Bachelor Project

Czech Technical University in Prague

F3

Faculty of Electrical Engineering
Department of Control Engineering

Vehicle slip ratio control system

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Field of study: Cybernetics and Robotics
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ZADÁNÍ BAKALÁŘSKÉ PRÁCE

I. OSOBNÍ A STUDIJNÍ ÚDAJE

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II. ÚDAJE K BAKALÁŘSKÉ PRÁCI

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[1] Edward M. Kasprzak, L. Daniel Metz, William F. Milliken Douglas L. Milliken, Race Car Vehicle Dynamics - Problems, Answers and Experiments, Premiere Series Books, 2015, ISBN-10: 0768011272.
[2] Prof. Dr.-Ing. Uwe Kiencke Prof. Dr. Lars Nielsen; Automotive Control Systems, Springer, 2005
[3] Denis Efremov, Unstable ground vehicles and artificial stability systems, 2018
[4] Marek Laszlo, Flight Control Solutions Applied for Improving Vehicle Dynamics, 2019

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III. PŘEVZETÍ ZADÁNÍ

Student bere na vědomí, že je povinen vypracovat bakalářskou práci samostatně, bez cizí pomoci, s výjimkou poskytnutých konzultací.
Seznam použité literatury, jiných pramenů a jmen konzultantů je třeba uvést v bakalářské práci.

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I would like to express my gratitude to my supervisor Ing. Tomáš Haniš, PhD. especially for his friendly attitude and for his time and advice.

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Also, I would like to thank my classmates and friends for their support throughout my studies.

And my final "thank you" is reserved for my dear parents and siblings for being in all aspects such a great family.

Declaration

I hereby declare that this master thesis was finished on my own and that I have cited all the used information sources in the list of references according to the Methodical guideline on the observance of ethical principles in the preparation of university graduate thesis.

In Prague, May 24, 2019

Abstract

The aim of this thesis is to utilize the concept of four independantly propelled wheels for enhancing handling characteristics of a small formula car. It is meant to be done by means of model-based design. The intention is to control the traction of the vehicle via control of the slip ratio of the vehicle's wheels.

For this purpose, a vehicle model is composed and then linearized. Subsequently, control algorithms are designed with the linear model. Then, the control system is virtually tested with the IPG Carmaker simulation environment. Finally, the software is tested in a real formula.

Keywords: vehicle dynamics, model-based design, single-track model, Pacejka magic formula, traction control, slip ratio, PI controller, root locus, linear parameter-varying system, IPG Carmaker, Formula Student

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Abstrakt

Cílem této práce je využít možností, které skýtají elektrická vozidla se čtyřmi nezávisle poháněnými koly, pro zlepšení jízdní charakteristiky malé formule. K tomu jsou použity postupy známé pod souhrnným názvem *model based design*. Záměrem je řídit podélnou trakci vozidla prostřednictvím řízení podélného skluzu kol vozidla.

Za tímto účelem je sestaven a zlinearizován model vozidla. Následně jsou na tomto lineárním modelu navrženy řídící algoritmy. Poté dochází k simulacím ve virtuálním simulačním prostředí IPG Carmaker. Na závěr jsou dané algoritmy implementovány do skutečné formule.

Klíčová slova: dynamika vozidla, modelově orientovaný design, jednostopý model, Pacejkova formule, kontrola trakce, skluz, PI regulátor, root locus, lineární systém s proměnným parametrem, IPG Carmaker, Formula Student

Překlad názvu: Řízení podélného skluzu kol vozidla

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Chapter 1
Introduction

1.1 General Introduction

As the first cars emerged in human history and the tale of automotive engineering began, an ancient urge resonated in mankind - the need to compete, to race, to surpass the others and win. Motorsport eventually settled at the top of the automotive industry and became the source of innovation and leading technology.

Although an electric drive has been known to engineers since the early times already, it has still played rather a minor role so far. The automotive industry has been ruled by the gasoline-engine technology.

However, the times have begun to change recently and electric cars are spreading wide. Unsurprisingly, the motorsport has to face the change, too. Luckily, electric drive brings a plethora of new options, including:

1. Independently propelled wheels.
2. Full torque from zero speed.
3. More simple, lighter gearbox.
4. No emissions.

Surely there are even more advantages of electric drive and also certain disadvantages, too. However, for a purpose of this thesis, the independently propelled wheels are crucial. With the independent wheel drive, algorithms of advanced *torque vectoring* and *traction control* can be implemented. A deeper insight in the topic of control systems used in vehicles can be acquired in [UK05].

1.2 Formula SAE

Aside from famous professional motorsport racing competitions (like Formula One, for example) there are also several student competitions. One of them is Formula SAE, of which our eForce FEE Prague Formula team is a member. The aim of this Formula SAE project is to enhance and test skills of engineering students. They earn an opportunity to participate in a challenging motorsport engineering project from the beginning to the end, from the design phase to testing. Students are required to construct a new car for every racing season.

During every Formula SAE racing event the teams undergo the following disciplines:

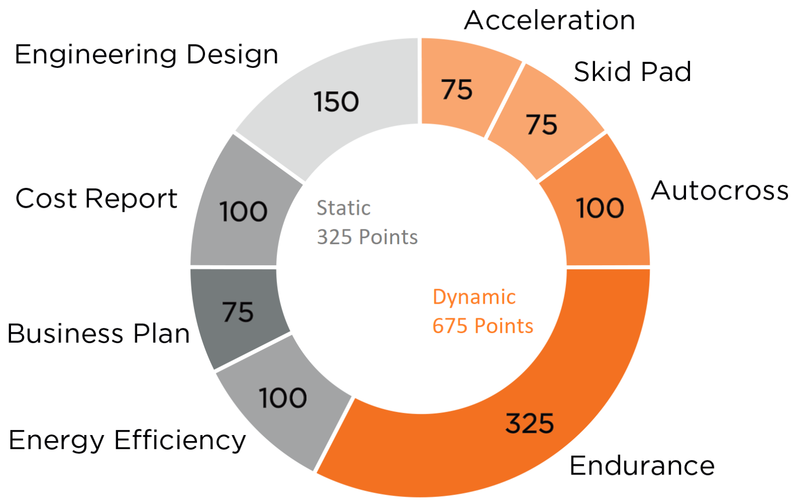

Figure 1.1: Formula SAE racing event disciplines

A part of the picture was adopted from `https://eforce.cvut.cz/formula-student/`.

- **Acceleration** takes place on a straight, 75m long track. Vehicle starts from zero speed and vehicle's velocity at the end of the track is measured.
- **Skid Pad** takes place on an 8-like shaped track. This discipline thoroughly examines vehicle's lateral acceleration ability.
- **Autocross** is actually a qualification for the Endurance race. The vehicle with the best lap time wins.
- **Endurance & Efficiency** is a 22km long race. Both the consumpted energy and the overall time are considered.

Concluding from the above, it is clear that a successful implementation of the traction control algorithm could cause a significant performance improvement and provide better racing results.

Traction control is a way to prevent a loss of traction. The traction preserved with no loss results in more predictable, safer behavior and energy loss reduction.

1.3 Objectives

The key objectives which should be accomplished are:

1. To adopt the vehicle model and the linearization tool and customize them for purpose of this thesis.
   Models and linearization and control methods are being developed by my senior colleagues at the Czech Technical University in Prague. The task is to adopt these methods and apply them to fulfill the assignment.

2. To develop traction control algorithms.
   The task is to implement an algorithm that controls wheels' slip ratio according to the slip ratio requested by driver.

3. To verify the acquired algorithms in virtual simulation environment.
   It is benefitial to utilize the virtual test driving before implementing the algorithms into a real car because this way many bugs can be faster discovered and repaired.

4. To verify the acquired algorithms in the real vehicle.
   It is necessary to know whether the algorithms work in the real car and could be possibly used at a racing event.

1.4 Outline

The thesis plan goes as follows:

1. Vehicle Model derivation - Chapter 2.
   A single-track model is introduced and then the individual parts of the single-track model are described.

2. Vehicle Model linearization - Chapter 3.
   This chapter describes the usage of LPV linearization.

3. Controller design - Chapter 4.
   Root locus method is used to design the control system.

4. Verification of the controller in a simulation environment and in the eForce formula - Chapter 5.
   Both the virtual testing in IPG Carmaker and the real testing with formula car are described and compared.

5. Conclusion and future work - Chapter 6.
   Achieved results are summarized and future improvements are proposed.

Chapter 2
Vehicle Model

A so-called *model-based design* approach is used in this thesis. It offers the following key advantages:

- The models are "reusable". A group of collaborators can develop various control designs with the same model, which facilitates mutual comparison of the designs.
- A full potential of making simulations can be utilized.
- Designers can choose models of various levels of complexity, which are suitable for different tasks.
- Error correction is facilitated, because the errors can be discovered in earlier stages of design.

There are plenty of vehicle models, let at least three of them be mentioned:

1. *Multi-body model*: a very complex model, taking weights of particular vehicle components into account. Has a behaviour most similar to a real car.
2. *Twin-track model*: The whole mass of the car is concentrated in a center of gravity of the vehicle. The model comprises four wheels and the center of gravity.
3. *Single-track model*: the simplest model from these three. Both front wheels and both rear wheels shrink into one wheel so the model actually resembles a motorcycle as there is only one wheel in the front part and one wheel in the rear part. Also, there is the whole mass of the vehicle concentrated in the center of gravity.

Further information about vehicle dynamics can be found in [EMK15].

The singlle-track model was considered appropriate to conduct the control design process with because it is the simplest one yet it still preserves a sufficient level of fidelity. The model was adopted from Denis Efremov and then customized. More information can be found in Denis Efremov's works [DEH19], [Efr18].

2.1 Introduction to single-track model

The conventional right-hand Cartesian coordinate system is used. In a 3D world, the z-axis heads up, "towards the sky", perpendicular to the ground.

Several assumptions and simplifications are made:

- Rolling and pitching motion is neglected.
- Only front wheel can be steered.
- The whole mass of the vehicle concentrates in the center of gravity.
- Aligning torque resulting from slip angles is neglected.

A schematic of the single-track model can be seen in the following figure:

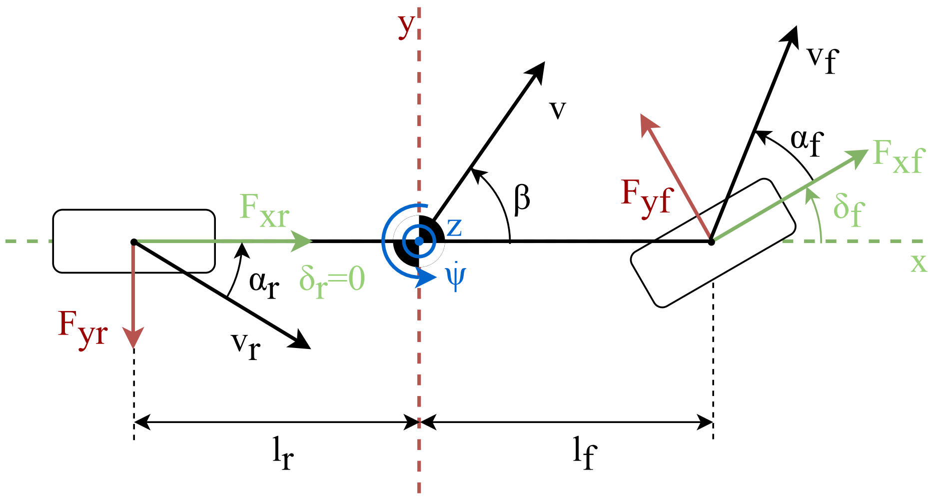

Figure 2.1: Single-track model schematic

Let the letters be explained:

- $\boldsymbol{F}_{xr}$ [N] is a component of a rear wheel force acting along the x-axis of the rear wheel.
- $\boldsymbol{F}_{yr}$ [N] is a component of the rear wheel force acting along the y-axis of the rear wheel.
- δ_r [rad] is a rear steering angle, it is the angle between the longitudinal axis of the vehicle and the x-axis of the rear wheel.
- $\boldsymbol{v}_r$ [$\mathrm{m\,s^{-1}}$] is a rear wheel velocity vector.
- α_r [rad] is a rear slip angle, it is the angle between the x-axis of the rear wheel and the rear wheel velocity vector $\boldsymbol{v}_r$.
- l_r [m] is a distance between the vehicle center of gravity and a rear axle.

- $\boldsymbol{v}$ [m s^{-1}] is a velocity of the vehicle's center of gravity.

- β [rad] is a vehicle's side-slip angle, it is the angle between a longitudinal axis of the vehicle and the velocity vector $\boldsymbol{v}$.

- $\dot{\psi}$ [rad s^{-1}] is a so called yaw rate, angular velocity of turning of the vehicle around the z-axis placed in the center of gravity of the vehicle.

- l_f [m] is a distance between the vehicle center of gravity and a front axle.

- $\boldsymbol{F}_{xf}$ [N] is a component of a front wheel force acting along the x-axis of the front wheel.

- $\boldsymbol{F}_{yf}$ [N] is a component of the front wheel force acting along the y-axis of the front wheel.

- δ_f [rad] is a front steering angle, it is the angle between the longitudinal axis of the vehicle and the x-axis of the front wheel.

- $\boldsymbol{v}_f$ [m s^{-1}] is a front wheel velocity vector.

- α_f [rad] is a front slip angle, it is the angle between the x-axis of the front wheel and the front wheel velocity vector $\boldsymbol{v}_f$.

Another important forces which influence the behaviour of the car and are included in the model are seen in the following picture:

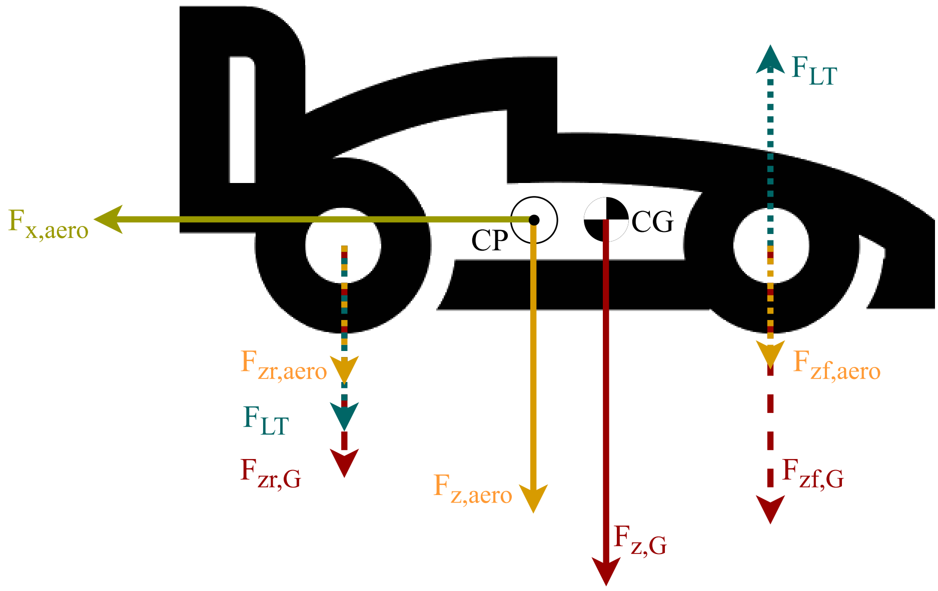

Figure 2.2: Formula side view

A part of the picture was adopted from `https://www.kisspng.com/png-formula-1-sports-car-auto-racing-computer-icons-4712795/`.

The variables in the picture:

- $\boldsymbol{F}_{x,aero}$ [N] is the aerodynamic drag force.
- $\boldsymbol{F}_{zr,aero}$ [N] is the component of aerodynamic downforce acting vertically on the rear axle.
- $\boldsymbol{F}_{LT}$ [N] is the load transfer force.
- $\boldsymbol{F}_{zr,G}$ [N] is the component of gravitational force acting vertically on the rear axle.
- CP is the center of pressure.
- $\boldsymbol{F}_{z,aero}$ [N] is the overall aerodynamic downforce acting on the vehicle in the center of pressure.
- CG is the center of gravity.
- $\boldsymbol{F}_{z,G}$ is the overall gravitational force acting on the vehicle in the center of gravity.

- $\boldsymbol{F}_{zf,aero}$ [N] is the component of aerodynamic downforce acting vertically on the front axle.

- $\boldsymbol{F}_{zf,G}$ [N] is the component of gravitational force acting vertically on the front axle.

Henceforth, only scalar values of the above introduced vector quantities are used. Thus, the bold notation of the variables is ommited.

The model has overall five states and six inputs. The states are: velocity v, side-slip angle β, yaw rate $\dot{\psi}$, the front wheel angular velocity ω_f and the rear wheel angular velocity ω_r. The inputs are: the front steering angle δ_f, the rear steering angle δ_r (it is always zero), the front wheel input torque τ_f, the rear wheel input torque τ_r, the front wheel input breaking torque τ_{Bf} and the rear wheel input breaking torque τ_{Br}.

The single-track model consists of the following parts:

- Aerodynamics.
- Load transfer.
- Steering angles projection.
- Vehicle dynamics.
- Slip ratios & slip angles.
- Wheel models.
- Tire models.
    - Pacejka magic formula
    - Traction ellipse

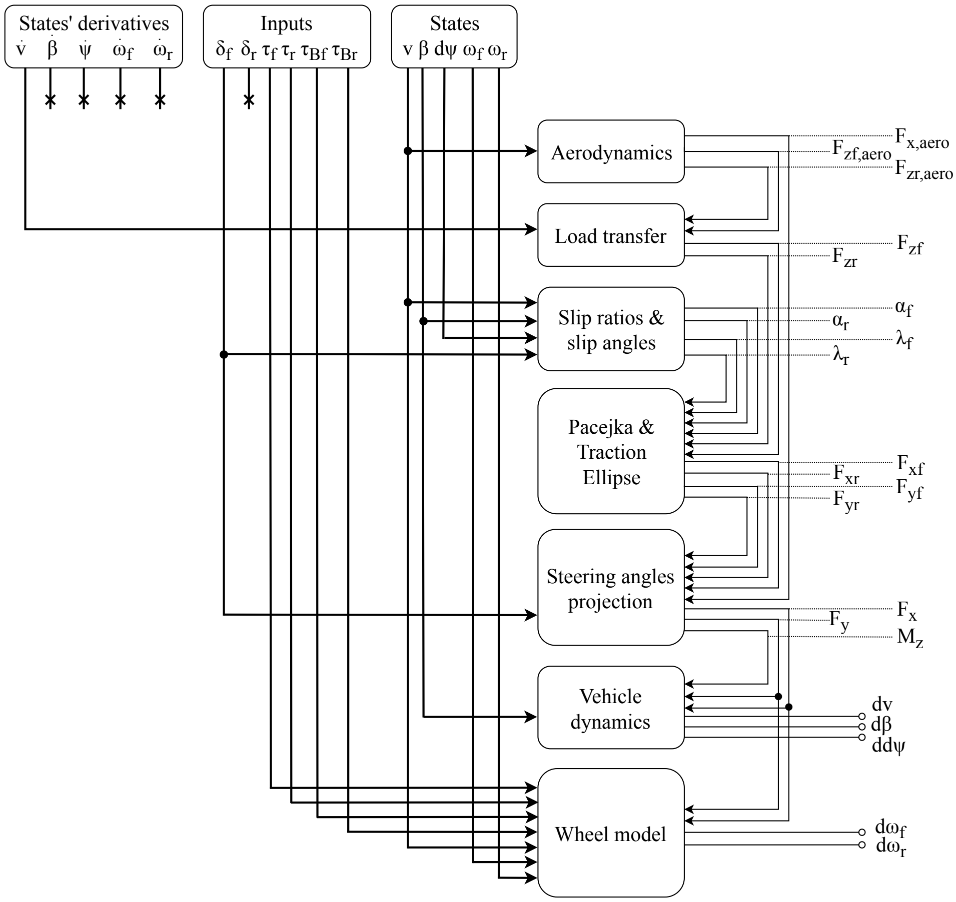

Figure 2.3: Block schematic of single track model

The figure 2.3 illustrates the dependancies among model quantities. On the left side there are quantities that step into the model and in the middle there are model blocks. On the right side there are quantities that are being interchanged among the model blocks and in the bottom-right corner there are output quantities. Now, let the parts be examined one by one.

2.2 Aerodynamics

As it were shown by virtual car simulations, the effect of aerodynamics cannot be omitted from the model. The aerodynamics exerts two forces: longitudinal (x-axis) and vertical (z-axis). The longitudinal force $F_{x,aero}$ [N] is given by so-called *drag equation*

$$F_{x,aero} = \frac{1}{2}\rho v^2 C_D A\,, \tag{2.1}$$

- ρ $[\mathrm{kg\,m^{-3}}]$ is the *air mass density*,
- v $[\mathrm{m\,s^{-1}}]$ is the *velocity of the car's center of gravity*,

- C_D $[-]$ is the *drag coefficient* related to the center of pressure,
- A $[\mathrm{m}^2]$ is the *reference area*.

The vertical force $F_{z,aero}$ results from

$$F_{z,aero} = \frac{1}{2}\rho v^2 C_L A\,, \tag{2.2}$$

where C_L $[-]$ is the *lift coefficient* related to the center of pressure and the rest are the same as above. The vertical force is ditributed between the front and the rear axle according to their distance from vehicle's center of pressure:

$$F_{zf,aero} = \frac{CoPR}{CoPF + CoPR}\,F_{z,aero}\,, \tag{2.3}$$

$$F_{zr,aero} = \frac{CoPF}{CoPF + CoPR}\,F_{z,aero}\,. \tag{2.4}$$

$CoPF$ [m] is the distance between the front axle and the vehicle's center of pressure, $CoPR$ [m] is the distance between the rear axle and the vehicle's center of pressure.

2.3 Load transfer

The aim is to achieve a good acceleration. In this specific case, the *load transfer* plays a very important role. Load transfer means that the vertical force resulting from the acceleration of the vehicle is acting on both axles in opposite directions dependant on the sign of the acceleration:

$$F_{LT} = \frac{h}{Wb}\,a\,m\,, \tag{3.1}$$

- F_{LT} [N] is the force which is subtracted from the vertical force acting on one axle and added to the other one according to the sign of acceleration.
- h [m] is the height of the center of gravity of the vehicle.
- Wb [m] is the *wheelbase*.
- a $[m\,s^{-2}]$ is the acceleration of the center of gravity of the vehicle.
- m [kg] is the vehicle's mass.

As only positive acceleration in the model is considered, the equations for vertical forces acting on the axles may be already written. Gravitational forces $F_{zf,G}$, $F_{zr,G}$ [N]:

$$F_{zf,G} = m \, \frac{l_r}{Wb} \, G \,, \tag{3.2}$$

$$F_{zr,G} = m \, \frac{l_f}{Wb} \, G \,. \tag{3.3}$$

where G [m s^{-2}] is the gravitational acceleration. Resulting vertical forces:

$$F_{zf} = F_{zf,G} - F_{zf,aero} - F_{LT} \,, \tag{3.4}$$
$$F_{zr} = F_{zr,G} - F_{zr,aero} + F_{LT} \,. \tag{3.5}$$

2.4 Steering angles projection

Steering angles have an impact on both the longitudinal and the lateral components of both the front and the rear forces acting on wheels. Results are the *longitudinal component of force acting in the center of gravity* F_x [N], the *lateral component of force acting in the center of gravity* F_y [N] and the *turning torque* M_z [Nm] acting around the z-axis:

$$\begin{bmatrix} F_x \\ F_y \\ M_z \end{bmatrix} = \begin{bmatrix} \cos\delta_f & -\sin\delta_f & \cos\delta_r & -\sin\delta_r \\ \sin\delta_f & \cos\delta_f & \sin\delta_r & \cos\delta_r \\ l_f \sin\delta_f & l_f \cos\delta_f & -l_r \sin\delta_r & -l_r \cos\delta_r \end{bmatrix} \begin{bmatrix} F_{xf} \\ F_{yf} \\ F_{xr} \\ F_{yr} \end{bmatrix} \,. \tag{4.1}$$

2.5 Vehicle dynamics

Vehicle dynamics tells how three of the states vary depending on side-slip angle β and yaw rate $\dot{\psi}$. It transforms the force and the turning torque acting in the center of gravity to velocity, side-slip angle and the yaw rate derivatives:

$$\begin{bmatrix} \dot{v} \\ \dot{\beta} \\ \ddot{\psi} \end{bmatrix} = \begin{bmatrix} \frac{\cos\beta}{m} & \frac{\sin\beta}{m} & 0 \\ -\frac{\sin\beta}{mv} & \frac{\cos\beta}{mv} & 0 \\ 0 & 0 & \frac{1}{I_z} \end{bmatrix} \begin{bmatrix} F_x \\ F_y \\ M_z \end{bmatrix} - \begin{bmatrix} 0 \\ \dot{\psi} \\ 0 \end{bmatrix} \,. \tag{5.1}$$

2.6 Slip angles & slip ratios

The calculation of slip angles reveals how velocity, yaw rate, side-slip angle and steering angles affect wheels' slip angles. Because this thesis is focused mainly on linear motion, the following slip angles are rather unnecessary in the model:

$$\alpha_f = -\arctan \frac{(v \sin\beta + l_f\dot{\psi})\cos\delta_f - v\cos\beta\sin\delta_f}{|(v\sin\beta + l_f\dot{\psi})\sin\delta_f + v\cos\beta\cos\delta_f|}\,, \tag{6.1}$$

$$\alpha_r = -\arctan \frac{(v \sin\beta - l_r\dot{\psi})\cos\delta_r - v\cos\beta\sin\delta_r}{|(v\sin\beta - l_r\dot{\psi})\sin\delta_r + v\cos\beta\cos\delta_r|}\,. \tag{6.2}$$

To the contrary, slip ratios are very important for the purpose of this thesis. First of all, the x-axis components of wheels' velocities have to be determined:

$$v_{xf} = v\cos(\beta)\cos(\delta_f) + (v\sin(\beta) + l_f\dot{\psi})\sin(\delta_f)\,, \tag{6.3}$$

$$v_{xr} = v\cos(\beta)\cos(\delta_r) + (v\sin(\beta) - l_r\dot{\psi})\sin(\delta_r)\,. \tag{6.4}$$

Subsequently, the equations for slip ratios λ_f $[-]$, λ_r $[-]$ can be written:

$$\lambda_f = \frac{\omega_f R_f - v_{xf}}{v_{xf}}\,, \qquad \lambda_r = \frac{\omega_r R_r - v_{xr}}{v_{xr}}\,. \tag{6.5}$$

2.7 Tire models

Understanding of a behaviour of a tire is crucial for vehicle control as all the interaction between the vehicle and the road happens through the tire. Unfortunately, the tire-road interaction embodies a highly nonlinear and complex system. To model a tire is not a simple task. A thorough insight to this topic provides the dissertation thesis of Lukas Haffner from TU Wien [Haf08].

2.7.1 Simplified Pacejka Magic Formula

The so-called *Pacejka Magic Formula* introduced by Hans Bastiaan Pacejka in [Pac05] was decided to be used. It enables computing of both the lateral and the longitudinal component of force acting on each wheel just from knowledge of vertical load and slip ratio λ or slip angle α. The longitudinal forces may be acquired as follows:

$$F_{xf}(\lambda_f) = D_x F_{zf}\sin(C_x\arctan(B_x\lambda_f - E_x(B_x\lambda_f - \arctan(B_x\lambda_f))))\,, \tag{7.1}$$

$$F_{xr}(\lambda_r) = D_x F_{zr}\sin(C_x\arctan(B_x\lambda_r - E_x(B_x\lambda_r - \arctan(B_x\lambda_r))))\,. \tag{7.2}$$

The lateral forces depend on the slip angles:

$$F_{yf}(\alpha_f) = D_y F_{zf} \sin(C_y \arctan(B_y \alpha_f - E_y(B_y \alpha_f - \arctan(B_y \alpha_f)))) \,, \tag{7.3}$$

$$F_{yr}(\alpha_r) = D_y F_{zr} \sin(C_y \arctan(B_y \alpha_r - E_y(B_y \alpha_r - \arctan(B_y \alpha_r)))) \,. \tag{7.4}$$

As it can be seen, the resulting forces are influenced by the *shaping coefficients* B, C, D, E. These coefficients are unique for both the longitudinal and the lateral forces.

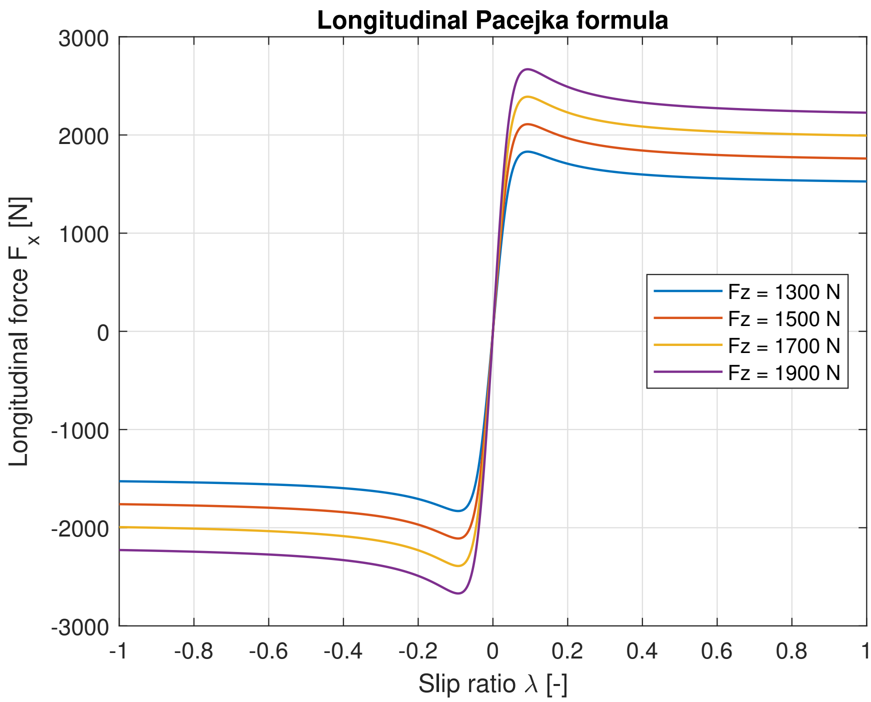

Figure 2.4: Dependancy of Pacejka formula shape on vertical wheel load

The values of forces in the figure 2.4 actually match with the values of acceleration of the vehicle 0, 4, 8, 12 $\mathrm{m\,s^{-2}}$, so the figure also demonstrates the effect of load transfer.

2.7.2 Traction Ellipse

It is commonly known that a car cannot reach the same maximal acceleration when cornering as it would have reached in the linear motion. This effect is modeled by a *traction ellipse*. It shows that only a certain combined force can be reached:

$$F_{combined} = \sqrt{\frac{F_x^2}{D_x^2} + \frac{F_y^2}{D_y^2}} \,. \tag{7.5}$$

If the resulting force F steps out of the ellipse, the wheel loses grip. Further details can be found in [MSC08].

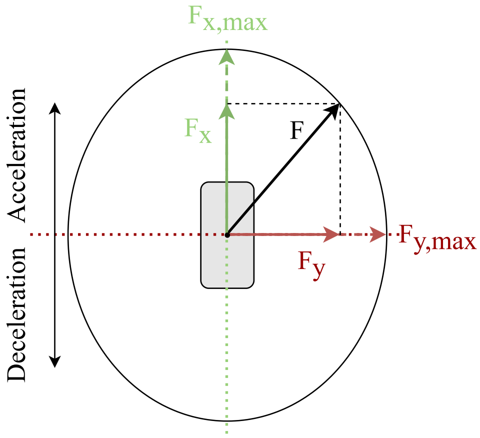

Figure 2.5: Traction ellipse

2.8 Wheel models

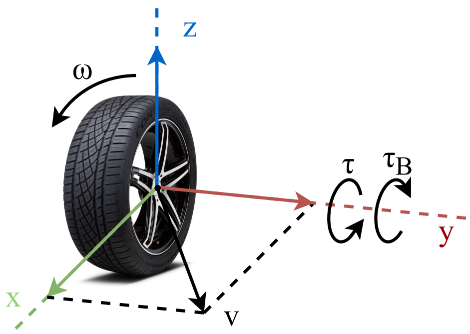

Figure 2.6: Wheel sketch with coordinate system and important variables.

A part of the picture was adopted from `https://images.tirebuyer.com/visual-aids/products/tires/continental/extremecontactdws06/continental_extremecontactdws06_bw_140843_vary.jpg`.

Resulting angular accelerations of the wheels depend on

- Wheels' moments of inertia $J_f,\ J_r$ [kg m^2].
- Torques $\tau_f,\ \tau_r$ [Nm] imposed on the wheels by motors.
- Braking torques $\tau_{Bf},\ \tau_{Br}$ [Nm] exerted on wheels by brakes.
- X-axis components of forces acting in the centers of wheels, $F_{xf},\ F_{xr},$ [N].

- X-axis components of the velocities of wheels' centers of gravity, $v_{xf},\ v_{xr}$ [$\mathrm{m\,s^{-1}}$].

The angular accelerations are given by:

$$\dot{\omega}_f = \frac{1}{J_f}(\tau_f - R_f F_{xf} - \mathrm{sgn}(\omega_f)\tau_{Bf} - k_f v_{xf})\,, \tag{8.1}$$

$$\dot{\omega}_r = \frac{1}{J_r}(\tau_r - R_r F_{xr} - \mathrm{sgn}(\omega_r)\tau_{Br} - k_r v_{xr})\,. \tag{8.2}$$

Chapter 3
Vehicle Model Linearization

3.1 LPV model

Because the nonlinear model was already acquired, a linearization needs to be conducted so the linear systems' theory can be utilized. The LPV model was decided to be used because a simple LTI linearization does not cover the behaviour of the system properly. The idea of such a linearization and the implemented algorithms alike were adopted from David Vošahlík. Detailed information on this topic can be found in his paper [DVH19]. The equations describing the LPV model are:

$$\dot{x}(t) = \boldsymbol{A}(p)\, x(t) + \boldsymbol{B}(p)\, u(t)\,, \tag{1.1}$$
$$y(t) = \boldsymbol{C}(p)\, x(t) + \boldsymbol{D}(p)\, u(t)\,. \tag{1.2}$$

In other words, now there is not only one operating point but a whole trajectory of operating points. The system's characteristics change overtime with a parameter. The goal is to create a grid of LTI systems which will be mutually switched according to grid parameters. The parameters are velocity and slip ratio, so a 2-D grid is obtained.

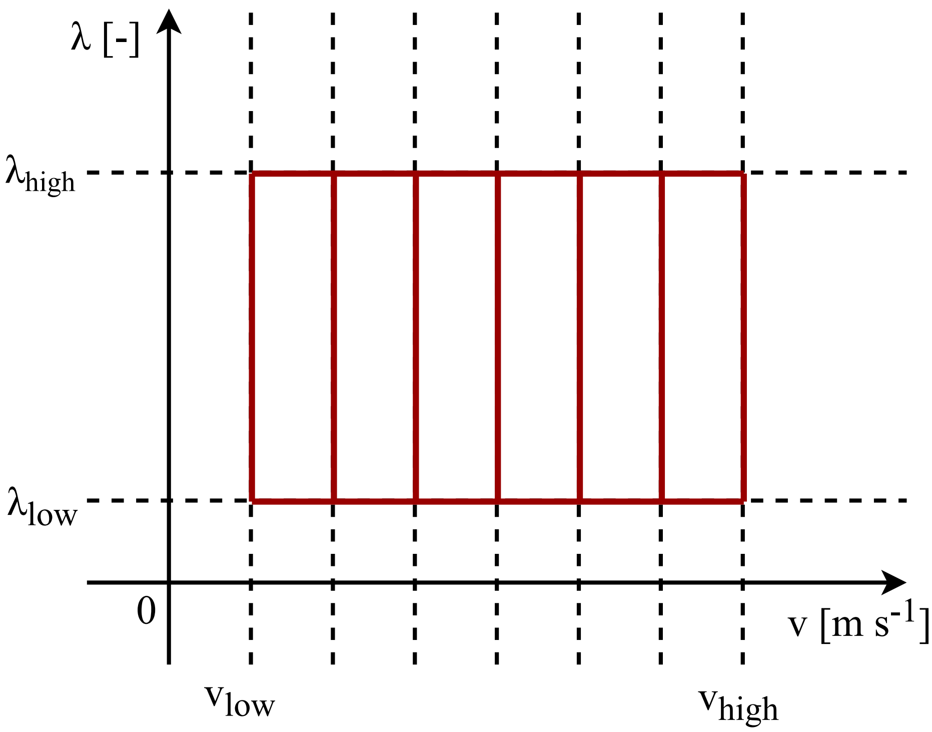

Figure 3.1: Slip ratio - velocity LPV grid

The process of linearization comprises the following steps:

1. Initialization.
2. Trimming.
3. Linearization.
4. Fitting.

3.2 Initialization

At the beginning of the algorithm, starting and stopping velocity are initialized, slip ratio, velocity increment and several other parameters. Also, the inputs, states and states' derivatives need to be initialized.

3.3 Trimming

An important question is how to choose an operating point in which the model could be linearized. The algorithm solves this problem with optimization tools, namely with Matlab function `fmincon`. The desired constraints are set and then `fmincon` finds the solution - a minimal cost function. The cost function is a weighted sum of squares of differences of desired states' derivatives and actual states' derivatives. The trimming tool also creates necessary offsets of inputs, states, states' derivatives and outputs because in the next step the state-space representation will be obtained and it is only valid as a deviation model, so it is always necessary to use the offsets to shift to the operating point.

3.4 Linearization

If the operating point has already been trimmed (the states and the inputs), Matlab's `linmod` can now be used to create the state-space representation.

The procedure continues as follows - for one slip ratio defined in the initialization the trimming starts at the starting speed. After the operating point is found, the system is linearized in this point. Subsequently, the velocity is incremented and both the trimming and the linearization repeat again. Finally, the stopping speed is reached. At the end, there are multiple state-space representations for various speeds and one slip ratio.

3.5 Fitting

After the last linearization is done, all the obtained operating points data are fitted with an eighth order polynomial in order to smoothen and facilitate the future switching between the models.

Chapter 4

Controller design

With the linearization described above two sets of state-space models are acquired, namely for slip ratios 0.01 and 0.09. The compensators are designed with `rltool` in Matlab for the following combinations of speeds and slip ratios:

- Low slip ratio and low speed.
- Low slip ratio and high speed.
- High slip ratio and low speed.
- High slip ratio and high speed.

All controllers are of a PI type. Altogether it makes four controllers for each wheel of the single track model, thus eight controllers for the whole system. A continuous switching happens among these controllers and is based on the two grid parameters - velocity and slip ratio. The switching actually means changing the values of P and I constants.

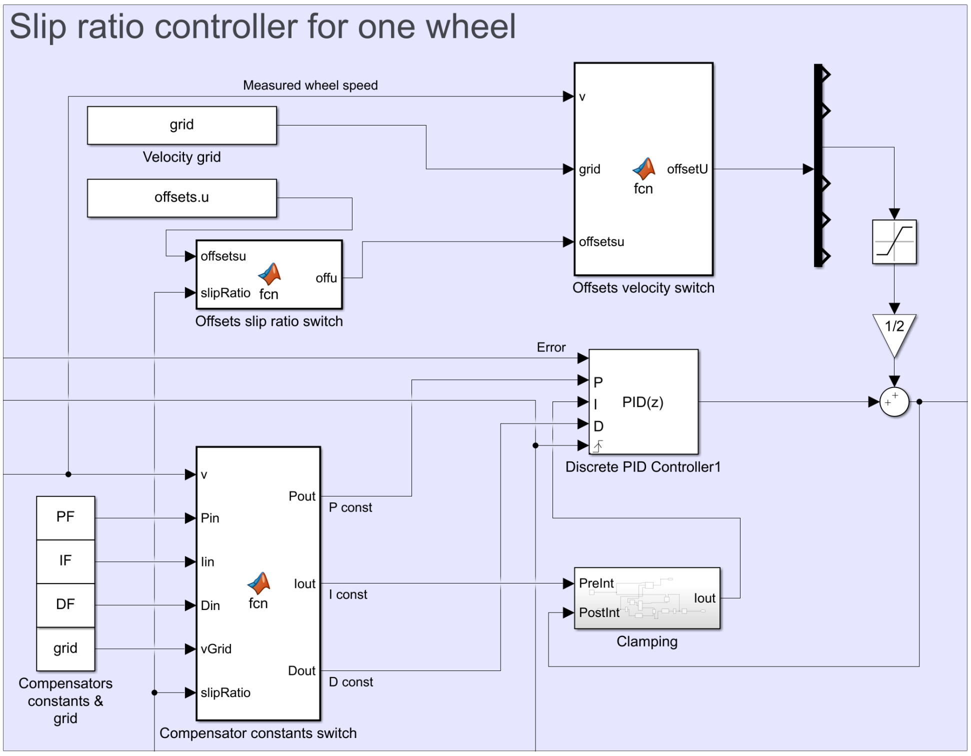

Figure 4.1: Slip ratio controller for one wheel

The picture 4.1 shows a slip ratio controller for one front wheel. Bottom-left there are constants P, I and D. As the controller is PI, D constant is always zero. Each of the constants PF (P front), IF and DF contain four numbers corresponding to the four combinations of slip ratios and velocities mentioned before. Further there are several switches. Clamping is a method of anti-windup. It is necessary to divide the input offsets by two because this is not the single-track model anymore so only half of the torque is required.

4.1 Implementation

Although the goal is to control the slip ratio, the compensator input is not actually slip ratio. However, with the slip ratio and measured velocity of the wheel it is possible to determine the request for angular velocity of the wheel. From this requested angular velocity the measured angular velocity is subtracted. That is the feedback. After this subtraction the resulting error becomes the input to the compensator.

Chapter 5

Experiments and results

5.1 Testing procedure

5.1.1 Virtual testing

Figure 5.1: IPG Carmaker simulation test run

The test driving software IPG Carmaker (`https://ipg-automotive.com/products-services/simulation-software/carmaker/`) was used. It enables creating various customized scenarios, test runs, roads, maneuvers, vehicles and much more. For purpose of formula testing, the simulations made by Marek Lászlo as a part of his work in eForce team were used. The practical differences between the sole Simulink and the Carmaker for Simulink (CM4SL) are:

- In Simulink there runs a continuous-time simulation. In Carmaker there runs a discrete-time simulation with a frequency of 1000 Hz.
- The custom control algorithms were designed with the single-track model. Carmaker uses much more complex multibody model.

- Various effects like rolling and pitching are disregarded in single-track model. Carmaker counts these effects in.

Generally, it can be stated that the Carmaker binds together almost realistic mechanical features of a formula with an ideal (unrealistic) world of Simulink simulations.

Also a so-called *driver in the loop* approach was used to test the control algorithms virtually. A closed loop consisting of the driver, Thrustmaster T300 RS force feedback steering wheel, pedals and a computer with IPG Carmaker and Matlab Simulink was made.

5.1.2 Real testing

Developed control algorithm was tested in the eForce 7th generation formula. The differences between the Carmaker software and the real formula are:

- Carmaker runs at frequency of 1000 Hz. Real formula is limited by CAN frequency 100 Hz.

- In Carmaker, all the data are available 100% of time and with no noise. In formula, the data are unreliable and noisy.

Generally speaking, the Carmaker and the formula are quite alike in terms of mechanical features and behaviour, but the Carmaker does not simulate real features of electronics, communication devices and sensors at all.

The code was implemented into formula's vehicle dynamics control unit as a part of control software made by Marek Lászlo. Further information can be found in his thesis [Lá19].

5.2 Test drives

The following experiments were carried on 12th May at the airfield of Panenský Týnec. The aim was to compare the behaviour of the car with traction control switched both off and on. The test drive comprised various accelerations in linear trajectory.

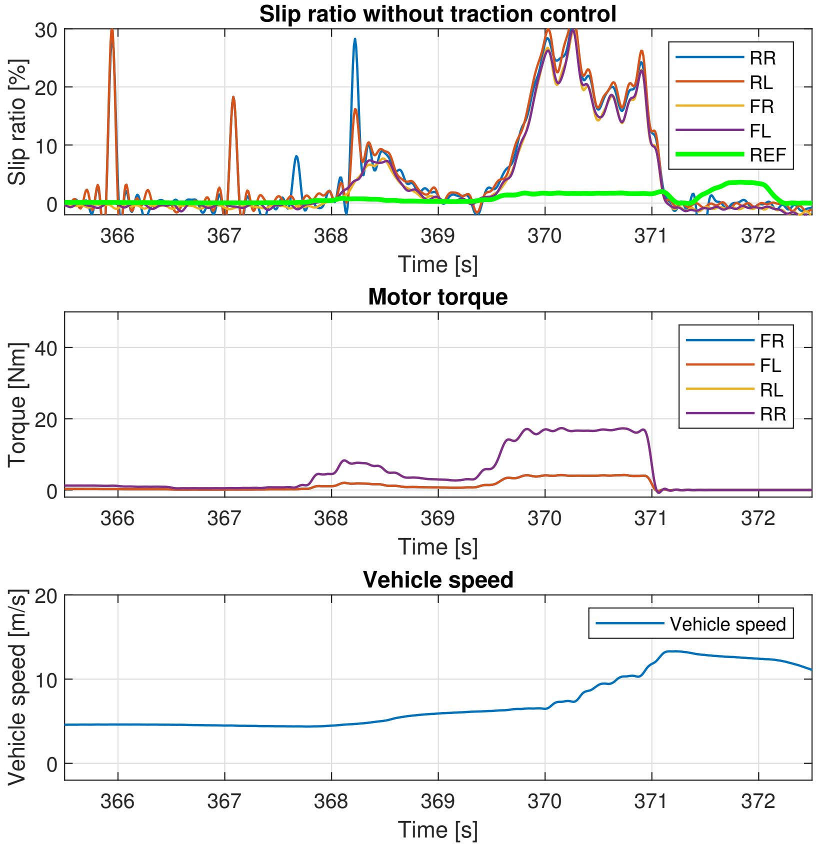

Figure 5.2: Traction control switched off

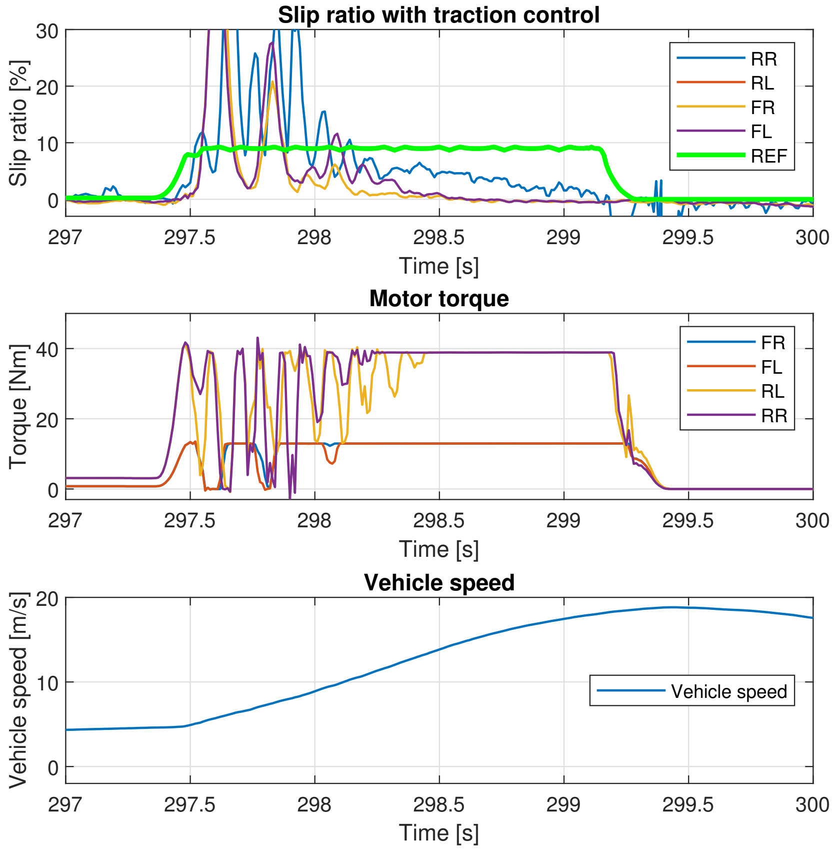

Figure 5.3: Traction control switched on

The abbreviations in the figures 5.2, 5.3 denote rear right wheel, rear left, front right, front left and a reference from pedal. Note: when the traction control is switched off, the control system does not interpret the position of the pedal as a request of slip ratio, but as a request of torque. Thus, the reference signal in 5.2 has not a meaning on its own, but only relatively in comparisson to the reference signal in 5.3. The figures above infer that:

1. In the real formula car, slip ratio is a quantity which is difficult to be measured. The measurements are very noisy and sometimes they are overall corrupt.

2. Switching the traction control on has a significant positive impact on system's behaviour. In 5.2, only a small values of torque caused such high values of slip ratios. In 5.3, the system can handle much higher torque, power, and slip ratios.

3. While accelerating, the limits of motors used in the formula are quickly reached. Surpassing the velocity of approximately 12 $\mathrm{m\,s^{-1}}$, the car can keep such high values of slip ratio no longer (it cannot reach such a high

acceleration in such a speed) because the motors are not able to provide necessary torque.

4. The whole system is extremely prone to oscillating, which makes the control even harder.

The inputs matching to the figures above were taken and used in Carmaker simulation:

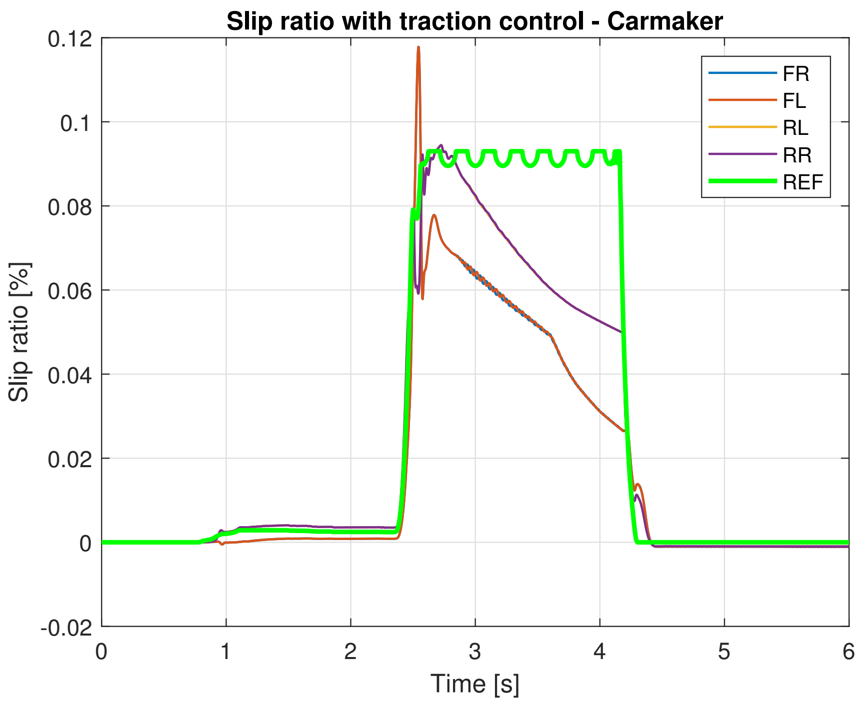

Figure 5.4: Carmaker simulation with inputs from real formula, 100% of road friction coefficient

The figure 5.5 reveals the major difference between the Carmaker and the real car. The system in Carmaker simulation oscillates much less than the real one. In this case, the simulation ran with road friction coefficient equal to 100%. Lowering the friction coefficient results in greater overshoot, but the impact on oscillations is rather minor.

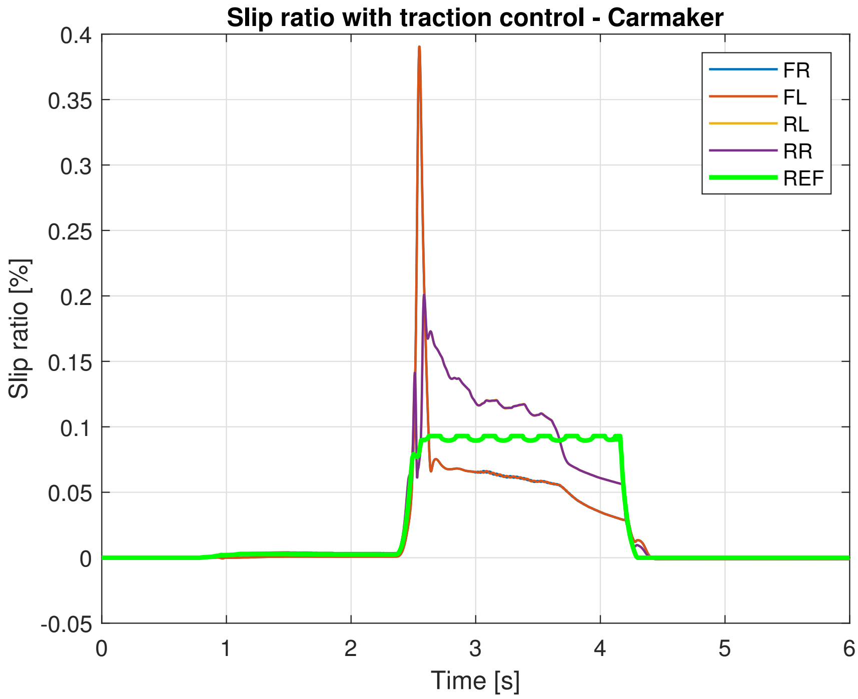

Figure 5.5: Carmaker simulation with inputs from real formula, 90% of road friction coefficient

Chapter 6

Conclusion & future work

6.1 Conclusion

1. It has been proved that it is possible to apply the approach of model-based design for developing a control system for a small racing formula. Although the real system is exceedingly nonlinear, it is still meaningful to develop linear models and use them for control design.

2. Virtual simulations and tests were utilized in a large extent and they are regarded crucial for the whole development process. Virtual testing embodies a vital link between simple single track model and the real car.

3. To control this system with such basic techniques like PI controllers is possible, yet the use of more advanced control designs would be, according to my estimation, more benefitial.

4. In comparisson with the previous control system that was used in the formula, the new algorithm presented in this thesis unfortunately has not proved a better performance so far. The previous algorithm can make the formula accelerate 0-100 km/h in 2.8 seconds. It can be seen in figure 5.3 that the acceleration of the new algorithm is slower. Currently the main obstacle is that with higher gains the whole system oscillates unacceptably.

5. The Carmaker model was compared with the real formula and it was discovered that the real vehicle oscillates aggresively which has to be taken into account when designing a control system using virtual simulations.

6. With the proposed algorithm, the car performance becomes better than there is no control algorithm. The assignment has been fulfilled.

6.2 Future work

In this section the possible future improvements are presented.

- To use more sophisticated control design techniques, for example LQI control. The expected advantage of this approach concludes from the fact that LQI allows to tailor a unique compensator for each state-space representation from the grid. The compensator is simply computed and it is not necessary to design it manually in `rltool`.
- To use a more complex model - twin-track model. For example, it would be benefitial for control of situations when the right and the left wheel are both on different surfaces.
- To add torque vectoring. This would also utilize the twin-track model. With torque vectoring, slip ratio control could be used in cornering maneuvers.
- To add longitudinal distribution. This would benefitial in situations when front and rear wheels are not on the same surfaces, or they are on surfaces with different friction coefficients, respectively.

Further improvements, not directly related to this thesis, could be sensors and measurement improvement, implementation of the control algorithm not in VDCU but directly in motor controller, increasing CAN frequency to 200 Hz, manufacturing a new motor controller with better characteristics, improving data filtering.

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Appendix A

Simulink schematics

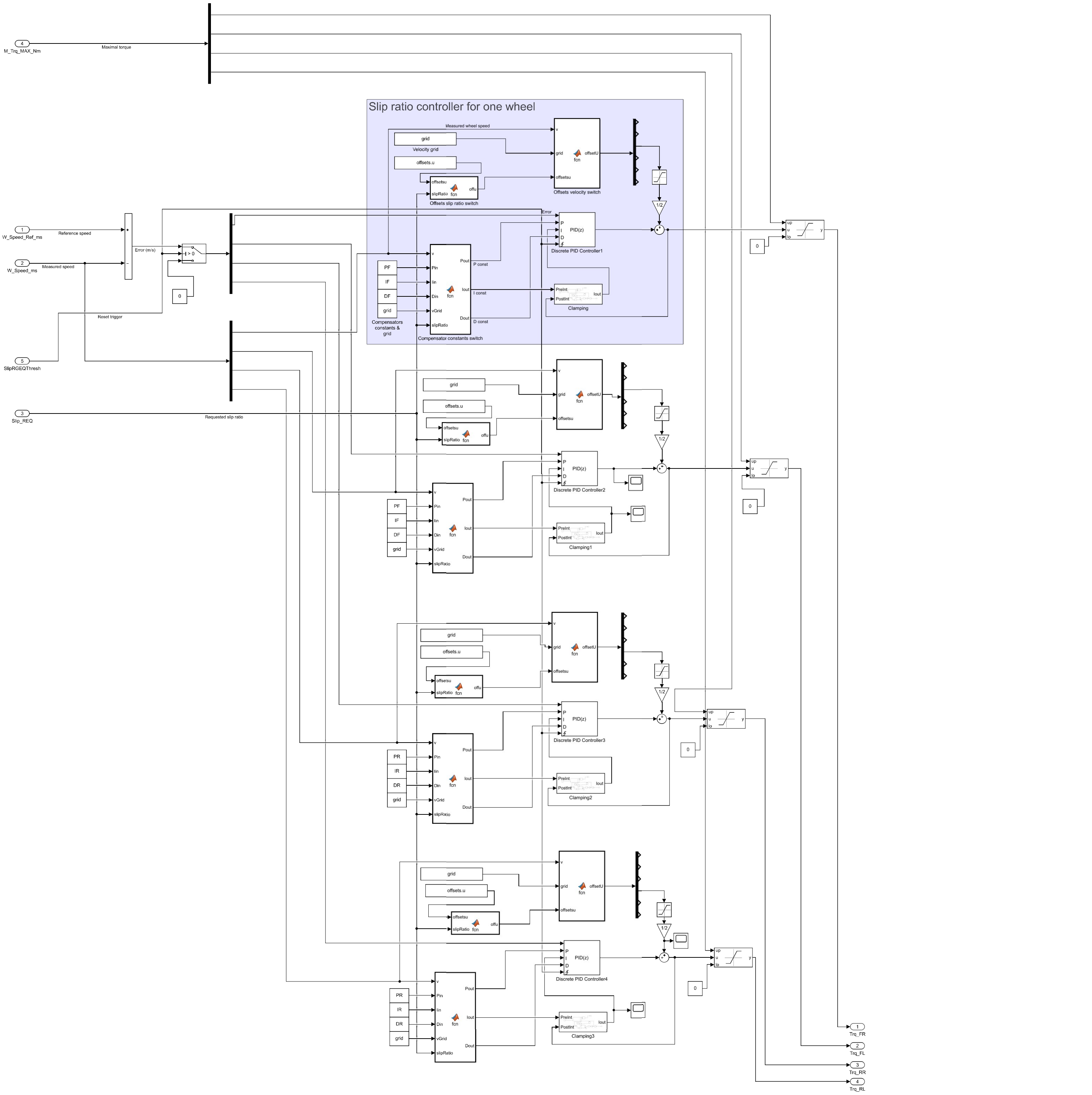

Figure A.1: Slip ratio controller block

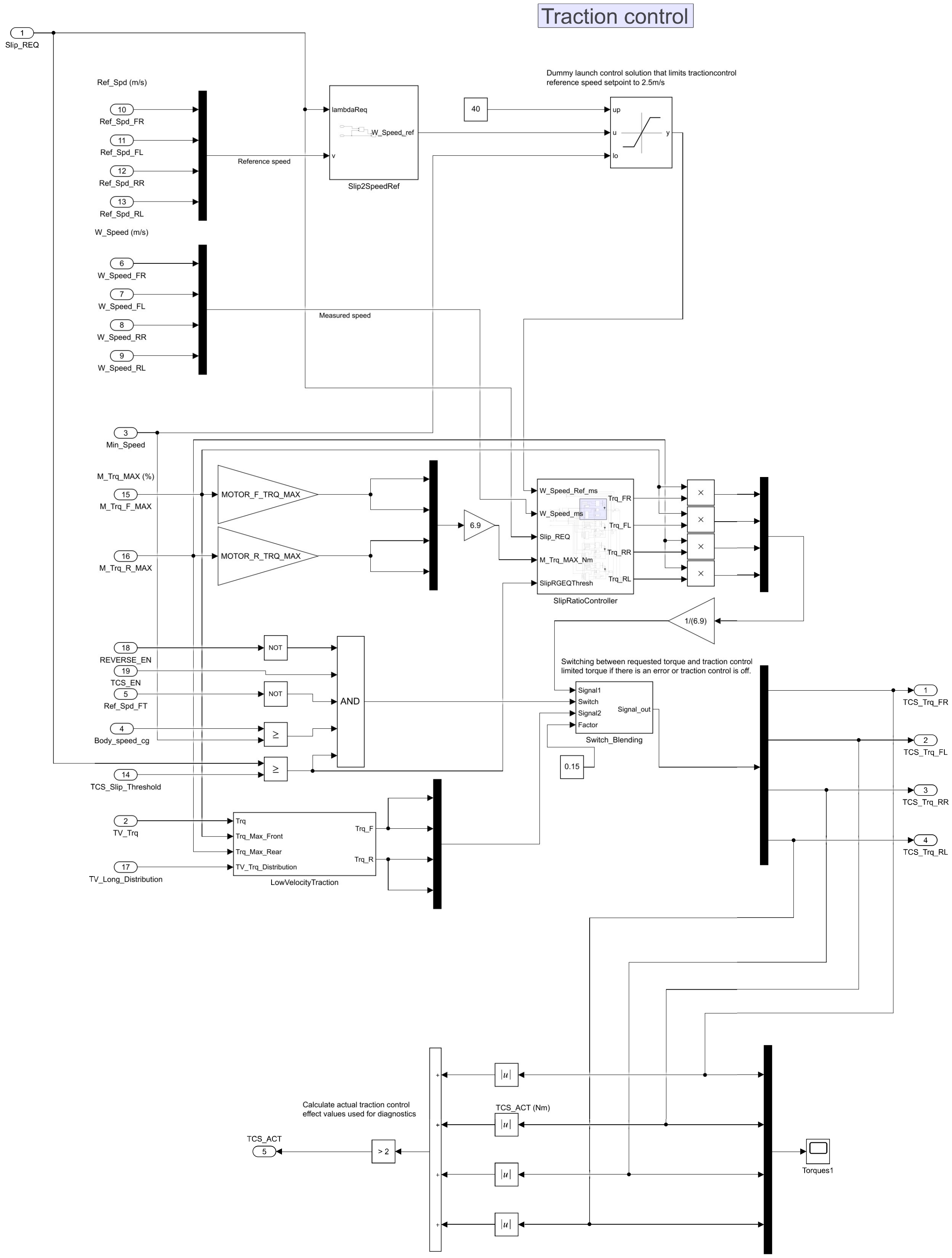

Figure A.2: Traction control block

Appendix B

Tables

Parameter name	Symbol	Value
Vehicle mass	m	285 kg
Vehicle principal axis moment of inertia	I_z	$120\ \mathrm{kg\,m^2}$
Front wheel moment of inertia	J_f	$0.1381\ \mathrm{kg\,m^2}$
Rear wheel moment of inertia	J_r	$0.1376\ \mathrm{kg\,m^2}$
Tire radius	R	0.2 m
Vehicle wheel base	Wb	1.54 m
Height of the center of gravity	h	0.27 m
Longitudinal distance of the front axle from the center of gravity	l_f	0.72 m
Longitudinal distance of the rear axle from the center of gravity	l_r	0.82 m
Longitudinal distance of the front axle from the center of pressure	$CoPF$	0.68 m
Longitudinal distance of the rear axle from the center of pressure	$CoPR$	0.86 m
Lift coefficient of vehicle body related to center of pressure	C_L	-3.5
Drag coefficient related to the center of pressure	C_D	-1.3
Aerodynamic reference area	A	$1.19\ \mathrm{m^2}$
Air density	ρ	$1.225\ \mathrm{kg\,m^{-3}}$

Table B.1: Important formula parameters (adopted from [Lá19])

Parameter name	Symbol	Value
Longitudinal shape factor	C_x	1.4
Longitudinal peak factor	D_x	1.4
Longitudinal stiffness factor	B_x	0.165
Longitudinal curvature factor	E_x	-1
Lateral shape factor	C_y	1.45
Lateral peak factor	D_y	1.4
Lateral stiffness factor	B_y	0.184
Lateral curvature factor	E_y	-0.3
Lateral self aligning factor	C_z	3.75
Lateral self aligning factor	D_z	-0.03
Lateral self aligning factor	B_z	0.11
Lateral self aligning factor	E_z	0.9

Table B.2: Formula tire parameters (adopted from [Lá19]).