Czech Technical University in Prague

Faculty of Electrical Engineering

BACHELOR THESIS

Bohdan Kashel

Development of methodology for vehicle parameters identification

Department of Control Engineering

Supervisor of the master thesis:	Ing.Denis Efremov
Study programme:	Cybernetics and Robotics

Prague 2020

BACHELOR'S THESIS ASSIGNMENT

I. Personal and study details

Student's name: **Kashel Bohdan** Personal ID number: **452850**

Faculty / Institute: **Faculty of Electrical Engineering**

Department / Institute: **Department of Control Engineering**

Study program: **Cybernetics and Robotics**

II. Bachelor's thesis details

Bachelor's thesis title in English:

Development of methodology for vehicle parameters' identification

Bachelor's thesis title in Czech:

Vývoj metodologie pro identifikaci jízdních parametrů vozu

Guidelines:

The goal of the thesis is to deliver suitable procedure for parameters identification of a vehicle dynamics using proposed maneuvers. Provide verification of such a methodology based on car models implemented in simulator "Live for Speed".
1. Familiarize with vehicle dynamics simulator "Live for Speed".
2. Adopt low-fidelity and high-fidelity vehicle models.
3. Suggest suitable procedure for identification of parameters of:
a. the rigid body;
b. tires;
4. Provide validation of identification procedure on different car models implemented in the simulator.

Bibliography / sources:

[1] Schramm, D., Hiller, M., & Bardini, R. (2014). Vehicle dynamics. Modeling and Simulation. Berlin, Heidelberg,151.
[2] Franklin, G. F., Powell, J. D., Emami-Naeini, A., & Powell, J. D. (1994). Feedback control of dynamic systems (Vol. 3). Reading, MA: Addison-Wesley.

Name and workplace of bachelor's thesis supervisor:

Ing. Denis Efremov, Department of Control Engineering, FEE

Name and workplace of second bachelor's thesis supervisor or consultant:

Ing. Tomáš Haniš, Ph.D., Department of Control Engineering, FEE

Date of bachelor's thesis assignment: **27.01.2020** Deadline for bachelor thesis submission: **14.08.2020**

Assignment valid until:
by the end of summer semester 2020/2021

Ing. Denis Efremov	prof. Ing. Michael Šebek, DrSc.	prof. Mgr. Petr Páta, Ph.D.
Supervisor's signature	Head of department's signature	Dean's signature

III. Assignment receipt

The student acknowledges that the bachelor's thesis is an individual work. The student must produce his thesis without the assistance of others, with the exception of provided consultations. Within the bachelor's thesis, the author must state the names of consultants and include a list of references.

.

Date of assignment receipt Student's signature

Acknowledgements

I would like to say a special thank you to my supervisor, Ing. Denis Efremov. His support, guidance, and overall insights in this field have made this an inspiring experience for me. And my biggest thanks to my family for all the support you have shown me through this research, the culmination of three years of distance learning. I am also grateful to the Department of Control Engineering, Faculty of Electrical Engineering, Czech Technical University in Prague. I am incredibly thankful and indebted for sharing expertise, and sincere and valuable guidance and encouragement extended to me.

I hereby declare that I have completed this thesis with the topic "Development of methodology for vehicle parameters identification" independently and that I have included a full list of used references. I have no objection to the usage of this work in compliance with the act §60 Zákon č.121/2000 Sb. (copyright law).

In .... date .... signature of the author

Abstrakt: Cílem této bakalářské práce je vytvoření možnosti testování ovladačů v prostředí simulátoru "Life for Speed". Podobné testování dovoluje vyvinout řídicí algoritmy pro vozidla typu "drive-by-wire". Výsledkem jsou přizpůsobené matematické modely vozidel, testy pro identifikaci parametrů modelu a proces identifikace parametrů modelu. Za základní modely byly vybrány modely "Single-Track" a "Twin-Track". Výhodou modelu Single-Track je možnost popisu podélné dynamiky vozů s dostačující přesností. Vzhledem k celkové jednoduchosti daného modelu ve srovnaní se Twin-Track, Single-Track navíc dovoluje rozpracovat metodu identifikace parametrů modelů. Model Twin-Track na druhou stranu umožĹuje testování identifikačních parametrů na komplexním modelu se složitějším fyzikálním popisem, jelikož zahrnuje deset stavů (oproti třem ve Single-Track). Tato práce přizpůsobuje existující Single-Track a Twin-Track modely s utvořením jejich kompatibilních vůči signálům z LFS verzí. Výsledkem této prace je také vytvoření návodu k samostatnému generovaní identifikačních dat, eventuálně vytvoření experimentů pro generovaní dat, vytvoření a / nebo přizpůsobování vzorců pro výpočet parametrů modelů a porovnání modelů s LFS. V rámci této bakalářské práce byly také vytvořené signály pro generovániĚ dat, soubory automatické inicializace a soubory pro výpočet parametrů modelů.

Klíčová slova: "Life for Speed", single-track, twin-track, identifikace, identifikace parametrů pneumatiky

Abstract: This thesis's general goal is to create a possibility to test controllers in the "Life for Speed" vehicle simulator environment that opens the door to develop control algorithms for drive-by-wire vehicles. Completing this goal are adaptation vehicle models, creating tests for identification of the model's parameters, and identification of model's parameters. Single-Track and Twin-Track models were chosen as vehicle models. This work uses an adopted Single-Track model because it has sufficient fidelity to describe lateral vehicle dynamics and to test identification methods because it is simpler than the Twin-Track models (three versus ten states). Adopted Twin-Track was used because it allows testing identification on the more complex model (ten states) with more complicated physics. This thesis includes an adaptation of existing Single-Track and Twin-Track models for compatibility with signals from LFS, guide how to generate data for identification, creating and/or adapting formulas for calculating model parameters, and comparisons between each model and LFS. They were also created presets for generating needful data, automatic initialization files, and mat-files that calculate parameters for models.

Keywords: "Life for Speed", single-track, twin-track, identification, tyre identification

Contents

1. Introduction

Technology has the power to do many things, and changing our world for a better place is one of them. We are privileged to be living in a time where science and technology can assist us, make our lives easier and safer, change the ways we think and solve daily problems. The technology we were already exposed to and accustomed to has paved the way for us to innovate further, and one of those technologies is a drive-by-wire technology.

At the moment when I heard that there is a chance to contribute to something that will bring the future closer, I realized that I could not lose this opportunity. This thesis is a part of a bigger project of adopting a vehicle dynamics simulator, "Life for Speed,[1] " for needing of automatic control department. The general goal of the team is to develop control strategies for the car of the future with fully drive-by-wire technology.

The general goal of this thesis is to create a possibility to test controllers in the "Life for Speed" environment, that opens the door to develop control algorithms for drive-by-wire vehicles. Completing this goal are adaptation vehicle models, creating tests for identification of the model's parameters, and identification of model's parameters. Single-Track and Twin-Track models were chosen as vehicle models. It includes the Single-Track model because this type of model allows a physically plausible description of vehicles' driving behavior without significant modeling and parameterization efforts. This fact means that a Single-Track is a perfect choice for the development of controllers. The Twin-Track model was chosen because of complexity. This complexity allows test developed controllers on the advanced model.

1.1 Outline

This work is divided into 8 parts.

In the first two parts, which are [**Introduction**] and [**Objectives**], work description and goals are stated.

The 3th and 4th parts, which are [**Nonlinear Single-Track Model**] and [**Nonlinear Twin-Track Model**] introduces used vehicle's and tire's models.

In part, which is named [**Model identification**], identification process for Single-Track and Twin-Track rigid bodies and tires is described.

The 6th chapter, which is [**Comparisons between LFS and Models**], presents to reader's attention a comparisons between various parameters from LFS and models.

The next part, [**Results**], lists reached goals.

The last part, [**Conclusion**] summarises this thesis.

2. Objectives

The primary objectives of this thesis are:

- Familiarize with vehicle dynamics simulator "Live for Speed"[1] (in text will be used abbreviation LFS).
- Adopt low-fidelity (Single-Track) and high-fidelity (Twin-Track) vehicle models
- Create experiments which generate data for lateral motion
- Create experiments which generate data for longitudinal motion
- Suggest suitable procedure for identification of parameters of:
  - the rigid body;
  - tires;
- Provide validation of identification procedure on different car models implemented in the simulator.

3. Nonlinear Single-Track Model

3.1 Main model

Nonlinear Single-Track model it is adopted Single-Track model from [2]. Vehicle coordinate system used in this thesis is shown in Figure [3.1]. It is the conventional right-hand Cartesian coordinate system. The x axis follows from the center of gravity to the front of the vehicle. The y axis goes towards the left side of the car, and z axis lies from the center of gravity to the top of the vehicle. The vehicle's yaw has a positive angle increment while turning to the left.

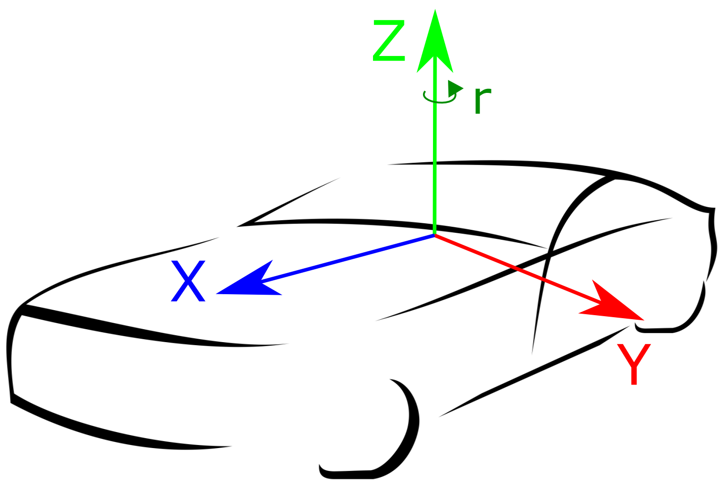


Figure 3.1: The vehicle coordinate system [2]

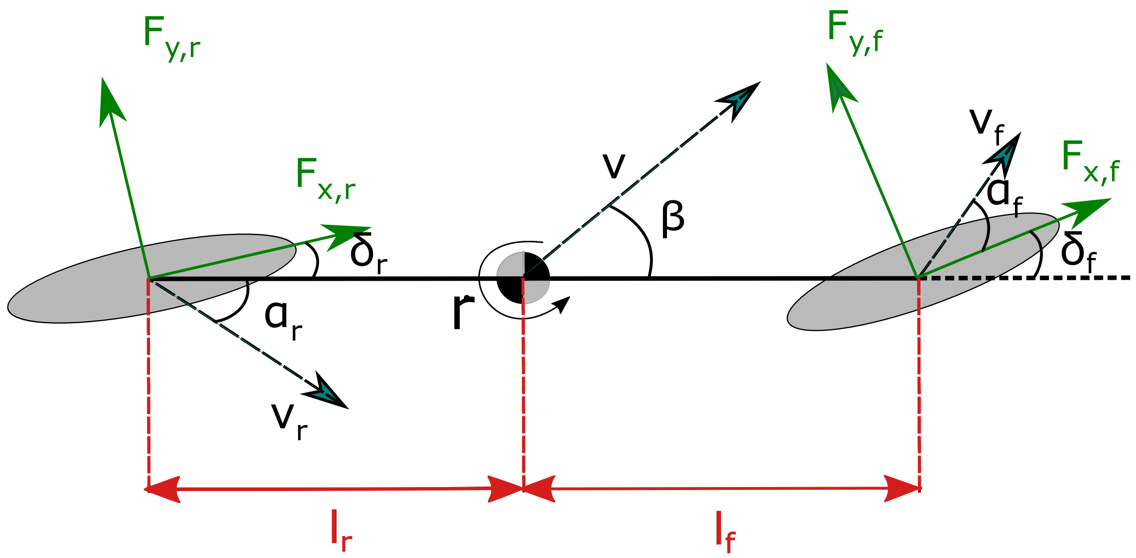


Figure 3.2: The single-track model [2]

The Single-Track model (Fig.3.2) has 3 degrees of freedom and is used to represent the planar translation and yaw motion of a vehicle and consists of 3 states and 4 inputs, which are listed in Table 3.1. All parameters used in the model are presented in Table 3.2. The block diagram of the nonlinear singe-track model is shown in Figure [3.3].

Table 3.1: States and inputs of the Single-Track

State/Input	Symbol	Units
Velocity of CG	v	$\mathrm{m\,s^{-1}}$
Side-slip angle	β	rad
Yaw rate	r	$\mathrm{rad\,s^{-1}}$
Steering angle of the front axle	δ_{f}	rad
Steering angle of the rear axle	δ_{r}	rad
Angular velocity of the front wheel	ω_{f}	$\mathrm{rad\,s^{-1}}$
Angular velocity of the rear wheel	ω_{r}	$\mathrm{rad\,s^{-1}}$

Table 3.2: Parameters of the Single-Track

Parameter	Symbol	Units
Vehicle mass	m	kg
Yaw moment of inertia	I_{yaw}	$\mathrm{kg\,m^{-2}}$
Front axle-CG distance	l_{f}	m
Rear axle-CG distance	l_{r}	m
Radius of wheels	p	m
Aerodynamic drag coefficient	c_{air}	-
Air density	ρ	$\mathrm{kg\,m^{-3}}$
Frontal area of the car	A	$\mathrm{m^2}$
Shaping coefficients for lateral dynamics	$c_{\mathrm{D,y}}$	-
	$c_{\mathrm{B,y}}$	-
	$c_{\mathrm{C,y}}$	-
	$c_{\mathrm{E,y}}$	-
Shaping coefficients for longitudinal dynamics for throttling	$c_{\mathrm{D,x,t}}$	-
	$c_{\mathrm{B,x,t}}$	-
	$c_{\mathrm{C,x,t}}$	-
	$c_{\mathrm{E,x,t}}$	-
Shaping coefficients for longitudinal dynamics for breaking	$c_{\mathrm{D,x,b}}$	-
	$c_{\mathrm{B,x,b}}$	-
	$c_{\mathrm{C,x,b}}$	-
	$c_{\mathrm{E,x,b}}$	-

3.1.1 Used Assumptions and Simplifications

The nonlinear single-track model is used to describe planar vehicle motion consist next simplifications [3]:

- Vehicle motion include only lateral, longitudinal motions and yawing.
- Vehicle mass is concentrated at the center of gravity.
- Tires are represented as two tires, first tire on front and second tire on rear axle, with imaginary contact points between tires and surface on the center of axles.
- Neglecting of pneumatic trail and aligning torque resulting from a side-slip angle.
- Mass distribution on the axles is constant.
- Longitudinal forces on tires, resulting from a normalized tire slip angle, are neglected. All longitudinal forces acting on each axle are assumed to be strictly from the engine.

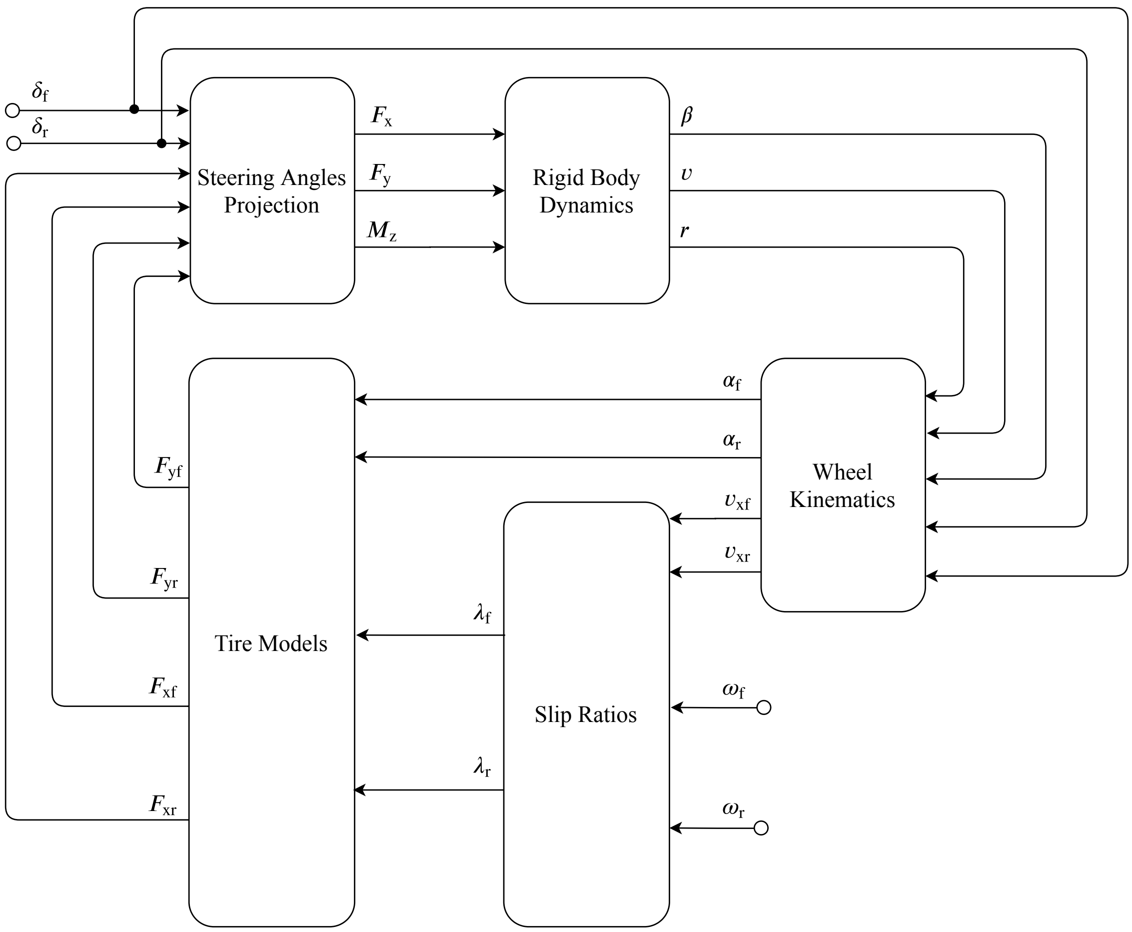


Figure 3.3: The block diagram of the single-track model [2]

3.1.2 Block Representation

The bloc representation of the nonlinear single-track model can be found in Fig. 3.3. Each block is described more detailed in the next sub-sections. The model consists of:

- **rigid body dynamics** block is for a chassis representation;
- **steering angles projection** block respond for computing of forces and angular momentum acting on the rigid body;
- **tire models** block calculates longitudinal and lateral forces from slip variables and restrictions made by traction ellipse;
- **wheel kinematics**, calculation part, which compute side-slip angles of the tires in the particular coordinate frame;
- **slip ratios** block for calculating how each wheel spins.

3.1.3 Rigid Body Dynamics

The rigid body dynamics have three degrees of freedom represented by translation motion (as the velocity of CG v), side-slip angle β, and rotation motion modeled by yaw rate r.

The longitudinal motion is derived as

$$F_{\mathrm{x}} = F_{\mathrm{x,tr}} + F_{\mathrm{x,rot}} = ma_{\mathrm{x}} - mrv_{\mathrm{y}} = m(v\cos\beta)t - mvr\sin\beta = m(\dot{v}\cos\beta - v\sin\beta(\dot{\beta}+r)). \tag{3.1}$$

The lateral motion is represented by equation

$$F_{\mathrm{y}} = F_{\mathrm{y,tr}} + F_{\mathrm{y,rot}} = ma_{\mathrm{y}} + mrv_{\mathrm{x}} = m(v\sin\beta)t + mvr\cos\beta = m(\dot{v}\sin\beta + v\cos\beta(\dot{\beta}+r)). \tag{3.2}$$

The rotational motion is represented by equation

$$M_{\mathrm{z}} = I_{\mathrm{yaw}}\dot{r}. \tag{3.3}$$

Aerodynamics are represented as a dissipate force which reacts against the velocity vector in both, longitudinal and lateral directions as

$$F_{air,x} = \frac{1}{2}c_{air}\rho A(v\cos\beta)^2, \tag{3.4}$$

$$F_{air,y} = \frac{1}{2}c_{air}\rho A(v\sin\beta)^2, \tag{3.5}$$

In a matrix form rigid body dynamics have the next look:

$$\begin{pmatrix}\dot{\beta}\\ \dot{v}\\ \dot{r}\end{pmatrix} = \begin{pmatrix}\frac{1}{mv} & 0 & 0\\ 0 & \frac{1}{m} & 0\\ 0 & 0 & \frac{1}{I_{\mathrm{yaw}}}\end{pmatrix}\begin{pmatrix}-\sin\beta & \cos\beta & 0\\ \cos\beta & \sin\beta & 0\\ 0 & 0 & 1\end{pmatrix}\left\langle\begin{pmatrix}Fx\\ Fy\\ Mz\end{pmatrix} - \begin{pmatrix}F_{air,x}\\ F_{air,y}\\ 0\end{pmatrix}\right\rangle - \begin{pmatrix}r\\ 0\\ 0\end{pmatrix}. \tag{3.6}$$

3.1.4 Steering Angles Projection

The steering angle projection translates forces acting on tires in wheel coordinate frames into forces and rotation momentum acting on a rigid body via transformation matrix as follows

$$\begin{pmatrix}F_{\mathrm{x}}\\ F_{\mathrm{y}}\\ M_{\mathrm{z}}\end{pmatrix} = \begin{pmatrix}\cos\delta_{\mathrm{f}} & -\sin\delta_{\mathrm{f}} & \cos\delta_{\mathrm{r}} & -\sin\delta_{\mathrm{r}}\\ \sin\delta_{\mathrm{f}} & \cos\delta_{\mathrm{f}} & \sin\delta_{\mathrm{r}} & \cos\delta_{\mathrm{r}}\\ l_{\mathrm{f}}\sin\delta_{\mathrm{f}} & l_{\mathrm{f}}\cos\delta_{\mathrm{f}} & -l_{\mathrm{r}}\sin\delta_{\mathrm{r}} & -l_{\mathrm{r}}\cos\delta_{\mathrm{r}}\end{pmatrix}\begin{pmatrix}F_{\mathrm{xf}}\\ F_{\mathrm{yf}}\\ F_{\mathrm{xr}}\\ F_{\mathrm{yr}}\end{pmatrix}. \tag{3.7}$$

3.1.5 Tire Models

The block representation of the tire models can be seen in Figure [3.4]. The block contains two tire models, which calculate via Simplified Pacejka Magic formula raw lateral and longitudinal forces for each wheel, then that forces are scaled in blocks named "Traction Ellipse".

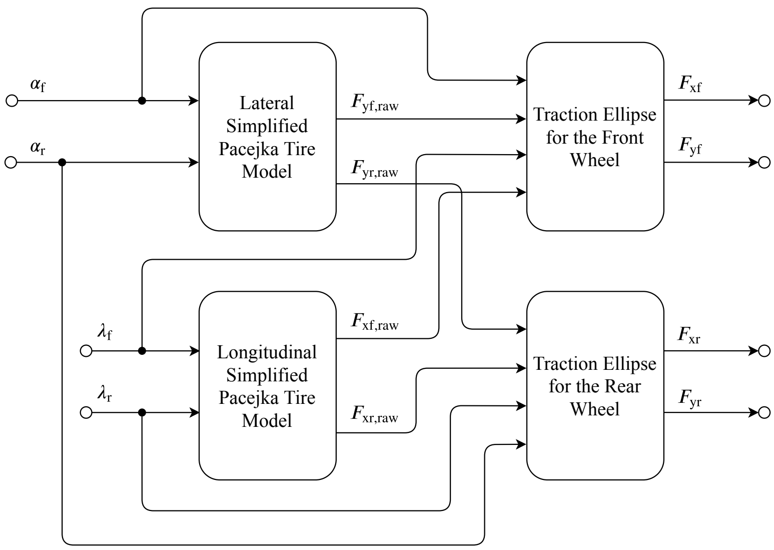


Figure 3.4: The block diagram of the tire models [2]

The tire model is the main part of any nonlinear car model because tires are primary system actuators of a vehicle dynamics. Simplified Pacejka Magic formula [4] is used for mapping a tire generated force on slip variable in both lateral and longitudinal directions defined as

$$F_{\mathrm{x,f,raw}}(\lambda_{\mathrm{f}}) = c_{D,x}F_{\mathrm{z,f}}\sin(c_{C,x}\arctan(c_{B,x}\lambda_{\mathrm{f}} - c_{E,x}(c_{B,x}\lambda_{\mathrm{f}} - \arctan(c_{B,x}\lambda_{\mathrm{f}})))), \tag{3.8}$$

$$F_{\mathrm{x,r,raw}}(\lambda_{\mathrm{r}}) = c_{D,x} F_{\mathrm{z,r}} \sin(c_{C,x} \arctan(c_{B,x}\lambda_{\mathrm{r}} - c_{E,x}(c_{B,x}\lambda_{\mathrm{r}} - \arctan(c_{B,x}\lambda_{\mathrm{r}})))), \quad (3.9)$$

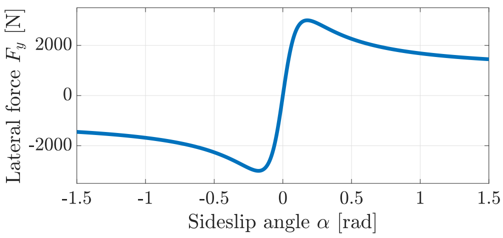
$$F_{\mathrm{y,f,raw}}(\alpha_{\mathrm{f}}) = c_{D,y} F_{\mathrm{z,f}} \sin(c_{C,y} \arctan(c_{B,y}\alpha_{\mathrm{f}} - c_{E,y}(c_{B,y}\alpha_{\mathrm{f}} - \arctan(c_{B,y}\alpha_{\mathrm{f}})))), \quad (3.10)$$

$$F_{\mathrm{y,r,raw}}(\alpha_{\mathrm{r}}) = c_{D,y} F_{\mathrm{z,r}} \sin(c_{C,y} \arctan(c_{B,y}\alpha_{\mathrm{r}} - c_{E,y}(c_{B,y}\alpha_{\mathrm{r}} - \arctan(c_{B,y}\alpha_{\mathrm{r}})))), \quad (3.11)$$

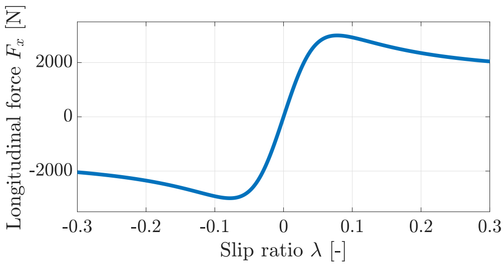
where load forces are constant and are calculated from the car parameters as follows

$$F_{\mathrm{z,f}} = gm\frac{l_{\mathrm{r}}}{l_{\mathrm{f}} + l_{\mathrm{r}}}, \quad F_{\mathrm{z,r}} = gm\frac{l_{\mathrm{f}}}{l_{\mathrm{f}} + l_{\mathrm{r}}}, \quad (3.12)$$

where $g = 9.81\,\mathrm{m\,s^{-2}}$ is a gravity coefficient of the Earth. For longitudinal force is used two sets of coefficients $c_{\mathrm{B,x}}, c_{\mathrm{C,x}}, c_{\mathrm{D,x}}, c_{\mathrm{E,x}}$, first one for throttling and second one for braking.



(a) Lateral force



(b) Longitudinal force

Figure 3.5: Example of lateral and longitudinal forces acting on a tire. [2]

3.1.6 Traction Ellipse

A tire cannot generate combined force (both, lateral and longitudinal) grater than the vertical force F_{z} acting on a wheel by the vehicle. Combined slip occurs when the vehicle is accelerating or braking in a cornering maneuver. That restriction is guaranteed by the traction ellipse (an example of the traction ellipse is showed in figure [3.6])

$$F_{\mathrm{tot}} = \sqrt{\frac{F_{\mathrm{x}}^2}{c_{\mathrm{D,x}}^2} + \frac{F_{\mathrm{y}}^2}{c_{\mathrm{D,y}}^2}} \leq \mu F_{\mathrm{z}}, \quad (3.13)$$

where μ is a friction coefficient of a road; $c_{\mathrm{D,x}}$ and $c_{\mathrm{D,y}}$ are parameters from Pacejka (in general, they are friction coefficients of the road in different directions).

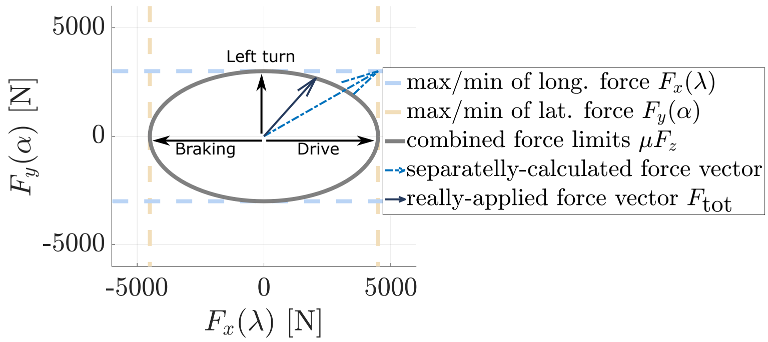


Figure 3.6: Traction ellipse and example of scaling of the force vector. [2]

The implementation of the traction ellipse can be adopted from [2]. The following algorithm (Eq. (3.14) - (3.18)) is applied to scale (if it is needed) the resulted force:

$$\beta^* = \arccos\left(\frac{|\lambda|}{\sqrt{\lambda^2 + \sin^2(\alpha)}}\right), \tag{3.14}$$

$$\mu_{\mathrm{x,act}} = \frac{F_{\mathrm{x,raw}}}{F_{\mathrm{z}}}, \qquad \mu_{\mathrm{y,act}} = \frac{F_{\mathrm{y,raw}}}{F_{\mathrm{z}}}, \tag{3.15}$$

$$\mu_{\mathrm{x,max}} = c_{\mathrm{D,x}}, \qquad \mu_{\mathrm{y,max}} = c_{\mathrm{D,y}}, \tag{3.16}$$

$$\mu_{\mathrm{x}} = \frac{1}{\sqrt{\left(\frac{1}{\mu_{\mathrm{x,act}}}\right)^2 + \left(\frac{\tan(\beta^*)}{\mu_{\mathrm{y,max}}}\right)^2}}, \qquad F_{\mathrm{x}} = \left|\frac{\mu_{\mathrm{x}}}{\mu_{\mathrm{x,act}}}\right| F_{\mathrm{x,raw}}, \tag{3.17}$$

$$\mu_{\mathrm{y}} = \frac{\tan(\beta^*)}{\sqrt{\left(\frac{1}{\mu_{\mathrm{x,max}}}\right)^2 + \left(\frac{\tan(\beta^*)}{\mu_{\mathrm{y,act}}}\right)^2}}, \qquad F_{\mathrm{y}} = \left|\frac{\mu_{\mathrm{y}}}{\mu_{\mathrm{y,act}}}\right| F_{\mathrm{y,raw}}. \tag{3.18}$$

3.1.7 Wheel Kinematics

The wheel kinematics of the single-track model includes calculation of the velocity vector of each wheel and side-slip angle of each tire in the particular wheel coordinate frame. Velocity vectors for the front and rear wheels are calculated as

$$\begin{pmatrix} v_{\mathrm{xf}} \\ v_{\mathrm{yf}} \end{pmatrix} = \begin{pmatrix} \cos\delta_{\mathrm{f}} & \sin\delta_{\mathrm{f}} \\ -\sin\delta_{\mathrm{f}} & \cos\delta_{\mathrm{f}} \end{pmatrix} \begin{pmatrix} v_{\mathrm{x}} \\ v_{\mathrm{y}} + l_{\mathrm{f}} r \end{pmatrix} = \begin{pmatrix} \cos\delta_{\mathrm{f}} & \sin\delta_{\mathrm{f}} \\ -\sin\delta_{\mathrm{f}} & \cos\delta_{\mathrm{f}} \end{pmatrix} \begin{pmatrix} v\cos\beta \\ v\sin\beta + l_{\mathrm{f}} r \end{pmatrix}, \tag{3.19}$$

$$\begin{pmatrix} v_{\mathrm{xr}} \\ v_{\mathrm{yr}} \end{pmatrix} = \begin{pmatrix} \cos\delta_{\mathrm{r}} & \sin\delta_{\mathrm{r}} \\ -\sin\delta_{\mathrm{r}} & \cos\delta_{\mathrm{r}} \end{pmatrix} \begin{pmatrix} v\cos\beta \\ v\sin\beta - l_{\mathrm{r}} r \end{pmatrix}. \tag{3.20}$$

Thus, the side-slip angles of each wheel can be calculated using the definition as

$$\alpha_{\mathrm{f}} = -\arctan\frac{v_{\mathrm{yf}}}{|v_{\mathrm{xf}}|}, \tag{3.21}$$

$$\alpha_{\mathrm{r}} = -\arctan\frac{v_{\mathrm{yr}}}{|v_{\mathrm{xr}}|}. \tag{3.22}$$

3.1.8 Slip Ratios

This block is used to calculate slip ratio per each wheel, which can be calculated using following definition:

$$\lambda_{\mathrm{f}} = \frac{\omega_{\mathrm{f}} p - v_{\mathrm{x,f}}}{|v_{\mathrm{x,f}}|}, \quad \lambda_{\mathrm{r}} = \frac{\omega_{\mathrm{r}} p - v_{\mathrm{x,r}}}{|v_{\mathrm{x,r}}|}. \tag{3.23}$$

4. Nonlinear Twin-Track Model

4.1 Main Model

This chapter describes a Nonlinear Twin-Track vehicle model that was used for fitting car parameters. Actually, it is modified Twin-Track [5]. The notation used in this thesis is the same as in [3].

Vehicle coordinate system used for the description of the Twin-Track model is shown in Figure 4.1. It is the conventional right-hand Cartesian coordinate system. The x axis follows from the center of gravity to the front of the vehicle. The y axis goes towards the left side of the car, and z axis lies from the center of gravity to the top of the vehicle. The vehicle's roll, pitch and yaw angles are measured conventionally (counter-clockwise to axis).

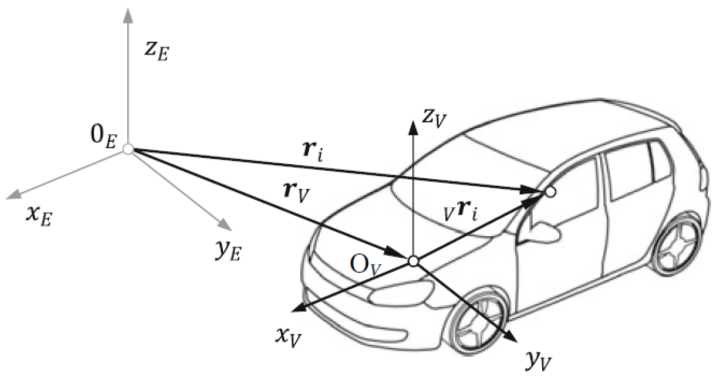


Figure 4.1: Inertial (earth-fixed) and Vehicle (body-fixed) coordinate systems. [3]

The Twin-Track model (Fig. 4.2 and Fig. 4.3) is used to represent motion of a vehicle in 3 dimensional space and consists of 10 states and 8 inputs, which are listed in Table 4.1. All parameters used in the model are presented in Table 4.2. Note that in mathematical description is used symbol Ω, where is $\Omega = \begin{bmatrix} \dot{\phi} \\ \dot{\theta} \\ \dot{\psi} \end{bmatrix}$. The block diagram of the Nonlinear Twin-Track model is shown in Figure 4.4.

Tire forces $F_{\mathrm{R_i}}$, forces from the tires acting on the chassis F_i, ${}^{V}r_{\mathrm{A_i}}$ position vector of the center of gravity of the chassis O_V from the pivot point A_i are showed on Fig. 4.2.

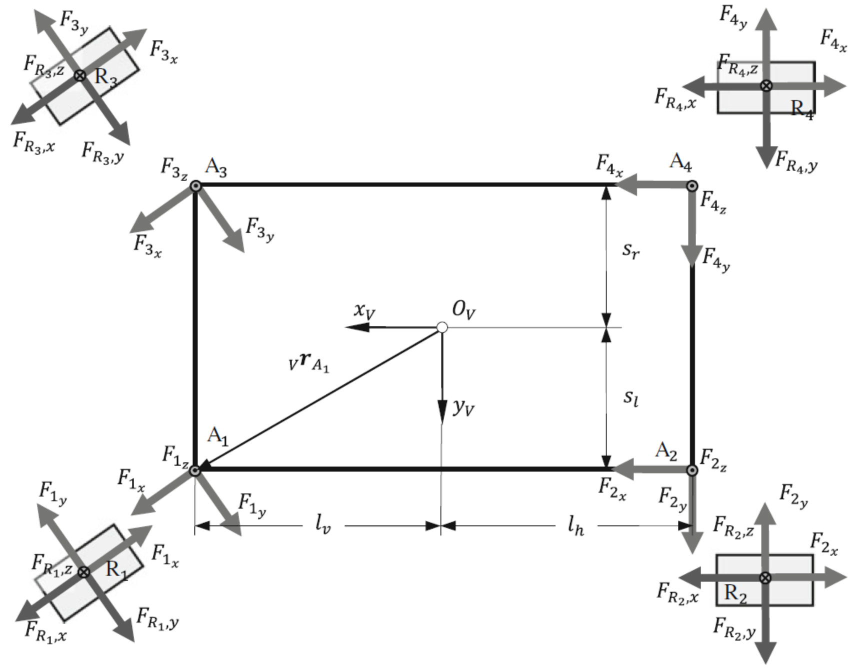


Figure 4.2: Free body diagram of a twin track model - Top view. [3]

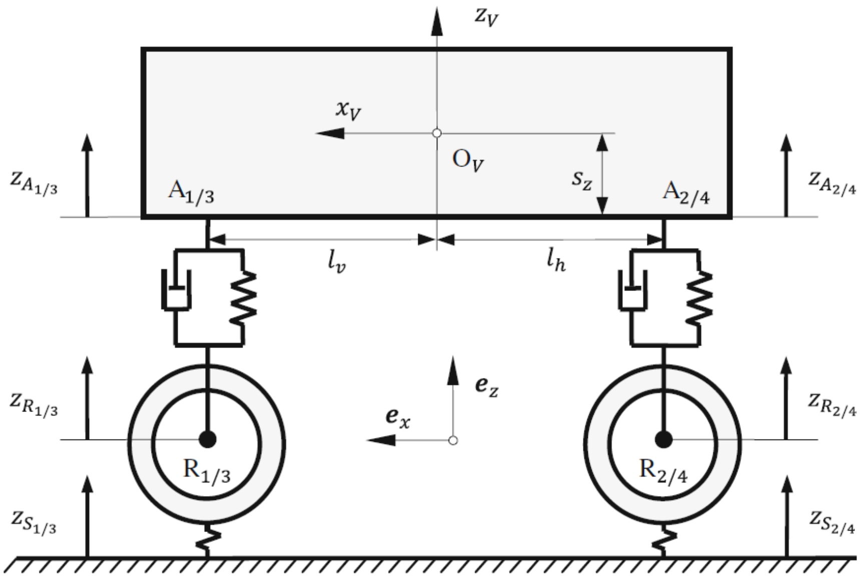


Figure 4.3: Free body diagram of a twin track model - Side view. [3]

State/Input	Symbol	Units
Position of the vehicle body in earth-fixed coordinate system on OZ axis	z	m
Velocity of CG in each dimension	$v_{\mathrm{x,y,z}}$	$\mathrm{m\,s^{-1}}$
Roll rate, pitch rate, yaw rate	$\dot{\phi}, \dot{\theta}, \dot{\psi}$	$\mathrm{rad\,s^{-1}}$
Euler's angles (earth-fixed coordinate system)	ϕ, θ, ψ	rad
Steering angle of each wheel	δ_i	rad
Angular velocity of each wheel	ω_i	$\mathrm{rad\,s^{-1}}$

Table 4.1: States and inputs of the Twin-Track

Parameter	Symbol	Units
Vehicle mass	m	kg
Moment of inertia (matrix 3×3)	I	$\mathrm{kg\,m^{-2}}$
Front/Rear axle-CG distance	$l_{\mathrm{v,h}}$	m
Radius of wheels	p	m
Distance from CG to left/right wheels on OY axis	$s_{\mathrm{l/r}}$	m
Distance from CG to the point where springs are anchored on OZ axis	s_{z}	m
Spring stiffness for front/rear wheels	$c_{a_{\mathrm{f/r}}}$	$\mathrm{N\,m^{-1}}$
Spring compression coefficient for front/rear wheels	$d_{a_{\mathrm{compression,f/r}}}$	$\mathrm{N\,m^{-1}}$
Spring rebound coefficient for front/rear wheels	$d_{a_{\mathrm{rebound,f/r}}}$	$\mathrm{N\,m^{-1}}$
Drag coefficient	c_{air}	-
Air density	ρ	$\mathrm{kg\,m^{-3}}$
Frontal area of the car	A	$\mathrm{m^2}$
Shaping coefficients for lateral dynamics	$c_{\mathrm{D,y}}$	-
	$c_{\mathrm{B,y}}$	-
	$c_{\mathrm{C,y}}$	-
	$c_{\mathrm{E,y}}$	-
Shaping coefficients for longitudinal dynamics for acceleration	$c_{\mathrm{D,x,t}}$	-
	$c_{\mathrm{B,x,t}}$	-
	$c_{\mathrm{C,x,t}}$	-
	$c_{\mathrm{E,x,t}}$	-
Shaping coefficients for longitudinal dynamics for breaking	$c_{\mathrm{D,x,b}}$	-
	$c_{\mathrm{B,x,b}}$	-
	$c_{\mathrm{C,x,b}}$	-
	$c_{\mathrm{E,x,b}}$	-

Table 4.2: Parameters of the Twin-Track

4.2 Scheme

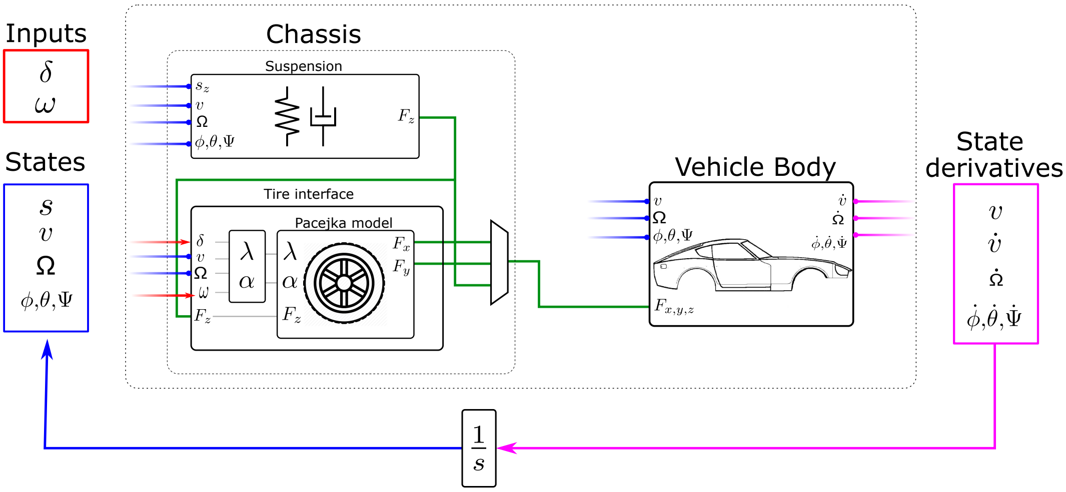


Figure 4.4: The block diagram of the Twin-Track model [6]

The model is separated into 2 main parts: Chassis and Vehicle Body.Chassis represent tires and suspension, the suspension then supports the Vehicle Body (sprung mass).

4.3 Vehicle Body

The vehicle body can be described by next matrix form

$$m\left(\begin{bmatrix}\dot{v}_{\mathrm{x}}\\ \dot{v}_{\mathrm{y}}\\ \dot{v}_{\mathrm{z}}\end{bmatrix}+\Omega\times\begin{bmatrix}v_{\mathrm{x}}\\ v_{\mathrm{y}}\\ v_{\mathrm{z}}\end{bmatrix}\right)=\sum_{i=1}^{4}\begin{bmatrix}\mathbf{F}_{i,\mathrm{x}}\\ \mathbf{F}_{i,\mathrm{y}}\\ \mathbf{F}_{i,\mathrm{z}}\end{bmatrix}-\mathbf{F}_{\mathrm{air}}\begin{bmatrix}v_{\mathrm{x}}\\ v_{\mathrm{y}}\\ 0\end{bmatrix}+{}^{V}\,\mathbf{T}_{E}\begin{bmatrix}0\\ 0\\ -mg\end{bmatrix}. \tag{4.1}$$

Forces $\mathbf{F}_i$ are in body-fixed coordinates, the matrix ${}^{V}\mathbf{T}_{E}$ transforms the earth-fixed gravitational acceleration to vehicle-fixed coordinates. The rest of the variables are in vehicle-fixed coordinate frame. $\mathbf{F}_{\mathrm{air}}=\frac{1}{2}c_{\mathrm{air}}\rho A\sqrt{v_{\mathrm{x}}^2+v_{\mathrm{y}}^2}\begin{bmatrix}v_{\mathrm{x}}\\ v_{\mathrm{y}}\\ 0\end{bmatrix}$

simulate resistance of air where c_{air} is drag coefficient, air density ρ and A is a frontal area of the car.

The rotational dynamics is described by:

$$I\dot{\Omega}+\Omega\times(I\Omega)=\sum_{i=1}^{4}r_i\times\mathbf{F}_i+r_{\mathrm{w}}\times\mathbf{F}_{\mathrm{w}}, \tag{4.2}$$

where I represent matrix of inertia, F_{w} are aerodynamic forces acting at the center of aerodynamics pressure r_{w}, the vector is w.r.t. center of gravity in vehicle coordinates . The vector r_i is the point of application of the force $\mathbf{F}_i$ from wheels and suspension, its values are determined from the dimensions of the body, e.g. for the first, front-left wheel, the value would be $r_i = \begin{bmatrix} l_\mathrm{f} \\ s_\mathrm{l} \\ -s_\mathrm{z} \end{bmatrix}$.

4.4 Chassis

4.4.1 Suspension

The suspension is modeled as spring-damper systems acting on each wheel individually.

Spring

Spring force acting on i^{th} wheel is defined as follows:

$$
{}^{V}\mathbf{F}_{Fi} = -(c_{\mathrm{a},i}\Delta l_{F_i})^{V}\mathbf{T}_E \begin{bmatrix} 0 \\ 0 \\ 1 \end{bmatrix}, \forall i \in \{1, 2, 3, 4\}, \tag{4.3}
$$

where $c_{\mathrm{a},i}$ is the stiffness coefficient of spring i. Δl_{F_i} is the compression of spring i, ${}^{V}\mathbf{T}_E$ is a rotation matrix, transforming Inertial coordinates to Vehicle coordinates. The multiplication by $\begin{bmatrix} 0 \\ 0 \\ 1 \end{bmatrix}$ means that the force acts only along the (inertial) z_E-axis (the spring is assumed to always point upwards with respect to the inertial coordinates).

Damper

Damping force acting on i^{th} wheel is defined as follows

$$
{}^{V}\mathbf{F}_{Di} = -(d_{\mathrm{a},i}\Delta \dot{l}_{F_i})^{V}\mathbf{T}_E \begin{bmatrix} 0 \\ 0 \\ 1 \end{bmatrix}, \forall i \in \{1, 2, 3, 4\}, \tag{4.4}
$$

where $d_{\mathrm{a},i}$ is the rebound coefficient of spring i for $\Delta l_{F_i} \geq 0$ and is the damping coefficient of spring i for $\Delta l_{F_i} < 0$.

4.4.2 Tire Interface

The model comes with the Simplified Pacejka tire model with constant coefficients and uses slip ratio λ and slip angle α as their inputs.

Slip variables

Longitudinal (circumferential) slip:

$$
\lambda_i = \frac{\dot{x}_{R_i} - p\omega_{R_i}}{|\dot{x}_{R_i}|}. \tag{4.5}
$$

Slip angle:

$$\alpha_i = -\arctan(\frac{\dot{y}_{R_i}}{|\dot{x}_{R_i}|}), \tag{4.6}$$

where $\dot{x}_{R_i}$ is velocity of the wheel center point along x in the wheel-fixed coordinate system, $\dot{y}_{R_i}$ is velocity of the wheel center point along y in the wheel-fixed coordinate system, p is wheel radius, ω_{R_i} Angular velocity of wheel i

Wheel center point velocity

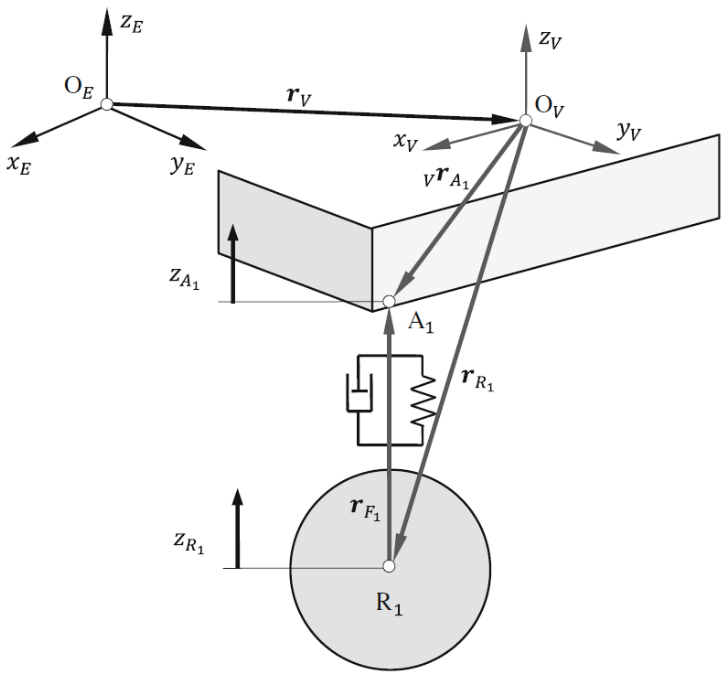


Figure 4.5: Clarification on the meaning of wheel center point vector r_{R_i} and the pivot point A_i . [3]

The velocities of the wheel center point are obtained as follows:

$${}^V\mathbf{r}_{R_i} = {}^V\mathbf{r}_{A_i} + {}^V\mathbf{T}_E \begin{bmatrix} 0 \\ 0 \\ -l_{F_i} \end{bmatrix}, \forall i \in \{1, 2, 3, 4\}, \tag{4.7}$$

where ${}^V\mathbf{r}_{R_i}$ is a position of the wheel center point with respect to vehicle coordinates, ${}^V\mathbf{r}_{A_i}$ is a position of the spring anchor with respect to the vehicle coordinates, l_{F_i} is length of the spring.

The point ${}^V\mathbf{r}_{A_i}$ is where the spring is anchored to the vehicle chassis and where the tire forces are applied to the chassis.

$$
{}^{V}\mathbf{v}_{R_i} = \begin{bmatrix} {}^{V}\dot{x}_{R_i} \\ {}^{V}\dot{y}_{R_i} \\ {}^{V}\dot{z}_{R_i} \end{bmatrix} = {}^{V}\mathbf{v}_V + {}^{V}\omega_V \times {}^{V}\mathbf{r}_{R_i} + {}^{V}\mathbf{T}_E \begin{bmatrix} 0 \\ 0 \\ -\dot{l}_{F_i} \end{bmatrix}, \forall i \in \{1,2,3,4\}, \tag{4.8}
$$

where ${}^{V}\mathbf{v}_{R_i}$ is wheel velocity with respect to vehicle coordinates, ${}^{V}\mathbf{v}_V$ is vehicle velocity with respect to vehicle coordinates, ${}^{V}\omega_V$ is vehicle angular velocity with respect to vehicle coordinates. Note: Vehicle = Chassis.

$$
{}^{R_i}\mathbf{v}_{R_i} = \begin{bmatrix} {}^{R_i}\dot{x}_{R_i} \\ {}^{R_i}\dot{y}_{R_i} \\ {}^{R_i}\dot{z}_{R_i} \end{bmatrix} = {}^{R_i}\mathbf{T}_V \; {}^{V}\mathbf{v}_{R_i}, \tag{4.9}
$$

where ${}^{R_i}\mathbf{v}_{R_i}$ wheel center point velocity with respect to wheel-fixed coordinates, ${}^{R_i}\mathbf{T}_V$ rotational matrix from vehicle coordinates to wheel-fixed coordinates.

Simplified Pacejka's Magic Formula

This model uses Pacejka's magic formula for calculating generated forces on tire:

$$
F(k) = c_D F_{\mathrm{z}} \sin(c_{\mathrm{C}} \arctan(c_{\mathrm{B}} k - c_{\mathrm{E}}(c_{\mathrm{B}} k - \arctan(c_{\mathrm{B}} k)))), \tag{4.10}
$$

where k is either sideslip angle α or longitudinal slip λ. F is either F_{y} or F_{x}, depending on the input argument.

Coefficients $c_{\mathrm{B}}, c_{\mathrm{C}}, c_{\mathrm{D}}, c_{\mathrm{E}}$, in this model, are constant for given F. So for calculating F_{y} and F_{x}, will be needed two sets of these parameters.

Dependency between forces F_{x} and F_{y} is expressed with traction ellipse 4.6:

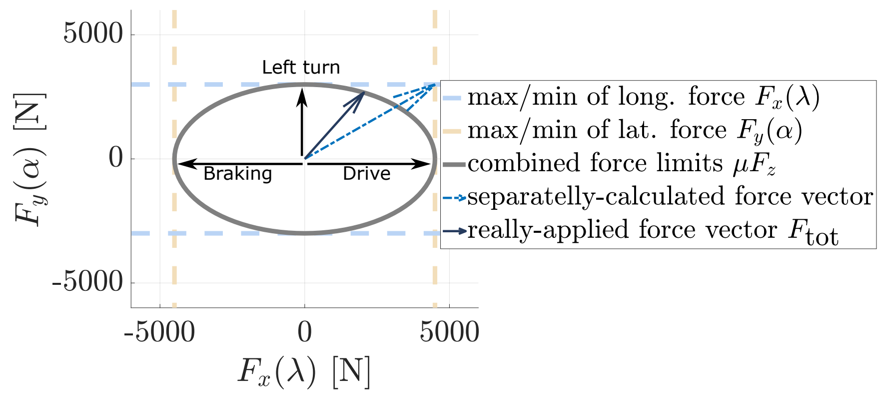


Figure 4.6: Traction ellipse for constant $c_{\mathrm{B}}, c_{\mathrm{C}}, c_{\mathrm{D}}, c_{\mathrm{E}}$.
[2]

And following algorithm (Eq. (4.11) - (4.18)) can be used to capture this dependency. Let's call the forces calculated from 4.10 $F_{\mathrm{x,max}}$ and $F_{\mathrm{y,max}}$ as in Figure 4.6.

$$\alpha^* = \mathrm{s}(\alpha), \tag{4.11}$$

$$\beta = \arccos\left(\frac{|\lambda|}{\sqrt{\lambda^2 + {\alpha^*}^2}}\right), \tag{4.12}$$

$$\mu_{\mathrm{x,act}} = \frac{F_{\mathrm{x,max}}}{F_{\mathrm{z}}}, \qquad \mu_{\mathrm{y,act}} = \frac{F_{\mathrm{y,max}}}{F_{\mathrm{z}}}, \tag{4.13}$$

$$\mu_{\mathrm{x,max}} = \frac{D_{\mathrm{x}}}{F_{\mathrm{z}}}, \qquad \mu_{\mathrm{y,max}} = \frac{D_{\mathrm{y}}}{F_{\mathrm{z}}}, \tag{4.14}$$

$$\mu_{\mathrm{x}} = \frac{1}{\sqrt{(\frac{1}{\mu_{\mathrm{x,act}}})^2 + (\frac{\tan(\beta)}{\mu_{\mathrm{y,max}}})^2}}, \tag{4.15}$$

$$\mu_{\mathrm{y}} = \frac{\tan(\beta)}{\sqrt{(\frac{1}{\mu_{\mathrm{x,max}}})^2 + (\frac{\tan(\beta)}{\mu_{\mathrm{y,act}}})^2}}, \tag{4.16}$$

$$F_{\mathrm{x}} = \frac{\mu_{\mathrm{x}}}{\mu_{\mathrm{x,act}}} F_{\mathrm{x,max}}, \tag{4.17}$$

$$F_{\mathrm{y}} = \frac{\mu_{\mathrm{y}}}{\mu_{\mathrm{y,act}}} F_{\mathrm{y,max}}. \tag{4.18}$$

Forces F_{x} and F_{y} are now respecting the traction ellipse from 4.6. The force F_{z} is load on the tire, the resultant force from the spring-damper system. Note that the shape of the ellipse is determined by $F_{\mathrm{x,max}}$ and $F_{\mathrm{y,max}}$.

4.5 Coordinate Transformations

In this section is described rotation matrices which are used for transformation between various coordinate systems (c() and s() is used instead of cos() and sin()).

Wheel-fixed to body-fixed:

$$^{V}\mathbf{T}_{R_i} = \begin{bmatrix} \mathrm{c}(\delta_i)\mathrm{c}(\theta) & -\mathrm{s}(\delta_i)\mathrm{c}(\theta) & -\mathrm{s}(\theta) \\ \mathrm{s}(\phi)\mathrm{s}(\theta)\mathrm{c}(\delta_i) + \mathrm{c}(\phi)\mathrm{s}(\delta_i) & -\mathrm{s}(\phi)\mathrm{s}(\theta)\mathrm{s}(\delta_i) + \mathrm{c}(\phi)\mathrm{c}(\delta_i) & \mathrm{s}(\phi)\mathrm{c}(\theta) \\ \mathrm{c}(\phi)\mathrm{s}(\theta)\mathrm{c}(\delta_i) - \mathrm{s}(\phi)\mathrm{s}(\delta_i) & -\mathrm{c}(\phi)\mathrm{s}(\theta)\mathrm{s}(\delta_i) - \mathrm{s}(\phi)\mathrm{c}(\delta_i) & \mathrm{c}(\phi)\mathrm{c}(\theta) \end{bmatrix}, \tag{4.19}$$

where δ_i is the steering angle of wheel i, ϕ and θ are Euler angles (roll and pitch).

Body-fixed to inertial:

$$^{E}\mathbf{T}_{V} = \begin{bmatrix} \mathrm{c}(\theta)\mathrm{c}(\psi) & \mathrm{s}(\phi)\mathrm{s}(\theta)\mathrm{c}(\psi) - \mathrm{c}(\phi)\mathrm{s}(\psi) & \mathrm{c}(\phi)\mathrm{s}(\theta)\mathrm{c}(\psi) + \mathrm{s}(\phi)\mathrm{s}(\psi) \\ \mathrm{c}(\theta)\mathrm{s}(\psi) & \mathrm{s}(\phi)\mathrm{s}(\theta)\mathrm{s}(\psi) + \mathrm{c}(\phi)\mathrm{c}(\psi) & \mathrm{c}(\phi)\mathrm{s}(\theta)\mathrm{s}(\psi) - \mathrm{s}(\phi)\mathrm{c}(\psi) \\ -\mathrm{s}(\theta) & \mathrm{s}(\phi)\mathrm{c}(\theta) & \mathrm{c}(\phi)\mathrm{c}(\theta) \end{bmatrix}, \tag{4.20}$$

where ϕ, θ and ψ are Euler angles. For more clarification on how these matrices were derived, see [3].

5. Model Identification

5.1 LFS Signals Preview

Thesis [7] describes methods for reading signals from ”Life for Speed” simulator and information from files. This methods allow reading of the following signals, which are important for us, from the LFS in online mode for each wheel:

Name of signal	Description
XForce	tire generated force in lateral direction $F_{\mathrm{y,raw}}$.
YForce	tire generated force in longitudinal direction $F_{\mathrm{x,raw}}$.
VerticalLoad	load force F_z.
AngVel	angular velocity of a wheel ω in radians per second
Steer	steering of a wheel in radians
SlipRatio	slip ratio of a wheel λ.
TanSlipAngle	tangent of wheel's sideslip angle α.

Wheels are marked in the next way: W1 - rear left, W2 - rear right, W3 - front left, W4 - front right.

Car_info.mat and car_info.txt files provide the following important information about chosen vehicle:

- **forceForward** represents a weight distribution on the front part of the vehicle;
- **Length of a wheelbase** l;
- **Total mass of the vehicle** m;
- **Moment of inertia of the rigid body** I as matrix 3×3;
- **Rim's radius** r for each wheel in metres;
- **Width of a tyre** w for each wheel in metres;
- **Sidewall height proportion** represents a ratio between tyre's height and tyre's width.
- **Aerodynamic drag coefficient** c_{air}

5.2 Single-Track Body Identification

The identification of the Single-Track rigid body parameters is determined as parsing of the values listed in the table 3.2. Car_info.mat can provide the following information: total mass of the vehicle m, aerodynamics drag coefficient c_{air} and yaw moment of inertia I_{yaw} as third column and a third-row element of matrix I, air density is chosen as a conventional constant $\rho = 0.015\ \mathrm{kg\,m^{-3}}$. Frontal area of the car A cannot be calculated within available information and empirically chosen as a constant $A = 3\ \mathrm{m}^2$.

Front axle-CG distance l_v can be calculated using the following formula:

$$l_\mathrm{v} = l \cdot \mathrm{forceForward}. \tag{5.1}$$

Rear axle-CG distance l_h is calculated as:

$$l_\mathrm{h} = l \cdot (1 - \mathrm{forceForward}). \tag{5.2}$$

Formula of wheel's radius p have the following look:

$$p = r + w \cdot \text{Sidewall height proportion}. \tag{5.3}$$

Actual radius of wheel's can vary because of F_z. It can be calculated using equation 5.4 in situation when the car is moving with constant velocity v_x and does not perform any lateral motion:

$$p = \frac{\lambda \cdot |v_\mathrm{x}| + v_\mathrm{x}}{\omega}. \tag{5.4}$$

5.3 Twin-Track Body Identification

The identification of the Twin-Track rigid body parameters is determined as parsing of the parameters listed in the table 4.2. Car_info.mat can help with identification of Twin-Track body by providing the following important information: total mass of the vehicle m, aerodynamic drag coefficient c_air and moment of inertia I. Car_info.txt can provide the following parameters: distance from the CG to left/right wheels on OY axis $s_\mathrm{l/r}$, spring stiffness for front/rear wheels $c_{a_\mathrm{f/r}}$, spring compression coefficient for front/rear wheels $d_{a_\mathrm{compression,f/r}}$, spring rebound coefficient for front/rear wheels $d_{a_\mathrm{rebound,f/r}}$. Air density is chosen again as a constant $\rho = 0.015\ \mathrm{kg\,m^{-3}}$. Frontal area of the car A and distance from center of gravity to the point where springs are anchored on OZ axis s_z, cannot be calculated within available information, so they are empirically chosen as a constant: $A = 3\ \mathrm{m}^2$ and $s_\mathrm{z} = 0.5$ m. Front axle-CG distance, rear axle-CG distance and wheel's radius are calculated in the same way as for the Single-Track (equations 5.1, 5.2, and 5.4).

5.4 Tire's Parameters Identification

The identification of shaping coefficients is done for general tire function, not for each tire. That and the fact that Single-Track and Twin-Track model use the same Pacejka Magic formula mean that shaping coefficients identification is the same for both Single-Track and Twin-Track.

Shaping coefficients can be calculated by using of nonlinear regression with Huber weight [8] function

$$\text{weight} = \frac{1}{\max(1, |\text{residual}|)}, \tag{5.5}$$

for robust fitting of unknowns parameters c_D, c_C, c_B, c_E in equations 3.8, 3.9, and 4.10.
Lateral dynamics uses equation 3.10 for forward wheels and 3.11 for rear wheels with signals XForce as F_y , arctan(TanSlipAngle) as α, W4_VerticalLoad + W3_VerticalLoad as $F_\mathrm{z,f}$, W1_VerticalLoad + W2_VerticalLoad as $F_\mathrm{z,r}$ in a same time .

Longitudinal dynamics is described by a next equation 3.8 for forward wheels or 3.9 for rear for both situation: braking and throttling. Signals used in equations are YForce as F_x , SlipRatio as λ, W4_VerticalLoad + W3_VerticalLoad as $F_\mathrm{z,f}$, W1_VerticalLoad + W2_VerticalLoad as $F_\mathrm{z,r}$ in a same time .

The Nonlinear regression needs initial coefficient values which could be calculated within nonlinear curve-fitting. Initial values for nonlinear curve-fitting is represented in the table 5.1

Table 5.1: Pacejka's magic formula coefficients

Parameter	Symbol	Limit	Initial guess
Shaping coefficients	c_D	0..2	1
	c_B	4..30	8
	c_C	1..2	1.5
	c_E	−30..1	−4.5

To collect the data for lateral dynamics parameter identification, the following maneuver is used: maximal turn of the steering wheel to the left and then to the right at maximum possible velocity for first gear. Fig. 5.1 shows inputs for this experiment.

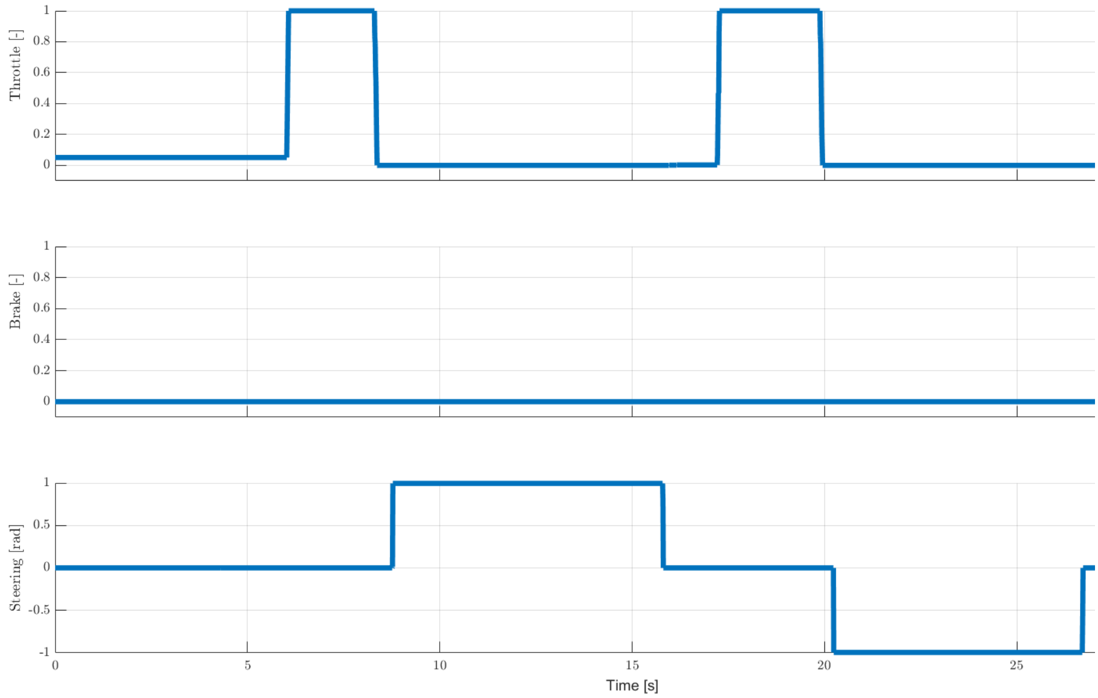


Figure 5.1: Inputs for data generating for lateral coefficients identification

Described experiment allows data generation of all possible range for slip angles (Fig. 5.2)

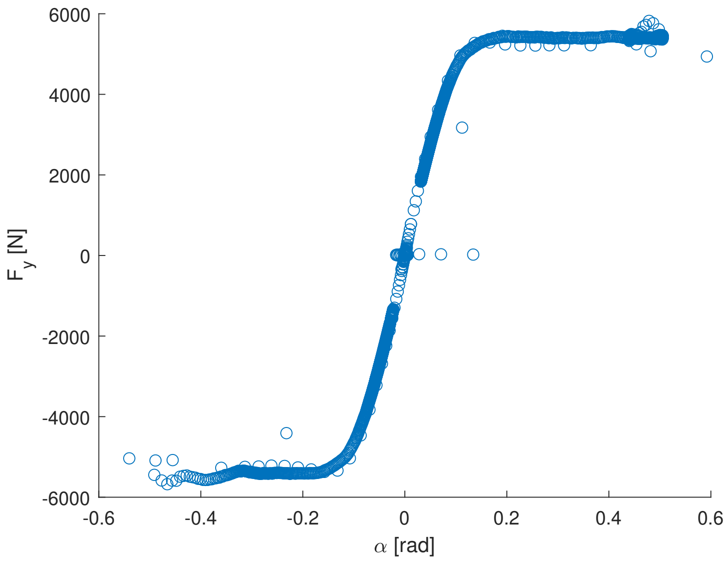


Figure 5.2: Measured data for lateral dynamics tire parameter identification

Data acquisition experiment for longitudinal dynamics includes sharp start on the first gear and sharp braking at the maximal speed for the first gear, it's done to generate the data for all possible range for slip ratios. The inputs for that experiment are shown in Fig. 5.3

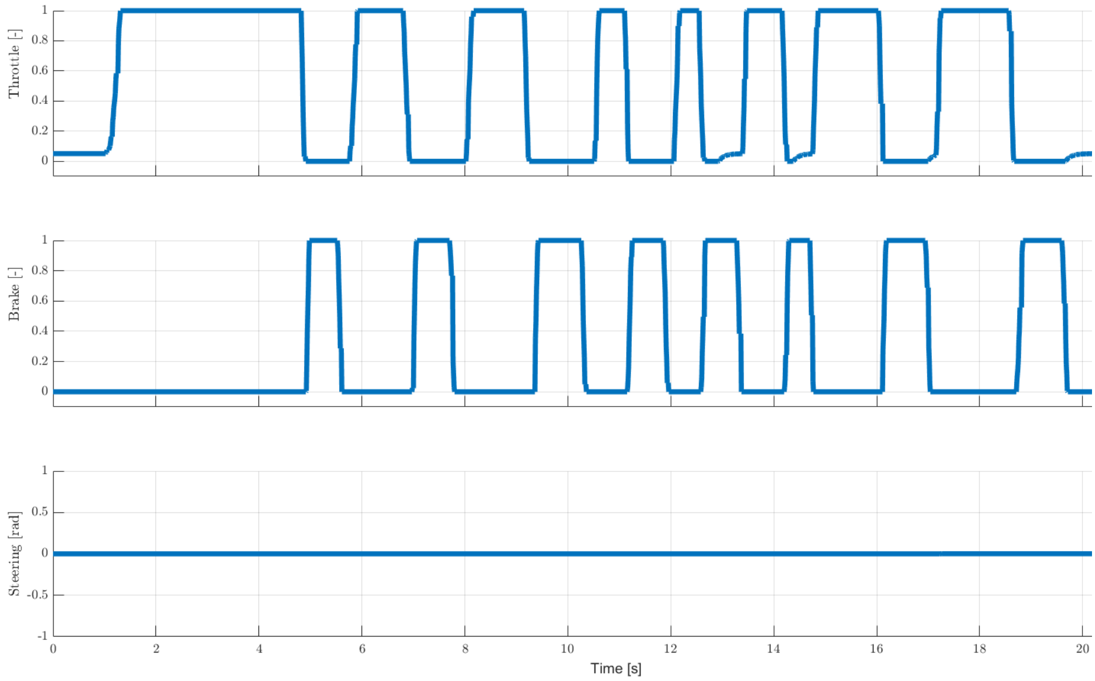
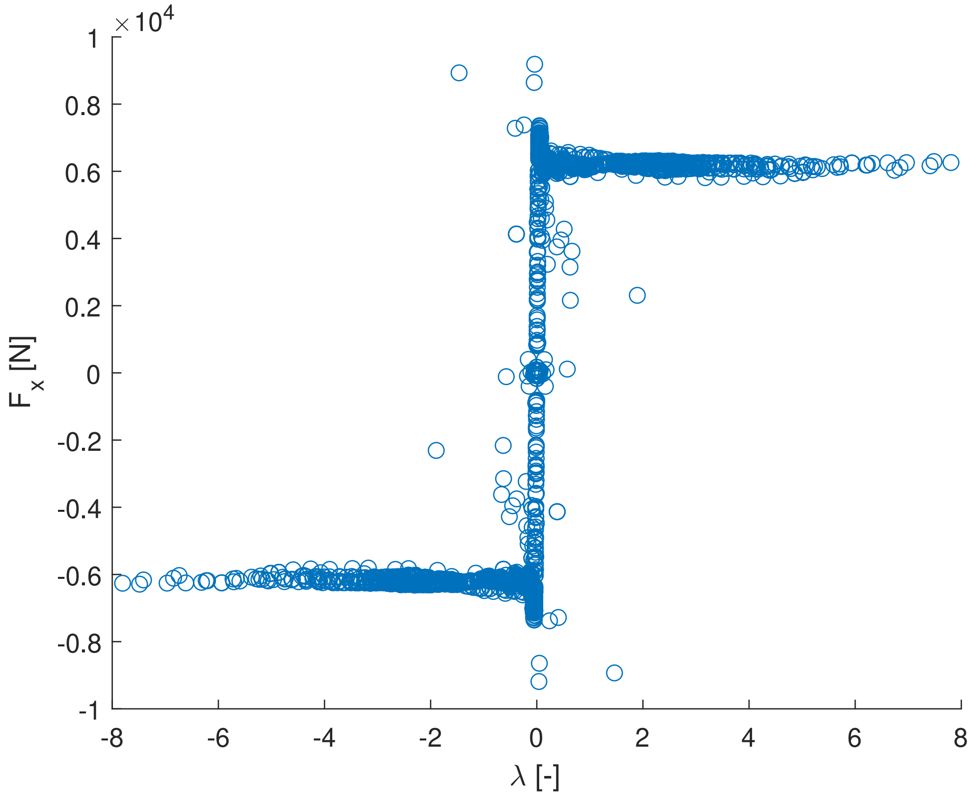
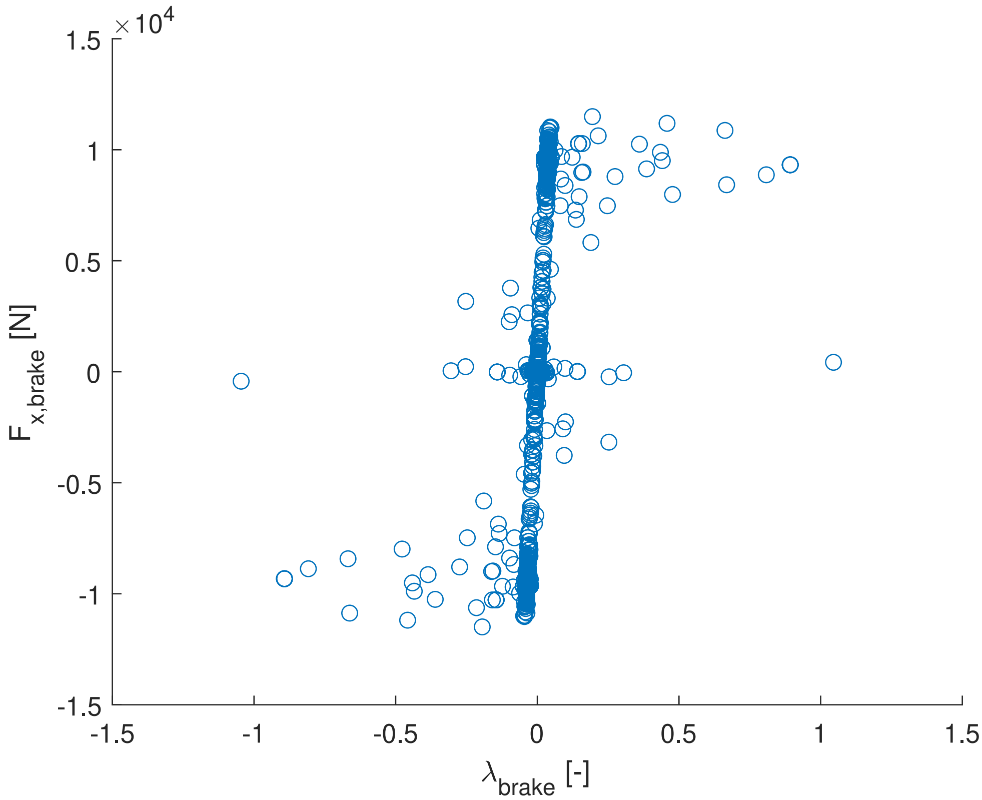


Figure 5.3: Inputs for data generating for longitudinal coefficients identification

Collected data are shown in Fig. 5.4:
Notice that the slip ratio has a saturation limit in -1. That means that for calculation of coefficients for braking data where slip ratio > -1 is used. The fitting is used to estimate shaping coefficients for nonlinear curves, which describe the dependency of the generated lateral force on tire's slip angle or the longitudinal force on tire's slip ratio. The fitted curves are shown in figures [5.5] , [5.6], and [5.7]. Fitting scripts allow manual changes of Pacejka's parameters and observing how it impact dependency.



(a) acceleration case



(b) braking case

Figure 5.4: Measured data for longitudinal dynamics tire parameter identification.

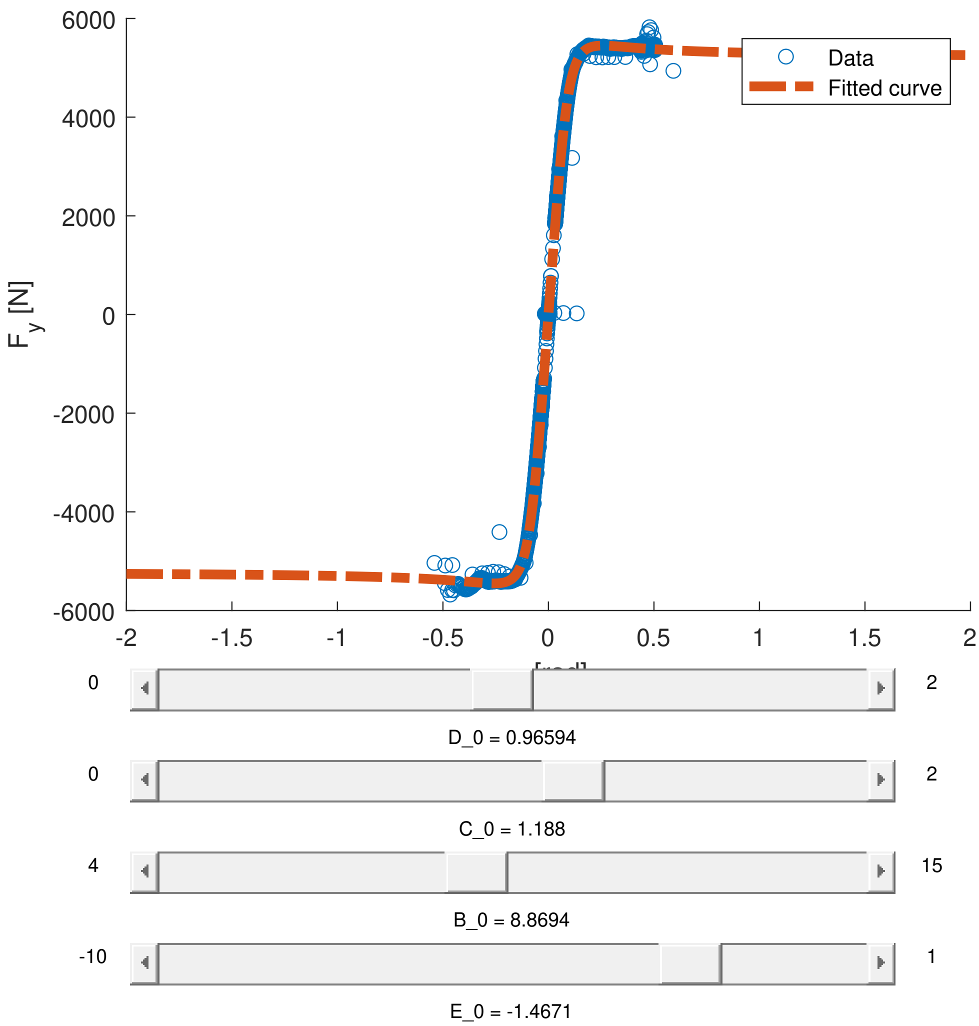


Figure 5.5: Curve fitting for the generated lateral force F_y

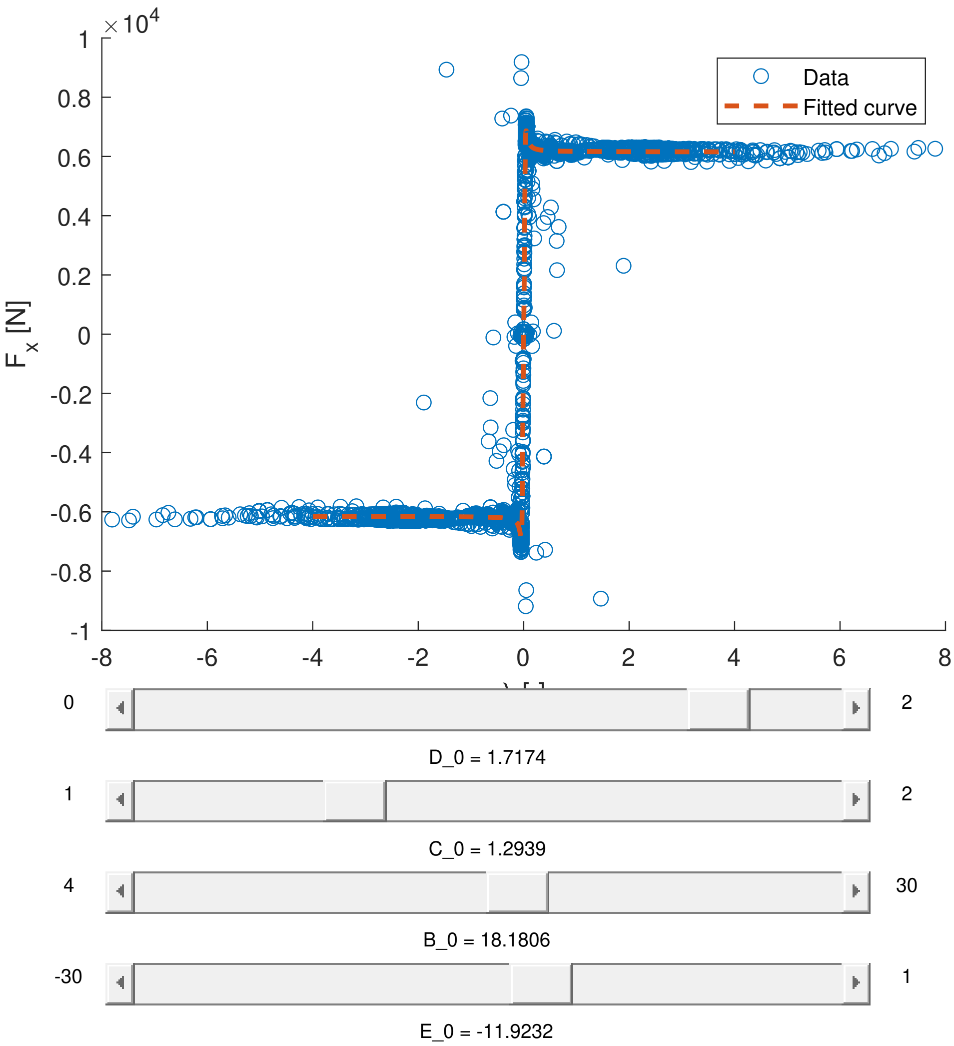


Figure 5.6: Dependence of the generated lateral force F_x on tire slip ratio λ for throttling

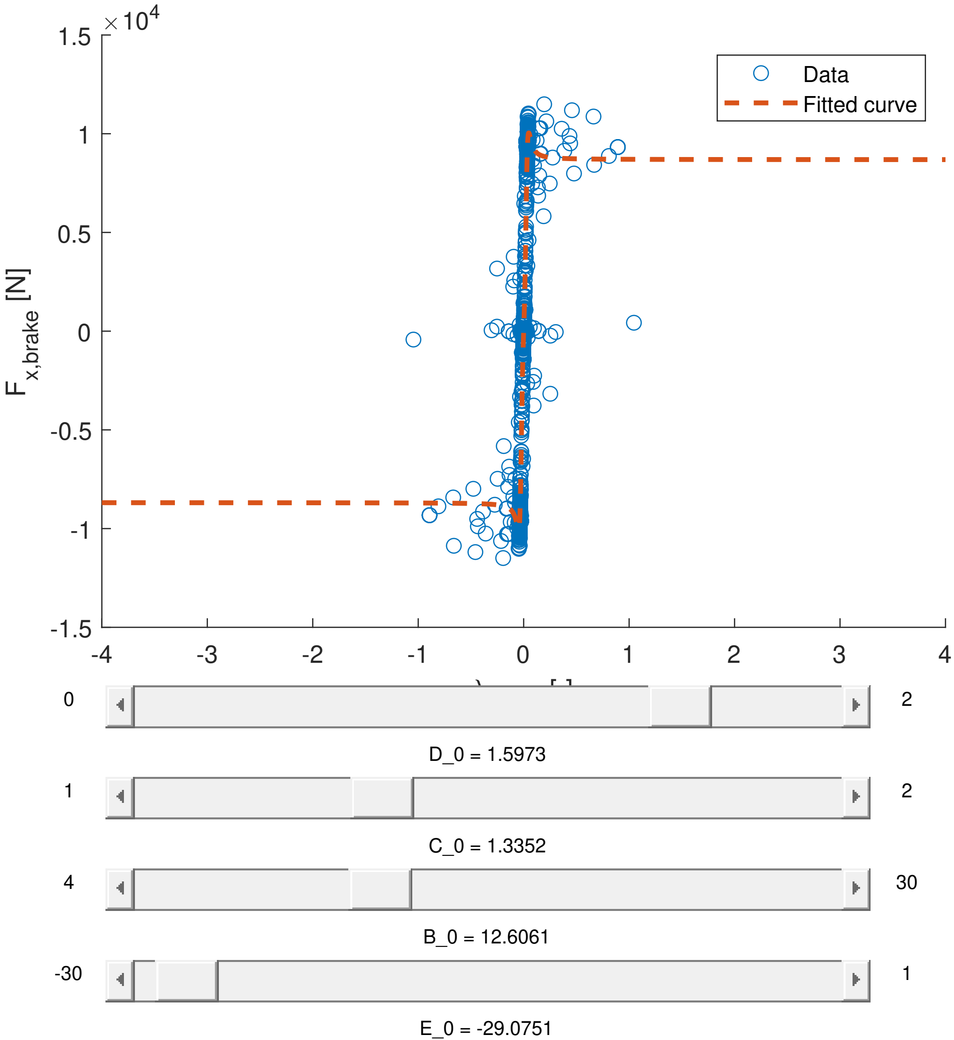


Figure 5.7: Dependence of the generated lateral force F_x on tire slip ratio λ for braking

6. Validation Ride Tests

Resulting identified model were tested on 3 cars:

- **GTI**, car with front drive, this car was used to develop identification method.
- **UF1**, car with front drive.
- **XRG**, car with rear drive.

6.1 Single-Track

Figures 6.1, 6.2, 6.3 demonstrate that Single-Track well follows LFS in the following signals: velocity v, yaw rate $\dot{\psi}$. Slip ratios λ and slip angles α are followed as well, but have some differences, because wheel radius in Single-Track model is constant, but in LFS tire are dynamically changed. This mean that parameters for longitudinal and lateral dynamics are calculated with sufficient accuracy.

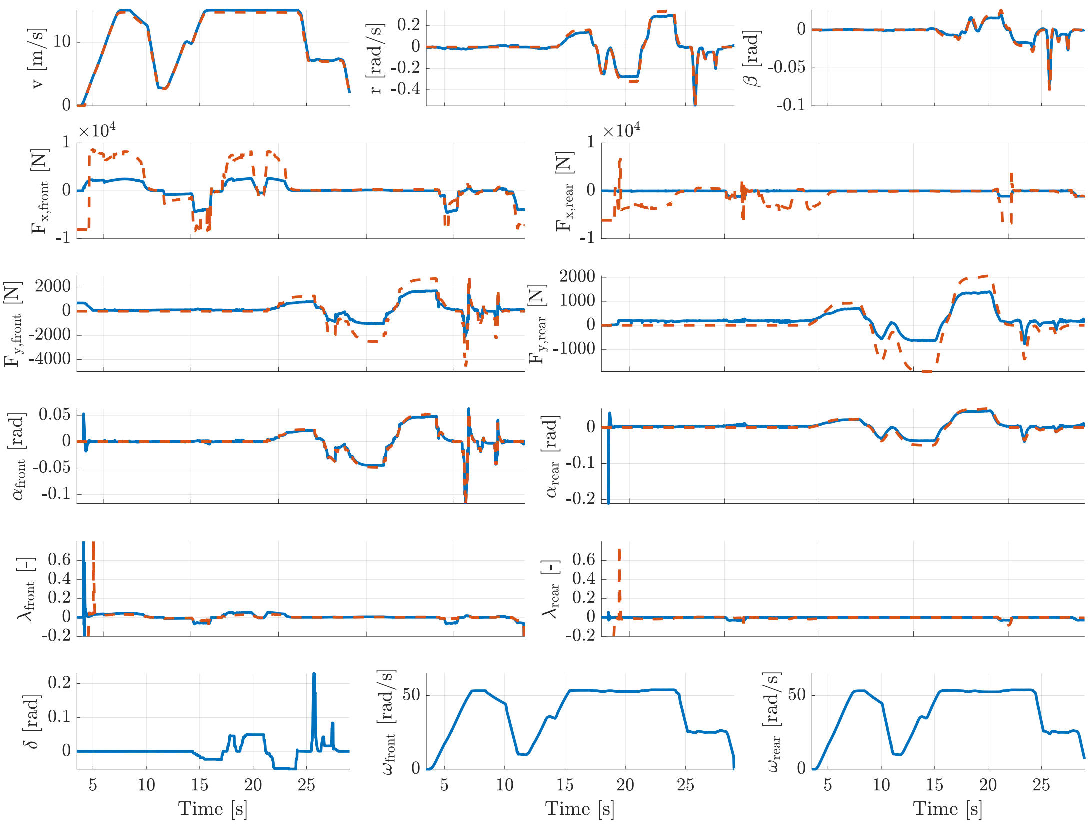


Figure 6.1: Comparison between LFS and the Single-Track model for GTI (blue line it is LFS, red line is the Single-Track)

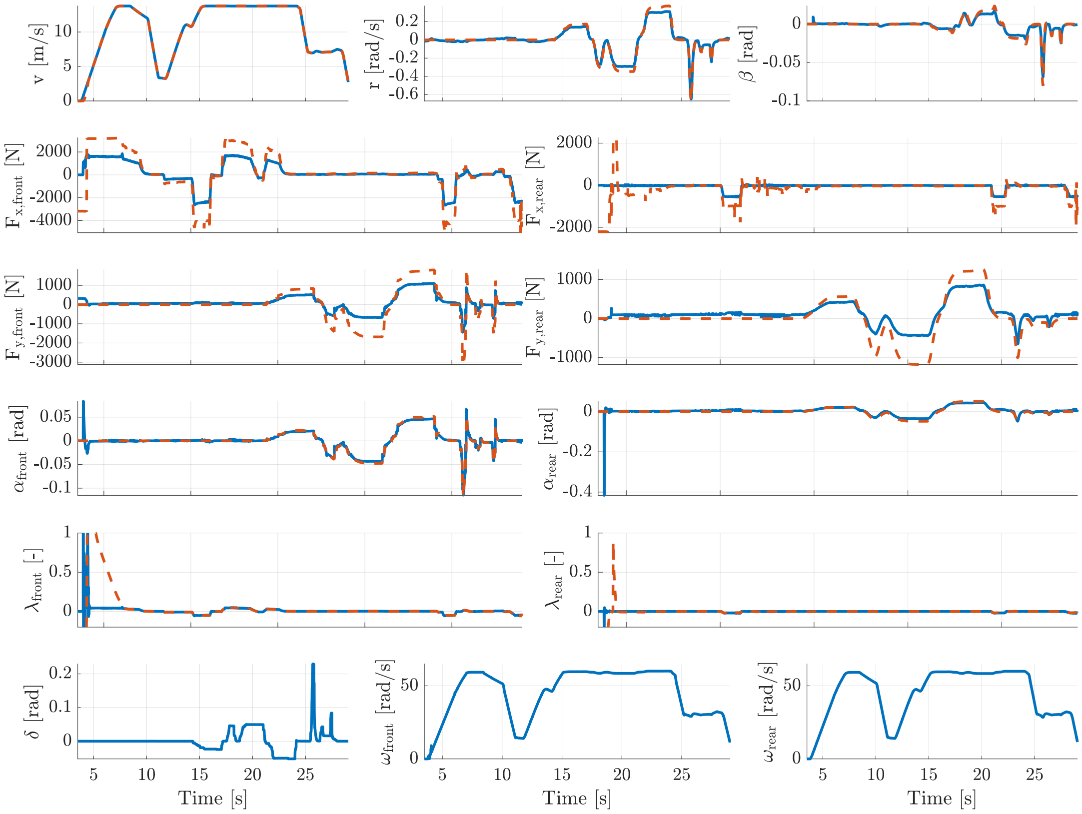


Figure 6.2: Comparison between LFS and the Single-Track model for UF1 (blue line it is LFS, red line is the Single-Track)

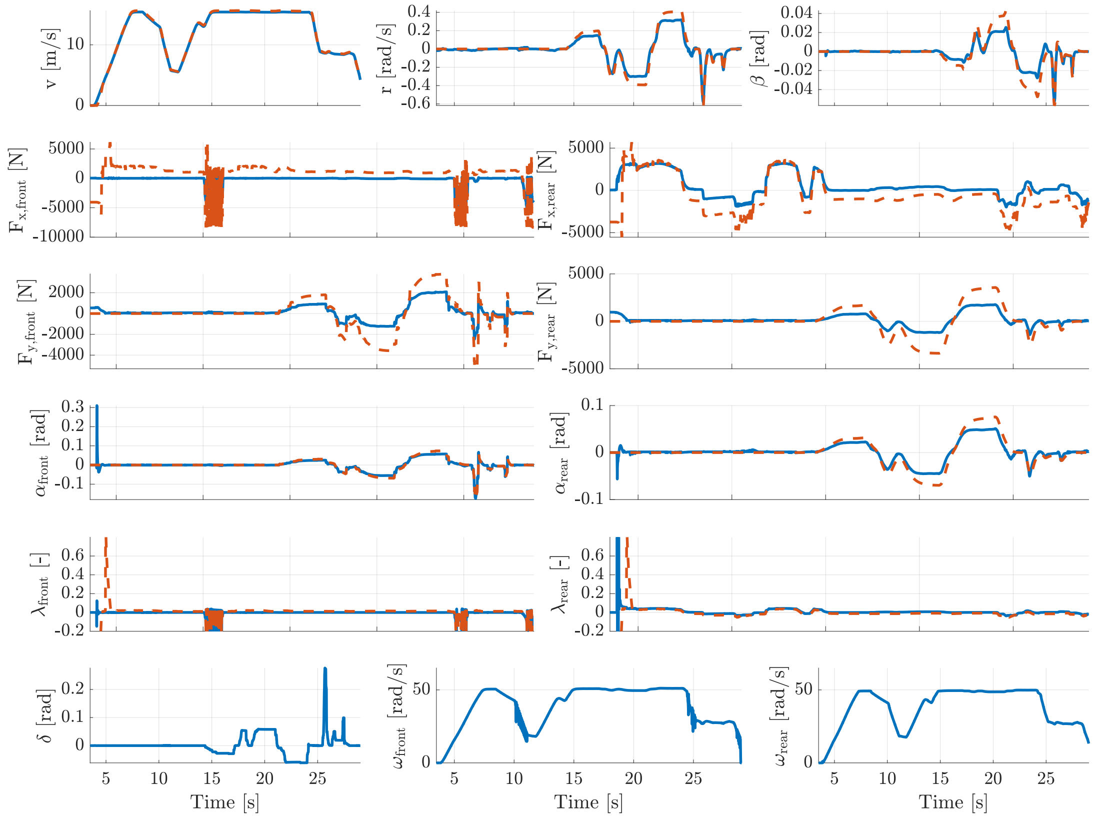


Figure 6.3: Comparison between LFS and the Single-Track model for XRG (blue line it is LFS, red line is the Single-Track)

6.2 Twin-Track

Figures 6.4,6.5,6.6 demonstrate that Twin-Track well follow LFS in signals. Lateral and longitudinal dynamics are similar to Single-Track. Slip ratio λ and slip angle α are well followed, but have some differences, because wheel radius in Twin-Track model are constant, but in LFS they are dynamically changed. Other parameters have more differences that described.

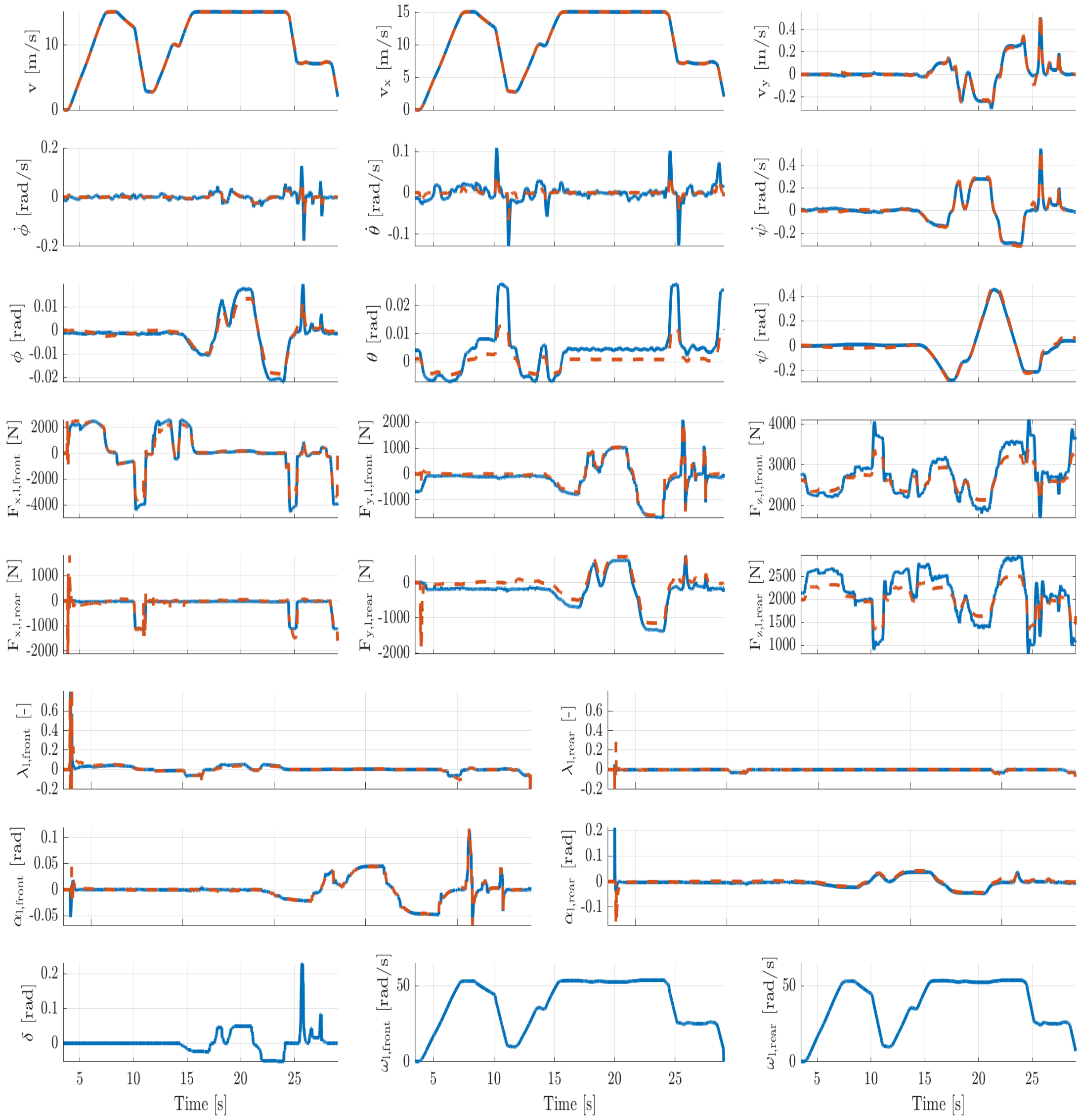


Figure 6.4: Comparison between LFS and the Twin-Track model for GTI (blue line it is LFS, red line is the Twin-Track)

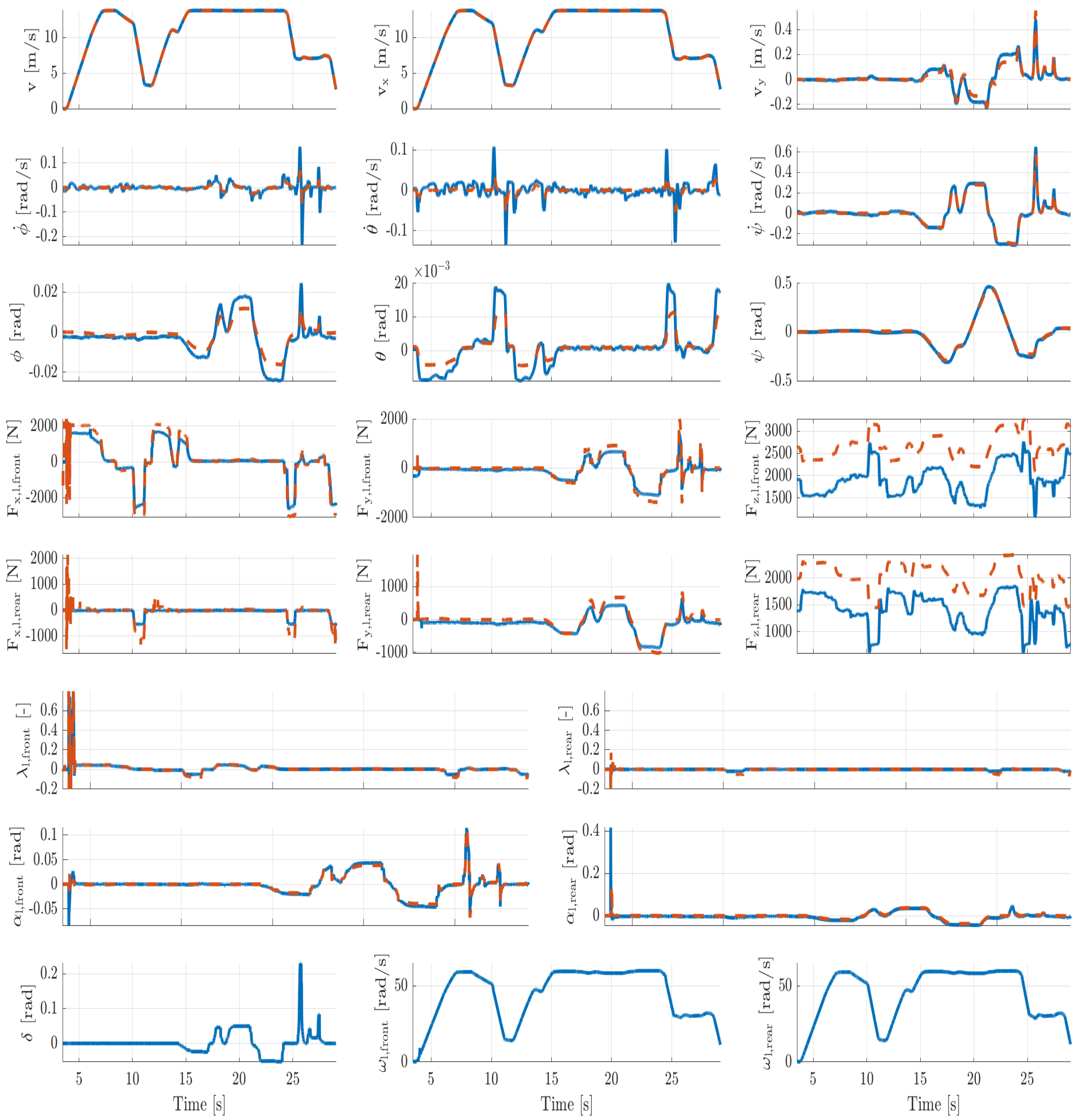


Figure 6.5: Comparison between LFS and the Twin-Track model for UF1 (blue line it is LFS, red line is the Twin-Track)

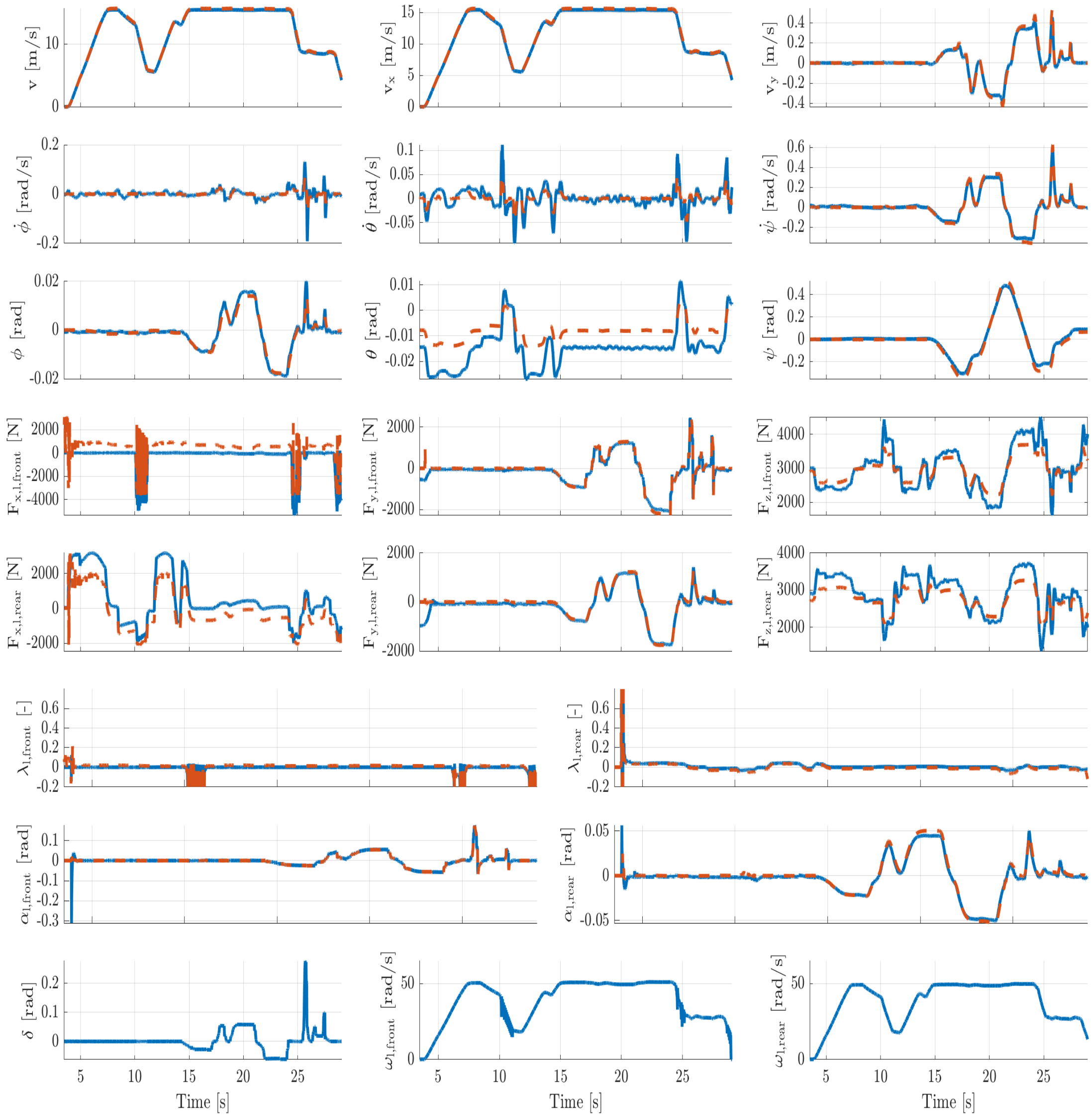


Figure 6.6: Comparison between LFS and the Twin-Track model for XRG (blue line it is LFS, red line is the Twin-Track)

7. Results

All of the goals, listed in chapter [**Objectives**], are done. Namely:

- Familiarization with vehicle dynamics simulator "Live for Speed" was completed. Describing of information that can be taken from the simulator is placed in [**LFS Signals Preview**].
- Low-fidelity and high-fidelity vehicle models were adopted and described in [**Nonlinear Single-Track Model**] and [**Nonlinear Twin-Track Model**].
- Experiments for longitudinal motion were created and prepared as prescript inputs for MATLAB. Inputs are described in the chapter [**Model identification**].
- Experiments for lateral motion were created and prepared as prescript inputs for MATLAB. Inputs are described in the chapter [**Model identification**].
- Procedure for identification of parameters is described in [**Model identification**].
- Validation of identification procedure on different car models implemented in the simulator is provided in chapter [**Comparisons**]

8. Conclusion

This thesis completely describes the identification process for parameters of Single-Track and Twin-Track models. Two main modeling techniques exist for creating vehicle models. These techniques are mainly used to describe the vehicle dynamics: Single-Track [[3], chapter 10], and Twin-Track models [[3], chapter 11]. In this work, adopted Single-Track model, described in the article "Nonlinear Single-Track Model" [2], was used because it has sufficient fidelity to describe a lateral vehicle dynamics, and to test identification methods because it is simpler than the Twin-Track models (three versus ten states). Twin-Track was used because it allows testing identification on more difficult model (ten states) to compare it in more "complicated" design.

This thesis includes an adaptation of the existing Single-Track and Twin-Track models for compatibility with inputs from LFS, guide how to generate data for identification, creation and/or adaptation formulas for calculating of models parameters, and comparisons between Single-Track/Twin-Track model and LFS.

Result is presets for generating needful data, automatic initialization file, mat-files that calculate parameters for models.

Bibliography

[1] LFS, "Live for speed's official website," Available online: https://www.lfs.net/, 2020.

[2] D. Efremov, "Single-track vehicle model," Available online: https://github.com/SDS-RC-FEE-CTU-in-Prague/SingleTrack, 2020.

[3] D. Schramm, M. Hiller, and R. Bardini, *Vehicle dynamics, Modeling and Simulation.* Springer, 2014.

[4] H. Pacejka, *Tire and vehicle dynamics.* Elsevier, 2005.

[5] V. Cibulka, "Twin-track vehicle model," Available online: https://github.com/SDS-RC-FEE-CTU-in-Prague/TwinTrack, 2020.

[6] ——, "Mpc based control algorithms for vehicle control," Ph.D. dissertation, Masterâ™s thesis, Czech Technical University in Prague, 06 2019, 2019.

[7] T. Tomáš, "Verification platform development for vehicle control system validation," B.S. thesis, České vysoké učení technické v Praze. Vypočetní a informační centrum., 2020.

[8] P. J. Huber, *Robust statistics.* John Wiley & Sons, 2004, vol. 523.

List of Figures

List of Tables