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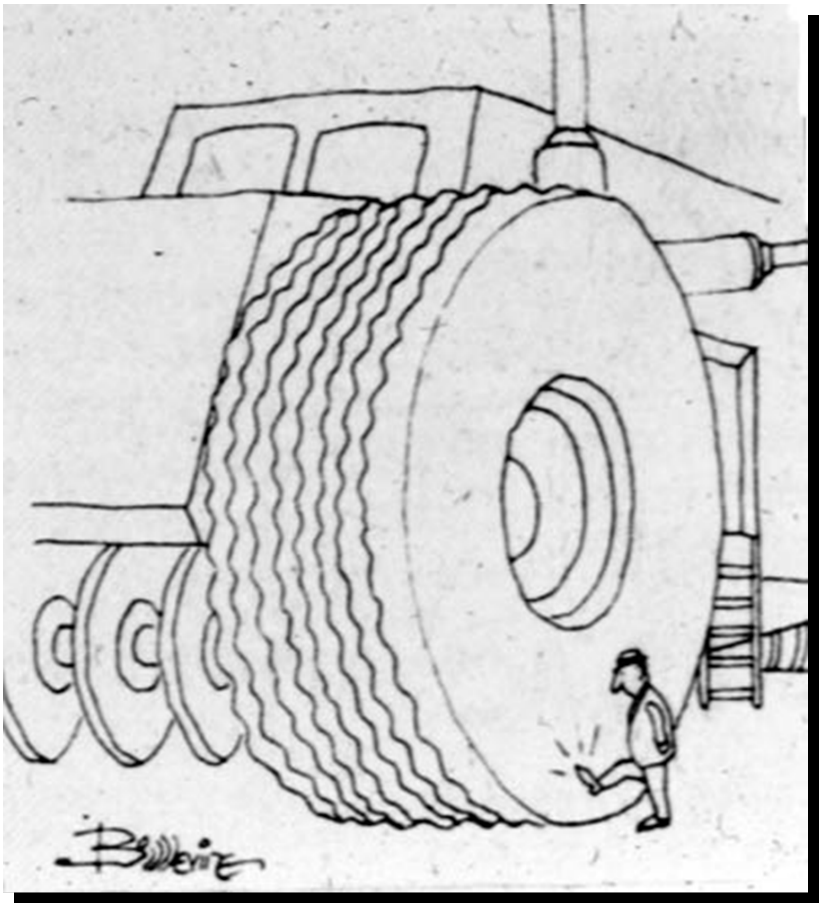

Tire Characteristics Sensitivity Study

Master's thesis in Automotive Engineering

FOAD MOHAMMADI

Department of Applied Mechanics
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Vehicle Engineering and Autonomous Systems
CHALMERS UNIVERSITY OF TECHNOLOGY
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Department of Applied Mechanics
Vehicle Dynamics Group
Vehicle Engineering and Autonomous Systems
Chalmers University of Technology
SE-412 96 Gothenburg
Sweden
Telephone: +46 (0)31-772 1000

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Abstract

Tires are the main contact between a vehicle and the road. Tire modelling for full vehicle simulations is a challenge due to tires' non-linear behaviour and strong coupling between their different degrees of freedom. Several approaches for tire modelling with regard to handling are available. Empirical models such as Magic Formula fitted to test data are the most widely used. The aim of this thesis is to find an alternative tire model with relatively few parameters that are physically meaningful and hence suitable for parametric sensitivity studies. The model should also be representative of tire behaviour by an acceptable fit to tire test data.
A survey of applications of tire models and suitable parameters for sensitivity analysis has been performed within Volvo cars with focus on handling. Simultaneously a literature study has been done to identify and categorize available tire models.
TMeasy was chosen to be used in the parametric sensitivity studies. This semi-physical tire model fits to test data relatively well and is based on fewer parameters compared to the Magic Formula. The main reason for the selection of TMeasy is that the parameters describing the model, represent tire characteristics that were identified in the application survey as target parameters for sensitivity analysis.
The effect of tire parameters on vehicle handling characteristics have been investigated by implementing TMeasy in Adams/Car and performing full vehicle simulations.

Keywords: Tire models, TMeasy, Sensitivity analysis, Vehicle dynamics, Handling

Preface

This thesis work has been conducted as a partial requirement for the Master of Science degree in "Automotive Engineering" at Chalmers University of Technology, Gothenburg, Sweden in cooperation with Volvo Car Corporation. All the stages of the project were performed at CAE vehicle dynamics team, CAE & Objective testing, Volvo Cars Research & Development during January – June 2012.

I would like to acknowledge and thank my supervisor at Volvo Cars, Sergio da Silva for his guidance and support and my examiner at Chalmers, Mathias Lidberg for sharing his knowledge and expertise. Special thanks to all the engineers that participated in the application survey, for it would have been simply impossible without their help. I also would like to thank all the staff of the CAE vehicle dynamics group for their inputs and unconditional help throughout this project.

Gothenburg, June 2012
Foad Mohammadi

"Automobiles and trucks are machines for using tires."

- Maurice Olley, 1947

Contents

1 Introduction

1.1 Motivation and background

In 1887, John Boyd Dunlop made the first practical pneumatic tire from rubber for his son's bicycle. Dunlop's patent was declared invalid due to a prior work done on rubber made tires. He was credited with "realizing rubber could withstand the wear and tear of being a tire while retaining its resilience." That was the beginning of a new field of research and a new chapter in the automobile history. Until the early 1920's extensive experimentation was done by tire companies to increase the durability of the rubber by adding different materials to it such as wood, leather and finally steel cords. Pneumatic tires started to be used in aircraft and automotive industry. This started a new era in vehicle design from ride and handling perspectives. The importance of tires became more and more clear to engineers and researchers as automotive industry evolved. Specially during the past three decades enormous progress has been made in chassis engineering and drivability. Today's vehicle are significantly safer and more comfortable compared to their predecessors. The chassis and suspension design and tuning has been improved by implementation of new analysis techniques and computer simulations. Evolution of electronic control systems has also provided new opportunities in chassis control, among which the most well known are the anti-lock brakes, active suspension, and traction control systems. In general, the product maturity increased in the automotive industry despite the short development times. This has brought new challenges for the engineers leading to more and more reliance on computer simulations and analysis of the components and systems. Being the only contact between the road and the vehicle, the importance of tire modelling became more evident. As depicted in Figure 1.1.1, the effect of tires on overall vehicle operational characteristics is significant.

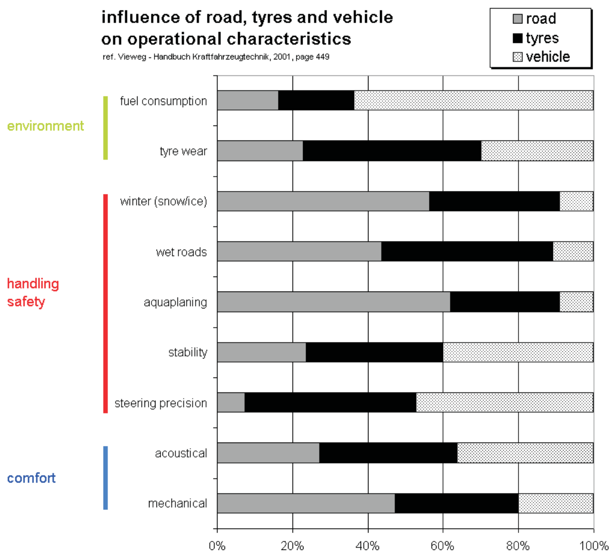

Figure 1.1.1: *Influence of road, vehicle and tires on overall operational characteristics [3]*

Still, after 125 years from their introduction, tires are treated as a black box. Tire behaviour is heavily dependent on its structure, construction and rubber decomposition. The later appears to be the most complex aspect. For many years, tire rubber compounding was driven by experimentation. The relation between a specified measure of a specific tire compound and its effect on the road forces is very hard to derive, if not impossible. Material and assembly variations occur during manufacturing and curing leading to the fact that even in the same batch of production, there are not two tires that are exactly the same. The other challenge in modelling tires is their non-linear behaviour and strong coupling between different degrees of freedom. There are several parameters that effect tire behaviour. These parameters in most of the cases have mutual influence on each other. Extensive research has been done to identify these relationships and isolate their effects.

The rise of precision electronics and measurement systems enabled tire and vehicle engineers to have a deeper insight into tire modelling by measuring tire force and moment characteristics using test rigs and test vehicles. These tests, apart from being extremely costly, are entitled to problems such as repeatability, tire wear, road condition variances and measurement errors. Furthermore, these tests are done for only a limited number of working conditions. This necessitates the need for a model of some sort to interpolate or extrapolate the force and moment characteristics for other working conditions. On the other hand, for tire testing it is required that an actual tire is built and measured in a testing facility. Due to short development times in automotive industry, prototype tire availability for testing is very costly and sometimes impossible. Another consequence of these short development times was more reliance on computer modelling and simulations rather than prototype testing. These were the driving forces for the introduction of tire models.

Today, understanding the tire behaviour and modelling techniques is vital for product development within the automotive industry. This is required to avoid costly prototypes and replace them with computer simulation models. It is needed to assist the vehicle-tire co-development process in which tire suppliers and vehicle manufacturers work in harmony to deliver a good overall system (the vehicle) rather than separately developed subsystems in isolation. The knowledge gained from studying tire behaviour in handling can assist the development of suspension and chassis systems leading to improved vehicle handling characteristics and resulting in more driving pleasure and safety for the end costumer.

1.2 Scope

At the start of the vehicle development there is none or very little data of the final tire available for modelling purposes. Furthermore, due to the aforementioned challenges it is hard to isolate the effect of tires in vehicle dynamic testing. To better understand the effect of tire parameters on vehicle handling, sensitivity studies can be performed by varying a certain tire parameter and investigating the effect on a certain vehicle dynamic handling characteristic.

Therefore there is a need for a tire model that is able to change the tire physical characteristics separately and be relatively easy to use in vehicle dynamics simulations in terms of computational effort and user-friendliness. Empirical models such as Magic Formula has been used for many years in vehicle dynamic handling simulations and incorporates some capabilities for such sensitivity studies. Although the large number of parameters and scaling factors makes it somewhat difficult to use. So there is a need for a simpler tire model that has physically meaningful parameters that are easy to change.

The requirements for such a model are set based on an application survey done in this work. In the survey the aim is to find the application of tire models within Volvo Cars Corporation (VCC) with a focus on vehicle dynamic handling applications.

The goal of this thesis work is to develop a method to study the effect of tire characteristics on vehicle handling through sensitivity analysis.

2 Targeting Tire Modelling Applications

So far the importance of understanding the effect of tire characteristics on vehicle handling have been mentioned. Sensitivity analysis is to be used to gain knowledge about tire characteristics influence on vehicle handling. Therefore it is essential to identify the cases in which the effect of tires are to be studied. To select the cases for the sensitivity study, an application targeting survey is performed to identify the following:

- **Use-cases:** In what handling scenarios are these sensitivity studies applicable?
  Example: Effect of tires in straight line braking performance of the vehicle.
- **Use-case metrics:** What indexes are used for those use-cases?
  Example: Stopping distance in meters in the straight line braking scenario.
- **Tire parameters:** For a particular use-case, the effect of which tire characteristics are needed to be investigated? These characteristics are represented by tire parameters.
  Example: Effect of maximum longitudinal force on the stopping distance in the straight line braking scenario.

The needed information was gathered by interviewing concept, development, calculation and test engineers as well as technical specialists and attribute leaders that work with tires, handling and steering. Moreover, to better understand the vehicle-tire co-development process in VCC, further investigation has been done on VCC-tire supplier relationship, tire selection procedure from start to the end of a project and last but not the least, tire models and modelling methods currently used in VCC.

2.1 Use-cases

In order to identify scenarios for the sensitivity analysis, various handling and steering tests and the corresponding simulations are investigated. The aim is to identify the cases in which the tires would play a role in determining the performance of the vehicle in that particular scenario.

The performance is measured by one or more quantity. This quantity is called the use-case metric and is used to assess the effect of tire parameters on the handling characteristics of the vehicle in the sensitivity analysis for each particular use-case. Table 2.1.1 presents the results of the application targeting survey. Note that some of the use-cases studied and their corresponding metrics are omitted in this version of the report due to confidentiality issues.

Straight line braking test is used to assess the steady-state braking performance of the vehicle as well as transient and steady-state pitch gradients of the vehicle. Stopping distance measured in meters is the metric used to describe vehicle's performance in this test. Tires' longitudinal force characteristics such as maximum force and stiffness can influence the metric in this test.

Constant radius cornering is used to characterize the steady-state turning performance of the vehicle. This test includes the full driving spectrum from normal to limit cornering situations by covering a range of lateral accelerations in a constant radius skidpad setup. Linear range understeer gradient measured in deg/g is the metric used to describe the vehicle cornering characteristics in this test. Tires' lateral characteristics such as maximum force and cornering stiffness are known to influence vehicle's performance in this test. The difference between these parameters for the front and rear tires will have an even greater effect on the metric.

The main objective of the **Yaw stability** test is to determine the transient behaviour of the vehicle by the means of a steering step input and observing the yaw overshoot as well as the maximum vehicle side slip angle. Tire's lateral force characteristics such as cornering stiffness, maximum force and corresponding slip angle can affect the metric. Since this test involves a transient state, tire's relaxation is also important in determining the yaw stability of the vehicle.

Power on in a corner mimics a real-world driving scenario where a driver tends to take a 90 degree turn under full acceleration (e.g. drive onto a main road from a side road with fast approaching traffic). This test is used to objectively determine a vehicle's directional response properties when a combined throttle and steer input is applied. Turning capability (deg) is used as a metric to identify vehicle's capability to stay on the road. Tires' combined force generation and the lateral and longitudinal force characteristics can influence the turning capability of the vehicle.

Brake in a turn test is used to objectively determine a vehicle's directional response properties when brakes are applied in a turn. The test is performed for two cases: one with a moderate lateral acceleration and

Use-case		
Name	**Metric**	**Tire Parameters**
Straight line braking	Stopping distance (m)	Maximum longitudinal force, Longitudinal stiffness, Shape (for ABS modulation).
Constant radius cornering	Linear range understeer gradient (deg/g)	Maximum lateral force, Cornering stiffness.
Yaw stability	Maximum side slip angle (deg)	Maximum lateral force, Slip angle at maximum force, Cornering stiffness, tire relaxation.
Power on in a corner	Turning capability (deg)	Maximum combined lateral force, Slip angle at maximum combined lateral force, Cornering stiffness, Maximum combined longitudinal force, Slip angle at maximum combined longitudinal force, Longitudinal stiffness.
Brake in a turn	Max. side slip rear delta (deg)	Maximum combined lateral force, Slip angle at maximum combined lateral force, Cornering stiffness, Maximum combined longitudinal force, Slip angle at maximum combined longitudinal force, Longitudinal stiffness.
Single lane change	Yaw rate settling time (s)	Maximum combined lateral force, Slip angle at maximum combined lateral force, Cornering stiffness, Maximum combined longitudinal force, Slip angle at maximum combined longitudinal force, Longitudinal stiffness, tire relaxation.

Table 2.1.1: Application targeting survey results

one with a high lateral acceleration. Maximum side slip angle difference in rear is the metric that is used to assess the vehicle performance in a sensitivity study for this particular use-case. Just as the power on cornering test, tires' lateral and longitudinal force characteristics as well as combined force generation will affect the metric.

Single lane change test is used to objectively determine a vehicle's transient response behaviour (yaw stability and response) under closely controlled test conditions similar to lane change manoeuvres in real traffic. Yaw settling time is used as a metric to measure the yaw stability of the vehicle in this test. Since the test is done in a transient combined slip situation, tires' lateral and longitudinal steady-state force characteristics, combined force generation and transient behaviours such as tire relaxation are known to influence the yaw rate settling time of the vehicle.

2.2 Tire parameters

Tire force and moment characteristics can be represented by one or more parameters. Therefore these parameters have physical significance in a sense that changing them results in changes in tire force and moment characteristics.

A summary of tire parameters identified in the application targeting survey is presented in Table 2.2.1. Tire parameters are grouped based on their effect on tires' lateral, longitudinal and self-aligning characteristics. Some other tire parameters are also considered that can not be grouped into these categories such as vertical load sensitivity (i.e. effect of vertical force on force and moment characteristics) and rolling radius.

2.3 Application targeting conclusion

These parameters together with the study done on the objective handling and steering test procedures suggest that the ideal model should be simple and user-friendly and run based on the physically meaningful parameters mentioned. It should have a good representation of tire force and moment characteristics in pure and combined

Longitudinal	Lateral	Self-aligning torque
Maximum force	Maximum force	Maximum torque
Slip ratio at max. force	Slip angle at max. force	Slip angle at max. torque
Shape of curve	Shape of curve	Shape of curve
Maximum combined force	Maximum combined force	
Slip ratio at max. combined force	Slip angle at max. combined force	**Other**
Longitudinal stiffness	Cornering stiffness	Vertical load sensitivity
Rolling resistance	Transient response	Rolling radius

Table 2.2.1: Application targeting survey: Tire Parameters Summary

slip situations. Also it should have the possibility of modelling camber and relaxation effects. Since all the tests are done on dry and flat surfaces, the model does not need to take effects such as belt deformations and enveloping properties into account.

Concerning the sensitivity study, it should be easy to change the parameters in the model and hence change the tire force and moment characteristics. Thereby one can change a single tire parameter and perform full-vehicle simulations and compare the results with the original tire. This way it is possible to investigate the sensitivity of a metric in a particular use-case to a certain tire parameter.

3 Review of Tire Models

During the past 50 years different approaches for modelling tires were explored. Semi-empirical tire models such as Magic Formula that fit to tire test data were developed to represent tires in vehicle dynamic simulations. By the improvement of computational power and simulation capabilities, complex physical tire models were developed to predict force and moment characteristics of the tire based on its physical characteristics and construction. While the later are widely used for ride, comfort and durability purposes, semi-empirical tire models such as Magic Formula based on tire force and moment measurements are the most common for vehicle dynamics handling simulations. There are also models that lay in between these two extremes: Tire models that use similarity method by manipulating the basic characteristics of the tire, and models based on simple mechanical representations of the tire structure. Figure 3.0.1 roughly summarizes various consequences that arise by taking each approach in tire modelling.

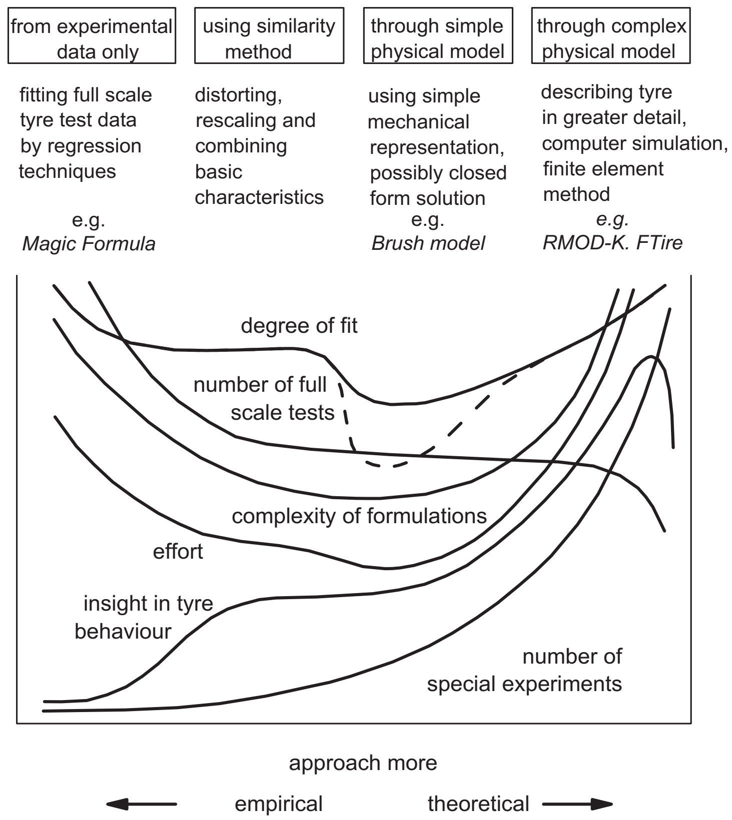

Figure 3.0.1: *Four categories of possible types of approach to develop a tire model [25]*

In this chapter two tire models are represented: Magic Formula which is used as the reference tire model and TMeasy which is the model selected for the sensitivity analysis. Appendix A includes all the other tire models that did not stand out to be used in this work, but were reviewed for the sake of comparison and categorization.

Throughout this thesis slip ratio (longitudinal slip) is defined as [25]:

$$s_x = -\frac{V_{cx} - r_e\Omega}{V_{cx}} \tag{3.0.1}$$

In which V_{cx} is the longitudinal velocity of the wheel center, r_e is the effective rolling radius and Ω is the rotational velocity of the wheel.

Slip angle is defined as:

$$s_y = -\tan^{-1}\left(\frac{V_{cy}}{V_{cx}}\right) \tag{3.0.2}$$

where V_{cy} and V_{cx} are lateral and longitudinal velocities of the wheel center, respectively.

3.1 Magic Formula

3.1.1 Description

Developed in 1987 by Egbert Bakker and Lars Nyborg from Volvo Cars and Hans B. Pacejka from Delft University of Technology, this semi-empirical tire model is the most commonly used in vehicle dynamics studies. Several versions of this model has been published as a result of this cooperation as in [6], [7],and [26]. TNO , a spin-off company from TU-Delft, and Pacejka developed this model even further with the commercial name of "MF-tyre". Several versions of this model were released (For more information of different releases from TNO refer to the backward compatibility graph in [21, p. 14]). MF-Tyre is able to model steady-state as well as transient behaviours up to 8 Hz on smooth surfaces which makes it suitable for use in handling and steering analyses. Magic Formula is named as such because of its unique formulation that can fit any test data regardless of physical properties of the tire through a set of mathematical parameters.

3.1.2 Force generation

The general form of the formula can be written as bellow:

$$y = D\sin[C\arctan\{(1-E)Bx + E\arctan(Bx)\}] \tag{3.1.1}$$

and

$$Y(X) = y(x) + S_V \tag{3.1.2}$$

$$x = X + S_H \tag{3.1.3}$$

Y(x) is the output and it could either be defined as lateral force $F_y = y(s_y)$ in which the input X becomes the slip angle s_y or it could be used for the longitudinal force $F_x = y(s_x)$ in which X becomes the slip ratio s_x. The remaining coefficients of the Magic Formula are described as

B	stiffness factor
C	shape factor
D	peak value
E	curvature factor
S_H	horizontal shift
S_V	vertical shift

The basic format of the Magic Formula in pure slip including camber effects is described here in detail. For the complete formulation of the model refer to Chapter 4.3 of [25].

Tire properties can be changed by the use of **"user scaling factors"**. Table 3.1.1 shows 13 of these scaling factors that are used to change tire characteristics for the pure slip condition.

The normalized change in vertical load is defined as:

$$df_z = \frac{F_z - F'_{zo}}{F'_{zo}} \tag{3.1.4}$$

in which F_z is the vertical load and F'_{zo} is the adapted nominal load which is roughly approximated by:

$$F'_{zo} = \lambda_{Fzo} F_{zo} \tag{3.1.5}$$

in which F_{zo} is the nominal rated load.

Scaling factor	Description
λ_{Fzo}	nominal (rated) load
$\lambda_{\mu x,y}$	peak friction coefficient
$\lambda_{\mu V}$	with slip speed V_s decaying friction
λ_{Kxs_x}	brake slip stiffness
λ_{Kys_y}	cornering stiffness
$\lambda_{Cx,y}$	shape factor
$\lambda_{Ex,y}$	curvature factor
$\lambda_{Hx,y}$	horizontal shift
$\lambda_{Vx,y}$	vertical shift
$\lambda_{Ky\gamma}$	camber force stiffness
$\lambda_{Kz\gamma}$	camber torque stiffness
λ_t	pneumatic trail (effecting aligning torque stiffness)
λ_{Mr}	residual torque

Table 3.1.1: Magic formula user scaling factors for pure slip

Longitudinal force (pure longitudinal slip)

$$F_{xo} = D_x \sin[C_x \arctan\{B_x s_{xx} - E_x(B_x s_{xx} - \arctan(B_x s_{xx}))\}] + S_{Vx} \tag{3.1.6}$$
$$s_{xx} = s_x + S_{Hx} \tag{3.1.7}$$
$$C_x = p_{Cx1} \cdot \lambda_{Cx} \qquad (> 0) \tag{3.1.8}$$
$$D_x = \mu_x \cdot F_z \cdot \zeta_1 \qquad (> 0) \tag{3.1.9}$$
$$\mu_x = (p_{Dx1} + p_{Dx2} df_z) \cdot \lambda^*_{\mu x} \qquad (> 1) \tag{3.1.10}$$
$$E_x = (p_{Ex1} + p_{Ex2} df_z + p_{Ex3} df_z^2) \cdot \{1 - p_{Ex4} sgn(s_{xx})\} \cdot \lambda_{Ex} \qquad (\leq 1) \tag{3.1.11}$$
$$K_{xs_x} = F_z \cdot (p_{Kx1} + p_{Kx2} df_z) \cdot \exp(p_{Kx3} df_z) \cdot \lambda_{Kxs_x} \qquad (= B_x C_x D_x = \partial F_{xo}/\partial s_{xx} \text{ at } s_{xx} = 0) \quad (= C_{Fs_x}) \tag{3.1.12}$$
$$B_x = K_{xs_x}/(C_x D_x + \varepsilon_x) \tag{3.1.13}$$
$$S_{Hx} = (p_{Hx1} + p_{Hx2} df_z) \cdot \lambda_{Hx} \tag{3.1.14}$$
$$S_{Vx} = F_z \cdot (p_{Vx1} + p_{Vx2} df_z) \cdot \{|V_{cx}|/(\varepsilon_{Vx} + |V_{cx}|)\} \cdot \lambda_{Vx} \cdot \lambda'_{\mu x} \cdot \zeta_1 \tag{3.1.15}$$

Lateral force (pure lateral slip)

$$F_{yo} = D_y \sin[C_y \arctan\{B_y s_{yy} - E_y(B_y s_{yy} - \arctan(B_y s_{yy}))\}] + S_{Vy} \tag{3.1.16}$$
$$s_{yy} = s_y^* + S_{Hy} \tag{3.1.17}$$
$$C_y = p_{Cy1} \cdot \lambda_{Cy} \qquad (> 0) \tag{3.1.18}$$
$$D_y = \mu_y \cdot F_z \cdot \zeta_2 \tag{3.1.19}$$
$$\mu_y = (p_{Dy1} + p_{Dy2} df_z) \cdot (1 - p_{Dy3} \gamma^{*2}) \cdot \lambda^*_{\mu y} \qquad (> 0) \tag{3.1.20}$$
$$E_y = (p_{Ey1} + p_{Ey2} df_z) \cdot \{1 - (p_{Ey3} + p_{Ey4} \gamma^*) sgn(s_{yy})\} \cdot \lambda_{Ey} \qquad (\leq 1) \tag{3.1.21}$$
$$K_{ys_yo} = p_{Ky1} F'_{zo} \sin[2 \arctan\{F_z/(p_{Ky2} F'_{zo})\}] \cdot \lambda_{Kys_y} \qquad (= B_y C_y D_y = \partial F_{yo}/\partial s_{yy} \text{ at } s_{yy} = \gamma = 0) \quad (= C_{Fs_y}) \tag{3.1.22}$$
$$K_{ys_y} = K_{ys_yo} \cdot (1 - p_{Ky3} \gamma^{*2}) \cdot \zeta_3 \tag{3.1.23}$$
$$B_y = K_{ys_y}/(C_y D_y + \varepsilon_y) \tag{3.1.24}$$
$$S_{Hy} = (p_{Hy1} + p_{Hy2} df_z) \cdot \lambda_{Hy} + p_{Hy3} \gamma^* \cdot \lambda_{Ky\gamma} \cdot \zeta_0 + \zeta_4 - 1 \tag{3.1.25}$$
$$S_{Vy} = F_z \cdot \{(p_{Vy1} + p_{Vy2} df_z) \cdot \lambda_{Vy} + (p_{Vy3} + p_{Vy4} df_z) \gamma^* \cdot \lambda_{Ky\gamma}\} \cdot \lambda'_{\mu y} \cdot \zeta_2 \tag{3.1.26}$$
$$K_{y\gamma o} = \{p_{Hy3} K_{ys_yo} + F_z(p_{Vy3} + p_{Vy4} df_z)\} \lambda_{Ky\gamma} \qquad (=\sim \partial F_{yo}/\partial \gamma \text{ at } s_{yy} = \gamma = 0) \quad (= C_{F\gamma}) \tag{3.1.27}$$

Self-aligning torque (pure lateral slip)

$$M_{zo} = M'_{zo} + M_{zro} \tag{3.1.28}$$
$$M'_{zo} = -t_o \cdot F_{yo} \tag{3.1.29}$$
$$t_o = t(s_{yt}) = D_t \cos[C_t \arctan\{B_t s_{yt} - E_t(B_t s_{yt} - \arctan(B_t s_{yt}))\}] \cdot \cos' s_y \tag{3.1.30}$$
$$s_{yt} = s^*_y + S_{Ht} \tag{3.1.31}$$
$$S_{Ht} = q_{Hz1} + q_{Hz2} df_z + (q_{Hz3} + q_{Hz4} df_z)\gamma^* \tag{3.1.32}$$
$$M_{zro} = M_{zr}(s_{yr}) = D_r \cos[C_r \arctan(B_r s_{yr})] \tag{3.1.33}$$
$$s_{yr} = s^*_y + S_{Hf} \qquad (= s_{yf}) \tag{3.1.34}$$
$$S_{Hf} = S_{Hy} + S_{Vy}/K'_{ys_y} \tag{3.1.35}$$
$$K'_{ys_y} = K_{ys_y} + \varepsilon_K \tag{3.1.36}$$
$$B_t = (q_{Bz1} + q_{Bz2} df_z + q_{Bz3} df_z^2) \cdot (1 + q_{Bz5}|\gamma^*| + q_{Bz6}\gamma^{*2}) \cdot \lambda_{Kys_y}/\lambda^*_{\mu y} \qquad (> 0) \tag{3.1.37}$$
$$C_t = q_{Cz1} \qquad (> 0) \tag{3.1.38}$$
$$D_{to} = F_z \cdot (R_o/F'_{zo}) \cdot (q_{Dz1} + q_{Dz2} df_z) \cdot \lambda_t \cdot sgnV_{cx} \tag{3.1.39}$$
$$D_t = D_{to} \cdot (1 + q_{Dz3}|\gamma^*| + q_{Dz4}\gamma^{*2}) \cdot \zeta_5 \tag{3.1.40}$$
$$E_t = (q_{Ez1} + q_{Ez2} df_z + q_{Ez3} df_z^2)\{1 + (q_{Ez4} + q_{Ez5}\gamma^*)\frac{2}{\pi}\arctan(B_t C_t s_{yt})\} \qquad (\leq 1) \tag{3.1.41}$$
$$B_r = (q_{Bz9} \cdot \lambda_{Ky}/\lambda^*_{\mu y} + q_{Bz10} B_y C_y) \cdot \zeta_6 \tag{3.1.42}$$
$$C_r = \zeta_7 \tag{3.1.43}$$
$$D_r = F_z R_o\{(q_{Dz6} + q_{Dz7} df_z)\lambda_{Mr}\zeta_2 + (q_{Dz8} + q_{Dz9} df_z)\gamma^* \lambda_{Kz\gamma}\zeta_0\} \cos' s_y \cdot \lambda^*_{\mu y} sgnV_{cx} + \zeta_8 - 1 \tag{3.1.44}$$
$$K_{z\alpha o} = D_{to} K_{ys_y o} \qquad (=\sim -\partial M_{zo}/\partial s_{yy} \text{ at } s_{yy} = \gamma = 0) \quad (= C_{Ms_y}) \tag{3.1.45}$$
$$K_{z\gamma o} = F_z R_o (q_{Dz8} + q_{Dz9} df_z)\lambda_{Kz\lambda} - D_{to} K_{y\gamma o} \qquad (=\sim \partial M_{zo}/\partial\gamma \text{ at } s_y = \gamma = 0) \quad (= C_{M\gamma}) \tag{3.1.46}$$
$$\tag{3.1.47}$$

For the longitudinal force in pure slip the Magic Formula coefficients, , coefficients B, C, D, E, S_H and S_V are defined by 14 constants as presented in Table 3.1.2.

For the lateral force in pure slip, coefficients B, C, D, E, S_H and S_V can be described by 18 constants as the result of the fitting process, Table 3.1.3.

As shown in Table 3.1.4, number of constants used to describe the aligning torque in pure slip conditions is 25.

Constant	Description
p_{Cx1}	Shape factor Cfx for longitudinal force
p_{Dx1}	Longitudinal friction Mux at Fznom
p_{Dx2}	Variation of friction Mux with load
p_{Ex1}	Longitudinal curvature Efx at Fznom
p_{Ex2}	Variation of curvature Efx with load
p_{Ex3}	Variation of curvature Efx with load squared
p_{Ex4}	Factor in curvature Efx while driving
p_{Kx1}	Longitudinal slip stiffness Kfx/Fz at Fznom
p_{Kx2}	Variation of slip stiffness Kfx/Fz with load
p_{Kx3}	Exponent in slip stiffness Kfx/Fz with load
p_{Hx1}	Horizontal shift Shx at Fznom
p_{Hx2}	Variation of shift Shx with load
p_{Vx1}	Vertical shift Svx/Fz at Fznom
p_{Vx2}	Variation of shift Svx/Fz with load

Table 3.1.2: Magic formula constants for longitudinal force in pure slip

Constant	Description
p_{Cy1}	Shape factor Cfy for lateral force
p_{Dy1}	Lateral friction Muy
p_{Dy2}	Variation of friction Muy with load
p_{Dy3}	Variation of friction Muy with squared inclination
p_{Ey1}	Lateral curvature Efy at Fznom
p_{Ey2}	Variation of curvature Efy with load
p_{Ey3}	Inclination dependency of curvature Efy
p_{Ey4}	Variation of curvature Efy with inclination
p_{Ky1}	Maximum value of stiffness Kfy/Fznom
p_{Ky2}	Load at which Kfy reaches maximum value
p_{Ky3}	Variation of Kfy/Fznom with inclination
p_{Hy1}	Horizontal shift Shy at Fznom
p_{Hy2}	Variation of shift Shy with load
p_{Hy3}	Variation of shift Shy with inclination
p_{Vy1}	Vertical shift Svy/Fz at Fznom
p_{Vy2}	Variation of shift Svy/Fz with load
p_{Vy3}	Variation of shift Svy/Fz with inclination
p_{Vy4}	Variation of shift Svy/Fz with inclination and load

Table 3.1.3: Magic formula constants for lateral force in pure slip

Constant	Description
p_{Bz1}	Trail slope factor for trail Bpt at Fznom
p_{Bz2}	Variation of slope Bpt with load
p_{Bz3}	Variation of slope Bpt with load squared
p_{Bz4}	Variation of slope Bpt with inclination
p_{Bz5}	Variation of slope Bpt with absolute inclination
p_{Bz9}	Slope factor Br of residual moment Mzr
p_{Bz10}	Slope factor Br of residual moment Mzr
p_{Cz1}	Shape factor Cpt for pneumatic trail
p_{Dz1}	Peak trail Dpt = Dpt*(Fz/Fznom*R0)
p_{Dz2}	Variation of peak Dpt with load
p_{Dz3}	Variation of peak Dpt with inclination
p_{Dz4}	Variation of peak Dpt with inclination squared
p_{Dz6}	Peak residual moment Dmr = Dmr/(Fz*R0)
p_{Dz7}	Variation of peak factor Dmr with load
p_{Dz8}	Variation of peak factor Dmr with inclination
p_{Dz9}	Variation of Dmr with inclination and load
p_{Ez1}	Trail curvature Ept at Fznom
p_{Ez2}	Variation of curvature Ept with load
p_{Ez3}	Variation of curvature Ept with load squared
p_{Ez4}	Variation of curvature Ept with sign of Alpha-t
p_{Ez5}	Variation of Ept with inclination and sign Alpha-t
p_{Hz1}	Trail horizontal shift Sht at Fznom
p_{Hz2}	Variation of shift Sht with load
p_{Hz3}	Variation of shift Sht with inclination
p_{Hz4}	Variation of shift Sht with inclination and load

Table 3.1.4: Magic formula constants for self-aligning torque in pure slip

3.2 TMeasy

3.2.1 Description

TMeasy was first conceived as a tire model for vehicle dynamic simulations of agricultural vehicles. It has been successfully used in real-time applications with cars and heavy vehicles in programs such as veDYNA and SIMPACK [18]. This semi-empirical tire model is developed to assist in situations where there is none or insufficient tire data available. The idea is that in these occasions an engineering estimation can be done to interpolate or extrapolate the properties from a similar tire. The aim of TMeasy is to give useful tire forces from little information with model parameters that have concrete physical meaning. TMeasy model construction is presented here briefly.

3.2.2 Force generation

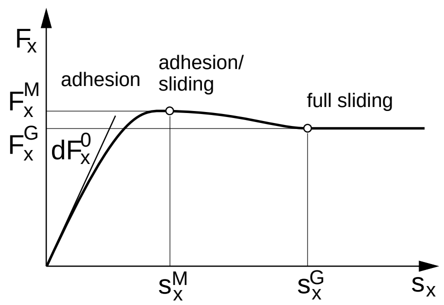

Figure 3.2.1: *Typical longitudinal force characteristics*

As shown in Figure 3.2.1, a typical longitudinal force F_x as a function of longitudinal slip s_x can be described by the following parameters[18]:

- Initial inclination (longitudinal stiffness) dF_x^0
- Maximum longitudinal force F_x^M
- Longitudinal slip at maximum force s_x^M
- Sliding force F_x^G
- Longitudinal slip at sliding force s_x^G

If the curve does not have a peak then F_x^M can be set equal to F_x^G. In that case s_x^M can be used to distinguish between adhesion and sliding region.

The acting point of the resultant lateral force depends on the distribution of the force over the contact area length L (Figure 3.2.2). For small slip angles, when all the tread particles stick to the road, an almost linear distribution of force on the contact area length appears. In such a case the resultant lateral force is exerted on a point behind the center of the contact area. At average slip angle values the particles at the end of the contact area start to slide. That moves the acting point of the resulting lateral force forwards, sometimes even before the contact area center point. And finally at high slip angles only a small portion of tread particles at the front of the contact area stick to the ground. In these extreme cases the resulting force is applied at the center of the contact area.

The self-aligning torque M_s is then obtained by multiplying the resulting lateral force F_y by the dynmic tire offset or pneumatic trail n.

$$M_s = -nF_y \tag{3.2.1}$$

Lateral force F_y and the dynamics tire offset are both functions of the lateral slip s_y as shown in Figure 3.2.3.

The parameters describing the lateral force are:

- Initial inclination (cornering stiffness) dF_y^0

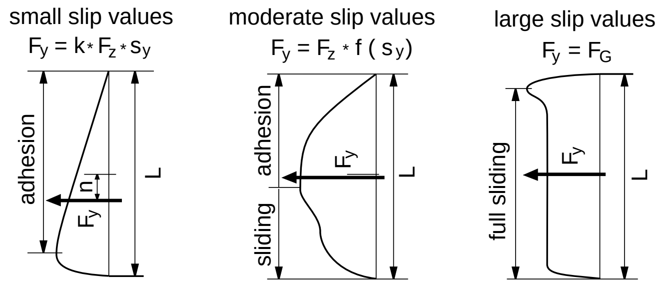

Figure 3.2.2: *Lateral force distribution over the contact area for different slip values*

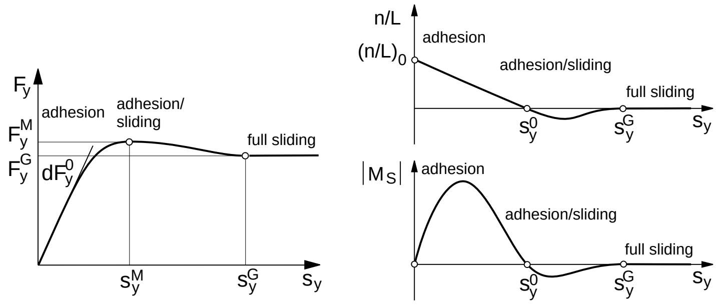

Figure 3.2.3: *Typical plots of lateral force, dynamic offset and self-aligning torque*

- Maximum lateral force F_y^M
- Lateral slip at maximum force s_y^M
- Sliding force F_y^G
- Lateral slip at sliding force s_y^G

The dynamic tire offset is normalized by the length of the contact area L. The initial value n/L_0 together with slip values s_y^0 and s_y^G describe the dynamic tire offset as a function of lateral slip.

TMeasy simulates the tire behaviour in combined slip situations (braking or accelerating in a curve, for instance) by generalizing the tire characteristics through a normalization process (see Figure 3.2.4). The longitudinal slip s_x and lateral slip s_y can be generalized by vector addition as:

$$s = \sqrt{\left(\frac{s_x}{\hat{s_x}}\right)^2 + \left(\frac{s_y}{\hat{s_y}}\right)^2} \tag{3.2.2}$$

For normalizing, the normation factors $\hat{s_x}$ and $\hat{s_x}$ are calculated based on the location of the maxima s_x^M, s_y^M, the maximum values F_x^M, F_y^M and the initial inclinations dF_x^0, dF_y^0.

$$\hat{s_x} = \frac{F_x^M}{dF_x^0} \quad \text{and} \quad \hat{s_y} = \frac{F_y^M}{dF_y^0} \tag{3.2.3}$$

The generalized tire parameters are then calculated with the corresponding values of the longitudinal and

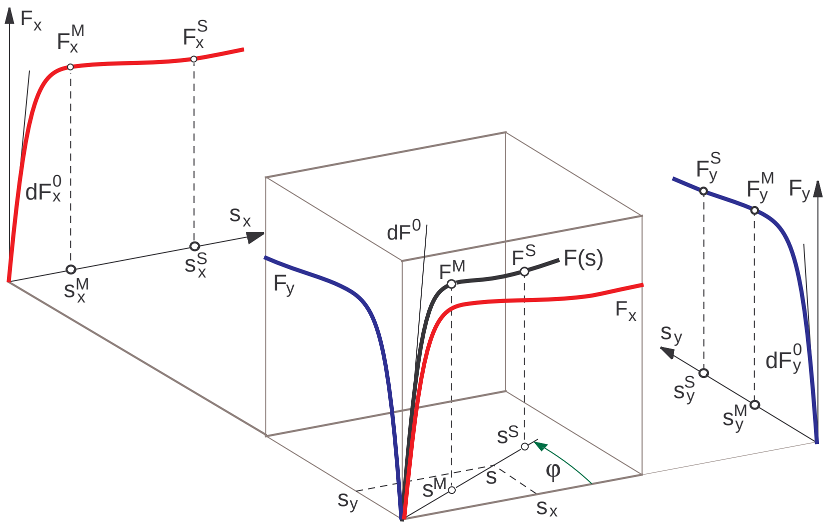

Figure 3.2.4: *Generalized tire characteristics [19]*

lateral tire parameters and normalization factors.

$$
\begin{aligned}
dF^0 &= \sqrt{\left(dF_x^0 \hat{s_x} \cos\varphi\right)^2 + \left(dF_y^0 \hat{s_y} \sin\varphi\right)^2}, \\
s^M &= \sqrt{\left(\frac{s_x^M}{\hat{s_x}} \cos\varphi\right)^2 + \left(\frac{s_y^M}{\hat{s_y}} \sin\varphi\right)^2}, \\
F^M &= \sqrt{\left(F_x^M \cos\varphi\right)^2 + \left(F_y^M \sin\varphi\right)^2}, \\
s^G &= \sqrt{\left(\frac{s_x^G}{\hat{s_x}} \cos\varphi\right)^2 + \left(\frac{s_y^G}{\hat{s_y}} \sin\varphi\right)^2}, \\
F^G &= \sqrt{\left(F_x^G \cos\varphi\right)^2 + \left(F_y^G \sin\varphi\right)^2}
\end{aligned}
\tag{3.2.4}
$$

The angular functions

$$
\cos\varphi = \frac{s_x/\hat{s_x}}{s} \quad \text{and} \quad \sin\varphi = \frac{s_y/\hat{s_y}}{s} \tag{3.2.5}
$$

grant a smooth transition from characteristic curve of longitudinal to the curve of lateral forces in the range of $\varphi = 0$ to $\varphi = 90°$.

The function $F = F(G)$ can be described in intervals by a broken rational function, a cubic polynomial and a constant F^G.

$$
F(s) = \begin{cases}
s^M dF^0 \dfrac{\sigma}{1 + \sigma\left(\sigma + F^0 \dfrac{s^M}{F^M} - 2\right)}, & \sigma = \dfrac{s}{s^M}, \qquad 0 \le s \le s^M \\
F^M - (F^M - F^G)\sigma^2(3 - 2\sigma), & \sigma = \dfrac{s - s^M}{s^G - s^M}, \qquad s^M < s \le s^G \\
F^G, & s > s^G
\end{cases}
\tag{3.2.6}
$$

When defining the curve parameters, one just has to make sure that the condition $dF^0 \ge 2\dfrac{F^M}{s_M}$ is satisfied, otherwise the function will have a turning point in the interval $0 \le s \le s^M$.

By projecting the generalized force in longitudinal and lateral directions, the corresponding forces can be obtained:

$$
F_x = F\cos\varphi \quad \text{and} \quad F_y = F\sin\varphi \tag{3.2.7}
$$

$\mathbf{F_z = F_{z_{nom}}}$					
Longitudinal force $\mathbf{F_x}$		**Lateral force $\mathbf{F_y}$**		**Self-aligning torque $\mathbf{M_z}$**	
dF_x^0	Initial slope	dF_y^0	Initial slope	$(n/L)_0$	Norm. pneumatic trail
F_x^M	Max force	F_y^M	Max force	s_y^0	s_y trail changes sign
s_x^M	s_x where $f_x(s_x) = F_x^M$	s_y^M	s_y where $f_y(s_y) = F_y^M$	s_y^G	s_y trail tends to zero
F_x^G	Sliding force	F_y^G	Sliding force		
s_x^G	s_x where $f_x(s_x) = F_x^G$	s_y^G	s_y where $f_y(s_y) = F_y^G$		
$\mathbf{F_z = 2 \times F_{z_{nom}}}$					
Longitudinal force $\mathbf{F_x}$		**Lateral force $\mathbf{F_y}$**		**Self-aligning torque $\mathbf{M_z}$**	
dF_x^0	Initial slope	dF_y^0	Initial slope	$(n/L)_0$	Norm. pneumatic trail
F_x^M	Max force	F_y^M	Max force	s_y^0	s_y trail changes sign
s_x^M	s_x where $f_x(s_x) = F_x^M$	s_y^M	s_y where $f_y(s_y) = F_y^M$	s_y^G	s_y trail tends to zero
F_x^G	Sliding force	F_y^G	Sliding force		
s_x^G	s_x where $f_x(s_x) = F_x^G$	s_y^G	s_y where $f_y(s_y) = F_y^G$		

Table 3.2.1: TMeasy tire parameters

The self-aligning torque can be calculated by Equation 3.2.2 via the dynamic tire offset. The dynamic offset is approximated as function of the lateral slip by a line and a cubic polynomial:

$$\frac{n}{L} = \begin{cases} (n/L)_0(1 - |s_y|/s_y^0) \ , & |s_y| \leq s_y^0 \\ -(n/L)_0 \dfrac{|s_y| - s_y^0}{s_y^0} \left(\dfrac{s_y^G - |s_y|}{s_y^G - s_y^0} \right)^2 \ , & s_y^0 < |s_y| \leq s_y^G \\ 0 \ , & |s_y| > s_y^G \end{cases} \tag{3.2.8}$$

The value of $(n/L)_0$ can be approximated. For small lateral slip values, $s_y \approx 0$, the distribution of lateral force over the contact area length can be approximated by a triangle, see Figure 3.2.2. The point that the resultant lateral force is applied then given by

$$n(F_z \to 0 \ , \ s_y = 0) = \frac{1}{6} L \tag{3.2.9}$$

This value can only serve as a reference point. The uneven distribution of pressure in longitudinal direction of the contact area results in the change of the deflection profile and the dynamic tire offset.

The list of parameters that are needed to model a tire in TMeasy are shown in Table 3.2.1. Note that tire parameters for only two normal loads are needed (Nominal normal load and 2 times the nominal load). Tire force and moment characteristics for other normal loads are calculated based on quadratic functions. Forces and moments due to camber are calculated separately and then added to the tire force and moment values. These effects are discussed in detail in [18].

3.3 Summary

Different types of tire models were reviewed in this chapter. The use of semi-empirical tire models is generally preferred instead of more complex tire models in handling simulations due to their higher accuracy in representing tire force and moment characteristics. Meanwhile the complex tire models are widely used in ride, comfort, durability and misuse scenarios. Among the semi-empirical tire models a simple model was needed that is representative of the tire behaviour and characteristics. That means it should fit well to tire test data and incorporate effects such as camber, relaxation and combined slip. At the same time it should provide the possibility of changing the tire parameters presented in the application survey (Chapter 2).

A MF-Tyre tire property file for use in Adams/Car has more than 130 parameters for model description. The large number of parameters makes Magic Formula rather difficult to use for sensitivity studies. Moreover, not all of these parameters have physically explicit meaning and can not even be measured on a real tire (parameters such as curvature factor p_{Ex4} or exponent p_{Kx3}). Tire parameters realized in the application targeting in most cases relate to more than one constant in the magic formula model. For example for changing the maximum longitudinal force for a particular camber angle and vertical load case, two parameters (p_{Dx1} and p_{Dx1}) and 2 user scaling factors ($\lambda_{\mu x}$ and λ_{Fzo}) have to be adjusted.

Magic formula's accuracy comes with the cost of increased number of parameters and complexity in formulation. Therefore it is desired to find an alternative tire model with less parameters that can be related to the tire parameters realized in the application survey.

However, because of its high accuracy in representing tire characteristics, Magic Formula is used as the reference tire model in this project.

It was shown that the parameters used in TMeasy are in most cases the same as the parameters realized in the application survey results (page 4). Also it is fairly easy to change these parameters and all of them have physical meaning. When test data is missing, model parameters can be pragmatically estimated by adjustment of the data from a similar tire. Later in the project when there is more data available, these parameters can be easily derived and replace the estimated values in the model.

TMeasy model presented in the literature however has limitations. The modelling of the self-aligning torque is inaccurate due to simplified contact area length and resultant force application point assumptions. The model is unable to capture asymmetries in tire force and moment behaviour. Also TMeasy is not able to model tire relaxation effects.

Despite these shortcomings, considering the direct relation of model parameters to the results of the application survey, its simplicity and good longitudinal and lateral force representation, TMeasy appears to be the perfect compromise suitable for sensitivity studies.

Figure 3.3.1 categorizes the models reviewed based on the approaches defined in Figure 3.0.1. Note that this is a rough categorization and some tire models can be considered to be in between two or three of these categories as they utilize various approaches to model specific tire characteristics.

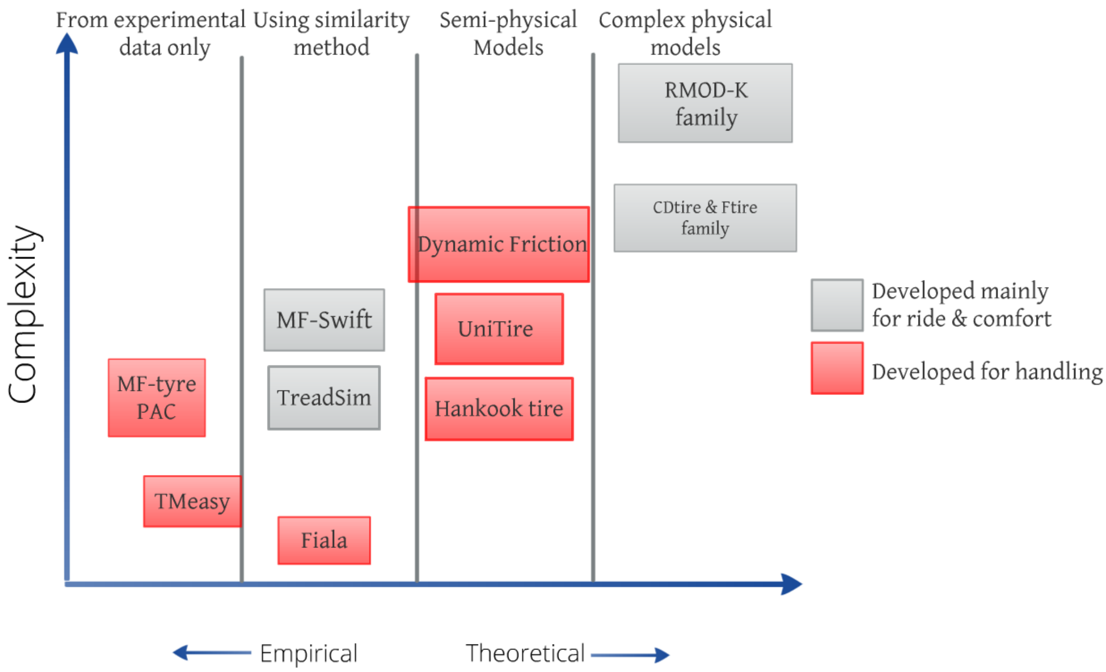

Figure 3.3.1: *Tire models categorization*

Requirement	MFtyre	TMeasy	MF-Swift	TreadSim	Fiala	Dynamic Friction	UniTire	Hankook	RMOD-K	FTire & CDtire
Tire Force and moment representation pure slip	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
Tire Force and moment representation combined slip	✓	✓	✓	✓		✓	✓	✓	✓	✓
Camber effects	✓	✓	✓	✓		✓	✓	✓	✓	✓
Relaxation effects	✓		✓	✓	✓			✓	✓	✓
Turn slip effect	✓		✓				✓	✓	✓	✓
Number of parameters (scaled: 5 highest, 1 lowest)	4	2	4	4	1	4	3	3	5	5
Correlation to tire parameters realized in the application targeting survey (scaled: 5 highest, 1 lowest)	3	5	1	1	4	1	1	1	1	1

Table 3.3.1: Tire model requirements summary

4 Model Verification

Upon selecting TMeasy as the suitable tire model for the sensitivity studies, it is necessary to verify the model. For this purpose TMeasy is verified against a MFtyre model (Magic Formula model by TNO). Full vehicle simulations are performed in three specific use-cases with TMeasy and MFtyre implementations in Adams/Car and their influence on the results and particularly the use-case metrics are investigated. The vehicle used in these simulations is the Adams/Car demo vehicle [1].

The simulation is set up in Adams/Car and done on the demo vehicle, Figure 4.0.1, once with the MFtyre model and once with TMeasy tire model. Note that for model verification and sensitivity studies the ISO coordinate system is used for tire force and moments, since it is standard in Adams/Tire module [2] and conforms with TYDEX tire testing and modelling standards [24]. The ISO coordinates and TYDEX C and W coordinate systems are reviewed in Appendix B.

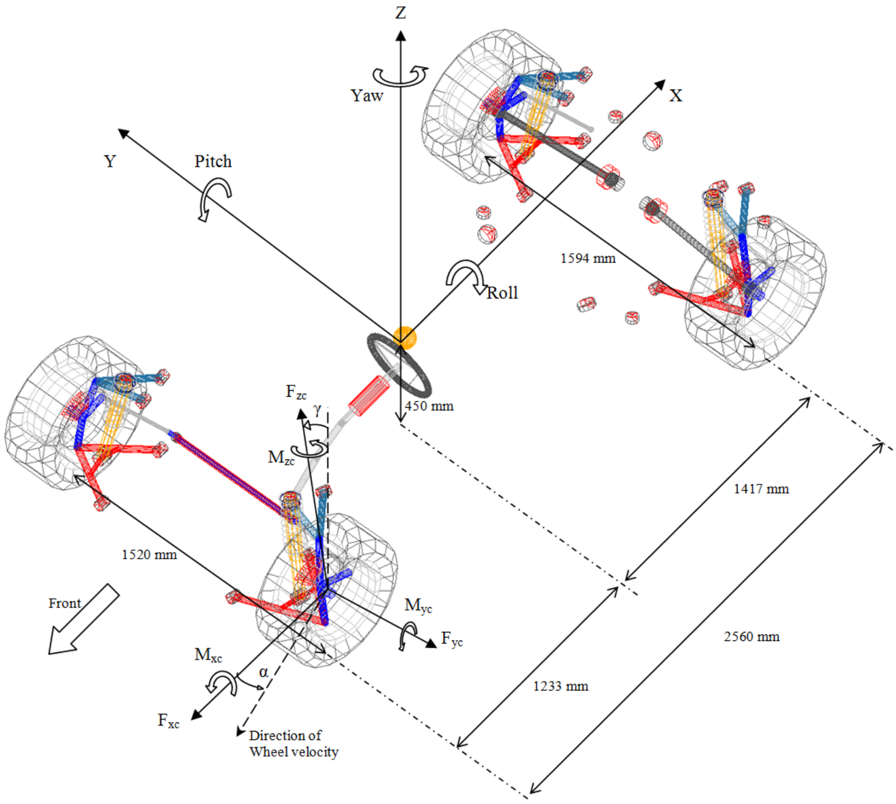

Figure 4.0.1: *Adams/Car Demo vehicle and the TYDEX-C axis system*

4.1 Comparison between TMeasy and Magic Formula

First, a TMeasy model is created based on the MFtyre model data for a generic tire called “tire 1”. This is done by plotting the longitudinal, lateral and self-aligning torque curves for the MFtyre model; and then measuring the TMeasy parameters on the curves and implementing them in the TMeasy tire property file. The TMeasy parameters for tire 1 are shown in Table 4.1.1.

Figures 4.1.1, 4.1.2 and 4.1.3 show the comparison between MFtyre and TMeasy longitudinal, lateral and self-aligning torque characteristics, respectively, for a nominal vertical load of $3000N$ and zero camber angle.

Figure 4.1.1 shows that TMeasy matches MFtyre reletively well with 2.9% maximum relative error in longitudinal force. The error is due to the fact that TMeasy, as expected, is unable to capture the asymmetric force behaviour seen at negative slip ratios for this tire.

In lateral force characteristics depicted in Figure 4.1.2, TMeasy matches MFtyre with maximum relative error of 3.7%. The tire asymmetric behaviour seen in positive slip angles are not realized by TMeasy.

Longitudinal force $\mathbf{F_x}$		Lateral force $\mathbf{F_y}$	
$F_z = 3\ kN$	$F_z = 6\ kN$	$F_z = 3\ kN$	$F_z = 6\ kN$
$dF_x^0 = 82.2\ kN$	$dF_x^0 = 236.2\ kN$	$dF_y^0 = 53.7\ kN$	$dF_y^0 = 95\ kN$
$F_x^M = 3.57\ kN$	$F_x^M = 6.57\ kN$	$F_y^M = 3.32\ kN$	$F_y^M = 6.08\ kN$
$s_x^M = 0.160$	$s_x^M = 0.100$	$s_y^M = 0.197$	$s_y^M = 0.196$
$F_x^G = 3.29\ kN$	$F_x^G = 6.01\ kN$	$F_y^G = 3.26\ kN$	$F_y^G = 5.83\ kN$
$s_x^G = 0.700$	$s_x^G = 0.500$	$s_y^G = 0.291$	$s_y^G = 0.349$
Self-aligning torque $\mathbf{M_z}$			
$(n/L)_0 = 0.17$	$(n/L)_0 = 0.25$		
$s_y^0 = 0.190$	$s_y^0 = 0.180$		
$s_y^G = 0.400$	$s_y^G = 0.350$		

Table 4.1.1: Tire 1 TMeasy tire parameters

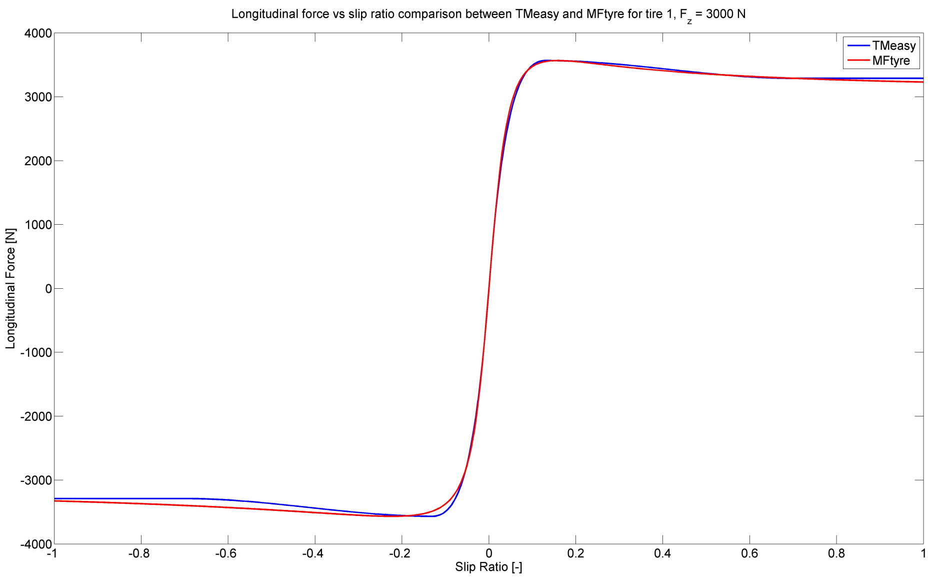

Figure 4.1.1: *Longitudinal force characteristics*

Figure 4.1.3 shows TMeasy's disadvantage in simulating self-aligning torque behaviour for moderate to high slip angles. This is due to the rather simplistic assumptions made in modelling of the lateral force distribution and the resultant force application point described in section 3.2.

Figure 4.1.4 shows the combined slip characteristics for TMeasy and MFtyre. Minor differences seen in the size of the friction ellipses are due to several reasons. First is the asymmetric tire behaviour not realized by TMeasy. The other reason is in the different approaches used in MFtyre and TMeasy for combined force calculations. While MFtyre fits some parameters to the test data obtained from combined slip tire tests, TMeasy relies on the data from pure slip tests and generates a general force curve based on normalizing the lateral and longitudinal forces. For higher vertical loads this difference increases where MFtyre "squares out" the elliptical shape resulting in higher lateral forces in the presence of a high longitudinal force. This phenomena is known to be MFtyre's disadvantage in modelling combined situations for very high vertical loads or on wet surfaces.

As mentioned in section 3.2, the current formulation of TMeasy does not include relaxation effects. Figure 4.1.5 compares TMeasy and MFtyre responses to a step input slip angle of 5 degrees in magnitude starting at $t = 0.5$ sec. It can be seen that MFtyre models the transient behaviour by including relaxation effects, while TMeasy simply switches the formula used to calculate the force (Equation 3.2.6).

Straight-line braking, Constant radius cornering and Single lane change simulations are performed to ensure TMeasy result correlation with the same simulations with MFtyre. These simulations and their results are

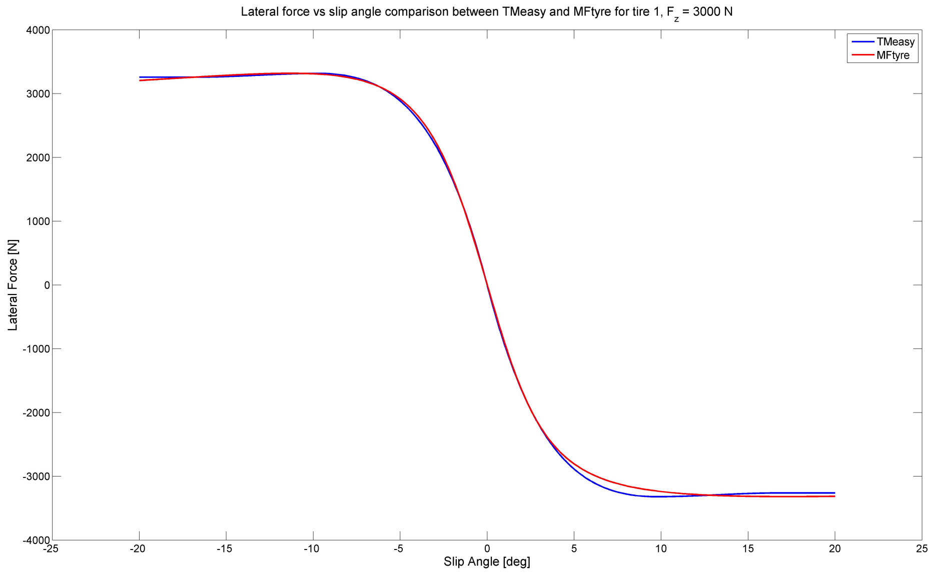

Figure 4.1.2: *Lateral force characteristics*

discussed in the following sections.

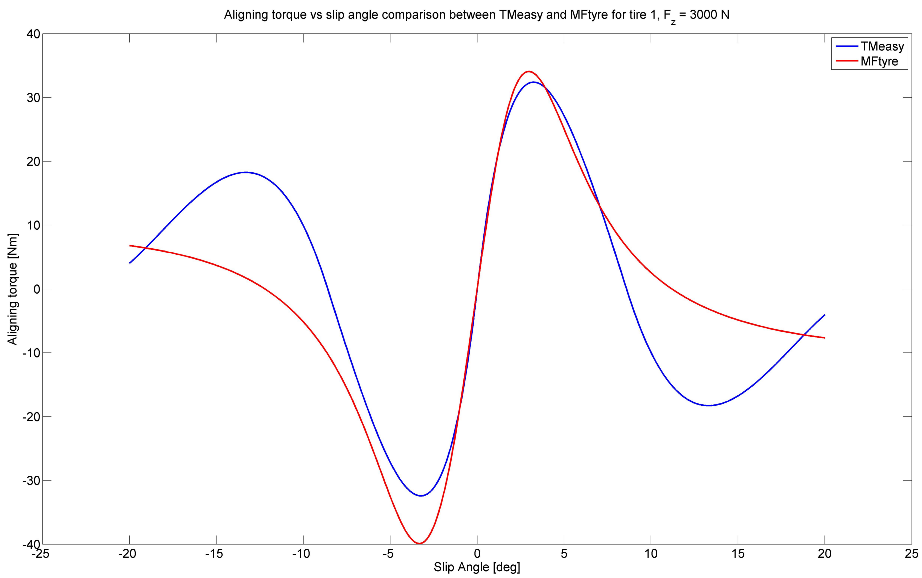

Figure 4.1.3: *Self-aligning torque characteristics*

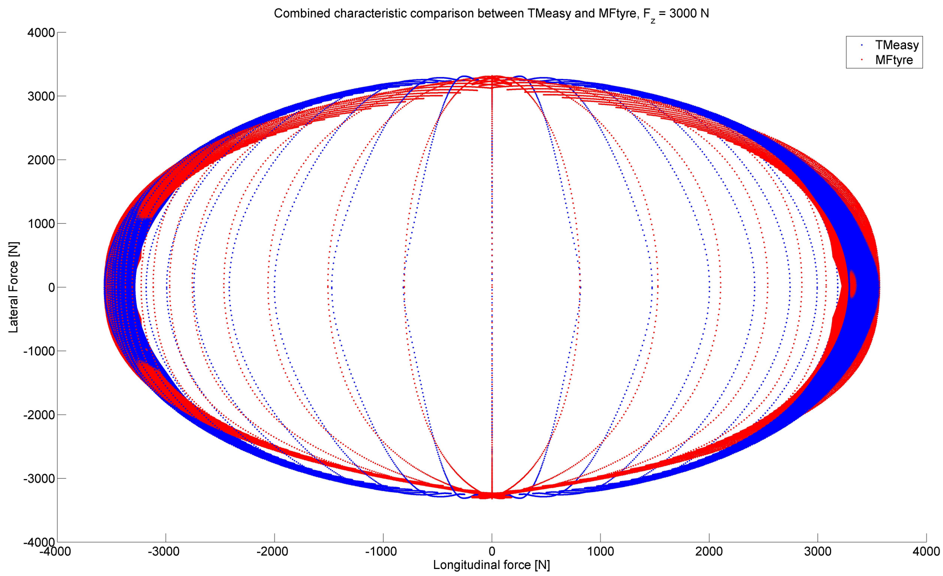

Figure 4.1.4: *Combined slip characteristics*

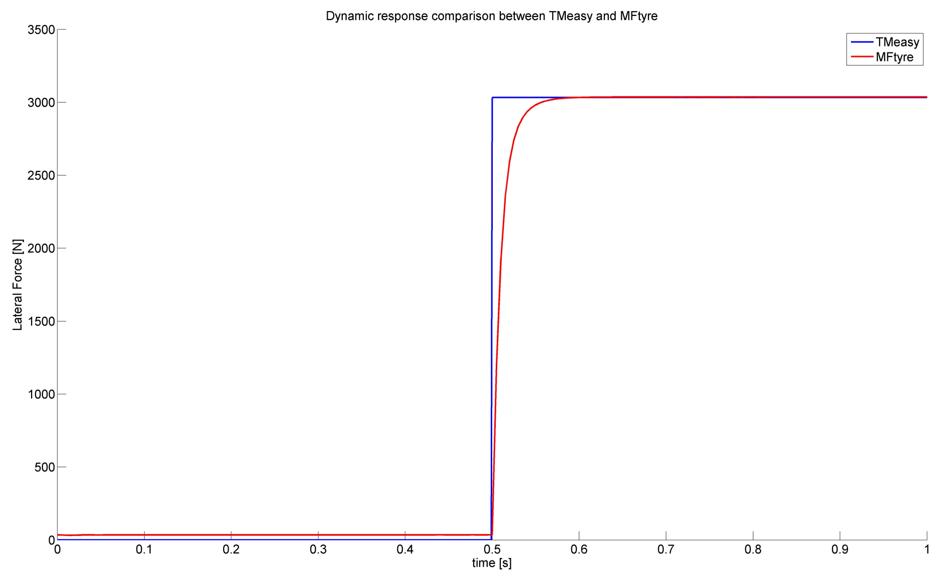

Figure 4.1.5: *Transient behaviour characteristics*

4.2 Straight line braking

The aim of this test is to measure the stopping distance of the vehicle in meters. While the vehicle is travelling on a straight line with 150 km/h, full ABS braking is performed (120 N on the brake pedal) with steering wheel fixed to 0 deg. The stopping distance is the distance travelled while the vehicle decelerates with a constant rate from 100 km/h to 2 km/h. It is measured from 100 km/h so that the dynamics of braking such as pitch angle changes are removed. It is measured to 2 km/h to remove the discrepancies in sensors due to standstill and pitch angle recovery. This test is selected to demonstrate the pure steady-state longitudinal force generation in TMeasy and to ensure its correlation to MFtyre results. The resulting metrics are shown in Table 4.2.1.

Tire model	**Stopping distance (m)**	Δ
MFtyre	34.4	
TMeasy	34.8	-0.4 m (-1.2%)

Table 4.2.1: TMeasy vs. MFtyre straight line braking results

Both cases reach 100 km/h at $t = 1.23$ sec. Figure 4.2.1 illustrates the chassis longitudinal accelerations and longitudinal forces on the front left and rear left tires for both cases (longitudinal forces on the right side tires have the same behaviour and are not showed here for simplicity). It is evident that TMeasy can easily capture the steady state pure longitudinal behaviour of the tire. There are however minor differences in the transient part (before $t = 1.23$ sec). These are due to the differences in rolling radius calculations between TMeasy and MFtyre. The rolling (dynamic) radius formulation in TMeasy is very simplified based on an approximate assumption as a first guess. This influences the normal load calculation which in turn effects the interpolation algorithms used to calculate the longitudinal force on the tire.

Comparing the results and considering the minimal error in calculation of the metric (stopping distance normally varies with a larger error when this test is done on a prototype vehicle several times), it can be concluded that the TMeasy model correlates well with the MFtyre model in longitudinal force characteristics.

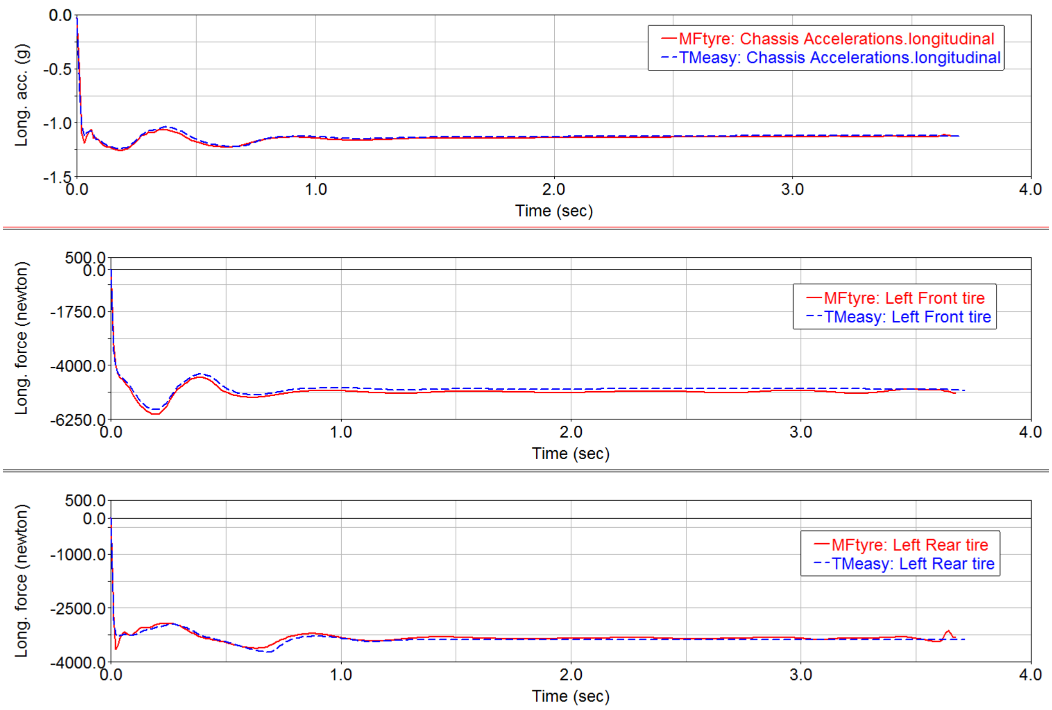

Figure 4.2.1: *TMeasy vs. MFtyre straight line braking results: (top) longitudinal acceleration (middle) longitudinal force on the left front tire (bottom) longitudinal force on the left rear tire*

4.3 Constant radius cornering

The aim of this test is to asses the pure steady-state lateral force characteristics of the TMeasy tire model and ensure its reliability by comparing it to MFtyre. The test is done on a skid-pad track with a radius of 40m for both right and left turns. The vehicle is then driven such that it covers a range of lateral accelerations from 0.1 to 1 g by slowly increasing the velocity. The linear range understeer gradient (deg/g) is the metric used for this particular use-case. (linear range = up to 0.4 g lateral acceleration)

The simulations are performed in Adams/Car with the demo vehicle and the results are shown in Table 4.3.1.

Tire model	Understeer gradient (deg/g)	Δ
MFtyre (left turn)	1.73	
TMeasy (left turn)	1.69	-0.04 deg/g (-2.3%)
MFtyre (right turn)	1.71	
TMeasy (right turn)	1.69	-0.02 deg/g (-1.2%)

Table 4.3.1: TMeasy vs. MFtyre constant radius cornering results

The difference seen in the right and left hand understeer gradient for MFtyre, although small, is due to tire asymmetric behaviour. This difference is not realized by the TMeasy model.

Figure 4.3.1 shows the relation between the front wheel angle and lateral acceleration for both left and right turns for TMeasy and MFtyre. The "equivalent" front wheel angle (bicycle model front wheel) is obtained by dividing the steering wheel angle by the steering gear ratio. The slope of any curve in the linear range is the understeer gradient. (Note that the values on the horizontal and vertical axes are absolute values for presentation purposes.)

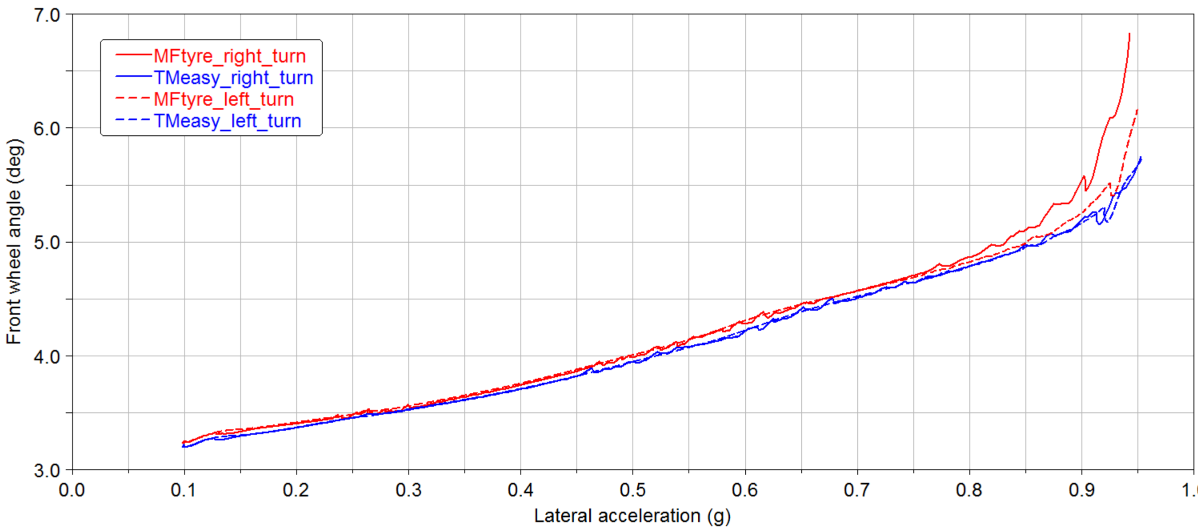

Figure 4.3.1: *TMeasy vs. MFtyre constant radius cornering understeer behaviour comparison*

In general it can be said that TMeasy correlates to MFtyre in the linear range. However MFtyre yields a higher progressiveness for the curve in higher lateral accelerations. Comparing left to right hand turns there is not much difference between the two TMeasy curves while MFtyre shows that the vehicle is more understeer in a right hand turn compared to a left hand turn for high lateral accelerations. This as mentioned before is due to tire asymmetric behaviour.

4.4 Single lane change

This use-case is selected to asses two different aspects of tire modelling. First is to compare TMeasy and MFtyre in a combined slip situation. Second is to show one of TMeasy model limitations which is replicating the tire transient behaviour.

This test is done by giving a sine input to the steering wheel of the vehicle travelling at 80 km/h, Figure 4.4.1. The speed is kept constant throughout the test by applying throttle if needed. The metric is the settling time of the chassis yaw rate after the steering sine input is finished and its value is zero again ($t = 3$ sec in the figure). The settling time is defined as the time it takes for chassis yaw rate to settle to a value within ±1% of its final value:

$$t_s = t_{(\dot{r} \to \dot{r}_{final} \pm 1\%)} - t_{steering\_end} \tag{4.4.1}$$

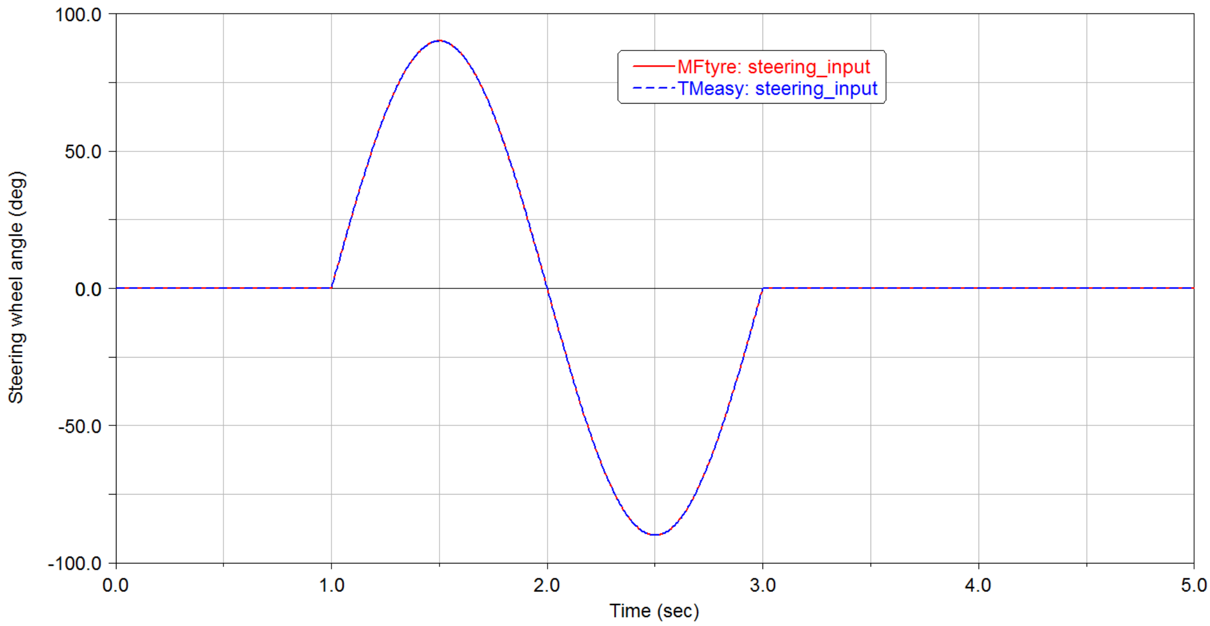

Figure 4.4.1: *TMeasy vs. MFtyre single lance change test steering input*

Table 4.4.1 compares the results obtained for two tire models. Figure 4.4.2 shows the yaw rate vs. time for the simulations done with TMeasy and MFtyre. The settling time for the TMeasy model is less than the one for the MFtyre model. The reason for that is that the MFtyre model needs time to build up the lateral force when the slip angle changes (relaxation effect) while the TMeasy model generates a the new lateral force for a changed slip angle instantly (Figure 4.1.5). Thus if we assume at $t = 3$ sec the slip angle on all tires become zero, it takes more time for MFtyre to remove the lateral force from the tires compared to TMeasy in which lateral forces become zero instantly.

Tire model	**Settling time (s)**	**Δ**
MFtyre	0.725	
TMeasy	0.505	-0.220 s (-30.3%)

Table 4.4.1: TMeasy vs. MFtyre single lane change results

When the steering input starts to deviate from 0 in the beginning of the sine input, just after $t = 1$ sec, a minor delay can be seen from the MFtyre model. This results in a slight relative error which accumulates throughout the steering input phase. Another source of this relative error, particularly seen at the peak values, is due to the different rolling radius calculation methods used in these models and its effect on the normal load on the tire which in turn effects the longitudinal and lateral force values generated.

Figure 4.4.3 summarizes the effect of the errors mentioned from another perspective, by observing the chassis roll angle. It can be seen that the vehicle running with the TMeasy tire model rolls less than the one running with MFtyre. This means that the vehicle with TMeasy model has lower lateral weight transfer, hence higher effective axle cornering stiffness. This causes the vehicle to stabilize faster compared the vehicle with the MFtyre tire model.

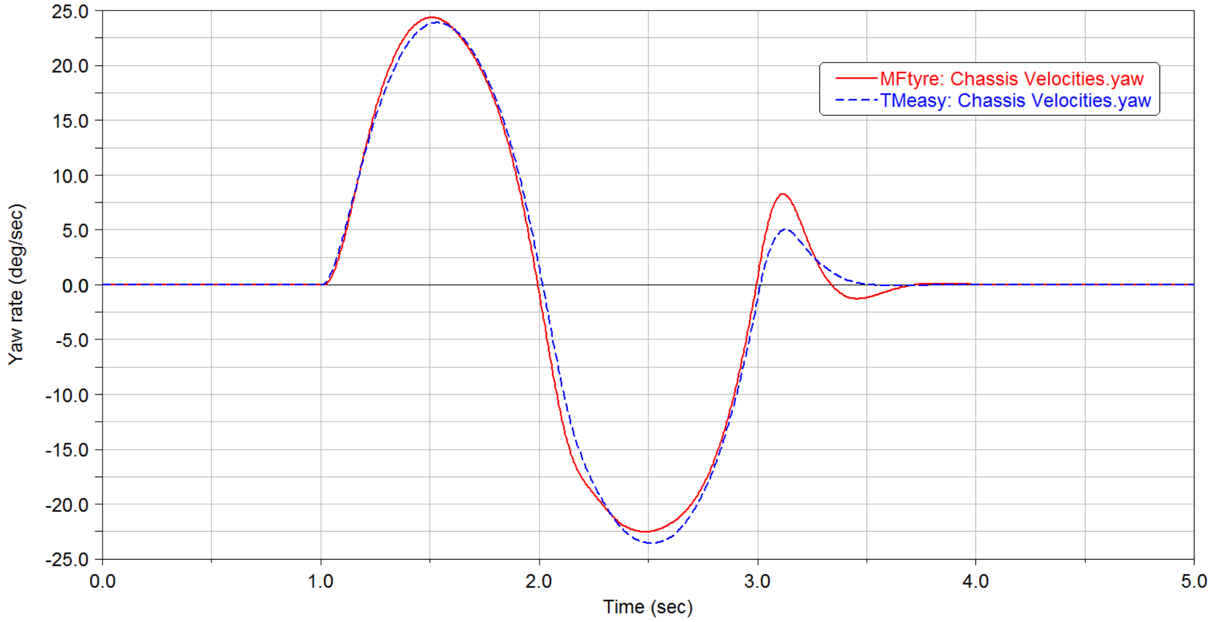

Figure 4.4.2: *TMeasy vs. MFtyre single lance change test chassis yaw rate*

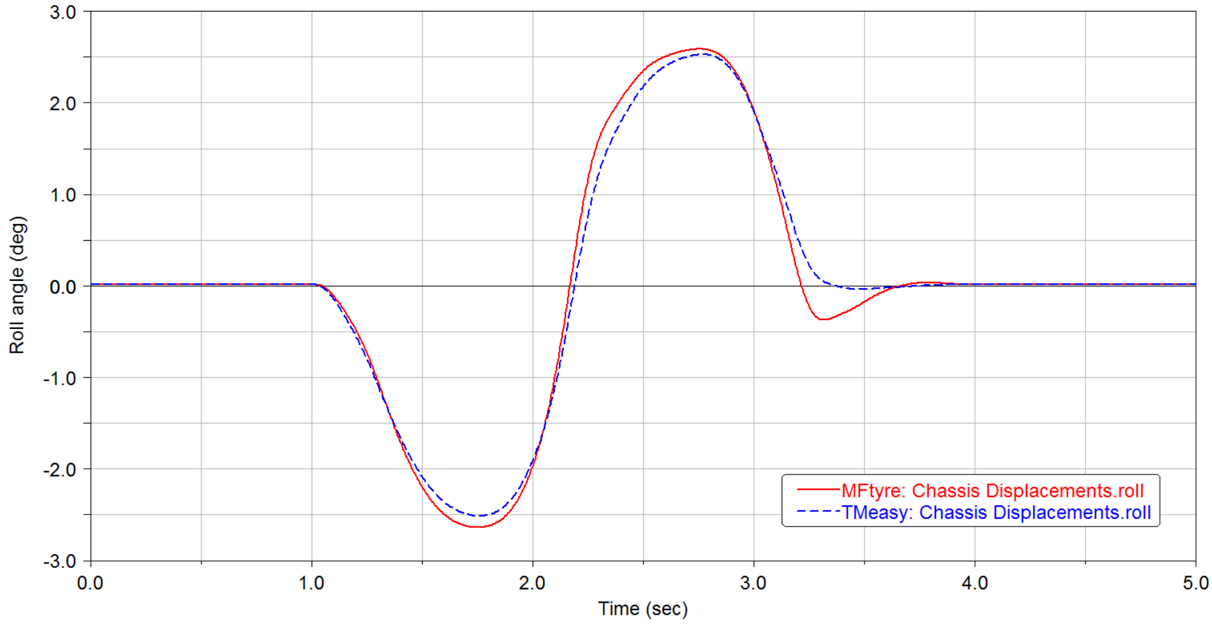

Figure 4.4.3: *TMeasy vs. MFtyre single lance change test chassis roll angle*

5 Sensitivity Analysis

The aim of the sensitivity analysis is to investigate the effect of the tire parameters selected in the application survey on the metrics defined for each particular use-case. Three use-cases are presented here: straight line braking with stopping distance as the metric, constant radius cornering with linear range understeer gradient as the metric and single lane change with yaw rate settling time as the metric.

This chapter strives to demonstrate the working method for a sensitivity study. Investigating the results of the analysis to draw conclusions and provide guidelines for tire development are outside the scope of this thesis work.

The reference tire is "tire 1" modelled by TMeasy. The tire is hence called "tmeasy_1" with parameters defined in Table 3.2.1 (page 16).

In each simulation a single tire parameter is changed and the results are compared to the simulations performed with the reference tire. The aim is to find out how much the metric changes in a particular use-case due to X % change in a single tire parameter. In some cases changing one tire parameter necessitates a change in other tire parameters due to physical or mathematical relations between them. For example changing the cornering stiffness without shifting the slip angle at maximum force, will affect the shape of the curve as well, which in some cases might not be a good representation of an actual lateral force behaviour for a tire.

5.1 Effect of maximum longitudinal force on stopping distance

For the straight line braking test only the effect of maximum longitudinal force on stopping distance is required. Two tires are made based on the reference tire. The maximum longitudinal force for the tire "tmeasy_1aa" is 10% more than the reference tire, while the same force for tire "tmeasy_1ab" is 10% less than the reference tire. As full ABS braking (120 N on pedal) is performed special care was taken to ensure the shape of the curve is preserved for all three tires. This was done by changing the sliding force along with the maximum force. These changes are done for both nominal vertical load and 2 × nominal vertical load.

Figure 5.1.1 compares the longitudinal force characteristics of these three tires. Table 5.1.1 compares the metric stopping distance for these three tire models.

Tire model	**Changes made**	**Stopping distance (m)**	**Δ**
tmeasy_1	Reference	34.8	
tmeasy_1aa	10% increase in F_x^M and F_x^G	32.5	-2.3 m (-6.6%)
tmeasy_1ab	10% decrease in F_x^M and F_x^G	38	+3.2 m (+9.2%)

Table 5.1.1: Straight line braking sensitivity analysis results

As shown in Figure 5.1.2, increasing the maximum longitudinal force have increased the steady-state longitudinal deceleration rate of the vehicle resulting in a decrease in the stopping distance. While decreasing the maximum longitudinal force decreases the steady-state deceleration rate of the vehicle and hence the vehicle should travel more to get from 100 to 2km/h.

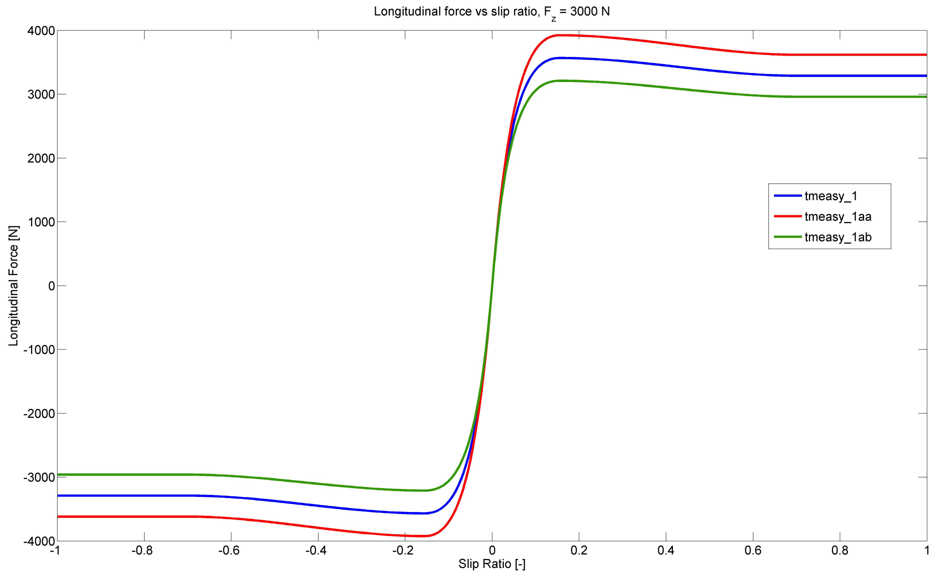

Figure 5.1.1: *Longitudinal force vs. slip ratio characteristics for tires tmeasy_1, tmeasy_1aa and tmeasy_1ab*

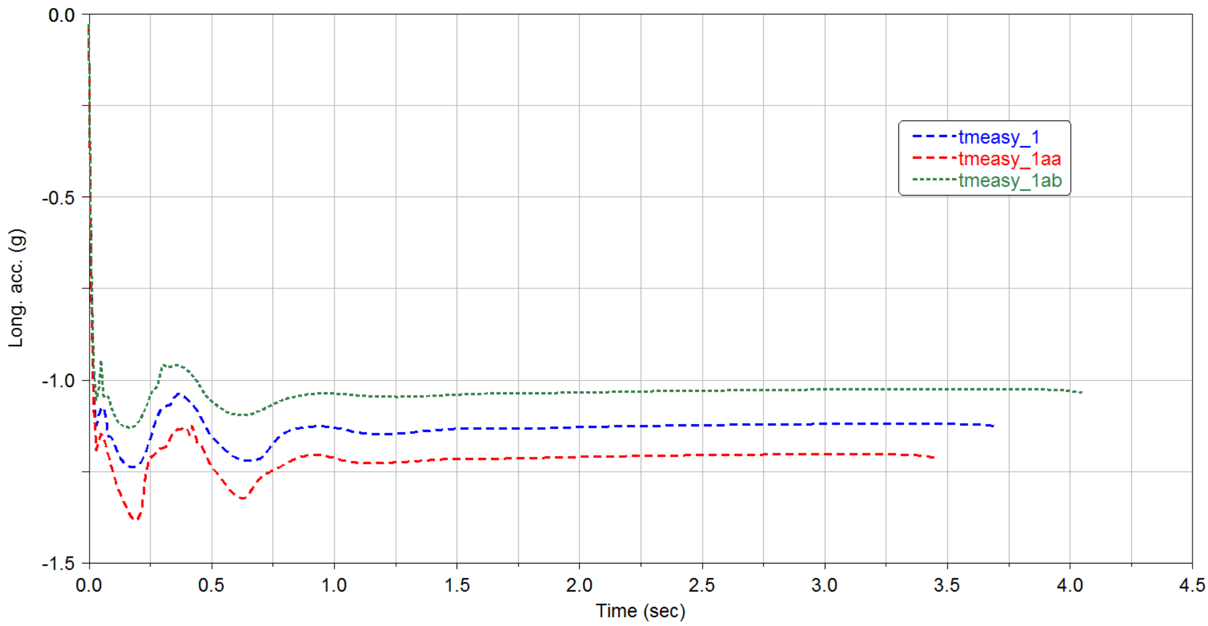

Figure 5.1.2: *Longitudinal acceleration sensitivity for straight line braking events with tires tmeasy_1, tmeasy_1aa and tmeasy_1ab*

5.2 Effect of maximum lateral force and cornering stiffness on linear range understeer gradient

For the constant radius cornering test two tires are introduced. Tire "tmeasy_1ba" has 10% increase in maximum lateral force compared to the reference tire. Tire "tmeasy_1bb" has 30% decrease in cornering stiffness compared to the reference tire. These changes are done for both nominal vertical load and 2 × nominal vertical load.

Figure 5.2.1 shows the lateral force characteristic for the tires used in this study. In case of tmeasy_1bb the slip angle at maximum lateral force is slightly increased to make a smooth transition in the curve just before the peak.

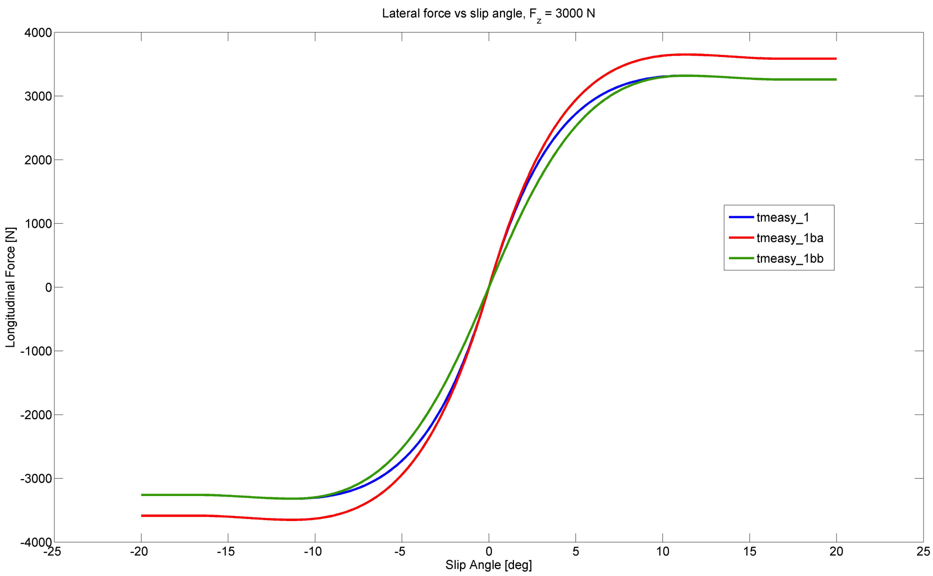

Figure 5.2.1: *Lateral force vs. slip angle characteristics for tires tmeasy_1, tmeasy_1ba and tmeasy_1bb*

Figure 5.2.2 compares the metric understeer gradient for all three cases. Table 5.2.1 shows the linear range understeer gradient for the simulations performed with these tires. It can be seen that the tire with less cornering stiffness (tmeasy_1bb) is more understeer in the linear range but becomes less understeer compared to the reference tire in high lateral accelerations due to the shift in the slip angle at maximum lateral force. Increasing the maximum lateral force (tmeasy_1ba) have caused the vehicle to be less understeer for all the lateral accelerations.

Tire model	**Changes made**	**Understeer gradient (deg/g)**	**Δ**
tmeasy_1	Reference	1.69	
tmeasy_1ba	10% increase in F_y^M and F_y^G	1.66	-0.03 deg/g (-1.8%)
tmeasy_1bb	30% decrease in dF_y^0	1.85	+0.19 deg/g (+11.4%)

Table 5.2.1: Constant radius cornering sensitivity analysis results

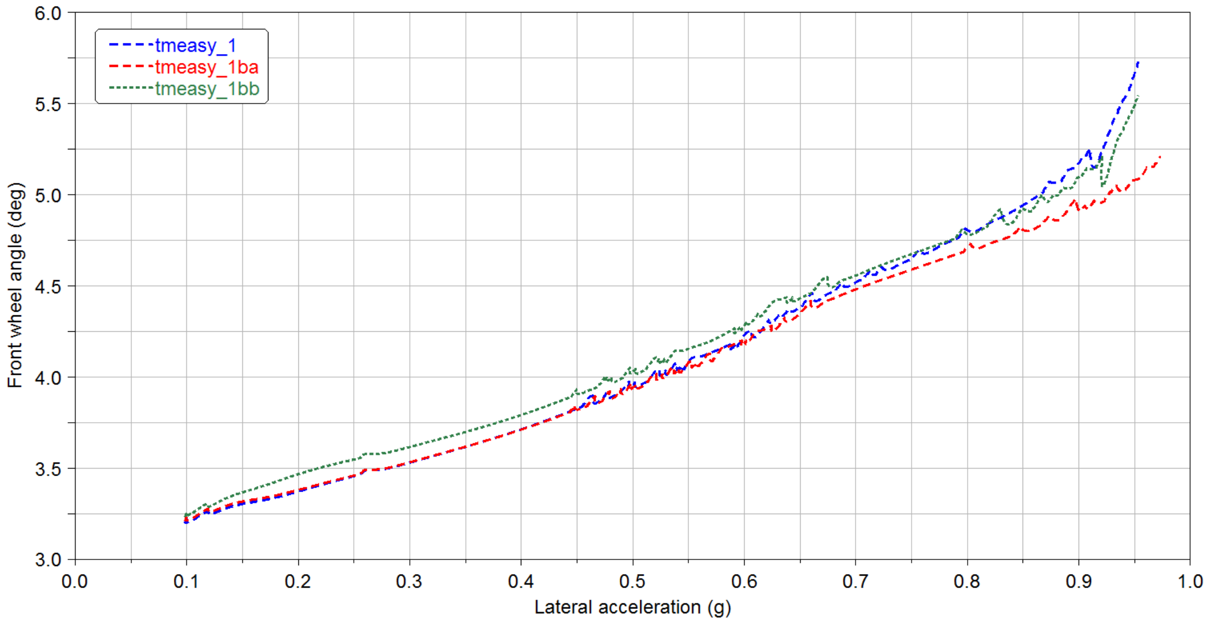

Figure 5.2.2: *Understeer gradient for constant radius cornering events with tires tmeasy_1, tmeasy_1ba and tmeasy_1bb*

5.3 Effect of maximum lateral force and cornering stiffness on yaw rate settling time

For the single lane change the same tires used in constant radius cornering test are utilized. Table 5.3.1 shows the yaw rate settling time for all three cases and their difference. Figure 5.3.2 shows the yaw rate response of the vehicle in all three cases just after the sine steering input is finished at $t = 3$ sec. It can be deduced that the settling time have increased for both tires tmeasy_1ba and tmeasy_1bb, compared to the reference case. The tire with the higher lateral force capability (tmeasy_1bb) results in a lower yaw rate during the whole manoeuvre, Figure 5.3.1.

Tire model	Changes made	Settling time (s)	Δ
tmeasy_1	Reference	0.505	
tmeasy_1ba	10% increase in F_y^M and F_y^G	0.905	+0.400 s (+79.2%)
tmeasy_1bb	30% decrease in dF_y^0	1.2	+0.695 s (+137%)

Table 5.3.1: Single lane change sensitivity analysis results

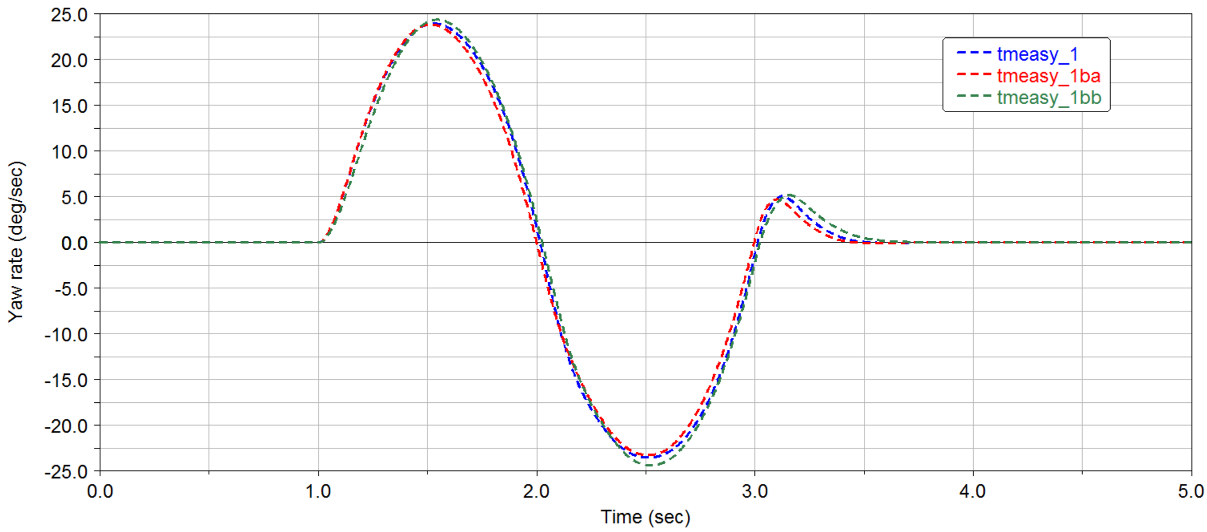

Figure 5.3.1: *Yaw rate for single lane change events with tires tmeasy_1, tmeasy_1ba and tmeasy_1bb*

Figure 5.3.3 shows these differences from the tire's perspective. The lateral forces on the left rear tire are selected to be presented because of the weight transfer to the left at the end of the manoeuvre and also in rear because the differences are more evident. It can be seen that tmeasy_1ba generates more force than the reference tire. tmeasy_1bb with less cornering stiffness generates the same lateral force as the reference tire but with a slower response which is due to the shift of the slip angle at maximum force. The effect of this shift can be seen at the end of the manoeuvre, where for the same slip angle the tire with the lower cornering stiffness generates less lateral force.

In section 4.4 it was stated that TMeasy does not incorporate the tire relaxation effects and hence it does not correlate with the results from MFtyre simulations. Based on that, special care should be taken in using the actual values generated by TMeasy for this special use-case where the metric is heavily influenced by the tire relaxation effects. However TMeasy can still be used to identify the trends and the direction of changes in these kind of sensitivity studies (with transient effect). This is particularly of importance as this shows that, despite the shortcomings mentioned with the model, the results can still be used to draw conclusions on what tire parameters should change and in what direction, in order to achieve a desired value for a use-case metric.

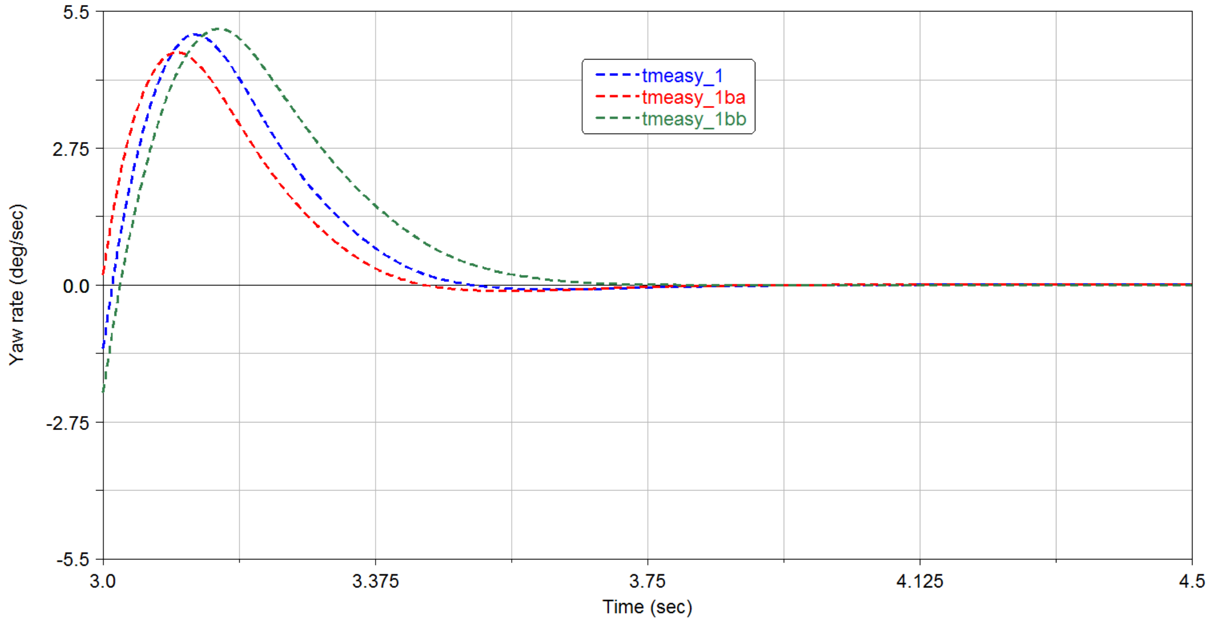

Figure 5.3.2: *Yaw rate settling after $t = 3$ sec for single lane change events with tires tmeasy_1, tmeasy_1ba and tmeasy_1bb*

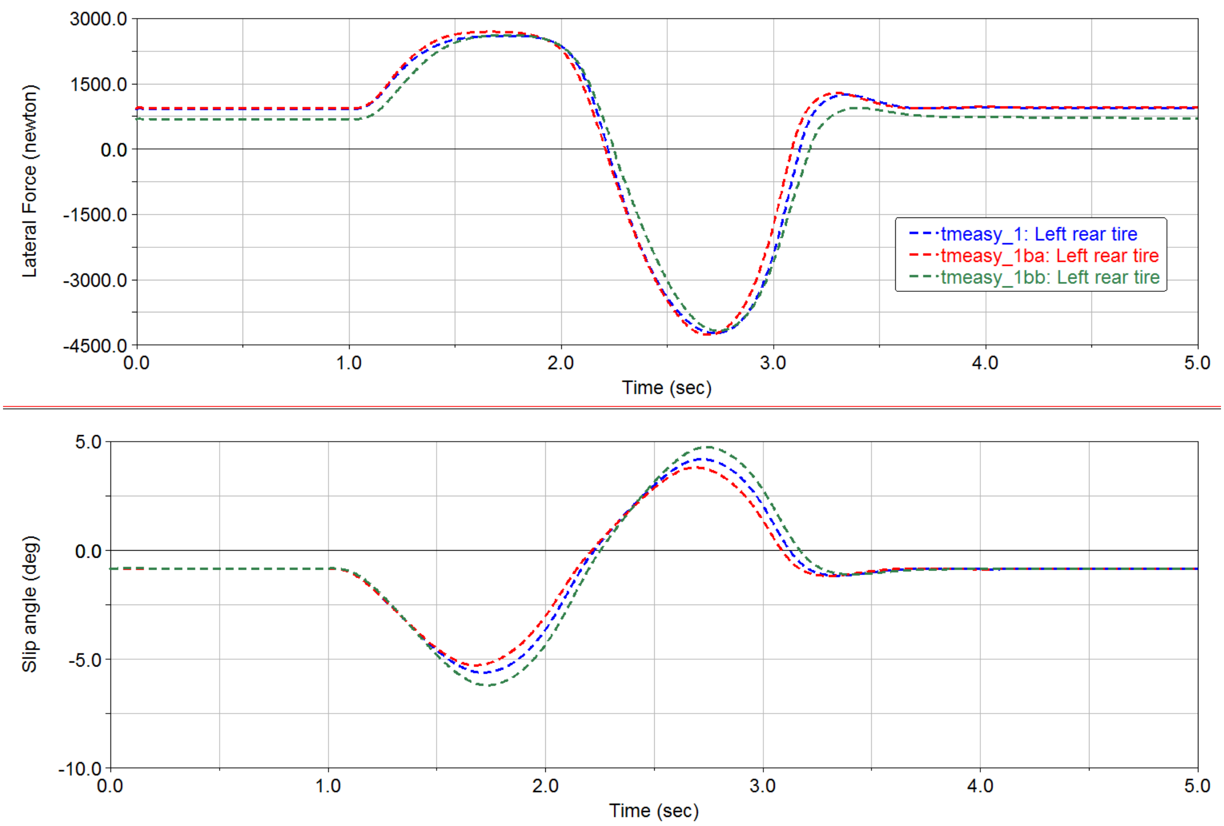

Figure 5.3.3: *Lateral force on the left rear tire for single lane change events with tires tmeasy_1, tmeasy_1ba and tmeasy_1bb*

6 Conclusions and Recommendations

6.1 Conclusion

The aim of this work was defined as “Developing a method to study the effect of tire characteristics on vehicle handling through sensitivity analysis”. For that purpose an application survey was done within Volvo Cars to identify different use-cases in handling field of vehicle dynamics. In each use-case a metric was defined as an index. A metric is used to describe the vehicle characteristics in a particular use-case or scenario. Also in each use-case, certain tire parameters were identified to be used in sensitivity studies.

A literature study was done to identify and categorize available tire models and to select one that is suitable for sensitivity studies based on the results of the application survey. TMeasy was chosen because it represents tire force and moment behaviour relatively well. Another reason for selecting TMeasy is that the model parameters have physical significance and are the same as the tire parameters realized in the application survey for sensitivity analysis. TMeasy, however, has certain limitations such as inaccuracy in self-aligning torque modelling, exclusion of relaxation effects and inability to capture tire asymmetric force and moment generation.

TMeasy was then verified against MFtyre in three particular use-cases. The use-cases were selected to assess different aspects of tire modelling as well as showing the limitations of TMeasy. In general, it was shown that the results of the metrics generated with TMeasy model correlates well the ones generated with MFtyre.

Sensitivity analysis was done in three use-cases in which the effect of different tire parameters where investigated. The physical significance of TMeasy model parameters made it easy to change the tire characteristics and study their effects on vehicle handling. Even in the use-case where the metric was influenced by relaxation effects, it was deduced that the method can still be applied with the current implementation of TMeasy to identify which tire parameters can be used and in what direction the changes can be made.

As a conceptual model TMeasy can be used when there is little or no test data available. In such a case tire parameters can be pragmatically estimated from another tire, thanks to the parameter set used in TMeasy model. Sensitivity studies can be performed to analytically investigate the effect of tire characteristics on handling attributes of the vehicle.

In general, all the chassis systems in a road vehicle are developed based on tire characteristics. Particularly suspension systems are designed around the tire properties such that they maximize the performance of the tire. This has always been done by a “black box” approach where tire is assumed to be unchangeable and the whole suspension design and geometry is altered to reach the targets set for different attributes. Fulfilling those numerous ride, handling, steering, durability, NVH and packaging targets always results in set of compromises depending on the importance and priority of each area and the corresponding attributes. So in some cases an attribute is sacrificed to achieve a target in another.

Engineers have been successfully able to optimize the vehicle for certain road and tire properties. Although it is not possible to make any changes on the road conditions, tire development is somewhat under the influence of car manufacturers. Having few physically meaningful parameters in the tire model makes it possible to perform optimization on the vehicle-tire combination rather than optimising each subsystem in isolation. Therefore having the possibility of changing tire parameters can open new opportunities for chassis and suspension design, specially in cases where there has to be a trade-off in the chassis design to accommodate a target for a particular attribute.

However, driving the tire development based on analytical methods like this has been unsuccessful in the past. Technically it is hard for the manufacturer to know the exact force and moment characteristic of a tire in advance. Although approximative methods are used, the actual force and moment characteristics are obtained by physically testing a tire after its production. This is very costly and in some cases impossible due to short lead times. Therefore, in the automotive industry, the tire selection process is mostly driven by subjectively testing the tires on a prototype vehicle. The method developed can be used to support this selection process. In most cases the macro-scale measures used in subjective testing can be related to physical tire parameters. Utilizing this method, it is possible to change those parameters analytically and investigate the effects on the metrics. This ultimately helps in reducing the testing hours needed for tire selection by providing an opportunity to do a pre-selection analytically using simulations. Another way this method can help in tire selection process and reducing testing time is by enabling the engineers to set targets for tire development using aforementioned vehicle-tire optimizations.

6.2 Recommendations

To increase the confidence in the method, certain tasks should be performed. Within the scope of this work, the TMeasy model was verified using the MFtyre model. It was assumed that MFtyre represents the tire force and moment characteristics precisely. This assumption, however, is not entirely correct. MFtyre has some limitations as well, for example in the combined force generation under high loads or on low friction surfaces. Also the quality of data used for obtaining the MFtyre for tire 1 is unknown. Therefore, it is recommended that the simulations performed with TMeasy be compared with physical objective tests, to be able to validate the tire model. It goes without saying that the vehicle model used in Adams/Car has to be validated beforehand. This can be done on any previous test and the corresponding validated Adams/Car model and hence it does not require any separate physical tests to be performed. The presence of a "good quality" test data for the tires is also vital for building accurate TMeasy tire models for the sake of validation.

The number of sensitivity study cases and tire parameters involved in this work were limited, due to time constraints. To improve the confidence in the method, it is highly recommended that more use-cases and tire parameters be involved in further analyses.

Regarding TMeasy, the behaviour of self-aligning torque could be improved by introducing more mature assumptions for lateral force distribution over the contact area length and calculation of the application point for the resultant force. Relaxation effects can also be included by introducing lagged longitudinal and lateral slip quantities, or by using a simple stretched string model, as is currently used in MFtyre for simulating tire relaxation effects. Also some scaling parameters could be used to model tire asymmetrical force and moment behaviour.

In the sensitivity studies performed, the tire parameters used were the ones that are known to have an effect on the use-case metrics. The next step would be investigating the influence of other tire parameters such as vertical stiffness, unloaded radius and tire width, on vehicle handling. Ultimately, it is desirable to use the method in a suspension-tire optimization process to further investigate the possibilities it might yield in chassis and suspension design as well as tire development.

A Appendix: other tire models

MF-SWIFT

Short **W**avelength **I**ntermediate **F**requency **T**ire model is constructed using the magic formula (MF) tire model plus a rigid ring which represents the tire belt. It is made to work in higher frequencies than the MF model, typically between 60-100 Hz as long as the wavelength is relatively short (>20 cm). It also include gyroscopic effects. The model is described in [25, Ch. 9] and [28]. It has evolved since then by TNO under the commercial name of MF-Swift. Different versions with added features have been released, refer to the backward compatibility graph in [21, p. 14]). MF-SWIFT is mostly used in ride and comfort applications.

TreadSim

This physical tire model was originally developed by Pacejka and presented in section 3.3 of [25]. The calculation of the forces and moments is based on the time simulation of the deformation history of one tread element while moving through the contact zone. In [29] Uil argues that a weak point of TreadSim is that the peak lateral friction coefficient for high vertical loads is generally too high and its dependency on inflation pressure does not match the behaviour found in tire F&M Measurements. Poorly modelled contact patch shape and pressure distribution are recognised as the cause for that. There are also some assumptions that are not realistic such as realization of carcass deformation of approximately 0.5 m. Although TreadSim is a physical tire model, the tire properties need to be determined from a tire measurement data set.

Dynamic tire friction model

Is a physical brush model based on LuGre friction modelling presented in [10]. The LuGre friction model was developed by Department of Automatic Control at Lund University (Sweden) and Laboratoire d'Automatique de Grenoble (France).It describes a dynamic force phenomena when frictional surfaces are sliding on each other. In the formulation for tires LuGre assumes that the friction surface is consisted of bristles that their movements are described by differential equations. Duer has feature developed the model by describing the dynamics of longitudinal lateral friction forces as well as self-aligning moment [11], [9] and [8]. The model parameters are highly dependent on the tire-road condition. Apart from being hard to measure, most of the parameters do not represent physical characteristics of the tire. These being mostly friction model parameters that are hard to relate to tire characteristics.

CDtire

CDtire was originally developed for comfort and durability purposes. The CDtire model family consists of:

- CDtire 20: a rigid ring model with (global) viscous-elastic side-wall, geometrically parametrized normal contact and partial differential equation based tangential contact,
- CDtire 30: a flexible belt, rod-type in-plane model with (local) viscous-elastic side-wall and brush type contact,
- CDtire 40: a flexible belt, shell-type 3D model with (local) viscous-elastic side-wall and brush type contact, [5].

In terms of construction and applications this model is very similar to FTire. For some results of model verification, see [4].

RMOD-K

RMOD-K is the name of a family of tire models that mostly reside on the "complex physical models" category. In its most sophisticated form (flexible belt) this detailed finite element model describes the actual shape of tire structure and can cover road undulation frequencies up to 100 Hz. The belt is attached to the rim with a side wall model that considers the effect of pressurized air in the tire. The belt can have more than one layer. These layers can interact with each other (frictional forces, etc.). The friction model includes both sliding and adhesion functions and also realizes the effect of temperature and contact pressure. The complexity can be reduced based on application. In other words, the model can switch to simplified representations or run in hybrid mode. RMOD-K is mainly designed for ride, comfort and durability applications and it has been implemented in ADAMS [23]. For a complete list of literature (mostly in German) refer to [22].

Hankook tire model

This semi-physical tire model is developed by Hankook tire Co. Ltd R&D center [14]. The steady-state behaviour is modelled through use of analytical formulation based on physical characteristics of the tire. Three forces and three application points are determined this way. For the transient behaviour different slip sweep rates are used to determine the transient tire characteristics. Although some of the parameters represent physical characteristics, the analytical formulation is mostly polynomial form with mathematical fitting parameters. In the literature it is stated that these parameters are to be experimentally derived but it is not clear how and from what kind of tests (slip sweeps supposedly).

Fiala

Fiala tire model is the standard tire model in all Adams/Tire modules [2]. It is a physical model, where the carcass is simulated as a beam with an elastic foundation for modelling the deformations in the lateral direction. The carcass-road contact is modelled by elastic brush elements. It is assumed that the contact patch has a rectangular shape with a uniform pressure distribution across the footprint. Based on these assumptions and neglecting the influence of camber angle, the expressions for steady-state slip conditions can be analytically derived. Based on these slip conditions the lateral and longitudinal forces are derived.

The nine inputs needed for the Fiala tire model are very simple and can be found in Table A.0.1 .

Parameter	**Description**
M_t	Mass of tire
Vertical_damping	Vertical damping coefficient
Vertical_stiffness	Vertical tire stiffness
CSLIP	Partial derivative of longitudinal force (Fx) with respect to longitudinal slip ratio at zero longitudinal slip
CALPHA	Partial derivative of lateral force (Fy) with respect to slip angle at zero slip angle
UMIN	Coefficient of friction with full slip (slip ratio = 1)
UMAX	Coefficient of friction at zero slip
Rolling_resistance	Rolling resistance coefficient
Relax_length_X	Relaxation length in X
Relax_length_Y	Relaxation length in Y

Table A.0.1: Fiala tire model parameters

In general the Fiala tire model is extremely simple and easy to use. But it is not suitable for sensitivity studies for several reasons. The main reason is that the representation of tire behaviour is extremely simplified. Important phenomena such as camber effects are not considered. The other reason is that not all the parameters have physical meaning. For example UMAX is a theoretical value and can not be obtained in reality because there is always some slip in the contact patch (i.e. zero slip ratio is practically impossible to achieve).

Due to these shortcomings the use of Fiala tire model in full vehicle simulations for sensitivity studies is rather questionable.

UniTire

UniTire is a semi-physical unified non-linear steady-state tire model made by Guo [20]. It works based on fitting a mathematical function to the test data in order to obtain the tire resultant force. This resultant force can then be divided to lateral and longitudinal forces. These forces are dependent on their respective friction coefficients that are calculated by a revised version of Savkoor's formula [27]. This formula is based on few friction parameters and slip velocities of the contact patch in lateral and longitudinal directions. This enables

UniTire to model the speed dependency of the tire forces and moments, as it is presented in [17]. In this paper, Guo argues that the effect of speed on the steady-state forces and moments generated by the tire are not negligible. These differences are due to the friction variation as a function of the sliding speed.

However, the friction parameters describing the UniTire model are hard to obtain from conventional tire tests. The rest of the parameters are mathematical fits and do not have physical meaning. Also it is extremely hard to connect these parameters to the tire characteristics realized in the application survey for sensitivity studies. The current version of the model does not cover the transient tire behaviour. Therefore UniTire is not suitable for parametric sensitivity studies.

FTire

FTire (**F**lexible Ring **Tire** Model) is designed as a "2.5-dimensional" non-linear vibration model. Although mostly used for ride and comfort purposes, it can be used in handling simulations as well. This model is reviewed here since it is implemented in Adams/Car and it is an example of the complex theoretical tire models. FTire is considered as a discrete element model and is a compromise between the computationally heavy finite element models and the simple pure in-plane models [15].

In Ftire the belt is represented by a slim ring that can be displaced and bent in any direction relative to the rim. As shown in Figure A.0.1, the flexible belt is represented by a number (50 to 200) of segments elastically attached to the wheel rim and to each other. Each segment is consisted of a number of tread blocks that have radial, tangential and lateral non-linear stiffness and damping properties. The radial deflection of these tread elements is dependent on the road profile and orientation of the belt elements. Sliding coefficient and sliding velocity on the ground determine the tangential and lateral deflections. The six components of the forces acting at the wheel center are calculated by integrating the forces in the elastic foundation of the belt.

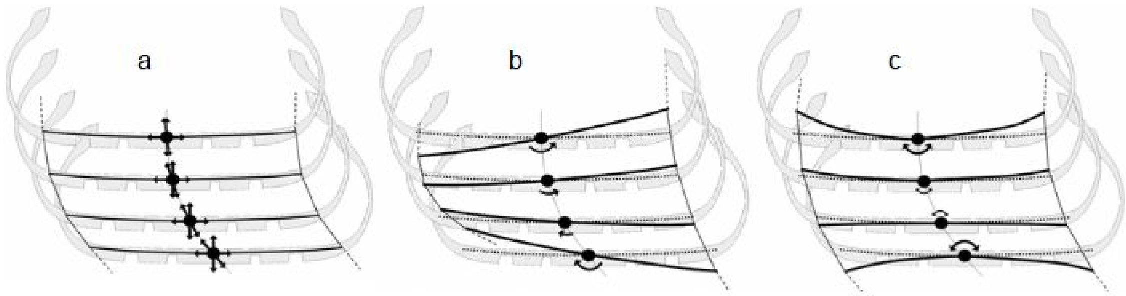

Figure A.0.1: *FTire belt segment degrees of freedom: (a) Translation, (b) Torsion and (c) Lateral bending, [16]*

Because of this construction FTire can be used with both short and long wavelength obstacles covering frequencies as high as 120 Hz. It can also cover complete stand-still without any model switch. And due to its high accuracy can be used in complex situations such as ABS braking on an uneven road.

Here a summary of the parameters needed for a FTire model is presented:

- Rolling circumference under normal running conditions
- Rim diameter
- Tread width
- Mass of the tire
- One out of:
  - Portion of tire mass that moves with the belt, OR
  - Tire radial stiffness
- Increase of overall radial stiffness at high speed with respect to the radial stiffness at stand-still and the wheel speed at which this dynamic stiffening reaches half of its value.

- Natural frequencies and respective damping moduli of first, second and fourth vibration modes of the inflated but unloaded tire with fixed rim.
- One out of:
  - Natural frequency of the fifth mode,OR
  - Belt in-plane bending stiffness
- One out of:
  - Natural frequency of the sixth mode,OR
  - Belt out-of-plane bending stiffness
- Profile height
- Rubber height over steel belt for zero profile height
- Tread rubber stiffness
- Percentage of contact area with respect to overall footprint area
- Tread rubber damping and elastic moduli
- Moment of inertia of the rim and other rotating parts
- Maximum and sliding friction coefficients

A detailed description of these parameters can be found in [12].

As it can bee seen the number of parameters that are needed for FTire to run is relatively large. These parameters are obtained from special measurements and test procedures described in detail in [13]. As mentioned before FTire is designed mostly for ride and comfort applications. The FTire parameters describe the structural properties of the tire and are not directly related to its handling characteristics. As an example it is hard, if not impossible, to change these parameters to reach a certain level of lateral force. The reason for this is that most of these parameters are completely interrelated to each other. For instance changing the profile height could change the natural frequencies or other parameters for different vibration modes. FTire, although presented as a compromise between the simple in-plane models and finite element models, is still computation heavy and time consuming specifically if it is compared to empirical tire models. Therefore FTire is not suitable for the sensitivity studies in the handling area.

B Appendix: Adams/Tire module axis system

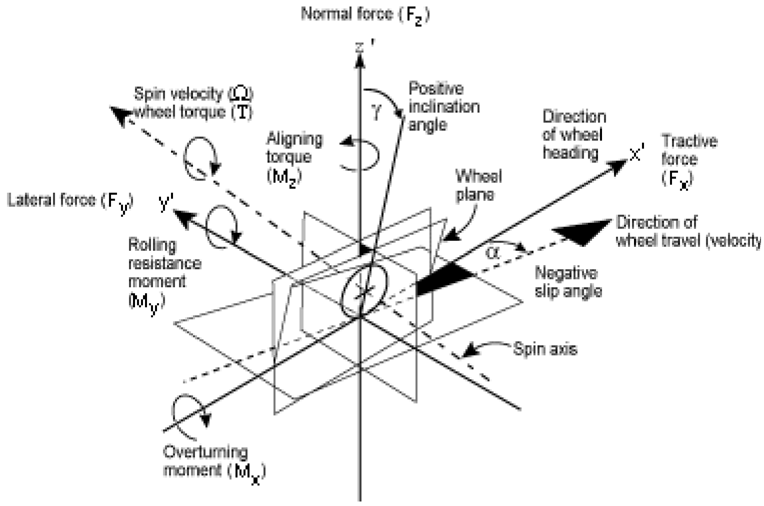

Figure B.0.1: *ISO coordinate system*

ISO-C (TYDEX C) Axis System

The TYDEX STI specifies the use of the ISO-C axis system for calculating translational and rotational velocities, and for outputting the force and torque at the tire hub. The properties of the ISO-C axis system are:

- The origin of the ISO-C axis system lies at the wheel center.
- The + x-axis is parallel to the road and lies in the wheel plane.
- The + y-axis is normal to the wheel plane and, therefore, parallel to the wheel's spin axis.
- The + z-axis lies in the wheel plane and is perpendicular to x and y (such as z = x × y).

ISO-W (TYDEX W) Contact-Patch Axis System

The properties of the ISO-W (TYDEX W) axis system are:

- The origin of the ISO-W contact-patch system lies in the local road plane at the tire contact point.
- The + x-axis lies in the local road plane along the intersection of the wheel plane and the local road plane.
- The + z-axis is perpendicular (normal) to the local road plane and points upward.
- The + y-axis lies in the local road plane and is perpendicular to the + x-axis and + z-axis (such as y = z × x).

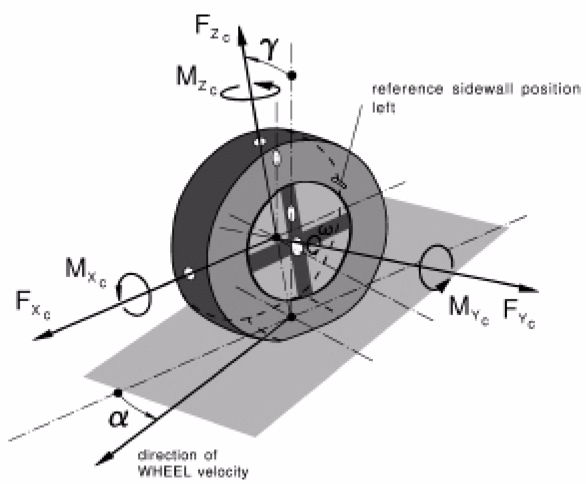

Figure B.0.2: *TYDEX-C axis system used in Adams/Tire*

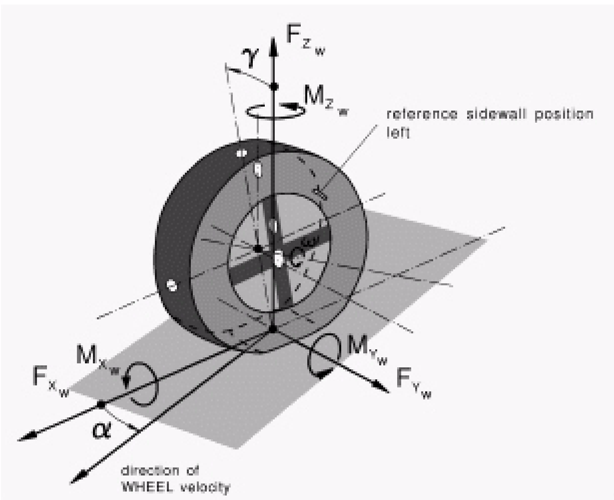

Figure B.0.3: *TYDEX-W axis system used in Adams/Tire*

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