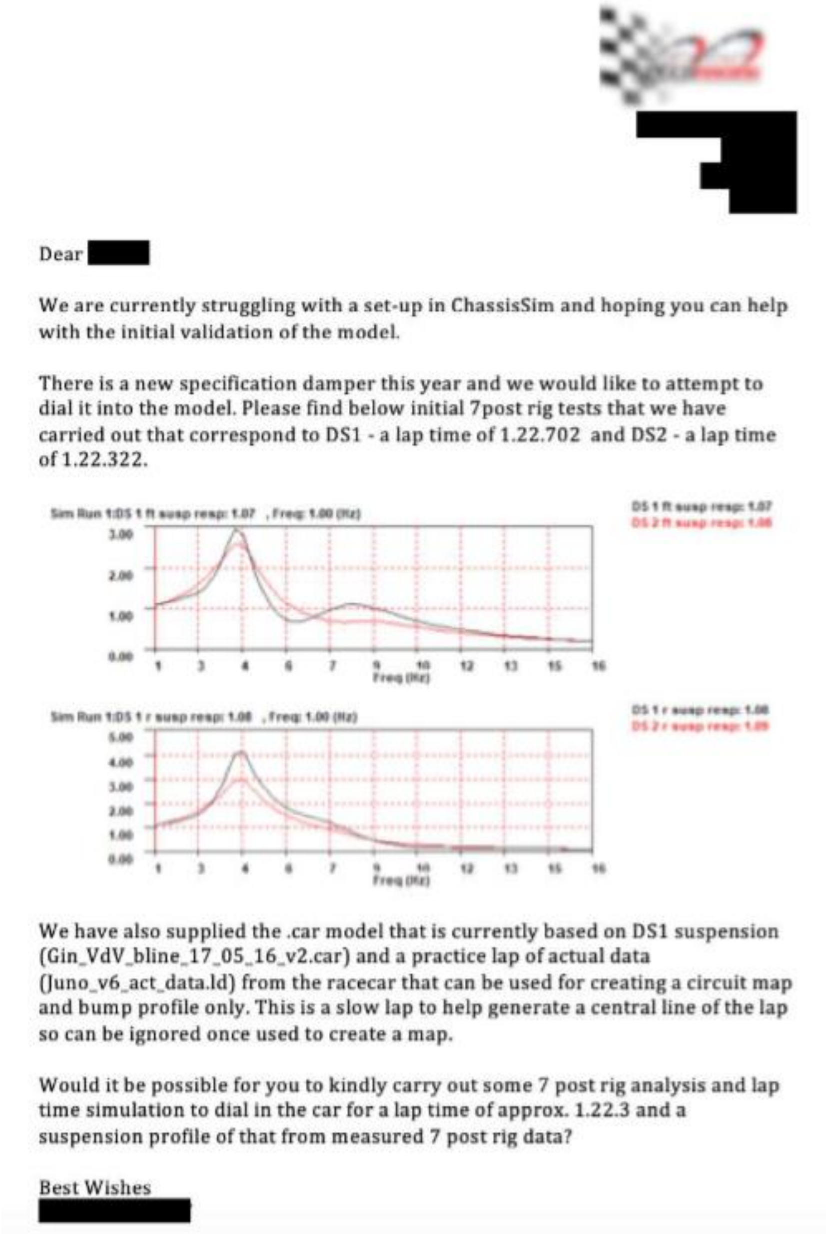
Front										
Velocidade	0	20	50	100	150	200	250	300	350	400
Delta Velocidae		20	30	50	50	50	50	50	50	50
Bump	0	142	236	394	552	710	868	1026	1183	1341
Delta Bump Force		142	94	158	158	158	158	158	157	158
Damping Rate Bump		7100	3133.333	3160	3160	3160	3160	3160	3140	3160
Damping Ratio Bump	0.76	0.33	0.34	0.34	0.34	0.34	0.34	0.33	0.34	0.00
Rebound	0	171	339	619	899	1179	1459	1739	2019	2299
Delta Rebound Force		171	168	280	280	280	280	280	280	280
Damping Rate Rebound		8550	5600	5600	5600	5600	5600	5600	5600	5600
Damping Ratio Rebound	0.91	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	

Front		
Velocity (mm/s)	**Damping ratio in bump**	**Damping ratio in rebound**
0	0.76	0.91
20	0.33	0.60
50	0.34	0.60
100	0.34	0.60
150	0.34	0.60
200	0.34	0.60
250	0.33	0.60
300	0.34	0.60

Figure 34. Damping ratio calculation tool.

Request for Damping Behavior Evaluation

Figure 35 shows the request where the optimization carried on this work was based on.



[redacted]

Dear [redacted]

We are currently struggling with a set-up in ChassisSim and hoping you can help with the initial validation of the model.

There is a new specification damper this year and we would like to attempt to dial it into the model. Please find below initial 7post rig tests that we have carried out that correspond to DS1 - a lap time of 1.22.702 and DS2 - a lap time of 1.22.322.

We have also supplied the .car model that is currently based on DS1 suspension (Gin_VdV_bline_17_05_16_v2.car) and a practice lap of actual data (Juno_v6_act_data.ld) from the racecar that can be used for creating a circuit map and bump profile only. This is a slow lap to help generate a central line of the lap so can be ignored once used to create a map.

Would it be possible for you to kindly carry out some 7 post rig analysis and lap time simulation to dial in the car for a lap time of approx. 1.22.3 and a suspension profile of that from measured 7 post rig data?

Best Wishes
[redacted]

Figure 35. Request for damping behavior evaluation used as baseline.

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