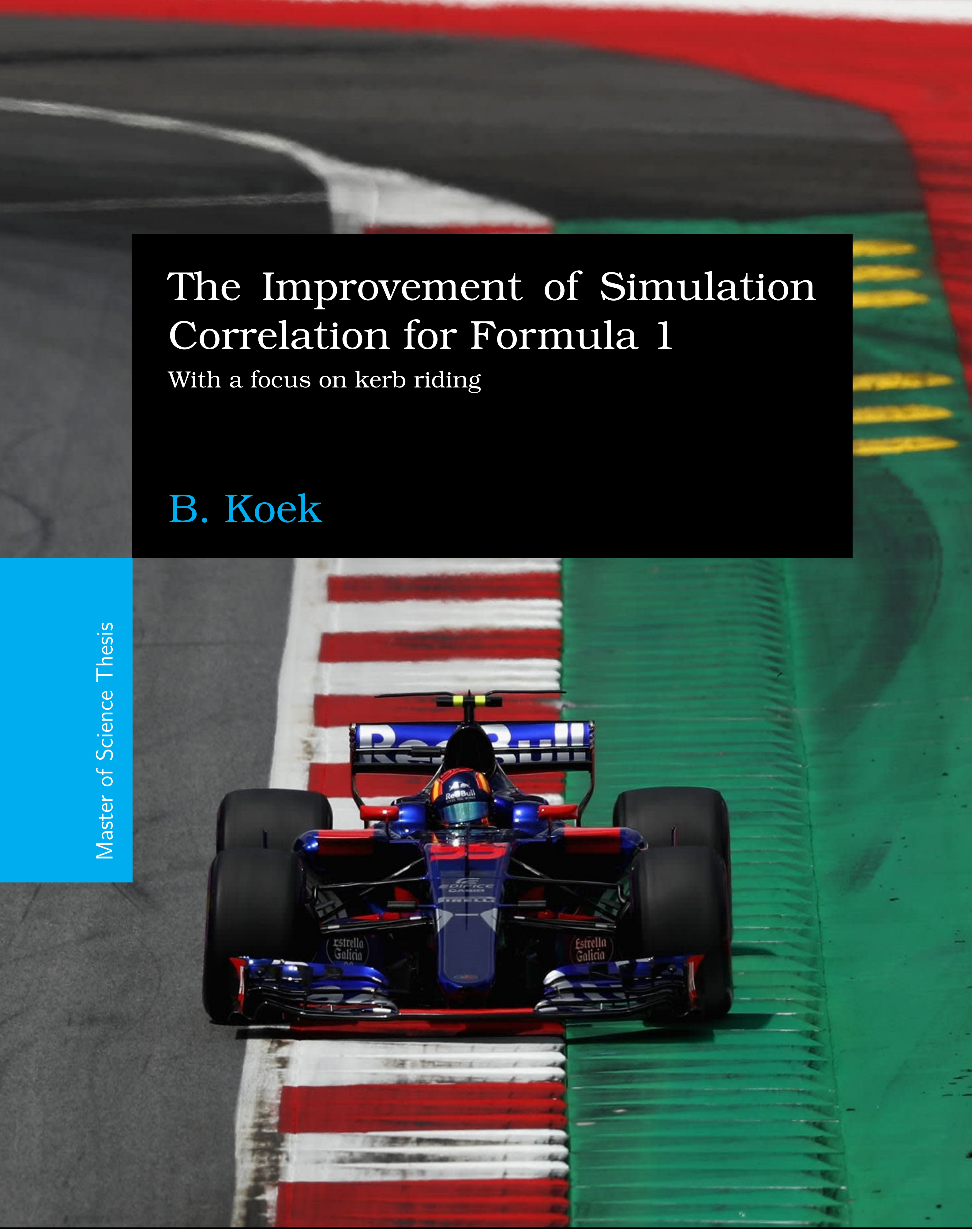

The Improvement of Simulation Correlation for Formula 1

With a focus on kerb riding

B. Koek

Master of Science Thesis

Delft University of Technology

Delft Center for Systems and Control

CONFIDENTIAL

The Improvement of Simulation Correlation for Formula 1

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Master of Science Thesis

For the degree of Master of Science in Systems and Control at Delft University of Technology

B. Koek

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Faculty of Mechanical, Maritime and Materials Engineering (3mE) · Delft University of Technology

A study conducted with Scuderia Toro Rosso.
Cover image: Getty Images / Red Bull Content Pool.

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The undersigned hereby certify that they have read and recommend to the Faculty of Mechanical, Maritime and Materials Engineering (3mE) for acceptance a thesis entitled

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Abstract

The highly competitive nature of Formula one makes every team eager to find areas of improvement on their cars. In the case of Scuderia Toro Rosso, it was found that there is a gap to improve performance while driving on kerb stones. Analysis of competitors showed that they can take wider driving lines with more speed. In order to improve this performance, the need for a high fidelity dynamic car model arose. This model can be used to understand the car dynamics, which can be used to choose a better suspension setup.

The baseline model used for this study was developed in MSC.Adams. After a discussion with field experts, two areas of model improvement were identified. In the baseline model, the tire is modeled using a MF-Tire model, where the vertical dynamics was modeled as a single spring and damper. In order to simulate the tire dynamics on kerb stones, a more advanced model is necessary. For this research, FTire was chosen as the best option. The other area of improvement is modeling the compliance of the suspension system. In the baseline model, all suspension members are modeled as rigid links, without any compliance. To model this compliance, an 'equivalent stiffness' model was proposed, where all stiffnesses are combined into flexible joints.

To use the FTire model, a set of parameters needs to be found for the Formula one tires. To find these parameters, measurements were done to capture the dynamics of the tire. A method is developed to filter out any rig disturbance, and make the data useful for parameter estimation. The estimation was done using Matlab, with MSC.Adams and FTire in-the-loop, and made use of a Genetic Algorithm (GA). Comparing the simulated behavior with the measurement showed good correlation for various conditions.

Since every part in the car contains some compliance, modeling and measuring every single part would be an extensive task. In order to reduce the complexity, all compliances are combined into a simplified model. In this simplified model, all compliances are modeled in the joints. A Pattern-Search algorithm in Matlab is used in combination with MSC.Adams to find the joint stiffnesses in such way the model matches measurements of the overall car compliance.

To measure the overall improvement, both models are combined into a full car model. The baseline and proposed model are validated against measured track data. From this compare, it can be concluded that the proposed model shows a significant better correlation on kerb simulations. This is mainly due to the fact that the baseline tire model is not capable of simulating the harsh road input. The proposed model provides a more powerful tool for engineers to better understand the car dynamics, and to find the best suspension setup for kerb riding.

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Chapter 1

Introduction

In Formula one, and motor sport in general, it is all about winning races. Every team tries to do the fastest lap to cross the start/finish line in front of the rest. Although this is the final target for everyone, from a team perspective there can be many reasons to do this [1]. Various teams, and their sponsors, race for promotional or R&D purposes.
Scuderia Toro Rosso is the second Formula one team part of Red Bull GmbH [2]. The team was established in 2006 after Red Bull bought the Faenza based Minardi team in 2005. The target of this satellite team to Red Bull Racing is to give a seat to talented youngsters of the Junior Driver Program of Red Bull. Since 2010 Scuderia Toro Rosso run independently doing all the car design and manufacturing in-house.

Scuderia Toro Rosso is always trying to maximize their performance. In this way research is done in the Vertical Performance group, with the focus on the suspension system. In this chapter we will discuss why this research is done, and what the target is.

1-1 Challenges in Vertical Vehicle Performance

After every race, the team evaluates the performance of the car. In this evaluation, engineers take a close look at their own teams car, but other cars are also sharply assessed. Video footage is analyzed and close-up pictures are studied on how other teams solve certain design problems. Another real valuable source of information is the GPS stream which every team receives from all cars on track. Using this data, engineers can estimate the car velocity (vCar) of all other cars compared to their own team.

In Figure 1-1 one of these analyses is shown. Here, one of the Toro Rosso cars is compared to a competitor from a top team. The figure is divided in three plots. On top, the track layout is depicted with the driving lines of both drivers overlayed. In the middle, one can see the speed trace. The lowest plot shows the tDiff. The tDiff is the difference in lap time compared to the other car versus the lap distance. A positive slope in tDiff means the competitor is

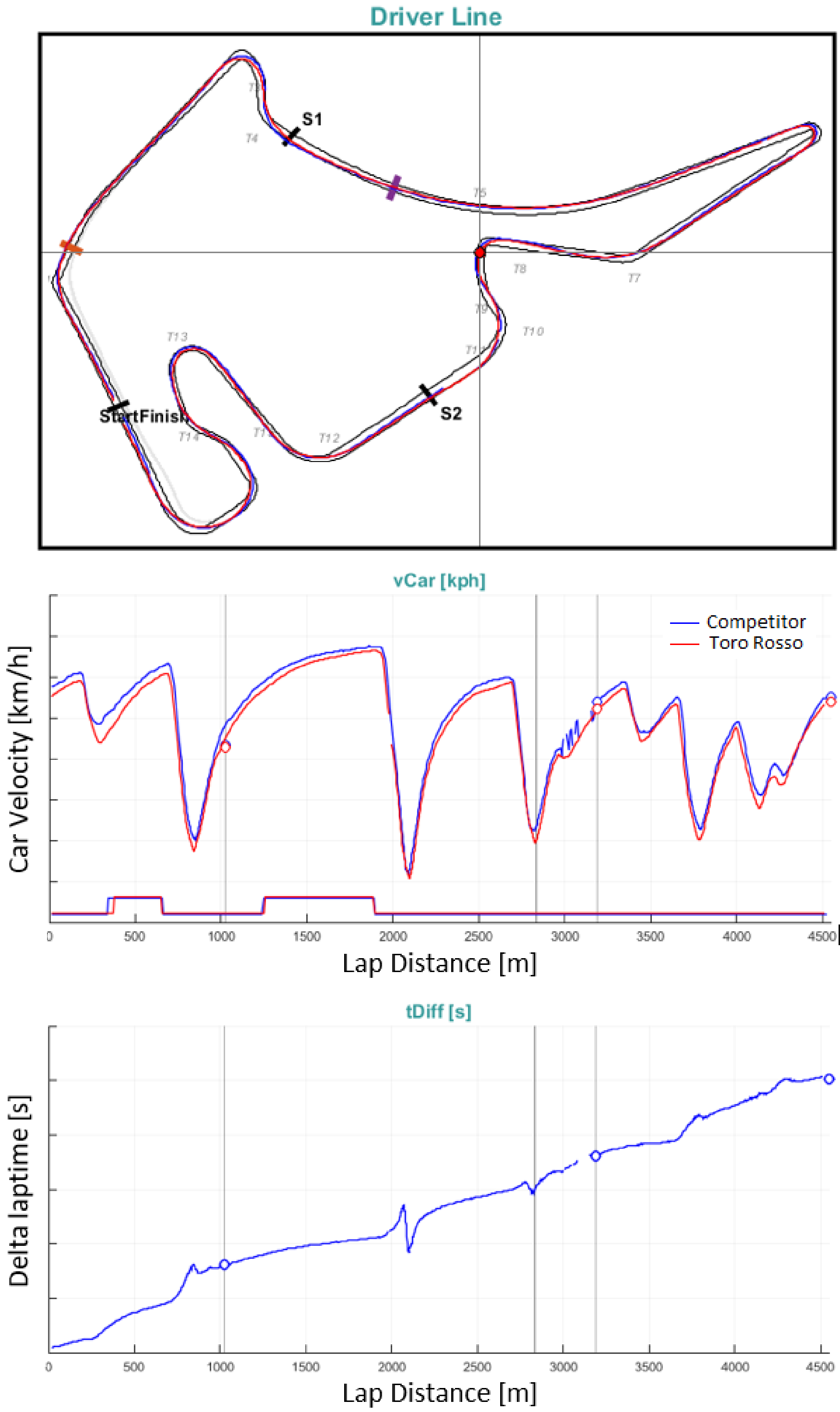


Figure 1-1: Car speed competitor analysis.

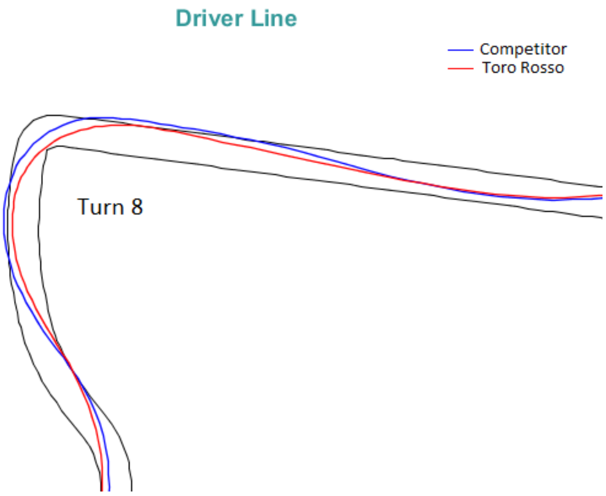


Figure 1-2: Difference in driving line at turn 8 shown on the track map.

quicker, periods of steep slope in this plot indicate phases where a lot of performance is won or lost. It can be seen that there is a performance difference in cornering. In Figure 1-2 a zoom-in is shown to one of these corners.

The blue trace is from a competitor of a top team, the quicker car in this turn. It can be seen that this car has a wider driving line compared to the red line (Scuderia Toro Rosso). It seems that driving a wider line can be beneficial for lap time. Choosing wider driving lines result in lower lateral force on the tires, which is often the limiting factor in a corner.
After noticing this performance difference, it is important to find out why the Scuderia Toro Rosso car cannot take this wider line. Understanding this limiting factor is a necessary step to improve the car to overcome this deficit.

Driving a wider line is not just a choice of the driver. Kerb stones are installed on the sides of a turn to restrict drivers taking this wider line. An example of kerb stones is shown in Figure 1-3. When a driver drives over these triangular shaped blocks, large vibrations in the tire and the rest of the car occur. It is up to the suspension to absorb these vibrations and make sure the tire keeps sticking to the ground.

In order to make the suspension work on all types of tracks, it is designed to allow a great variety of tuning parameters. These tuning parameters are achieved by changing parts of the suspension. Having a better understanding of the car dynamics during kerb riding can improve the decision making process in order to select the best parts on the car.

1-2 Simulation environment

A powerful method to predict the dynamics is using multibody simulation (MBS). MSC.Adams [4] is a powerful MBS software platform, and works with several tire models. This makes MSC.Adams suitable for this project.

To get a baseline car model, the Formula one car is modeled as a rigid body with solid suspension links attached to it. The springs/dampers/inerters are modeled using lookup

Figure 1-3: Photo of a kerb stone at the Red Bull Ring, Austria [3].

tables for non-linear behavior. To model the tires, an MF-Tire model is used where the vertical force is modeled using a spring and damper.

The baseline car model will be used as the benchmark model. Possible improvements can be measured comparing the benchmark and the measured data from track or test benches.

1-3 Research Objectives and Contributions

This research aims to contribute to the development of accurate modeling of a Formula one car, with a focus on the vertical dynamics on kerb stones. Using a scientific approach, Formula one teams can get a better understanding on what happens with the car while driving on kerbs. With this understanding, teams can engineer solutions to overcome their performance deficit.

After critical analysis with domain experts, the following fields of research are identified to improve the accuracy of the modeling:

1. **Tire modeling**, at the moment a simple model called the Magic Formula is used to simulate the tire dynamics. This model performs well in longitudinal and lateral direction, but does not model the dynamics in the vertical direction. Since the tire is highly non-linear in all directions, its dynamics can be much better represented using a more physical model. In the literature review [5] various alternatives are assessed, of which a summary can be found in Section 2-2-2. From this assessment, FTire seems the most suitable model for this project. In order to use this model, a set of parameters needs to be found to match the Formula one tire.

2. **Suspension compliance**, in the baseline MBS model, the suspension contains only rigid parts. Real car parts are not infinitely stiff and joints contain some compliance. From literature [6] it is found that compliance has a significant influence on the dynamics of the suspension. In the literature review [5], several methods to estimate compliance

values are covered. In Section 2-3 these methods are summarized. The objective is to find a method to model the compliance, and estimate its parameters.

The general problem statement for this simulation improvement study contains the following research objectives:

- Create a baseline model to which the improvements made can be benchmarked;
- Define Key Performance Indicators to benchmark the modeling accuracy;
- Implement a more physical tire model, and find a set of parameters that match the Formula one tire closely;
- Develop a method to estimate compliance values from test bench measurements, and implement this on the baseline model;
- Combine the baseline model, the physical tire model and the compliance values to benchmark the improvement of modeling accuracy comparing it to on-car measurements.

1-4 Structure

The parameterization of the FTire model and the estimation of virtual bushing stiffnesses can be done separately and do not interfere each other. This means the project can be split up into three parts:

1. Parameterize the FTire model based on historic tire data and dynamic measurements,
2. Estimate the virtual bushing stiffnesses from full car compliance measurements,
3. Apply the proposed tire and compliance model on the full car model and benchmark the improvement using on-car dynamic measurements.

An overview of this can be seen in Figure 1-4.

Before covering the parameterization of the FTire model and the estimation of virtual bushing stiffnesses, a summary of the Literature review is covered in Chapter 2.

After that, the split in three parts of the research can also be found in the structure of this work. In Chapter 3, the FTire parameterization process is covered. In this chapter the process is described from the tire testing to the estimation of a set of parameters. In Chapter 4, the estimation of the compliance is addressed. The chapter will start by defining the targets based on the measurements done. After this a simplified model is proposed, and its parameters are identified. Chapter 5 will combine the FTire and the Compliance model. This full-car model will be used to benchmark the effect on the overall car dynamics, and the improvement in simulation to measured track data. In this case an outer-kerb will be used for the benchmarking.

The thesis will be concluded in Chapter 6, where we will give the conclusions and advise for future work.

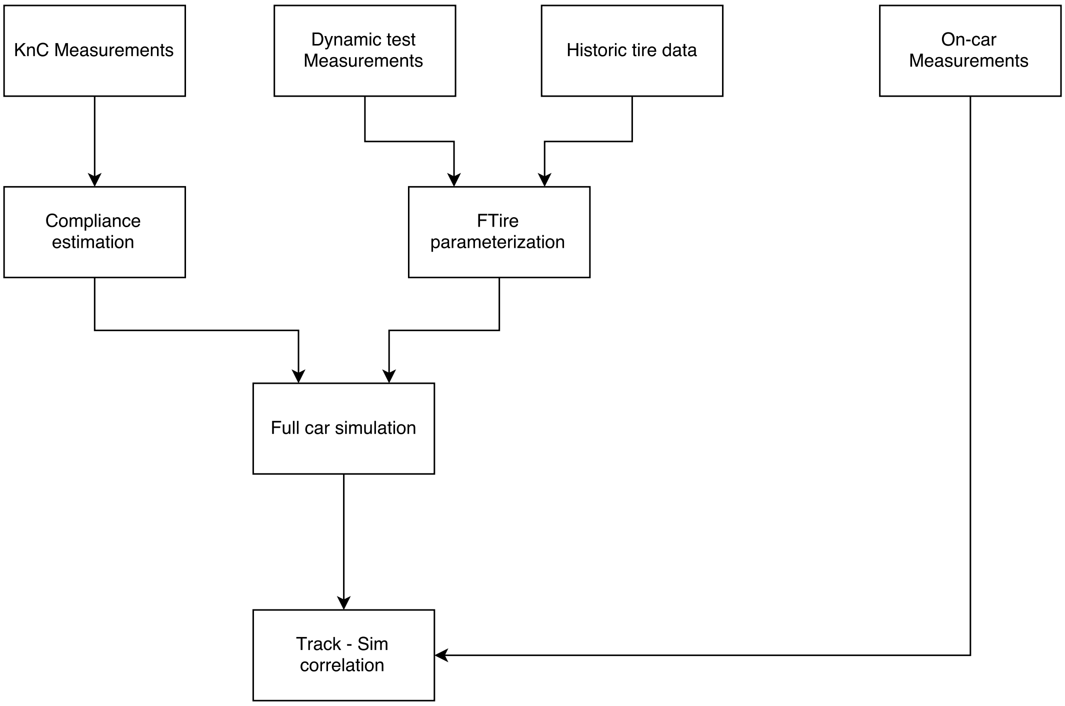


Figure 1-4: Workflow of the project.

Chapter 2

Literature Summary

2-1 Introduction

Before starting the research, a literature review was conducted [5]. In this review, literature relevant to the project was summarized and reviewed. The main findings and conclusions are written in this chapter.

2-2 Tire dynamics correlation

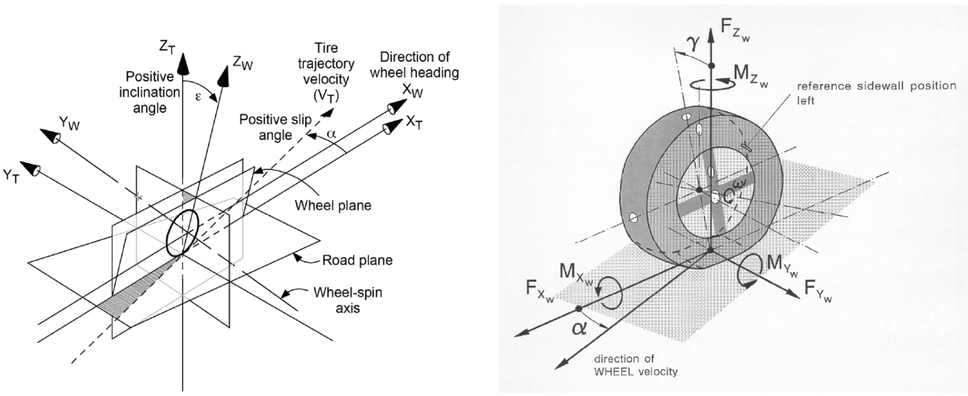
Tires are the main source of lateral and longitudinal force on the car. Since this lateral force is a key for the car to take a turn, it is really important to understand the behavior of the tires. In order to get a better understanding of a tire, several tire models were considered in the literature review. Basic definitions, and a selection of tire models are summarized in this section.

2-2-1 Definitions

Before discussing several models, we need to define various variables. Several standardizations of these variables are published. Two well-known organizations that publish these definitions are SAE [7] and TYDEX [8]. These variables are defined relative to two reference systems that are shown in Figure 2-1.

In these coordinate systems, some symbols are often used:

- F_x, the Longitudinal (Tire) Force.
- F_y, the Lateral (Tire) Force.
- F_z, the Vertical or Normal (Tire) Force.



(a) SAE reference system [7]. **(b)** TYDEX-W reference system [8].

Figure 2-1: Coordinate reference systems.

- M_x, the Yaw moment or Aligning torque.
- M_y, the Overturning moment.
- M_z, the Rolling resistance.
- γ, the Camber angle.
- α, the Side-slip angle.
- λ, the Slip angle.

The slip angle [7] can be determined as follows:

$$\lambda = \frac{v - \omega r}{\max(v, \omega r)}, \tag{2-1}$$

where

- v [m/s] is vehicle speed.
- ω [rad/s] is angular speed of the wheel.
- r [m] is the wheel radius.

2-2-2 Models

In order to choose the right tire model, one must clearly know what the use case for the simulation is. The target is to propose a model that more accurately represents the dynamic behavior in simulation during kerb riding, and get more insight into the dynamics of the system [9].

The baseline MF-Tire model [10] is good in estimating the lateral and longitudinal forces on the tire on smooth roads and runs in real-time [11]. This is very much suitable in the case of handling simulations. However, the MF-Tire model is not capable of estimating the vertical behavior accurately on short-wave obstacles. This is due to the fact that the vertical dynamics are modeled using a single spring and damper. These vertical dynamics are the main point of interest for this study, so the MF-Tire model is not the right model to use.

A good alternative might be MF-Swift [12]. Due to the modeling of a rigid ring, it can simulate high frequency content [13], and it can cope with short-wave obstacles [14]. This model can give a good representation of dynamic behavior on the kerb stones. Since it is an (semi-)empirical model, it cannot give insight information on the tire dynamics [15]. This is one of the requirements, so MF-Swift is not the best tire model for this use case.

FTire is modeled using a flexible belt [16]. This belt model is the core of the FTire model. On each belt element, a number of mass-less 'tread blocks' are associated. These blocks are in contact with the ground and contain nonlinear stiffness and damping properties. The deflections are determined by the sliding velocity and ground pressure. This creates a representative contact patch and an estimation of the forces and moments on the tire.

This extensive representation of the tire gives an accurate estimation of longitudinal, lateral and vertical behaviors, even at higher frequencies and short-waves obstacles. A disadvantage is that it is more computational expensive than the other models mentioned. The real-time performance is not the requirement of this load case since it will not be used in a simulator. This makes the FTire model the best candidate for improving the simulation results on kerb riding.

Another aspect of choosing the tire model is the possibility to find the right parameters. In Formula one, the teams are not allowed to test their tires freely. The parameterization of the MF-Swift tire model needs a lot of (predefined) tests [17]. Since FTire has a more physical modeling approach, the FTire parameters can be identified more easily with less testing [18, 19]. This advantage gives a preference to the FTire model in this study.

In Table 2-1, a summary is given of the selection criteria for the tire model. From this it can be seen that both MF-Swift and FTire are suitable for the kerb riding study. Since FTire needs less testing, and the parameters have a physical representation, this tire model is the favorite option.

	Frequency Correlation	Correlation on irregular roads	Physical testing needed (Vertical)	Parameters
MF-Tire	Low	Poor	Simple test	Empirical
MF-Swift	High	Good	Predefined set of dynamic tests	Semi-Empirical
FTire	High	Good	One type of dynamic test	Physical representation

Table 2-1: Comparison of the Tire Models.

2-3 Suspension Kinematics and Compliance correlation

The suspension is the connection between the tires and the chassis, and the particular design of it has several purposes. One of the goals is to make sure that the tire has the optimal orientation to the ground. Metrics, such as Camber and Toe for example, are defined by the suspension, and influence the behavior of the car. These metrics are described in Section 2-3-1. Another purpose is absorbing impacts from the road, and keep the tire on the road as stable as possible. Having a stable tire to ground interaction improves the potential of the tire. The last purpose is lift the car from its own weight and the additional aerodynamic down force. Since the wings have an angle on which they generate down force in the most efficient way, the suspension can be used to keep the wings in this optimal angle.

Suspensions are defined with hard-points that are located at locations of the joints in the suspension mechanism. These hard-points define geometric properties, which can be described with metrics, such as Camber, Toe, Kingpin and Caster [20]. Such metrics are explained in Section 2-3-1. Most of these properties are not constant, but are functions of wheel travel.

Another dependency is the load on the wheels. Since there is always some play in the joints of the suspension mechanism, and its links are not infinitely stiff, the suspension becomes compliant. The play and stiffness cause the hard-points to move, which results in a change of the geometric metrics [6].

The compliance affects the vehicle dynamics, especially in higher loads, so a simplified model with a limited set of parameters is necessary to cover the compliance of the suspension system. The right parameters need to be found for the proposed model in order to match the compliance of the specific car.

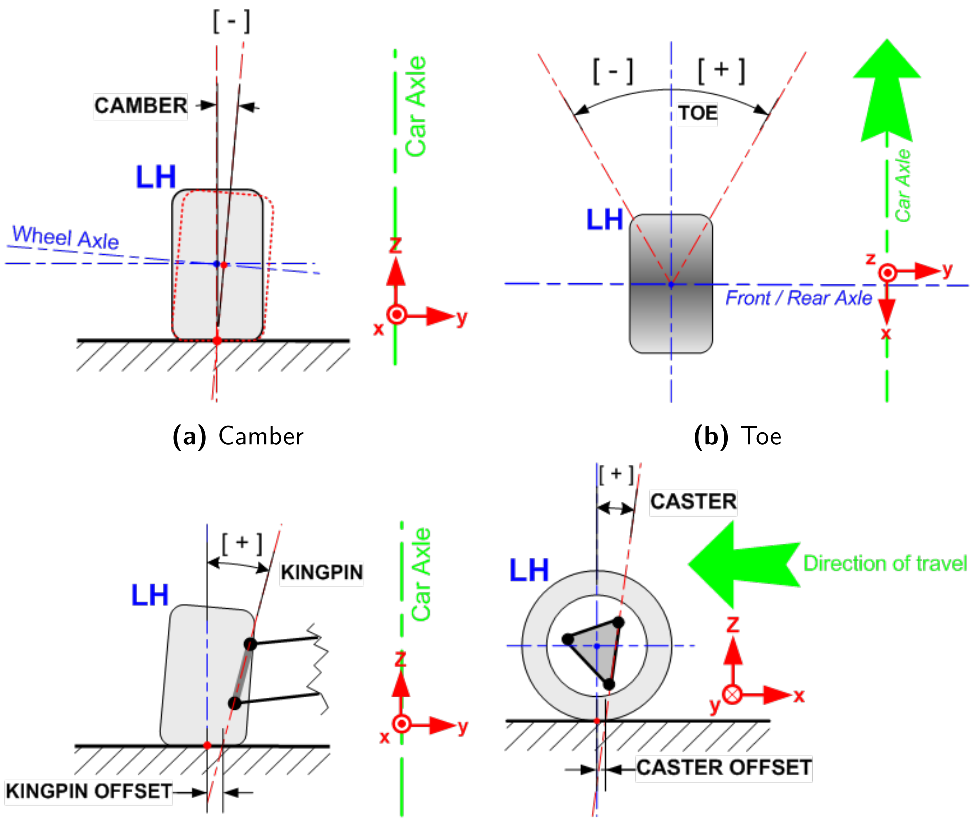
2-3-1 Definitions

In order to have a more intuitive interpretation of all the hard-points, geometric suspension metrics are introduced [20]. These numbers can help engineers to better understand dynamics of the car. In this section four of the most used geometric properties are discussed.

Camber angle is the angle between the wheels and a vertical plane as seen from the front of the car (see Figure 2-2a). If the top of the wheel is closer to the car axle than the bottom part, then we talk about negative camber. If it is the other way, it is called positive camber. Camber angle can be adjusted on the car for different tracks since it has a great influence on the handling of the car. A negative camber generally improves grip when cornering. A neutral camber can help for straight line acceleration, while a positive camber reduces the steering effort for example [1].

Toe angle is the angle of the wheel, measured from the top of the car (see Figure 2-2b). If the wheel points to the car axle, it is called positive toe, while pointing outwards is a negative toe. Positive toe, or toe in, can help the car to have better straight line stability. A disadvantage of a toe in is having a slower response in turning [1]. The toe angle is easily adjustable on the car, which allows making trade-offs for each track.

Kingpin is defined as an offset or angle. In order to get these values, one has to draw a line through the rotation points of the wheel (see Figure 2-2c). The kingpin offset is defined as the distance between the intersection of this line and the ground, on one side, and the center



(a) Camber

(b) Toe

(c) Kingpin

(d) Caster

Figure 2-2: Definition of the main geometric metrics [21].

of the wheel on the ground, on the other. The kingpin angle is the angle of this line seen from the front of the car.

Caster is similar to kingpin but measured from the side of the car (see Figure 2-2d). This measure has a big influence on the steering feel. Having a positive caster makes the car automatically steer straight, which gives the driver more steering feel. A trade-off is that it will take more steering effort. It also lifts the inner wheel and pushes the outer wheel in a corner.

2-3-2 Optimization-based Techniques for Compliance Estimation

The above mentioned properties are important for the dynamics of the car and the interaction of the tire with the road. To have a better understanding of the dynamics in kerbing, this compliance needs to be taken into account in simulations. A common way to do this is by modeling the joints as bushings [22]. A bushing connects two parts together by placing six springs for the six degrees of freedom. The three springs in x, y and z direction model the compliance in the system. The other three (torsional) springs model the friction in the joint.

Having six degrees of freedom in each joint creates a large amount of variables. Possibly, a smaller set of virtual bushings can be used to have an equivalent stiffness. This will be

Figure 2-3: A racing car on a K&C machine [24].

investigated in Section 4-1. Since all the compliances are combined in virtual bushings, the stiffness of these bushings cannot be measured. That is why a method has to be developed to estimate the values of these stiffness.

It is difficult to measure and model the compliance of every part of the car, but it is possible to measure the compliance of the complete vehicle. To measure compliance of a vehicle, a so-called Kinematics and Compliance (K&C) or Suspension Parameter Measuring Machine (SPMM) test rig can be used[23]. On this rig, as shown in Figure 2-3, the complete car can be fitted. In order to test the compliance, the springs inside the suspension are locked using a rigid bar. This is done such as to measure the compliance only, removing the free vertical movement of the suspension. The rig applies forces on the contact surface between the wheel and the ground. These forces induce forces and moments in the suspension. The induced compliance movement is measured and used for the virtual bushing stiffness estimation.

In order to find a set of bushings stiffnesses that makes the proposed compliance model fit the measurements, optimization techniques can be used. An optimization algorithm minimizes a given cost function by adjusting tuning parameters appearing in that function. A basic form of this objective cost function, is given by [25]:

$$\begin{aligned} \underset{x}{\text{minimize}} \quad & f(x) \\ \text{subject to} \quad & x \in \Omega. \end{aligned} \tag{2-2}$$

In this equation, x is the decision variable. This is a vector containing the parameters that can be tuned in order to minimize the function $f(x)$. The domain Ω, also called the constraint set or feasible set, defines the boundaries of this decision variable. The constrains can be used to set bounds on the stiffness values to get feasible results. A negative stiffness for a bushing for example might also be a result of an optimization algorithm which is physically not possible.

Several optimization methods are described in literature, each developed to solve a certain optimization problem. Main distinctions between different methods are suitability for problems of convex/non-convex and linear/quadratic/non-linear nature with/without boundaries and/or local/global optimum.

Four often used optimization methods are addressed in this literature survey:

- Levenberg-Marquardt [26, 27],
- Nelder-Mead [28, 29],
- Sequential Quadratic Programming (SQP) [28, 30],
- Genetic Algorithm [28].

In order to choose the right algorithm, the type of problem needs to be clarified. This will be part of the individual research problems. Optimization can not only be used for compliance estimation, but can be also used in the tire parameter estimation.

2-3-3 Conclusion

In order to apply the optimization methodology on a certain problem, an implementation needs to be found that is compatible with the problem. Since MSC.Adams is used in this study, only optimization methods that are compatible with MSC.Adams are considered.

Since Matlab is a software platform which is equipped with different optimization routines, one can explore if this platform can efficiently be used in combination with MSC.Adams. Using Software-In-The-Loop (SIL), Matlab can be used to apply the optimization routine. In this implementation, Matlab will change parameters of the Adams model and call the simulation. With these results, Matlab can compute the next set of parameters and iteratively find the best set of parameters.

Chapter 3

Tire dynamics correlation

In the literature review various tire models were studied and it was concluded that FTire is the most suitable tire model for this project. Since the tire has several dynamic vibration modes, it is not sufficient to model the tire as a simple spring as it was done in the baseline model. FTire should be able to model these modes, which can give the engineers a better insight in the tire dynamics.

In this chapter the parameterization of the FTire model is covered. To find the parameters, a set of measurements is needed. The first section will cover the test, and the signal processing related to that. The second section covers the estimation of the parameter set based on these measurements. This chapter will be concluded with the final results, and the correlation between the measurement and the simulations with the obtained parameter set.

3-1 Tire testing

In order to find a set of parameters that match the FTire model with the Formula one tire several sources of data are needed. Partially this data is supplied by the tire manufacturer, but one additional test is needed to capture the dynamic behavior of the tire. This additional test is a dynamic rolling rig test. During this test, the wheel is loaded statically on a rotating drum. On this drum a small obstacle, cleat, is fitted to excite the modes of the tire. A load-cell on the rig measures the vertical force and rolling resistance, and the rotating speed of the wheel is measured using a toothed wheel and a hall effect sensor.

3-1-1 Data selection and Filtering

Before using the measurements for parameter estimation, the data needs to be selected and filtered. During a run in the test, three different speeds are evaluated. A speed trace of a typical run is shown in Figure 3-1. Each individual cleat hit causes a fluctuation in the speed profile, this is filtered out by applying a 1Hz low-pass filter. The three constant speeds are

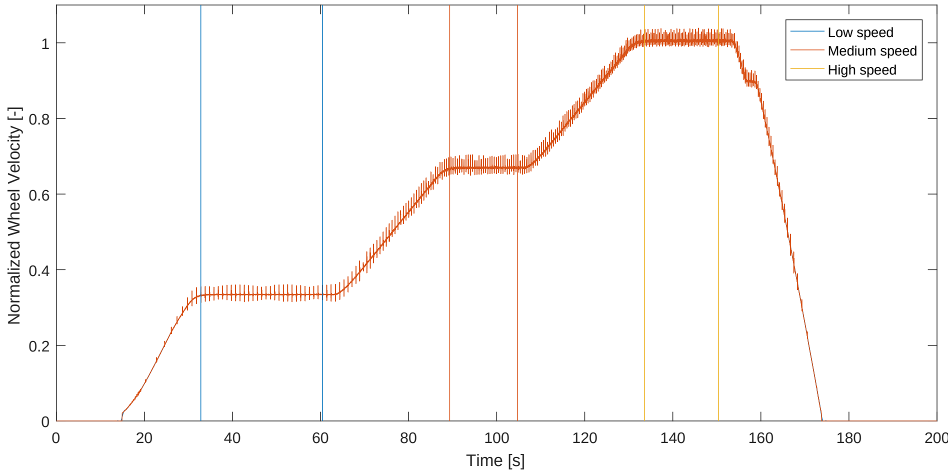


Figure 3-1: Speed profile during the test, with the selection marked.

isolated by selecting all data where the speed is within a defined margin to the target speeds. After that, a certain amount of samples on the beginning and end are removed to be sure the selection has a constant velocity.

Each individual cleat passage is isolated by computing the auto-correlation function $R_x(\tau)$ on the vertical force F_z [31]:

$$R_x(\tau) = \int x(t) \cdot x(t+\tau)\mathrm{d}t \tag{3-1}$$

The distance between maxima of this auto-correlation function correspond to the signal period. This signal period can be used to cut every passage. Now all individual cleat hits are identified, the signals need to be precisely aligned. This alignment is done using the cross-correlation function $R_{xy}(\tau)$:

$$R_{xy}(\tau) = \int x(t) \cdot y(t+\tau)\mathrm{d}t \tag{3-2}$$

The maximum of the cross-correlation function between each cleat passage and the first one corresponds to the phase time delay between the two signals. This time delay is used to shift and align the individual passages. In Figure 3-2, 26 Low-speed, 29 Medium-speed and 49 High-speed cleat impacts from a single test are aligned and shown.

From this figure, one can see that the test is highly repeatable, all individual passages overlay closely. In order to get a white noise free measurement, all individual cleat passages are averaged. Due to this high repeatability, averaging the signals does not affect the deterministic part of the signal [31].

In order to get a better understanding of the vibration modes of the tire, the Power Spectral Density (PSD) $S_{xx}(f)$ of the force signals can be used. An example of this PSD is shown on the right plot in Figure 3-2.

3-1-2 Reduction of rig disturbance

During the design of the rig, the load cell appeared to be a limiting factor for the maximum vertical load. Using the 'real' race car rims, the attachment from the rim to the rig is on

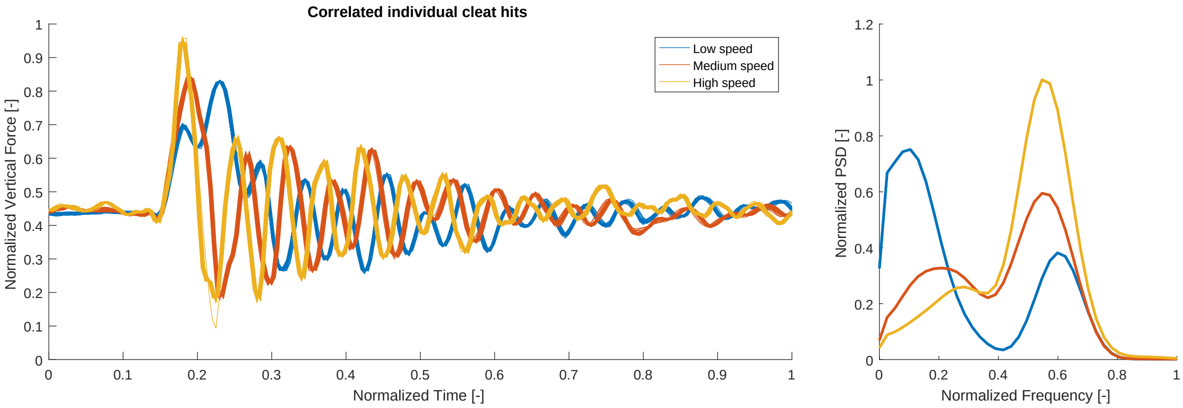


Figure 3-2: Multiple individual hits overlayed at three different speeds.

the center plane of the tire. This forces the load-cell to be fitted with an offset to this plane. In Figure 3-3 a cross section of the designed rig can be seen. On top, one can see that the load-cell is not in line with the center of the tire. This offset causes a moment on the load-cell. Due to the maximum allowed moment on this load-cell, the maximum allowed vertical force is reduced. With this reduction of maximum vertical force, no measurements can be conducted in the normal working range of the tire.

In order to increase the maximum vertical force, an alternative design was proposed. In this design, the 'real' rim is replaced by a 'dummy' rim. The 'dummy' rim has its mounting to the rig more outside the center line of the tire. This creates space to fit the load-cell on the center line of the tire, reducing the moment around the load-cell. This alternative design is shown on the bottom of Figure 3-3. In this figure, one can also see that the spacer between the base(green block on the right) and the load-cell is extended in this 'dummy rim' design. This is done to keep the tire centered on the drum without the need to move the base.

To investigate the effect of different rims on the dynamic behavior, both designs are realized. To measure the difference, a set of load cases was defined based on the maximum vertical force for the 'real rim' setup. These load cases contained a variety of speeds, temperatures, obstacle heights and tire pressures. This set of load cases was applied on both setups, and the results are compared.

In Figure 3-4 one can see the measurement results of nine load cases for both 'Real rim'(Blue) and 'Dummy rim'(Red) setups. The difference appeared to be significant, and an investigation is done on the rig compliance. Not only the rim is different between the 'Dummy' and 'Real' setup, but also the length of the spacer between the base and the load-cell. To measure the vibration modes of the rig, a Dynamometric hammer can be used. Using this hammer, an impulse is given to the system. From the force measurement in the hammer and from the load-cell of the rig, one can find a Frequency Response Function (FRF). In Figure 3-5 the response of the 'Dummy rim' setup(Red) and the 'Real rim' setup(Yellow) is shown. From this it can be seen that the 'Dummy rim' setup amplifies vibrations in the region of $\sim$ 80 - 110Hz, and damps the $\sim$ 110 - 150Hz range. This means that the compliance of the rig will influence our measurements in that frequency range. Since this is within the frequency region of interest, data of the 'Dummy rim' setup needs to be processed before it can be used to do model validation.

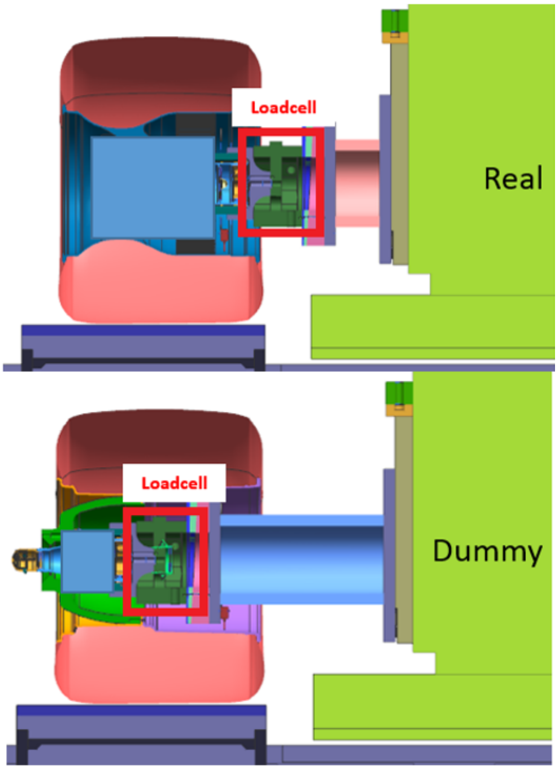


Figure 3-3: Comparison of the different configurations.

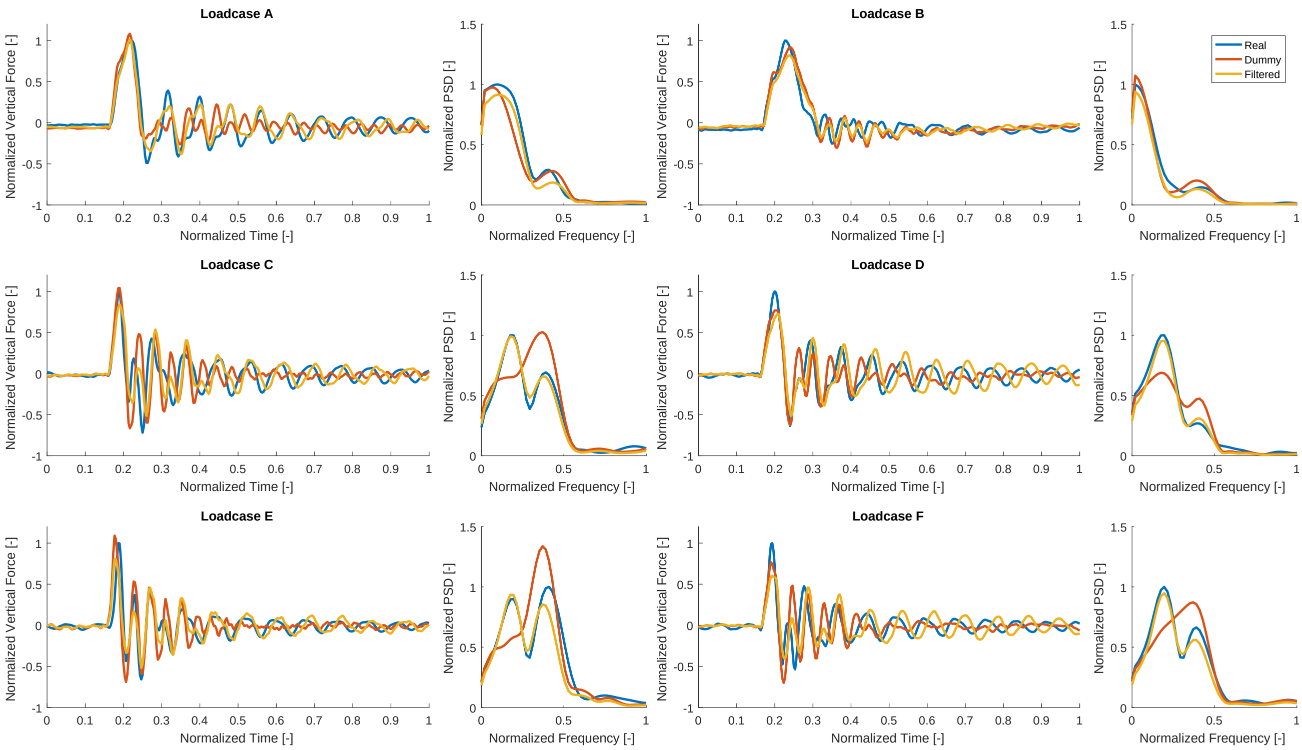


Figure 3-4: Validation set for the rig disturbance filter. Load case C was used for optimization.

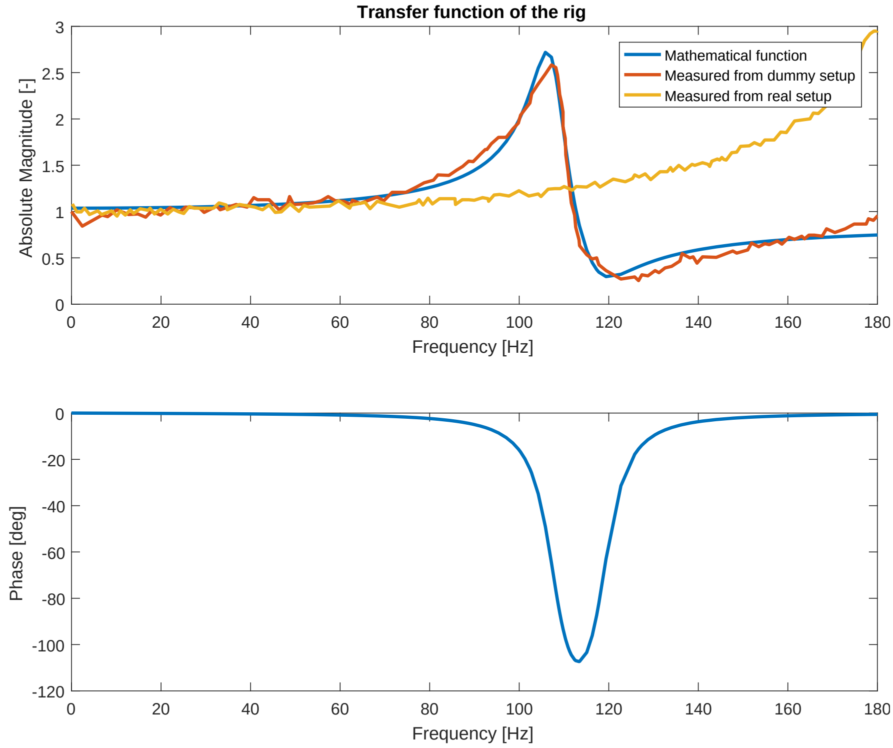


Figure 3-5: Frequency Response of the Rig compared to the found Mathematical Function.

From the FRF, a Transfer Function (TF) is found in the form:

$$\mathrm{TF}(s) = \frac{A}{s^2 + 2f\zeta s + f^2} + c. \tag{3-3}$$

In this TF, the parameters A, f, ζ and c are tuned by hand to match the measured FRF. The bode plot of this TF is plotted in Figure 3-5 in Blue.

The hypothesis is that this TF can be used as a filter to reduce the rig disturbance from the measurements done using the 'Dummy rim'. This is important since the measurements using the 'Real rim', which do not contain this disturbance in the frequency of interest, are only taken at low forces.

The performance of the filter can be measured by comparing the filtered 'Dummy rim' and the 'Real rim' data. Directly using the TF found in equation (3-3) did not yield good results. To improve correlation, a non-linear least square method was used. This optimization method tries to minimize the error between the filtered 'Dummy rim' and the 'Real rim' measurements, by tuning the parameters in the TF. This tuning was on a single load case. In order to validate that this filter is valid for all data, and not only the data used in the optimization, the results are computed for various load cases. The result of this filter is shown in Yellow on Figure 3-5. Load case C was used to optimize the parameters of the filter.

From this figure, one can see that the filtered 'Dummy rim' data matches the 'Real rim' measurements more closely. In Table 3-1 the Root Mean Square (RMS) error is displayed of both direct measurements and filtered data from the 'Dummy rim' setup, compared to the 'Real rim' measurement. This is done both for the complete time-series, as the PSD.

Table 3-1: Overview of improvement due to filtering.

	Time Series			PSD		
	RMS Error		Delta[1]	RMS Error		Delta
	Dummy	Filtered		Dummy	Filtered	
Load case A	440.80	243.21	-45%	7.32	4.53	-38%
Load case B	186.54	171.25	-8%	2.65	2.97	12%
Load case C	774.88	562.18	-27%	13.31	2.89	-78%
Load case D	346.04	372.05	8%	7.90	2.78	-65%
Load case E	578.65	380.33	-34%	12.01	7.04	-41%
Load case F	523.64	481.39	-8%	8.89	3.17	-64%
		Average	**-19%**			**-46%**

It can be concluded that using the filter, one can transform the 'Dummy rim' measurements, and reduce the rig compliance to an acceptable level. This transformed data can be used to find the parameters of the FTire model.

3-2 FTire parameter estimation

The processed measurements can be used to estimate the parameters of the FTire model. This estimation can be done using optimization techniques, where parameters of the model are tuned to minimize a defined cost function. To define this cost function a certain benchmark methodology need to be used.

3-2-1 Benchmark methodology

To measure how well the model correlates to the measurements, a benchmark methodology needs to be defined. This benchmark will be translated into a cost function to do the optimization. For this problem, there are mainly two benchmarks to be chosen from:

- Error based on the time series: the RMS error between the response of the simulation and the measurement can be taken. This metric has an advantage that the absolute values are compared. A constant offset or a different amplitude of a peak will change the benchmark. A disadvantage is that when the signals have a delay, this also will affect the benchmark.
- Error on the Frequency Response: another option is taking the RMS error of the PSD. Using this benchmark, the phase of the signals does not influence the metric. This can be an advantage when we expect a slight phase offset between simulation and the measurement after a number of oscillations. The disadvantage of this method is that

[1]Delta defined as the change in error divided by the initial error.

the absolute error is not benchmarked. A different amplitude of the initial impact for example is not penalized.

As stated, benchmarking using the PSD does not take a constant offset into account. This can be a disadvantage for various use-cases, but for this problem it might be an advantage. The static value of the response comes from the preloaded compression of the tire. Since this compression is only roughly measured on the test bench, the fixed deflection in simulation is also an approximation. Benchmarking using the frequency response is not influenced by that error. Another advantage is that benchmarking using the PSD is not influenced by the phase offset. On kerb stones, the input to the tire is high frequent. This means the overall tire frequency response needs to match, a phase offset after a number of oscillations is less important. This is why the simulation performance will be benchmarked using the RMS error between the PSD of the measurement and the simulation.

Another aspect of benchmarking is the selection of load cases. In order to have a representative tire model, a selection of load cases is made. This selection contains a set of different camber angles, static loads, cleat sizes, cleat angles and velocities. This combination gives a good impression of how well the model correlates to the measurements.

3-2-2 Estimation methodology

By changing the parameters of the FTire model, the goal is to find a set of parameters that gives the least error on the benchmark. The first step in the parameterization process is supplying the model with known data. This data contains the mass, inertia and geometric properties, and is supplied by the tire manufacturer.

The second step is selecting parameters to tune. The FTire parameter set contains numerous parameters to model various effects, for example heat transfer or tread wear. Since this is not relevant for the study, the parameters to model these effects are ignored. From the documentation, twelve parameters are selected as tuning parameters. These parameters combined define stiffness and damping values for static deflection and the different dynamic tire modes.

Since these parameters cannot be measured directly, an optimization technique will be used to estimate the values based on dynamic measurements. In order to do this, a combination of Matlab, MSC.Adams and FTire is used in an iterative way. A schematic of this Software-In-The-Loop (SIL) solution is shown in Figure 3-6.

The optimization starts in Matlab, where an optimization algorithm determines the starting values for the tuning parameters. With these initial values, a Tire Property File is generated which can be read by FTire. Then the simulations are started using a batch file which starts MSC.Adams. In this batch file, the various load-cases are defined to emulate the dynamic tests. MSC.Adams is used to simulate the test-rig, and calls FTire to model the tire in co-simulation. Once the simulation is done, the results are compared to the measurements in Matlab. To compare the results, the benchmark is computed: the RMS error in frequency domain. Based on this metric, the optimization algorithm computes a new set of tuning parameters. This process is repeated until the maximum number of iterations is reached.

The Genetic Algorithm (GA) is chosen as the optimization algorithm. The reason is that the method can work without an initial guess, is robust against local optima and can simply be implemented in Matlab.

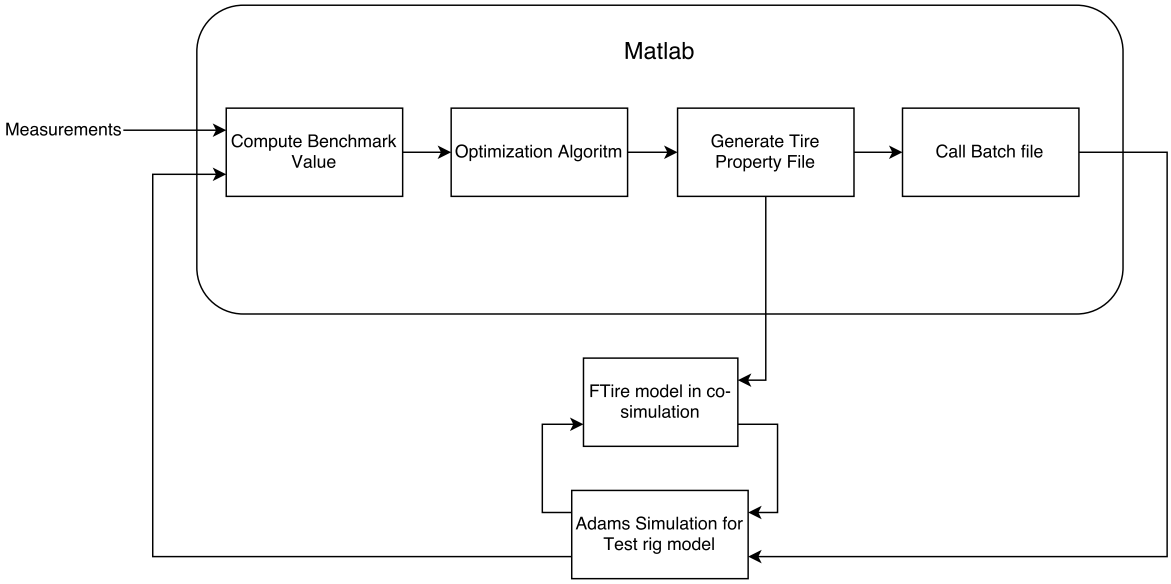


Figure 3-6: Diagram of the SIL architecture.

3-3 Results

From the filtered measurements and the benchmark methodology, a set of parameters can be found using the selected tuning parameters and optimization algorithm. In order to avoid over fitting, only two load cases are used in the optimization routine. The other load cases are used to validate the result. From this optimization, a resulting optimum was found. The results are shown in Figure 3-7. As can be seen, the model matches the measurements closely for all load cases.

Comparing the FTire model response and the Measurements with the Baseline tire model, the improvement can be seen. In Figure 3-8 a comparison is shown from a cleat impact. It can be seen that the Baseline tire model does not contain the full dynamic response of the tire. This shows that the Baseline tire model is not capable of simulating the response on short-wave obstacles, where the FTire model has good correlation.

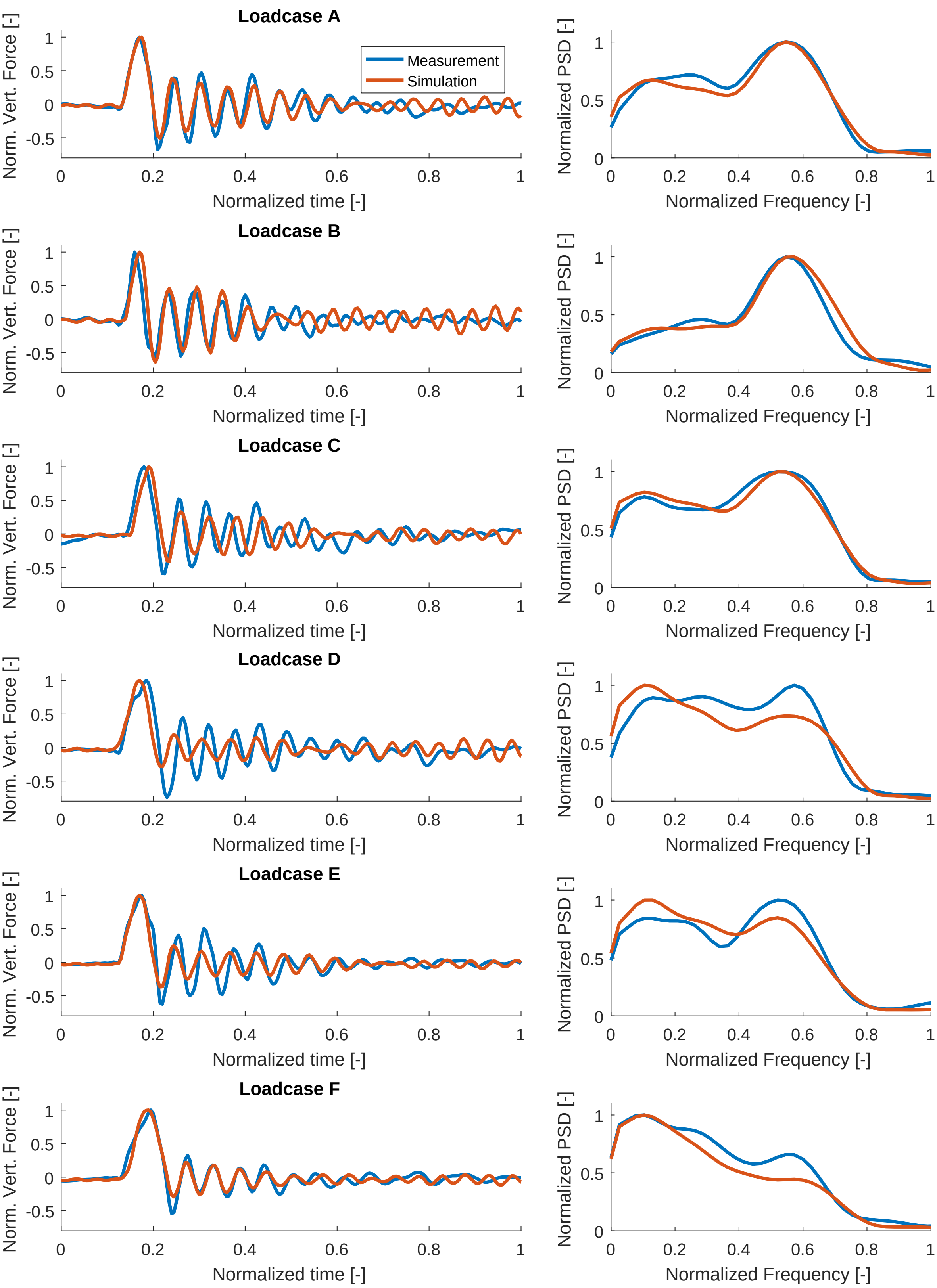


Figure 3-7: A compare of the FTire simulation to the measurement.

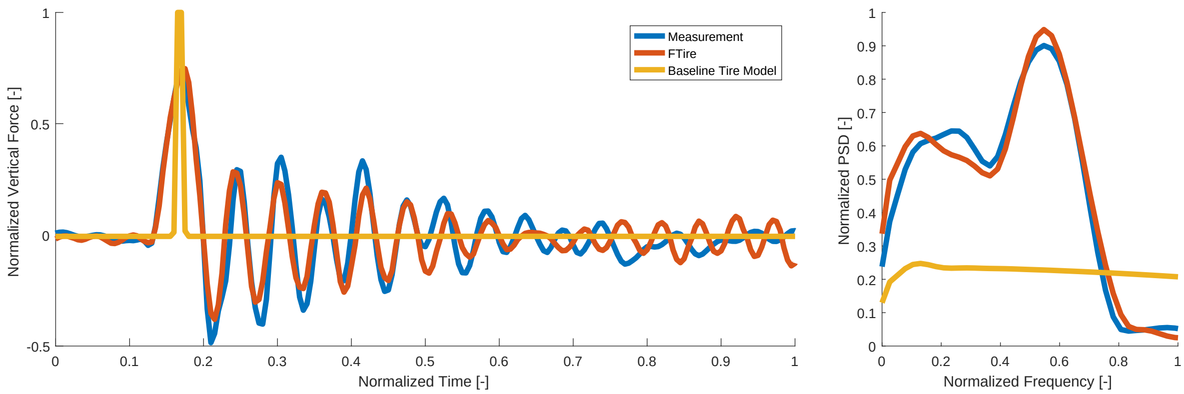


Figure 3-8: A compare of the Baseline model compared to the Proposed model and the Measurement.

Chapter 4

Suspension Kinematics and Compliance correlation

When a model is created, several assumptions are made. Often these assumptions are well substantiated, and can be made for specific simulations. In the baseline model used for this study, the assumption is made that all suspension members are rigid. This means that the orientation of the wheel compared to the chassis is defined by the suspension geometry only. In the real world, all mechanical parts have a certain stiffness. The large loads on a Formula one car while cornering will cause the suspension members to deflect. This deflection is called 'Compliance', and is an effect which cannot be ignored in a detailed kerb simulation.

In the first section, a model is proposed that can estimate this deflection on the car in MSC.Adams. In order to find the parameters, the overall stiffness need to be measured. This method is shown in the second section. To find the parameters of the proposed model that matches the measurements, optimization will be used. The methodology to do this will be covered in Section 4-3, after which the results are shown in the last section.

4-1 Modeling compliance

The compliance of the wheel compared to the chassis comes from a series of various stiffnesses, it is a combination of the deflections of all mechanical parts. This makes measuring and modeling all individual stiffness a difficult task. However, measuring the stiffness of the complete vehicle can be done using a Kinematics and Compliance (K&C) or Suspension Parameter Measuring Machine (SPMM) test rig, which is shown in Section 4-2. In order to model this overall compliance, a simplified model is suggested.

4-1-1 Model concept and DoF selection

The suspension can be divided by two main parts, the inboard and outboard suspension. The inboard suspension is defined as the springs and dampers, and are physically inside the

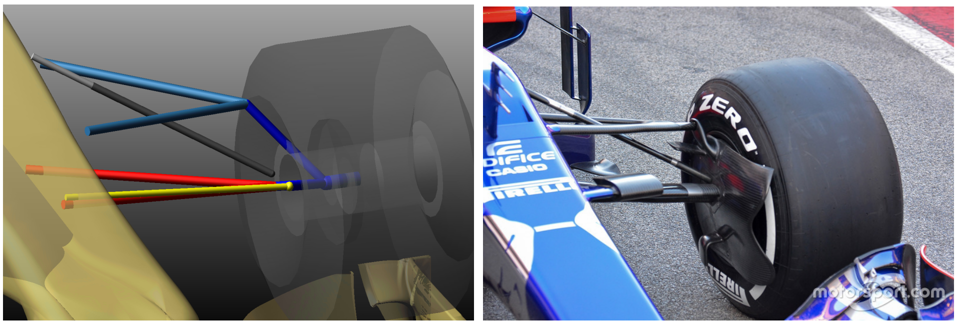


Figure 4-1: Image of the outboard suspension (Left: Screenshot MSC.Adams, Right: Photo [3]).

chassis. The outboard suspension is the part that connects the wheel to the chassis and the inboard suspension. A schematic drawing of this outboard suspension is shown in Figure 4-1.

The wishbones, represented by the Light Blue and Red rods in MSC.Adams, restricts the wheel to only two degrees of freedom, in vertical direction, and in rotation around the steer axis. The push-rod, represented by the gray rod, connects the inboard suspension to the wheel. The yellow rod represents the track-rod, this rod is connected to the steering system. The vertical motion of the wheel is constrained by the push-rod, and the rotation is constrained by the track-rod. In this figure, the inboard suspension is not shown for confidentiality reasons.

To reduce the complexity of the compliance model, it is proposed to capture all compliances in the bushings. If the joints are modeled as relatively stiff springs, all forces flow through these springs. It is assumed that the model is able to find a combination of joint-stiffness, that match the global stiffness of the car.

4-2 Compliance measurements

4-2-1 Test methodology

The global stiffness can be measured using a so called K&C or SPMM test rig. Figure 2-3 shows one of these test rigs. In this machine, the inboard suspension and the chassis is locked. This means that the arrow in Figure 4-1, connecting the inboard suspension to the push-rod, is rigidly attached to the chassis. This is done to isolate suspension deformations, and removes all degrees of freedom. Only the compliance is measured when an external load is applied.

4-2-2 KPI's

In Section 2-3-1 Camber, Toe, Caster and Kingpin were introduced. These metrics give an indication on how the tire is rotated with respect to the chassis. These numbers are not only dependent on kinematic hard-points, but also by wheel travel and compliance. In order to quantify how much these metrics change under compliance, a metric stiffness or compliance can be computed.

From these four, the Camber, Toe and Caster compliance are used as Key Performance Indicator (KPI) of the suspension. Since the Kingpin moves with the Camber, this metric is not considered. Additionally, the vertical movement due to compliance is taken into account. This gives us four KPI's:

- Vertical stiffness [kN/mm]
- Camber stiffness [kN/deg]
- Toe stiffness [kN/deg]
- Caster stiffness [kN/deg]

In order to quantify the compliance in all directions, multiple tests are done. During these tests, various combinations of loads in Vertical, Lateral, Longitudinal direction, and in rotation are applied on the wheels. For each test, the four stiffness KPI's can be computed.

4-3 Benchmarking and Optimization

For the compliance model, several stiffness values need to be estimated that makes the modeled compliance match the measured compliance on the test bench. In other words, it is necessary to find a combination of stiffnesses that provides a minimal cost. For the cost function, a weighted sum is used. Each individual metric gets a weight, based on the priority and the relevancy to the test. The vertical stiffness in a vertical test gets a greater weight than the caster stiffness in the same test for example. This is due to the fact that the vertical compliance is excited in the vertical test, where the caster compliance is almost unaffected.

$$\text{cost} = \sum_{i=1}^{n} w_i |k_i - \hat{k}_i|. \tag{4-1}$$

In this equation, n is the amount of metrics taken into account, and w_i is the weight for each metric. k_i and $\hat{k}_i$ are the measured and the simulated metric stiffness.

The cost function is used by an optimization algorithm that estimates the bushing stiffnesses. This optimization algorithm is implemented in Matlab and uses MSC.Adams using Software-In-The-Loop (SIL). A schematic of the implemented method is shown in Figure 4-2. Initially, measurements from the test bench are loaded. From these measurements, an input file for MSC.Adams is generated to replicate the test. Also the metric stiffnesses are computed from the test bench measurements. These metric stiffnesses are used in the cost function for the optimization algorithm. Matlab will call MSC.Adams, which will replicate all selected measurements with the iteratively estimated stiffness values from the optimization algorithm. The stiffness values are defined in the Bushing Property files generated by Matlab. When the simulation is done, Matlab will compute the resulting metric stiffnesses and the corresponding result from the cost function. This will be looped until an optimum is found.

To get an initial guess for the stiffness values, estimated bushing stiffness values from a Finite Element Method (FEM) are used provided by the design office. With these estimated stiffnesses, an initial value and upper and lower bounds for the optimization algorithm can be set with a certain confidence.

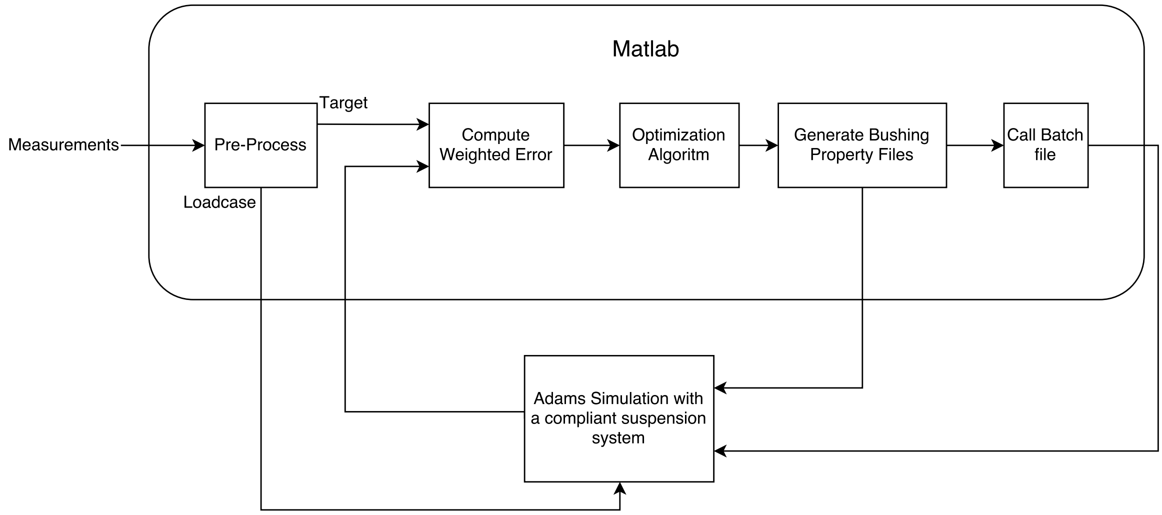


Figure 4-2: Diagram of the SIL architecture.

The PatternSearch method is used as the optimization algorithm. This method searches the bounded space from the initial value, without the need to have a derivative. This is preferable since we do not have an algebraic description of the model. Like the Genetic Algorithm (GA) for the tire parameter optimization, the PatternSearch algorithm can easily be implemented in Matlab.

In order to do this optimization in a user-friendly way, a tool is created in Matlab. A screenshot of this tool is shown in Figure A-1 in the appendix.

4-4 Results

With the described methodology, promising results are obtained. These results are shown in Table 4-1. For confidentiality reasons, no absolute stiffnesses can be shown. This is why all results are represented as normalized values to the Target value. The targets have value one, where a stiffer result has a value above one, and a softer result is below one.

In the 'Initial' column, the compliance is shown with the initial guess for the optimization. This guess comes from estimations of the design group based on measurements and FEM results. It can be seen that almost all compliances using this 'Initial' stiffness are higher than the target. This makes sense, since the overall measured compliance also contains un-modeled stiffnesses.

It can be seen that the simulated compliances are much closer to the measurements after optimization compared to the initial values. In only three cases, the initial guess is closer to the measurement than after optimization. In these cases, the loadcase was not exciting the KPI, hence the weight was close to zero. This indicates that the method of compliance modeling and parameter estimation gives a reliable representation of the car compliance.

Table 4-1: Final results of the Compliance Estimation, normalized to Target value.

		Weight	Target	Initial	Optimized
Load case A	**dZ**	w_1	1.00	3.68	1.00
	Camber	w_2	1.00	5.02	1.00
	Toe	w_3	1.00	2.89	1.16
	Caster	w_4	1.00	6.10	0.85
Load case B	**dZ**	w_5	1.00	5.06	1.15
	Camber	w_6	1.00	3.20	1.00
	Toe	w_7	1.00	1.29	0.65
	Caster	w_8	1.00	0.92	0.26
Load case C	**dZ**	w_9	1.00	8.70	0.80
	Camber	w_{10}	1.00	11.38	0.16
	Toe	w_{11}	1.00	3.17	0.49
	Caster	w_{12}	1.00	14.12	0.51
Load case D	**dZ**	w_{13}	1.00	6.62	1.77
	Camber	w_{14}	1.00	1.07	0.62
	Toe	w_{15}	1.00	1.71	0.99
	Caster	w_{16}	1.00	1.82	1.01

Chapter 5

Full car Benchmarking

With the individually identified models for the tire and compliance, both proposed extension can be applied to the full car model. In order to validate the correlation of the model, the exit kerb of turn 15 in Barcelona is taken. Figure 5-1 shows a map of this track.

The exit kerb is on the outside of the corner, where the apex kerb is on the inside. The kerb on this particular corner is on the left side of the car. This type of kerb is taken as reference since the speed on these kerbs is higher, and are more severe to the car, compared to apex kerbs. Another factor is that the car goes almost straight, reducing the complexity of transient dynamics and steering.

The baseline model and the used road profile are covered in the first section. Before adding complexity of the proposed compliance and tire models, the baseline model is validated. After that, the assumptions made for the validation will be covered. Since it is hard to compare the measurements from the track, and the simulation results, assumptions need to be made. The chapter will be concluded by the final results, and a benchmark between the baseline and the extended model.

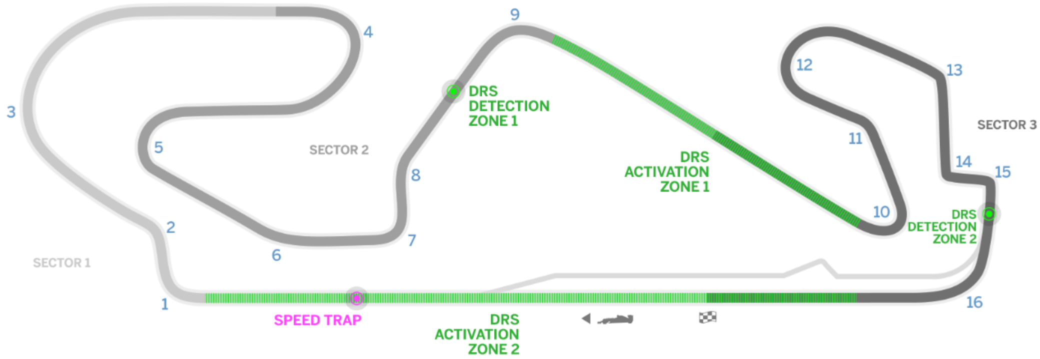


Figure 5-1: Barcelona track map [32].

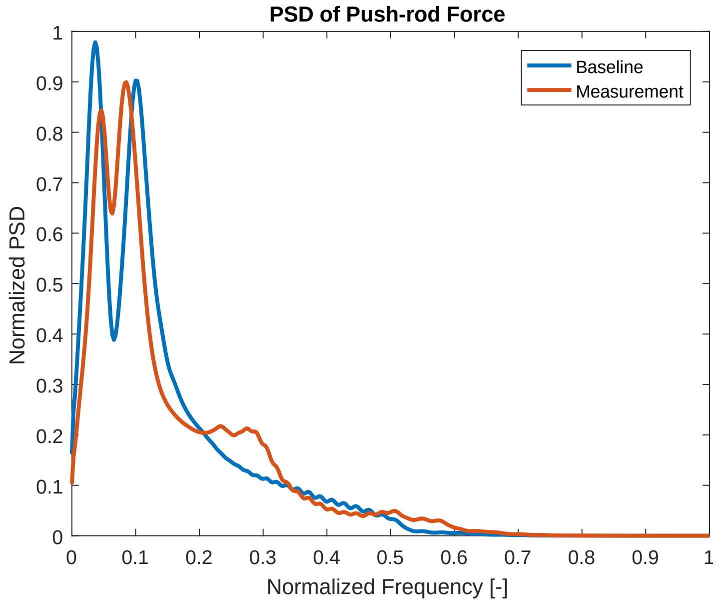


Figure 5-2: Response of the baseline model and the measurement in a simple test.

5-1 Baseline Model

Before introducing the FTire model and compliances, the baseline model was developed and validated. During this process, all basic properties of the model were reviewed and tuned where necessary. In order to do this, the model was compared with measurement data from a dynamic rig. On this rig, flat actuators under the tires excite the dynamics of the car. Since this test is easily reproducible in simulation, a good compare can be made.

Due to confidentiality reasons, this full validation process can not be shown. To show that there is a global match between the baseline model and the measurement, a PSD of the push-rod is shown in Figure 5-2. From this plot it can be seen that the baseline model with the standard tire model correlates well when a simple input is applied. This aligns with the literature review, where it was high-lighted that the MF-Tire model is valid for low-frequency road input.

5-2 Simulation and Assumptions

In order to have a representative model of the road, a geometric model of the kerb stone was made. The exit kerb taken for this simulation is 800x800mm in size, and 25mm deep. MSC.Adams reads the road as connected nodes in triangles. These nodes are defined using 3D coordinates. A Matlab script was created in order to compute all nodes and triangles, and export a road geometry file that can be read by MSC.Adams. An example is shown in Figure 5-3.

Formula one cars generate about 4-5 times their own weight in down force. This down force is a significant factor of the normal force of the road to the tire, and need to be taken into account in the simulation. In order to do this, three virtual 'aeroloaders' are added in the model. The

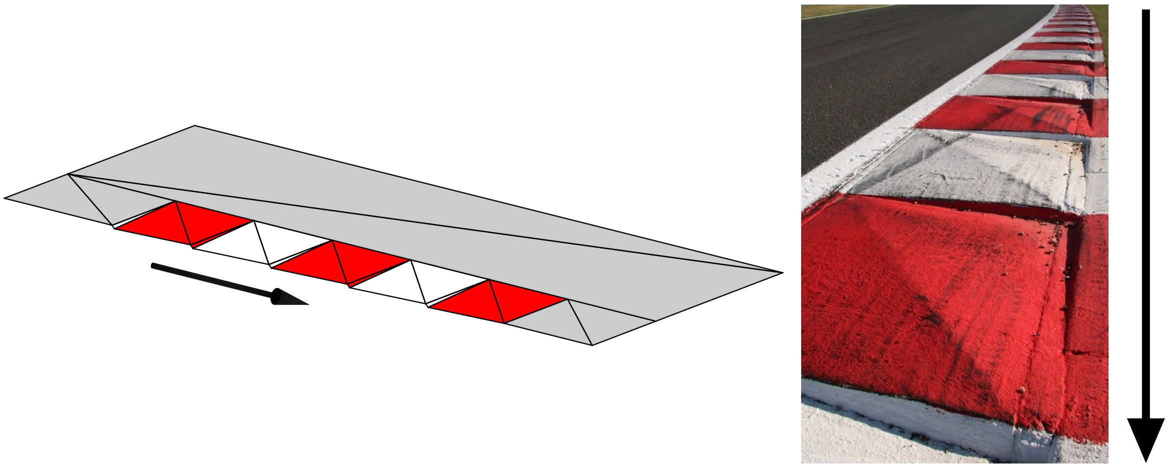


Figure 5-3: Geometry of the Kerbstone, on the left the 3D model, on the right a photo [33].

forces of these aeroloaders are found using an internally used, confidential, methodology. This methodology finds the three aeroloader forces based on the four (low-pass filtered) Push-Rod force measurements on the car.

Since it targets all four Push-Rod forces, the pitch and roll moment due to the longitudinal and lateral acceleration are replicated by the aeroloaders. This simplifies the simulation since the steering does not need to be modeled in simulation, but the roll moment comes directly from the aeroloaders. This gives more repetitive, and hence more comparable, simulation results.

Also the pitch moment due to the longitudinal acceleration is captured in the aeroloaders. This restricts the possibility to accelerate the car in simulation since this would cause a difference in normal force on the tires, and hence a different response from the tire model. This is why the assumption is made to model the car at a constant velocity. It is know that this assumption has an impact on the input frequency of the kerb stones. To see the sensitivity of this assumption, simulations were done on three different velocities. All simulations are done with the FTire model and the compliance already enabled.

As can be seen in Figure 5-4, the velocity certainly has an impact on the results. Especially on the PSD's in the range of 0.4-0.6 times the normalized frequency. This frequency comes directly from the velocity of the car and the size of the kerb. For further simulations, velocity A is always taken as reference.

Another uncertainty is the driving line of the car, it is not exactly known where the wheels are on the kerb stone. To see the sensitivity on the driving line, three different lines are simulated. Lines A to C represents these different lines, where A takes the least kerb, and C the most.

In Figure 5-5 the simulation results of this sensitivity study is shown. It can be seen that taking more kerb results in a more powerful signal. From now on, driving line B will be taken as the reference.

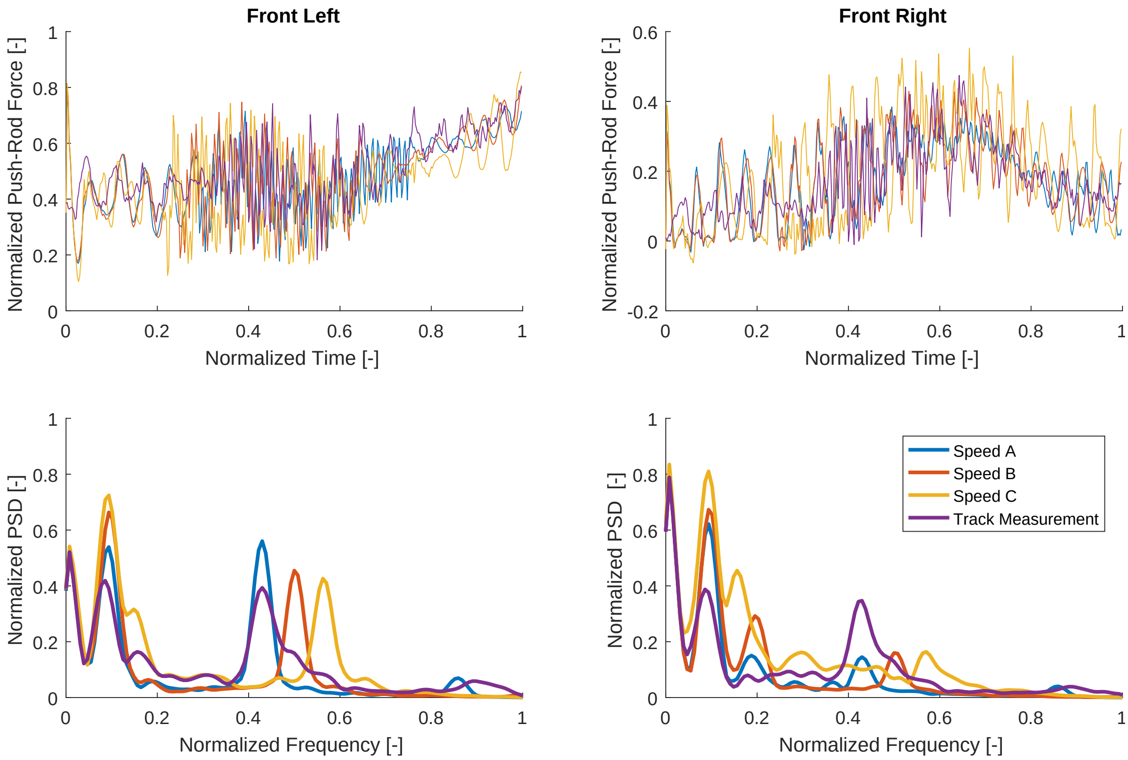


Figure 5-4: Sensitivity on the Car Velocity.

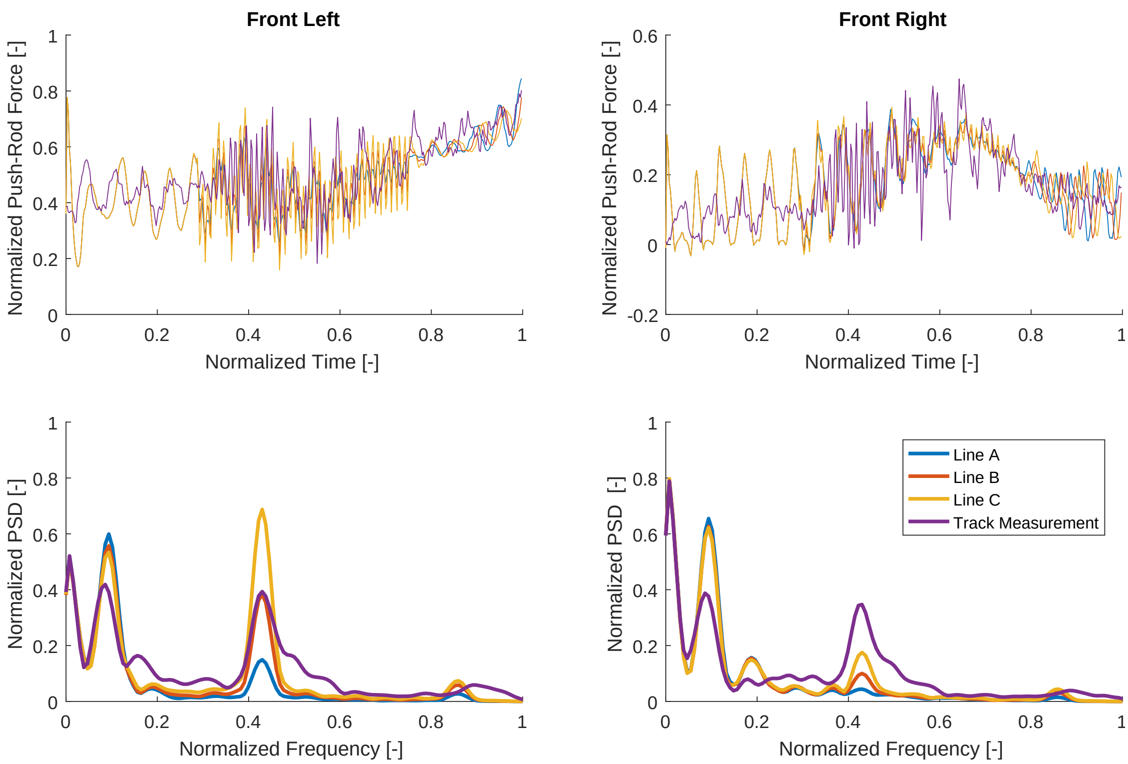


Figure 5-5: Sensitivity on the Driving line.

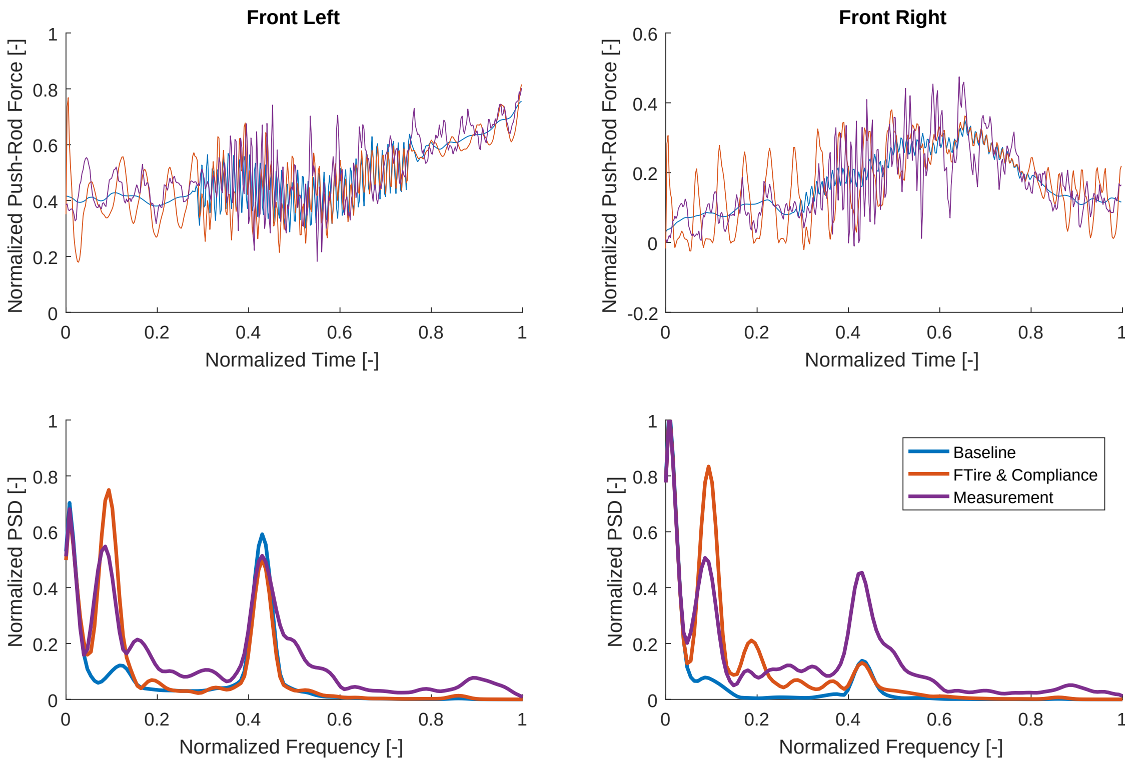


Figure 5-6: Plots of the baseline and final model compared to Track Measurements.

5-3 Results and Benchmark

With the defined assumptions in mind, the final results can be analyzed. Since the results are extremely sensitive to a difference in driving line, an absolute comparison cannot be made. In order to give a relative comparison, the frequency domain can provide interesting information. A comparison between the baseline model, the proposed model and the measurement on track is shown in Figure 5-6.

As it can be seen, there is a peak in the 0.4-0.6 times the normalized frequency range. Where the Baseline and the Extended model have a sharp peak, the measurement has a peak over a wider frequency range. This difference is due to the velocity of the car as mentioned in Section 5-2. If the velocity in simulation would follow the real car velocity, the sharp peak would be more spread out like the on-car measurement.

In the low frequency content, under 0.2 times the normalized frequency, different behavior can be seen. The baseline model does not correlate with the measurement, where the proposed model has the same oscillation peak. This is due to the fact that the tire model used in the baseline model is not capable of simulating high frequency road input. This clearly shows the benefits of the proposed model including FTire and the compliance model over the Baseline tire model.

Chapter 6

Conclusions and Future work

6-1 Conclusion

The goal of this research is to improve the dynamic car model to have a better understanding of the car dynamics during kerb riding and help the engineers to make better decisions on car setup. This can improve the performance of the car on the track, and can lead to better results in the races. In order to achieve this improvement, two areas of research were identified in the literature survey. The first area is the Tire Dynamics correlation. In the baseline model a relatively simple tire model was used to simulate the vertical force of the tire. FTire was found as suitable replacement, which is an advanced dynamic tire model. The other area of improvement is the modeling of the Compliance. Since it is relatively easy to measure the overall compliance of the car, an 'equivalent stiffness' model was proposed. In this simplification, the various stiffnesses of all parts are combined into a combination of joint stiffnesses. These stiffnesses can be found using optimization techniques.

In order to use the FTire model, its parameters for the specific Formula one tires need to be found. This was done in two steps. First, the dynamic response of the tires was measured using a special designed test rig. From these measurements it appeared that the rig contains compliance which cannot be neglected. A method to filter the data was proposed that reduces this effect (see Section 3-1). In the second step, the filtered data was used as target for the parameter estimation. For this parameter estimation, Matlab was used to run an optimization algorithm, which called MSC.Adams and FTire to simulate the tire response. The optimization method chosen was the Genetic Algorithm (GA) since it does not need initial values, and is robust against local optima.

The model of the compliance was covered in Section 4. In this chapter a simplified model was proposed to capture all stiffnesses of all parts into a combination of joint stiffnesses. This reduced the number of stiffness values dramatically, and makes it possible to use optimization techniques to identify these joint stiffnesses. The optimization algorithm runs again in Matlab, calling MSC.Adams in which the compliance is modeled. The optimization has the target to reduce the error between the model and the compliance measurements from the car. The error measurement is defined from standard car parameters, like Camber and Toe, that change by external loads due to compliance. Results show that the proposed model is capable of giving a good estimation of the suspension compliance.

To measure the overall improvement of the simulation correlation on the kerb stones, the FTire model and the compliance model were combined into a full car model. The model was validated by comparing the baseline and proposed model to measured track data on an exit kerb. Due to a high sensitivity on driving line and car velocity, an objective compare cannot be made. What can be seen from the frequency domain, is that the baseline model does not well replicate the car dynamics. This is due to the incapability of the baseline model to simulate non-smooth road-inputs. This certainly is an indication that the FTire model and the compliance model improve the correlation of the simulation on the kerb stones.

It can be concluded that the baseline model, using MF-Tire model and no compliance, is useful for general simulation. Furthermore, it shows good correlation to measurements on the car for smooth road inputs, and can be used for overall tuning. However, the baseline model shows poor correlation with the measurement of driving on the kerb. In this case, the proposed model shows a significant improvement compared to the model and provides a better approximation of the car dynamics.

Furthermore, the proposed FTire model provides better insight of the tire dynamics. Since the complete belt of the tire is modeled, its dynamics can be analyzed. This give the engineers more information about the contact patch shape for example, or about the vibration modes of the belt itself. Information that the engineers did not have available so far.

6-2 Future work

With this model, the engineers can better understand the car dynamics on kerb stones, and use this understanding to improve the setup on the car. The dampers can be tuned to have a more constant and bigger contact patch, and the driver can be coached to find the best driving line for example. These relatively small changes can help the team to improve their general performance, and reduce their lap times. Ultimately this can lead to better results in the race.

From a modeling perspective, a number of topics are highlighted on which further research would be beneficial:

Apply the methodology on the rear of the car

In this research only the front suspension and tires of the car are considered. With the information gathered in this study, it should be possible to apply the same methodology for the rear suspension and tires. This would give the opportunity to do detailed analysis on the full car. It might also affect the results of the overall analysis, since both sides influence the dynamic behavior of the full car.

Validate the model on other types of kerbs

For now, the model is only validated on an Exit kerb. To see how the model behaves on other types of kerb stones, extra validations need to be done. Different kerb stones have various geometries, and are taken with different velocities.

Introduce chassis compliance to the model

In this study, the car is divided in a front and rear part. An assumption is made that the main compliance comes from the suspension, and that the compliance of the chassis can be neglected for now. In order to improve the overall compliance correlation, this chassis compliance can be added.

Appendix A

Compliance Estimation tool

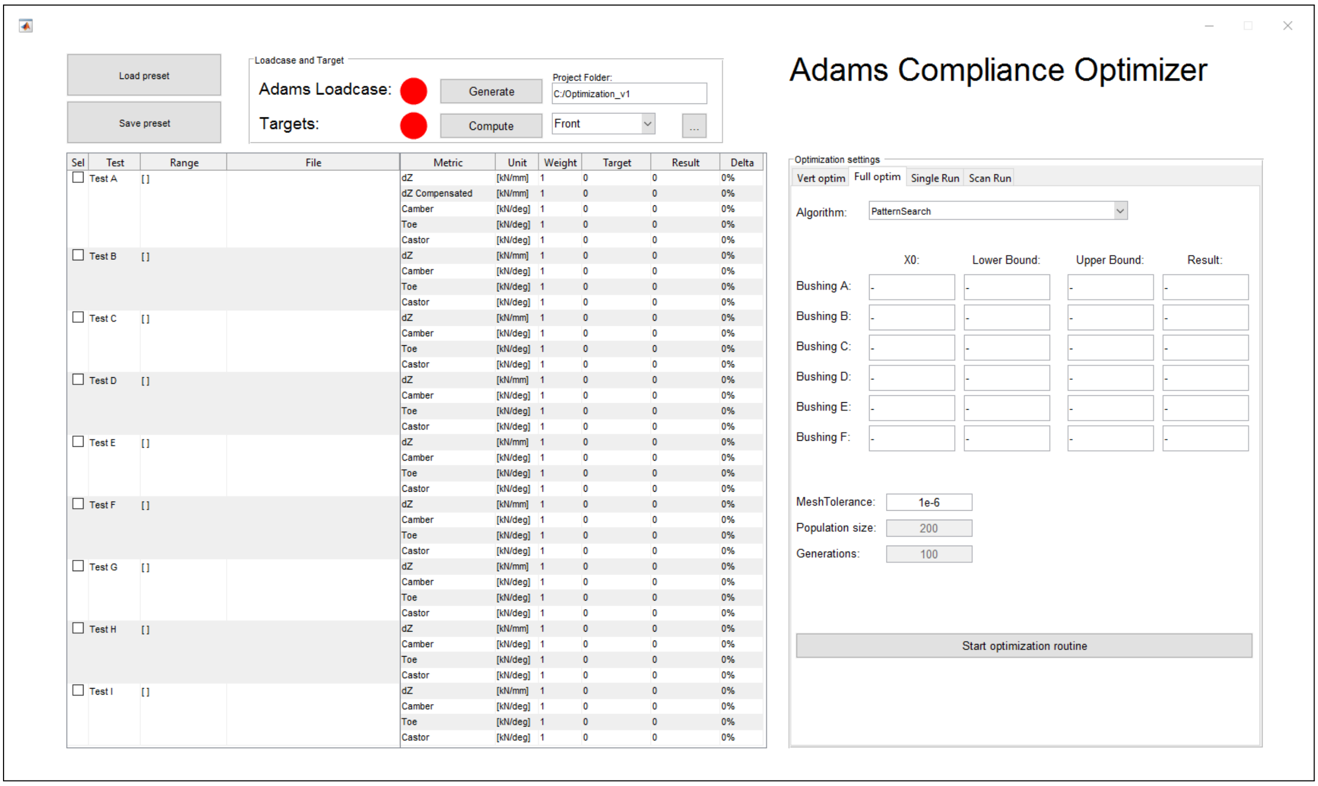


Figure A-1: Screenshot of the tool developed to do the optimization.

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Glossary

List of Acronyms

PSD	Power Spectral Density
FRF	Frequency Response Function
TF	Transfer Function
RMS	Root Mean Square
vCar	car velocity
MBS	multibody simulation
K&C	Kinematics and Compliance
SPMM	Suspension Parameter Measuring Machine
SIL	Software-In-The-Loop
GA	Genetic Algorithm
KPI	Key Performance Indicator
FEM	Finite Element Method